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ACCELERATED SCREENING METHODS FOR PREDICTING LUBRICANT PERFORMANCE IN REFRIGERANT COMPRESSORS

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ABSTRACT

As part of an overall testing program to identify more effective bench testers for screening oils for compressors, a comparison is made in this report between data obtained from a Falex specimen tester and those obtained from a high pressure tribometer (HPT). The Falex data were supplied by three compressor manufacturers. These data were mainly obtained by "bubbling" refrigerants through the oil, whereas, the data from the HPT are based on environmental (temperature and pressure) found in specific contacts in compressors. The same contact geometries and materials are used for both the Falex and the HPT tests. The main purpose of this comparison is to see if the controlled environment and the lower loads used with the HPT produces different rankings than that obtained from the Falex tests. In general, the results obtained from the HPT did not always correlate with each other and those obtained from the Falex tester. These findings are discussed in detail in this report.

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CHAPTER I

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. Among these replacements are R-134a or (1,1,1,2) tetrafluoroethane and various blends. These new refrigerants pose new problems for manufacturers, both for their thermodynamic properties as well as their tribological properties with new oils. Therefore, manufacturers of compressors are forced to re-evaluate the tribological behavior of critical contacts in compressors, lubricated by these new oil/refrigerant mixtures, under various operating and environmental conditions.

1.2 Research Objective

Due to the rapid changes in the air-conditioning and refrigeration industry, there is an increasing interest in a more effective means of screening lubricants for compressors. Present approaches, mainly using a Falex tester, are not universally accepted since correlation between specimen and component testing is often times not very reliable.

The primary scope of this work is to do an overall study on the methodology of specimen testing. The approach taken is to tribologically simulate critical contacts found in compressors. The first part 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 the second part, the HPT will be used to obtain friction and wear data under conditions which simulate, as closely as possible, the conditions existing in compressors. Upon completion of this study, a bench tester will be recommended which will aid in the screening of lubricants used in compressors of air-conditioning and refrigeration systems. The following approach will be taken to meet this objective:

Part I

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

Part II

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, materials in contact, contact geometry and surface topography.
3. Select representative contact geometries to be tested.
4. With input from manufacturers of compressors, select oils to be screened and refrigerants to be used.
5. Test the contact geometries with selected oil/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. Analyze the tribological effectiveness of each oil/refrigerant mixture and compare with data obtained by the manufacturers from component tests.

Recommendations concerning a specific bench test facility which can be used for accelerated oil screening will be based on the correlation obtained in step 6 above, as well as the correlation obtained in Part I of this study.

1.3 Summary of Overall Research Program

The 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 will be made between the data obtained from the Falex pin and vee-block tester and the HPT to those obtained from component testing. The main thrust of this research has been governed by specific

material pairings, contact geometries and lubricants of interest to the sponsors.

The accuracy of the results obtained from this research are contingent upon the extent to which the actual conditions of the compressor are modeled. With the assistance of manufacturers of compressors, critical contacts are identified, as are their appropriate operating conditions. Included among these conditions are environmental (refrigerant) pressure and temperature, contact temperature, load, velocity, materials, geometry, surface finish and lubricant type. A description of the high pressure tribometer will be presented in full detail in subsequent chapters as well as the procedures used in the testing program. Based upon the data obtained from the specimen testing program and those obtained from the components tests, recommendations will be made about a bench tester which will aid in screening lubricants for compressors of air-conditioning and refrigeration systems.

The main focus of this report will be on Part I of this study. In Part II, to be completed by August 1994, data from the HPT will be compared to available component data.

CHAPTER II

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 of the critical components. This behavior can be examined by testing compressors under conditions that they experience in service. Such a testing program will ensure that all contacts closely reproduce expected operating and environmental conditions. The main drawback to 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 oils or materials contact pair.

To reduce cost and time to screen various materials and/or lubricants, specimen testing can be considered. With specimen testing, it is also likely that a more fundamental understanding of contact behavior will be obtained since there is more control of the numerous variables which the contact experiences in practice. 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 oil-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) oil/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.

One of the problems in conducting simulative specimen testing is the scaling factors. This factor should compensate for the differences between the geometrical and material composition which surround the component. This may be critical because the heat dissipating characteristics of the assembly surrounding the specimen and those surrounding the component might be quite different. As a result, even though the heat generation rates due to friction might be the same, the

temperatures experienced by the specimen and component contacts might be quite different. Due to this temperature difference, the tribological behavior of the two contacts might not be the same. In simulative specimen testing, therefore, it is important to estimate the contact temperature in the component and, by external temperature controls, conduct specimen tests at approximately the same contact temperature as that of the component.

Another difficulty encountered in simulative specimen testing is the requirements of generating measurable wear in a relatively short period of time. If the load and speed experienced by the component to be simulated are used in the specimen testing program, it is likely that the operating conditions are such that no measurable wear occurs during the relatively short duration of a test. If a different load and/or speed is used in order to obtain measurable wear, again it is necessary to keep the specimen contact temperature approximately the same as that experienced by the component.

The purpose of the discussion above is to emphasize the importance of simulative testing to determine the tribological characteristics of tribo-contacts. Simulating critical contacts in compressors is further complicated by the fact that specimen tests should be conducted under the environmental pressure experienced by the component since this pressure affects the amount of refrigerant dissolved in the oil and, therefore, the lubricating characteristics of the oil/refrigerant mixture. The solution to an effective oil screening specimen test facility for compressors, therefore, might not be a simple one. An approach which has a good chance of success is described in detail in subsequent chapters.

2.2 Historical Background

The tribological evaluation of components in compressors of air-conditioning and refrigeration systems, has historically been to use standard specimen test equipment, such as Falex or four-ball tests, and bubble refrigerant through the lubricant to achieve an oil/refrigerant mixture. Huttenlocher [1] used just such an approach to determine tribological characteristics of mineral oil/CRC-12 using a Falex machine. Although this method seems to have gained popularity with equipment manufacturers and oil suppliers to screen prospective lubricants, the validity of these tests in modeling actual compressor conditions has yet to be established with any degree of confidence. If the oil and refrigerant are miscible, increasing the pressure and decreasing the temperature will increase the amount of refrigerant which will saturate into the oil medium. The limitations of the more standard testing equipment is in their inability to accurately model the typical environmental conditions found in actual compressors. This generally results in oil/refrigerant mixtures being simulated at atmospheric pressure, where the possibility of other gases saturating into the mixture is very likely. The other 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 in many tribo-contacts within real compressors.

The use of actual compressors would be an alternative to specimen testing. Sundaresan [2,3] has completed many long-term tests in which various PAGs were used with HRC-134a in a reciprocating piston compressor. The results of these data are extremely useful in lubricant screening for that one compressor, but results for other types of compressors as well as specific variables cannot be accurately inferred. Additionally, this type of testing requires long-term tests which are not cost effective. For these and other reasons, the trend in the tribological evaluation for tribo-contacts in compressors has been towards the use of pressurized friction and wear machines. Such machines have been used by Sanvordenker and Gram [4] and Sanvordenker [5] who utilized the modified Falex machine to test CRC-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 oil/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. [6], Komatsuzaki and Homma [7], and Komatsuzaki, et al. [8], for the tribological evaluation of CRC-12/mineral oil, CRC-12/alkylbenzene, and HRC-134a/PAG combinations, respectively. The apparatus used in their investigation was a modified four ball tester equipped with a pressure chamber surrounding the contact. However, the pressure capability of the machine was not adequate for some of the pressures found in compressors.

Since the pressure directly affects the composition of the oil/refrigerant mixture (i.e., the amount of refrigerant saturated into the oil), it is, therefore, essential that any testing apparatus adequately simulates 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, as well as the ability to control wide temperature ranges at and around the contact. To test under these conditions, 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 simulating environments found in most all compressors.

The HPT will play an integral part in the process of designing and evaluating an accelerated screening method for candidate lubricants for compressors of air-conditioning and refrigeration systems. From this evaluation and component testing by the manufacturers of compressors, a bench test facility will be recommended for the accelerated screening of lubricants.

2.3 Technical Approach

The decision making process used to recommend a bench facility for the accelerated screening of oils 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 more standard testing procedures, 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 materials and their microstructure and hardness, oils and their additives, refrigerants, contact geometry, interfacial temperature, contact kinematics, load, initial surface roughness and environmental pressure and temperature. In many testing machines, oils cannot be exposed to environments at controlled pressure and temperature, usually only one contact geometry can be tested, the kinematic conditions are restricted to only rotational or oscillatory motion and the contacts experience very high friction which generates very high interfacial temperatures not found in the operation of the component being simulated. Tribological data obtained from these testing machines might not predict lubricant performance in a compressor since (a) the amount of refrigerant in an oil, which is a function of pressure and temperature, can significantly affect the lubricating characteristics of the oil, (b) the contact geometry affects the state of stress, generation of lubricating films as well as local surface deformation which, in turn, determines lubrication regimes, i.e., hydrodynamics, elastohydrodynamics or boundary, (c) the generation of lubricating films as well as surface damage, under boundary lubrication conditions, is affected by the kinematic conditions of the contacts; for example, unidirectional and cyclic motion might give different results and (d) 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.

One of the main goals of the proposed research program is to work closely with a few compressor manufacturers who are willing to share in-house data on the rankings of a number of oils as to their tribological performance in specific contact. This ranking would be for both specimen Falex tests as well as component tests. Operating and environmental conditions under which the Falex and component data are obtained will also be sought from compressor manufacturers.

CHAPTER III

Experimental Apparatus

3.1 Overview - High Pressure Tribometer (HPT)

The design of the facility for the tribological evaluation of critical contacts in compressors, in 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 HPT system schematic, shown in Figure 3.1, is located in the Tribology Laboratory in the Mechanical Engineering Building (MEB) on the campus of The University of Illinois at Urbana-Champaign.

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 servo motors 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 applied load, frictional forces and moments during a test. The feedback from this transducer, as well as other sensors, provide the HPT with an excellent control system.

The design of the HPT system began in the Spring of 1990 and consists of five sections: the HPT, a purging facility, a charging facility, a sampling facility and a data acquisition unit. A thorough description of the design and operation of the complete high pressure tribometer system follows.

3.1.1 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 3.2. 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 thus the unit may then be purged or pressurized.

Figure 3.1 - The High Pressure Tribometer (HPT)

The pressure chamber is sealed at the joint where the two halves meet by a custom made six-inch telescopic seal. The θ -axis motor shaft is sealed by a custom made high pressure seal, designed for dry sealing in refrigerant environments up to a pressure of 250 psi and angular velocities of up to 2000 rpm. The design of this seal is such that when a positive pressure is generated within the chamber, the lip of the seal is held securely against the wear sleeve on the shaft. The higher the pressure, the tighter the seal. However, in a vacuum, the lip is actually pulled away from the shaft, making it necessary to incorporate a V-ring seal, used in conjunction with the rotary seal to establish the required vacuum. When the chamber is subjected to a vacuum, the lip of the V-ring seal is pulled down against the shaft and forms a tight seal between the shaft and the upper bearing housing.

The spindle also has a unique internal passage. This passage allows a heat transfer fluid to be pumped through the spindle. This fluid, along with the high thermal conductivity of the aluminum spindle, basically allow the spindle to remain at a constant temperature throughout the test. The temperature of both the upper specimen and the oil, therefore, are also maintained at an approximate constant temperature throughout a test.

Figure 3.3 - End of Spindle (Specimen Mounting)

The spindle serves as the mounting face for the upper specimen holder. Figure 3.3 shows the specimen mounting hole pattern in the spindle face. It has four 10-32 UNF self-locking inserts on a 2.50 in. diameter and a fifth center hole insert for mounting specimens.

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. The hole pattern, as shown in Figure 3.4, permits the holder to be securely mounted by up to seven 10-32 UNF self-locking inserts. 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. Since the Pyrex sleeve is entirely within the pressure chamber, there is no pressure gradient across it, and only the hydrostatic pressure of the oil requires sealing. Three sight ports, machined in the wall of the cup, corresponding to sight ports in the chamber walls, allow for the viewing of the contact during testing. Also shown in Figure 3.4 is a small hole (sampling hole) which communicates with the sampling port on the outside of the chamber and allows an oil/refrigerant mixture to be drawn-off during a test.

Figure 3.4 - Specimen Mounting Hole Pattern on Cup

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 box outside the pressure chamber. There is also a thermal sensor installed in the transducer 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.

3.1.2 Thermal Systems

Thermally, the test chamber has several important features that are described in this section. Virtually all internal surfaces of the chamber can be heated. This is required to prevent condensation of refrigerant on these surfaces at the 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 is outfitted with a 400 W cartridge and the lower half contains two 500 W cartridges. The temperature of each half can be controlled from the main control panel.

The temperature of the rotary spindle, and therefore the upper specimen, is controlled from -30°C to 150°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 fluid temperature. A mixture of Union Carbide Heat Transfer Fluid (UCAR) and distilled water serve as the recirculating fluid. By using the appropriate concentrations of this fluid, different temperature ranges can be tested.

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.

3.1.3 Rotational and Axial Motions

Motion in the tribometer is generated by two independent dc servo motors. 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.

3.1.4 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 3.5. Also shown in Figure 3.5 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 3.5 - 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 $\pm 1^\circ\text{C}$.

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.

3.2 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. 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.

CHAPTER IV

Experimental Procedure

4.1 Specimen Preparation

Prior to testing and/or 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. The specimens are then handled with clean forceps during the installation process to prevent the transfer of any contaminants from the skin to the specimens which may result in inaccurate wear and/or weight measurements. The same procedures are followed upon completion of testing, after which the wear is measured.

4.2 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 4.1, and the lower specimen holder is shown in Figure 4.2. 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 4.1 - Upper Specimen Holder

Figure 4.2 - Lower Specimen Holder

4.3 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, both 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 4.3). 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 4.3 - HPT Purging Facility

4.4 Charging Procedure

Before the HPT can be charged with a refrigerant vapor, the 8.0 lb pressure vessel (Figure 4.4) 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 4.4 - HPT Charging Facility

4.5 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. In most cases, the test is conducted for one hour, unless measurable wear is not achieved under the prescribed load and speed, in which case extended time durations become necessary. Once the test has begun, the computer collects instantaneous information and from these data computes an average coefficient of friction. 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 4.5.

Figure 4.5 - Graphic Representation of Computerized Data Collection

CHAPTER V

Friction and Wear Measurements

Some common data collection schemes were used throughout this study. These included, wear scar width measurements, wear volume calculations, weight measurements, surface roughness and hardness measurements. The procedures for collecting these data are subsequently discussed in detail.

5.1 Wear Scar Width Measurement

The width of the wear scar was measured with a Nikon SMZ-2T stereoscopic optical microscope. 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 in and then averaged and recorded as the average wear scar width. A typical wear scar is shown in Fig. 5.1.

Figure 5.1 - A Typical Wear Scar on the Pin

5.2 Wear Volume Calculation

The geometry shown in Fig. 5.2 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 5.2 - Geometrical Representation for Calculation of Wear Volume

The volume worn was calculated according to the two formulae:

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

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

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.

5.3 Surface Profiles

Upon completion of an oscillatory or unidirectional test, the wear on the plate specimen (Fig. 5.3) is determined quantitatively by analyzing the wear scar depth via the DEKTAK stylus surface profiler. Some sample printouts from the DEKTAK are shown in Fig. 5.4. The wear depth is determined by computing how far below the virgin (unworn) surface the mean wear scar depth falls.

Figure 5.3 - Wear Scar on Al Pad Specimen Caused by an Oscillatory Test

Figure 5.4 - Surface Profile of Wear Scar and Virgin Surface

5.4 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, approximately equidistant from each other (Fig. 5.5).

Figure 5.5 - 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 $\mu\text{m Ra}$. 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.

5.5 Surface Hardness and Weight Measurements

The surface hardness is measured using a digital micro-hardness tester (Vickers), according to the standard procedures. All surface hardness values used in this study are averages of at least three separate readings per specimen.

The weight of the specimens is measured with an analytic balance. The accuracy of the measurements is 0.00001 g. The weight of the specimen is taken immediately before and after testing to provide an additional method for determining weight loss and thus volume worn.

CHAPTER VI

Results and Discussion

6.1. Overview

As stated before, 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 identical conditions in both Falex and HPT testers. It should be noted that tests on the Falex are conducted at room temperatures, atmospheric pressures and at relatively high contact loads. The loads used to conduct tests with the HPT are smaller than those used on the Falex and the environments in the HPT are similar to those found in compressors.

6.1.1 Material Pairings and Contact Geometry

The specimens used to conduct tests with the HPT should accurately simulate the geometries and materials used in the Falex tests. Table 6.1, on the following page, establishes the various contact geometries and material pairings used in the HPT tests. These contact geometries and materials are the same as those used to obtain the Falex data by the manufacturers. The first two companies supplied information for only one material pairing and one contact geometry, whereas, company C offered data for two different contact material pairings. The contact geometries for the specimens used are shown in Figure 6.1a and Figure 6.1b. Figure 6.1a shows the contact geometry for the unidirectional test while Figure 6.1b shows the geometry for the oscillatory tests. The loads used to conduct the Falex tests are given in Table 6.2. 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. As previously stated, tests on the HPT are conducted under environmental conditions existing in critical contacts of compressors. These data were supplied by the manufacturers and are shown in Table 6.3.

Table 6.1 - Specimen Data (contact geometry and material pairings)

Description (interface)	Company A (counterformal)	Company B (counterformal)	Company C (counterformal)	
Geometry: - upper specimen - lower specimen	3" dia. Flat Disk 0.25" dia./ 0.375" long Pin	3" dia. Flat Disk 0.25" dia./ 0.375" long Pin	1"x1" Flat 2.25" radius 0.375" long Pin	3" dia. Flat Disk 2.25" radius 0.375" long Pin
Materials: - upper specimen - lower specimen	Gray Cast Iron SAE 333 Die Cast Aluminum	SAE 356 Die Cast Aluminum Hardened Drill Rod	SAE 380 Die Cast Aluminum Carburized 1018 Steel	Gray Cast Iron Carburized 1018 Steel
Hardness: - upper specimen - lower specimen	187 Hv 113 Hv	67 Hv 886 Hv	83 Hv 817 Hv	187 Hv 817 Hv
Surface Topo: - upper specimen - lower specimen	Ground Ground	Ground Ground	Ground Ground	Ground Ground
Surface Finish: - upper specimen - lower specimen	0.047 μm Ra 0.603 μm Ra	0.406 μm Ra 0.286 μm Ra	0.188 μm Ra 0.170 μm Ra	0.042 μm Ra 0.101 μm Ra

Table 6.2 - Load and Speed used in the Falex Tester

Description	Company A	Company B	Company C
Applied Load on the Lever Arm (lbf)	350	141	350
Contact (Normal) Load at Interface (lbf)	2,100	854	2,100
Max. Initial Contact Stress (psi)	307,000	222,600	129,000
Speed (in./s)	3.8 (290 rpm)	3.8 (290 rpm)	3.8 (290 rpm)

Figure 6.1a - Contact Geometry of Specimens for Unidirectional Motion. Note that both the upper and lower specimen are secured in place by specimen holders.

Figure 6.1b - Contact Geometry of Specimens for Oscillatory Motion. Note that both the upper and lower specimens are mounted in specimen holders.

6.1.2 Environmental and Operating Conditions

One of the major advantages of the HPT over the Falex tester is its ability to control the atmosphere of the chamber. This controlled environment provides a more realistic situation experienced by actual components. The different environmental and operating conditions used with the HPT are tabulated in Table 6.3. The environmental pressures and temperatures approximately correlate with those used in actual compressors. The environmental conditions of company A simulate a critical contact in scroll compressors, whereas, conditions for company B and C simulate a critical contact in reciprocating compressors. It should be noted that the maximum temperature that the HPT can accommodate is 266 °F, somewhat lower than the temperature experienced by some of these critical contacts. In order to compare results at different loads and speeds, tests were conducted at various combination of loads and speeds, except for part 1 of company C. Due to the inavailability of SAE 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 $\pm 15^\circ$ and mean radius of 0.975 in., gives an average speed of 3.8 in./s, the unidirectional speed used with the Falex tests. In part 2 of the results for company C, a speed of 15.9 in./s was used in order to obtain measureable wear in the test duration time of one hour.

Table 6.3 - Environmental and Operating Conditions for HPT

Operating Conditions	Company A	Company B	Company C	
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	—	—	$\pm 15^\circ$	—
Env. Pressure, psig	85	30	7	7
Env. Temperature, ° F	70	266	266	266

6.1.3 Oil/Refrigerant Mixtures

The various oil/refrigerant combinations studied are given in Table 6.4. The blend shown in the table consists of 30% R-32, 10% R-125, and 60% R-134a.

Table 6.4 - Oil/Refrigerant Combinations

Company	Refrigerant	Lubricant
Company A: (oil/refrigerant combinations)	R-22 (baseline)	Mineral 1 (baseline)
	R-134a	Ester 1
	R-134a	Ester 2
	R-134a	Ester 3
	Blend	Ester 1
	Blend	Ester 2
Company B: (oil/refrigerant combinations)	Blend	Ester 3
	R-12 (baseline)	Mineral 2 (baseline)
	R-134a	Ester 4
	R-134a	Ester 5
Company C: (oil/refrigerant combinations)	R-134a	Ester 6
	R-12 (baseline)	Mineral 2 (baseline)
	R-134a	Ester 7
	R-134a	Ester 8
	R-134a	Ester 9

6.2 Results - Company A

The contact geometry and material pairings given in Table 6.1, with the specified oil/refrigerant mixtures (Table 6.4) used for Company A were tested in the HPT under the environmental and operating conditions given in Table 6.3. Comparative tests for HFC-134a and blend refrigerants with various esters have been completed for the same contacts and operating conditions as the baseline tests. In general, these tests show worse wear results than the baseline tests. The baseline tests were conducted with R22/mineral oil mixtures. Friction and wear data were collected by means of the HPT. The wear data obtained were compared with the Falex wear data supplied by the company.

6.2.1 Friction and Wear Data

Figures 6.2a - 6.2f illustrate the friction and wear data collected by means of the HPT for two different speeds and loading conditions. For each operating condition, two tests were conducted for each oil/refrigerant mixture. The reported wear data were obtained by averaging the wear scar widths of the aluminum pin as discussed in Chapter 5. 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. Average surface roughnesses of the plates used are also shown below the abscissa of the wear plots. 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.

Figure 6.2a - Wear Results (3.8 in./s at 185 lbf)

Figure 6.2b - Coefficient of Friction (3.8 in./s at 185 lbf)

Duration of Each Test = 1 Hour

Figure 6.2c - Wear Results (23 in./s at 185 lbf)

Figure 6.2d - Coefficient of Friction (23 in./s at 185 lbf)

Duration of Each Test = 1 Hour

Figure 6.2e - Wear Results (23 in./s at 370 lbf)

Figure 6.2f - Coefficient of Friction (23 in./s at 370 lbf)

Duration of Each Test = 1 Hour

A comparison between the data obtained from the Falex tester and those obtained from the HPT is shown in Table 6.5. 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 oils determined by use of the HPT are compared for the different conditions.

Table 6.5 - Ranking of Oils by Wear (Company A)

Oil	Ref	Falex (3.8 in./s at 2,100 lbf)	HPT (3.8 in./s at 185 lbf)	HPT (23 in./s at 185 lbf)	HPT (23 in./s at 370 lbf)
Ester 1	R-134a	not available	Worst	Intermediate	Best
Ester 2	R-134a	not available	Best	Best	Intermediate
Ester 3	R-134a	not available	Intermediate	Worst	Worst
Ester 1	Blend	Worst	Worst	Worst	Worst
Ester 2	Blend	Best	Intermediate	Intermediate	Intermediate
Ester 3	Blend	Intermediate	Best	Best	Best

Data from the HPT show that there exists a consistent ranking for the oils/blend mixtures under all the different test conditions, whereas, there exists a slight discrepancy in the ranking for the oils/R-134a mixtures (Figs. 6.2a, 6.2c and 6.2e). The ranking of the oils/R-134a mixtures is not available for the Falex test. The results obtained from the Falex tester indicate that wear increases in the order Ester 1 > Ester 3 > Ester 2 when mixed with the blends, while the wear results from the HPT show that wear consistently increases in the order Ester 1 > Ester 2 > Ester 3 at the various testing conditions. Even though the wear data for the three testing conditions are not appreciably different, they do indicate the pitfall of selecting a lubricant for a component whose operating conditions are significantly different than those used for the screening tests. 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 6.2g. 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 noted that, in general, there is no intuitive relationship between friction and wear, and each property must be evaluated separately. For example, Ester 3 with the blend refrigerant shows the best wear resistance, however, its friction coefficient is among the worst.

Test 47FT (23 in./s at 185 lbf)

Test 65FT (23 in./s at 370 lbf)

Figure 6.2g - Coefficient of Friction vs. Time

6.3 Results - Company B

The contact geometry and material pairings given in Table 6.1, with the specified oil/refrigerant mixtures used by Company B, were tested in the HPT under the environmental and operating conditions given in Table 6.3. Wear and friction data from the HPT and those provided by Company B from their Falex tests are compared.

6.3.1 Friction and Wear Data

Figures 6.3a - 6.3f illustrate the friction and wear results obtained by means of the HPT. Tests are conducted at low speed (3.8 in./s) and low load (50 lbf), at low speed (3.8 in./s) and high load (100 lbf), at high speed (23 in./s) and low load (50 lbf) in order to compare results under different testing conditions. High and low conditions given above are from a relative standpoint to differentiate the various tests. The reported wear data were obtained by measuring the weight of the aluminum plates before and after the test. Because the mating piece, the drill rod pin, is very much harder than the aluminum, only very small polishing marks are observed on the pin, therefore wear was not measured on the pin. Figures 6.3a and 6.3c show that the HPT data do not give a consistent ranking at the different testing conditions. It should be noted, however, that the wear difference obtained for the three lubricants is not very large. The wear results shown in Figure 6.3e are almost identical so that it would not be appropriate to rank these lubricants based on the data obtained under this testing condition. From Table 6.6, the Falex data indicates that wear increases in the order of Ester 4 > Ester 6 > Ester 5. A consistent oil ranking, based on the HPT and Falex data, does not exist. Again, it should be emphasized that the wear data obtained for the three lubricants are not that different and, therefore, the ranking given might not be very significant.

Table 6.6 - Ranking of Oils by Wear (Company B)

Oil	Ref	Falex (3.8 in./s at 854 lbf)	HPT (3.8 in./s at 50 lbf)	HPT (23 in./s at 50 lbf)	HPT (3.8 in./s at 100 lbf)
Ester 4	R-134a	Worst	Intermediate	Worst	*
Ester 5	R-134a	Best	Best	Intermediate	*
Ester 6	R-134a	Intermediate	Worst	Best	*

* differences were negligible.

Some sample plots of the friction vs. time are given in Figure 6.3g. More fluctuations and a reduction of the friction coefficient are observed at the end of all the tests conducted at a speed of 23 in./s and a load of 50 lbf. The tests conducted at this operating condition produced more wear than the tests conducted at the other operating conditions. A more thorough examination of the worn surface will be done in the near future to try to understand this frictional phenomenon.

Figure 6.3a - Wear results (3.8 in./s at 50 lbf)

Figure 6.3b - Coefficient of Friction (3.8 in./s at 50 lbf)

Duration of Each Tests = 1 Hour

Figure 6.3c - Wear Results (23 in./s at 50 lbf)

Figure 6.3d - Coefficient of Friction (23 in./s at 50 lbf)

Duration of Each Test = 1 Hour

Figure 6.3e - Wear results (3.8 in./s at 100 lbf)

Figure 6.3f - Coefficient of Friction (3.8 in./s at 100 lbf)
Duration of Each Tests = 1 Hour

Figure 6.3g - Coefficient of Friction vs. Time

6.4 Results - Company C

Again, the contact geometries and material pairings given in Table 6.1, with the specified oil/refrigerant mixtures used for Company C, were tested in the HPT under the environmental and operating conditions given in Table 6.3. The data obtained for Company C are divided into two parts with different sets of the materials. For part 1, the materials were a steel pin on an aluminum pad and for part 2, the materials were a steel pin on a gray cast iron plate. Again, wear and friction data were obtained using the HPT for both parts. These data were compared to the Falex data supplied by the sponsor (Company C).

6.4.1 Friction and Wear Data - Part 1

As previously stated, oscillatory motion was used in this part of the testing program due to the inavailability of SAE 380 die cast aluminum in a size required for unidirectional tests. The reported wear data were obtained by averaging the wear depths on the aluminum pad measured by the DEKTAK stylus surface profiler. The wear depth is determined by tracing across the wear scar at three different locations and taking average of these measurements. Figure 5.3 and Figure 5.4 show a typical wear scar on the plate specimen and its surface profile, respectively. Wear and friction data obtained from the HPT are given in Figures 6.4a and 6.4b, respectively. From Fig. 6.4a, wear increases in the order Ester 9 > Ester 7 > Ester 8. However, from Table 6.7, the Falex data show that wear increases in the order Ester 7 > Ester 9 > Ester 8. Once again the ranking of the lubricants based on the Falex data and the data from the HPT do not necessarily match.

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 6.4c. 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 6.4a - Wear results (± 3.8 in./s at 50 lbf)

Figure 6.4b - Coefficient of Friction (± 3.8 in./s at 50 lbf)

Duration of Each Test = 1 Hour

Test 50FC (± 3.8 in/s at 50 lbf)

Figure 6.4c - Coefficient of Friction vs. Time

6.4.2 Friction and Wear Data - Part 2

Friction and wear data for part 2 of this testing program are shown in Figs. 6.4d-6.4h. The wear data given in Figs. 6.4d and 6.4f are obtained by measuring the wear depth on the gray cast iron plates. The wear depth is determined by the same method described in Section 6.4.1. 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.

From Figs. 6.4d and 6.4f, 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, wear increases in the order Ester 8 > Ester 9 > Ester 7. The wear differences are appreciable between the oils tested. Table 6.7 shows that a consistent lubricant ranking is obtained from both the Falex and HPT testers

Table 6.7 - Ranking of Oils by Wear (Company C)

Oil	Ref	Falex (part 1) (3.8 in./s at 2,100 lbf)	Falex (part 2) (3.8 in./s at 2,100 lbf)	HPT (part 1) ($\pm$ 3.8 in./s at 50 lbf)	HPT (part 2) (15.9 in./s at 300 lbf) (15.9 in./s at 400 lbf)
Ester 7	R-134a	Worst	Best	Intermediate	Best
Ester 8	R-134a	Best	Worst	Best	Worst
Ester 9	R-134a	Intermediate	Intermediate	Worst	Intermediate

It is interesting to note that when tests are conducted with an aluminum/steel contact (part 1), Ester 7 showed the worst wear characteristics, while with a gray cast iron/steel contact (part 2), 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 6.4h.

Figure 6.4d - Wear Results (15.9 in./s at 300 lbf)

Figure 6.4e - Coefficient of Friction (15.9 in./s at 300 lbf)
Duration of Each Test = 1 Hour

Figure 6.4f - Wear Results (15.9 in./s at 400 lbf)

Figure 6.4g - Coefficient of Friction (15.9 in./s at 400 lbf)
Duration of Each Test = 1 Hour

Figure 6.4h - Coefficient of Friction vs. Time

CHAPTER VII

Conclusions

Various materials and contact geometries lubricated with different oil/refrigerant mixtures were tribologically evaluated with the HPT. The Falex tests were conducted at room temperature, atmospheric pressures and relatively high contact loads by bubbling refrigerant through the oils. Data obtained from the Falex tests were supplied by the participating companies. 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.

Based on the data obtained, the following conclusions can be drawn:

1. For a given refrigerant, wear data obtained by using the HPT, under various speeds and loads, generally do not give the same oil ranking.
2. The same oil ranking was also not obtained when the HPT data were compared to those obtained with the Falex tester.
3. Generally, no correlation exists between friction and wear. A tribo-contact which gives relatively low friction can experience relatively high wear and vice versa.
4. The ranking of the oils can also be a function of the material pairs in contact.

Most of the data supplied by the manufacturers were qualitative (best, intermediate, worst) and, therefore, the relative wear difference between the best and worst is not known. In this report, quantitative wear data are obtained by means of the HPT, and the oils are ranked based on these data. By their very nature, wear data, supposedly obtained under the same conditions, can show considerable scatter. One of the major questions not answered in this report is the statistical significance of the different wear rates obtained for the oils under the various conditions. If, in fact, the differences obtained are statistically significant, then the operating and environmental conditions used to screen oils must be approximately those experienced by the component, especially if accurate screening of oils which have about the same quality is desired.

CHAPTER VIII

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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 EFFORT

From 1 May 1993 to 31 December 1993, Cristino Cusano (Principal Investigator) has denoted approximately 350 hours to this project. During the same period, Carl Poppe and Hyung Yoon have each devoted approximately 700 hours to this project.

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