Measurement of the Relationship Between Oil Circulation and Compressor Lubrication in a Mobile A/C System: Part Two

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Measurement of Oil Circulation and Compressor Lubrication in a Mobile Air Conditioning System
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Abstract

This study used previously developed instrumentation to determine the state of compressor lubrication in an automotive A/C system during three transient tests. The primary technique used in the study was measurement of the electrical contact resistance at the shoe and swashplate interface. The secondary technique was dynamic measurement of the swashplate cavity pressure.

Modifications were made to the existing test stand to allow the transient testing. The first test was an oil run down test. This test was conducted until compressor failure occurred. The second test was a liquid refrigerant slugging test, and the third test was a dry start test. These tests demonstrated that the instrumentation techniques were able to detect poor compressor lubrication conditions.

This work outlines the modifications made to the test loop and presents the results of these transient tests. A conjecture is put forth to explain test results and possible future work is discuss.
Chapter 1: Introduction

1.1 Background

In refrigeration applications, knowledge of the amount of oil in circulation is critical to understanding the phenomenon of system component failure. Loss of proper lubrication for an extended period of time will cause compressor failure. In fact, failures in mobile A/C applications are most likely to occur at the compressor. Compressors are the only part of the A/C system with complex moving parts and, therefore, are the most expensive parts of the system. In order to reduce warranty costs in the most effective manner, it is desirable to monitor the compressor and predict imminent compressor failure well before it occurs.

A common mode of failure of a swashplate type automotive compressor is due to loss of lubricating oil at the shoe and swashplate interface. While the compressor is being properly lubricated, a film of oil is maintained between the shoe and the swashplate. This oil film prevents the shoe and swashplate surfaces from making contact with one another. Failure at this interface is commonly caused by a phenomenon known as scuffing. When the shoe and swashplate come into contact, scuffing can be the result. Scuffing is localized damage caused by the occurrence of solid-phase welding between sliding surfaces without local surface melting. For the mobile A/C compressor used in this study, the swashplate is made of 390 Al coated with tin and the shoe is made of 52100 Steel. Once the tin coating on the swashplate is worn through, scuffing can occur. During scuffing in this type of compressor, small bits of the softer aluminum swashplate are transferred to the hard steel shoe. Once material transfer occurs, the friction between shoe and swashplate surfaces dramatically increases, eventually causing compressor binding and failure.

To date, the link between lubrication circulation and the state of compressor lubrication is not well understood. In mobile air conditioning systems, the lubricating oil circulates with the refrigerant. Hence, knowledge of how the lubrication moves throughout the loop is a necessary step in the understanding of compressor lubrication.

Current mobile A/C systems use Refrigerant 134a with a Polyalkylene Glycol (PAG) lubricant. Based on design experience, companies within the automotive industry charge A/C loops with enough PAG oil to maintain a steady state oil mass flow rate between 3 to 5%. The automotive system manufacturers use the ASHRAE standard method [1] for determining the oil and refrigerant concentrations. The ASHRAE standard method requires capturing a sample of fluid from the liquid line of the loop. The volatile refrigerant in the sample is gradually boiled off
over a 24-hour period, leaving only the oil. Mass of the sample is measured both before and after the boiling process, thereby allowing the concentration to be calculated.

This sampling technique only gives a good representation of oil mass flow rate when the system is operating under steady state conditions and oil mass flow is the same at all locations within the loop. During periods of transient operation, system components such as the evaporator and condenser can build up oil, resulting in time varying oil mass flow at different points within the loop. Unfortunately, transient operation has been demonstrated to be the major culprit in compressor failure based on laboratory tests in the automotive industry. With this fact in mind, measurement of steady state oil flow rate is not a good indication of what is occurring during possible failure conditions. It is necessary to have a real time measurement of the mass flow rate of oil at the inlet of the compressor to determine the link between compressor lubrication and oil circulation.

In the past, In-Situ concentration measurement of mixture components has been accomplished in several different ways. It was demonstrated by Baustian et al. [2], and Meyer and Jafardo [10] that solution concentrations may be determined by measuring the speed of sound through the fluid. This ultrasonic technique measured the time-of-flight of a sound pulse moving through a small distance in the fluid. Another method of concentration measurement was through density correlations. As shown by Bayani et al. [3], a high sensitivity density meter was capable of refrigerant/oil concentration measurements. Both of the methods described previously require relatively expensive and bulky measuring equipment.

Optical refractometry can determine the concentration of components in a solution through changes in refractive index. Refractive index of a solution was correlated to the oil and refrigerant mixture concentration Newell [11]. This technique was shown to measure concentrations in a refrigeration loop in situ by Wandell et al. [16]. This sensor developed by Newell was restricted to fluids with low refractive indices and was unable to measure high percentage oil in refrigerant mixtures. The sensor used in the current study was developed with the ability to measure fluids with high refractive indices. Hence, it was able to measure the full concentration range of oil in refrigerant.

Investigation of transient oil movement in an automotive test loop has been accomplished in the liquid line with optical refraction sensors developed by Wandell [17]. During Wandell’s study, mass flow rate of oil has been shown to vary as much as 1-23% for transient conditions such as quasi-steady state clutch cycling. Although the technique used in Wandell’s study can give an accurate measurement of oil return rate to the compressor during in the steady state, it is
necessary to modify this technique to capture oil mass return rate during transient periods. The measurement point in Wandell’s experiments was on the opposite side of the loop from the compressor. Measurement of concentration in the liquid line is not a good indication of oil mass return rate to the compressor during transients, because of oil holdup in the evaporator.

Once the ability to measure oil mass return rate is obtained, compressor lubrication during transient conditions can be studied. Due to the number and complexity of moving parts in the compressor, instrumentation is difficult. Neither a direct nor an indirect measurement of the lubrication of the compressor during transient conditions has been accomplished at this point in time.

External instrumentation of the compressor housing using an Acoustic Emissions sensor was attempted by Drozdek [7]. Surface waves at the compressor housing were measured with this sensor; however, no link with shoe/swashplate interactions could be established with this measurement. Drozdek also attempted internal instrumentation through the use of a contact resistance sensor and a dynamic pressure transducer. Initial indications showed that internal instrumentation is more successful in determining the state of compressor lubrication, this study only included steady state conditions.

1.2 Objectives and Scope of Work

The objectives of this project are to investigate compressor failure and system lubrication. The mechanism by which compressor failure occurs is well documented, but the lubrication conditions that cause failure are not well understood. This investigation will serve to help determine the link between oil return and compressor lubrication conditions.

Oil return to the compressor is studied with newly developed concentration sensors in conjunction with previous existing compressor and loop instrumentation. All three devices are based upon the sensor developed by Newell [11]. Oil mass flow rate is determined at three points within the loop, in the liquid line, at the exit of the evaporator, and at the entrance to the compressor.

Once oil mass flow is understood, it is correlated with the state of compressor lubrication. Compressor lubrication state is determined with two sensors: a contact resistance sensor and a dynamic pressure transducer. These internal sensing techniques investigated by Drozdek are applied to transient conditions. These sensors measure contact resistance at the shoe/swashplate interface and dynamic pressure in the swashplate cavity. The contact resistance and dynamic
pressure transducers are applied to several conditions where compressor failure is likely to occur. The conditions studied are Oil Shut Down, Slugging, and Dry Start conditions.
Chapter 2: Experimental Facility

2.1 Test Loop Description

The testing facility used for experimentation is described in detail by Weston [18] and Wandell [16] and is shown in Fig. 2.1. The test loop consists of two main parts: the air transport loops for the evaporator and condenser, and the refrigeration loop. The air transport loops are specially constructed and instrumented ducts. The refrigeration loop consists of the evaporator, condenser, expansion valve, compressor, and piping between components. The evaporator, condenser, expansion valve, liquid separator, and are standard production components from a 1994 Ford Crown Victoria. The compressor tested is a current model FS-10 compressor.

Figure 2.1- Original Mobile A/C Loop Schematic
2.1.1 Air Loops

Air loops are built around the evaporator and condenser. Air-side pressure and temperature measurements are made with Omega type T thermocouples and differential pressure transducers at the inlet and outlet of both evaporator and condenser. To obtain an average temperature over the cross-section of the ducts, 9-thermocouple grids are used at evaporator and condenser inlet and outlets. A 7 kW electric duct heater equipped with a proportional-integral-derivative (PID) controller is installed in the evaporator loop downstream of the evaporator. This heater insures that the evaporator is experiencing a constant air inlet temperature. Evaporator air outlet temperature can be controlled to an accuracy of 1°F with the electric heater. In addition, flow mixers induce mixing of the airflow. Mixing the evaporator inlet air stream insures that the air is at a uniform temperature at a given cross section. Reheating of evaporator exit air allows driving conditions to be accurately simulated.

Also, a relative humidity system was installed by Whitchurch [19]. This system can control humidity from ambient to 99% through the use of Vaisala HMP35A humidity probes upstream and downstream of the evaporator, electrical PID controller, and electronically actuated valve. This system is not used in these experiments.

A damper valve was installed in the condenser air loop downstream of the condenser to provide control over condensing temperature. This valve allows for recirculation of warm air from the exit of the condenser to the incoming airflow at the condenser inlet. Controlling the proportions of recirculated and fresh air does not provide full control over condenser inlet temperature, but allows sufficient control within the normal operational range. Airflow rates over evaporator and condenser are controlled variable frequency drives on the blower motors. The range of evaporator and condenser airflows achievable is well within the range present during normal driving conditions.

2.1.2 Refrigerant Loop

Compressor speed is controlled with a 7.5 HP motor and variable frequency drive. In automotive applications, the compressor is belted directly to the engine in an approximately 1-to-1 ratio, so normal operational speed range from idling operation to maximum engine speed is between 800 and 6000 RPM, respectively. The operational range of the variable frequency drive and motor combination can be greater than the normal operational range.

Mass flow can be determined in the liquid line of the loop. As condensed liquid exits the condenser, the flow passes through parallel sampling sections shown in Fig. 2.2. This sampling
section allows for refrigerant and oil concentration to be measured at the exit of the condenser. A Micromotion™ Coriolis type mass flow meter is installed between the test sections and the expansion valve. This mass flow meter determines the overall refrigerant and oil mass flow in the system. This test setup uses an orifice tube type expansion valve. Orifice size is fixed and can not be varied without replacing the orifice tube.

2.2 Loop Modifications

2.2.1 Oil Separation Section

Initial tests to determine the lowest threshold of oil mass return rate for good compressor lubrication were conducted by Drozdek et al [7]. For these experiments, the suction line of the loop was modified from its original configuration. First, the standard suction line accumulator was removed from the loop. Next, the original automotive refrigeration test loop was augmented with a specially designed oil separation section shown in Figure 2.3.

The oil separation section causes liquid to be separated from the flow after the exit of the evaporator. Refrigerant and oil exiting the evaporator is passed through a helical liquid separator. This separator uses centrifugal acceleration to remove liquid from the flow.

Figure 2.2- Liquid Line Sampling Sections
Refrigerant and oil flow exiting the evaporator enters the separator at the top and is passed from top to bottom of the separator through a helix shaped passage as seen in Figure 2.4. The liquid component of the flow is flung to the outside of the helical separator and impacts on the outer wall. The liquid slowly drips down the walls to the bottom of the separator where it exits out through the liquid return line. The vapor is allowed to pass through the helix, up the center vapor flow tube, and back out the top of the separator through the vapor return line.

Another Micromotion™ mass flow meter is installed in the liquid return line to measure liquid mass flow. In the separated liquid, there is still some refrigerant dissolved in the oil. This must be accounted for to determine true oil mass flow. An oil return concentration sensor is installed at the liquid exit of the separator in series with the mass flow meter to determine this amount of dissolved refrigerant in the liquid flow. All of the components in the liquid return line are connected with 6.35mm copper tubing and Swagelock brand compression fittings.
A sampling section was included in the liquid return line downstream of the mass flow meter so that an In-Situ calibration could be performed on the concentration sensor. The sampling section was constructed with 6.35mm (1/4") and 22.23mm (7/8") copper tubing and expansion and contraction sections as shown in Figure 2.5. Two ball valves were installed on both ends of the sampling section. The ball valves insured that both the sampling section and either end of the liquid return line could be sealed. This prevented oil from leaking while the refrigerant sample was being removed.
Two sight glasses are installed in the liquid return line to allow visual inspection of the liquid return. The first is installed upstream of the concentration sensor and the second is installed downstream of the mass flow meter. Vapor bubbles passing through the liquid return line give erroneous voltage readings from the concentration sensor and will give bad calibration data if the sampling section were to contain a large portion of vapor. Visual inspection through the sight glasses allowed bad calibration data points to be identified and eliminated.

Due to the slow flow rate of liquid flow down the walls of the separator and through the liquid return line, a significant amount of heat transfer occurred between the liquid and external environment. Ideally, a more accurate simulation of an actual system would be achieved with recombined refrigerant and oil at the same temperature as at the exit of the evaporator. If significant heat transfer occurred, the liquid returning out of the liquid return line would be several degrees Fahrenheit warmer than the temperature at the exit of the evaporator. On the other hand, the refrigerant vapor tends to maintain the same temperature as the fluid at the exit of the evaporator. The warm oil and the cooler refrigerant are recombined when the liquid return line and vapor line come together downstream of the separator. In order to avoid this problem, the separator was wrapped in 12.7 mm (½”) polypropylene insulation. In addition, the liquid return line copper tubing was insulated with 12.7 mm (½”) pipe insulation cut down and taped together to fit the 6.35 mm (¼”) tubing. The additional thermal resistance imparted by the insulation allowed the liquid to remain at approximately the same temperature as the fluid exiting the evaporator.
2.3 Compressor Modifications

2.3.1 Contact Resistance Sensor

One promising technique to determine the state of compressor lubrication is measurement of electrical contact resistance at the shoe/swashplate interface. Previous work by Yoon and Cusano [20] with a pin and disk machine has demonstrated that the contact resistance between aluminum and steel sliding contacts is on the order of 10kΩ while an oil film is present. When the oil film becomes thin and the metal asperities at the shoe and swashplate surfaces come into contact with one another, the resistance was shown to drop dramatically to values less than 1Ω. A contact resistance sensor applied to an automotive swashplate compressor was developed by Drozdek et al. [7]. This work has shown that measurement of contact resistances in steady state conditions is possible and gives insight into oil film conditions while the compressor is well lubricated.

In order to create a circuit through the moving parts of the compressor, a specially designed set of brushes was created to make electrical contact with the piston. The contact resistance brushes are mounted directly over the top of a specially instrumented piston as seen in Figure 2.6. This piston has been machined with axial grooves and plated with gold along the contact surface. The grooves insure that the brush fingers maintain contact with the piston throughout the rotation of the compressor. Gold plating reduces the contact resistance between brush and piston. This helps to insure that this resistance remains very small in comparison with the resistance at the shoe and swashplate interface.

Approximately 0.5 mm of material was machined from the surface of the bridge of the piston. Teflon™ insulating pads were fixed in this space with epoxy. During the rotation of the compressor, the piston bridges sometimes make contact with the side of the swashplate due to a moment placed on the piston through the shoe. These insulating pads were necessary in order to insure that the piston bridge and side of the swashplate would not make electrical contact. If the bridge were to contact the side of the swashplate, the electrical circuit would be shorted, and the contact resistance measurement would give a false indication of damaging conditions.

The electrical circuit continues from the shoe and swashplate interface up through the shaft. A slip ring was placed on the end of the shaft to close the circuit. This slip ring allowed for relative rotation between the shaft and data acquisition system wire connections while maintaining the necessary electrical contact.
2.3.2 Dynamic Pressure Transducer

The dynamic pressure sensor was mounted over the swashplate cavity as seen in Figure 2.6. The pressure transducer will measure pressure variations close to the swashplate as a function of time. This technique is a less direct measurement of compressor lubrication than is the contact resistance measurement, but pressure signatures have been shown to be indications of lack of lubrication by Drozdek et al. [8]. Pressure signatures will serve as a predictor of lubrication loss before it can be detected by the contact resistance measurement. These pressure signatures can be correlated with the more direct contact resistance measurement during transient periods of poor lubrication.

![Compressor Instrumentation Schematic](image)

Figure 2.6- Compressor Instrumentation Schematic

Instrumentation with a pressure transducer is a much more practical method for detection of lubrication loss for a production vehicle. If compressor lubrication state can be correlated with pressure signatures, a simple and reliable technique suitable for long term testing or even production might be developed.

2.4 Concentration Sensors

Three concentration sensors are installed in the loop. The liquid line concentration sensor developed by Wandell [17] is still present upstream of the expansion valve. The sensor used by
Wandell was replaced by a newer version of this type of sensor. The new liquid line sensor was less susceptible to erroneous measurements from vapor bubbles passing through the liquid line. The two new optical refraction sensors include oil return sensor and film thickness sensor. The oil return sensor is located at the liquid exit of the liquid separator as previously discussed. The film thickness sensor is located in the suction line immediately upstream of the compressor. The film thickness sensor is installed downstream of a straight section of 36cm long, 15.88mm (5/8") copper tubing in the suction line. In the suction line, the flow is often 2-phase in nature. Refrigerant is in vapor phase and the oil is in liquid phase. The refrigerant forms a vapor core while the oil is pulled along the walls. Small changes in flow area or flow direction can change the character of this 2-phase flow. The length of straight tubing was installed to insure that the oil and refrigerant 2-phase flow was fully developed after making a turn as the flow exited the evaporator.

A second sight glass is installed in the suction line immediately downstream of the film thickness sensor. This glass allows for inspection of the flow before entering the compressor. The film sensor electronics housing was mounted on the Uni-strut™ suction line support approximately 7.62cm from the sensor. The wire leads to the sensor sight glass were made as short as possible in order to avoid electric noise induction in the circuit.

2.5 Data Acquisition

Two separate data collection systems are in the experimental facility. The first is a Hewlett-Packard data acquisition system (DAS) 1300A VXI mainframe. The system includes a HP E13226B 5-1/2 digit scanning multi-meter and three HP 1395 relay multiplexers. Data from loop instrumentation, such as pressures, temperatures, mass flows, and sensor voltages were acquired with this system. The loop parameters changed relatively slowly with time, so a high sampling rate was deemed unnecessary. Loop data was taken at approximately 2 seconds per sample. The file management was performed with a PC running HP-VEE software.

The second data-logging device was a SoMat™ Model 2500. This device allows much higher sampling rates in comparison with the HP system. The SoMat™ device can acquire data up to speeds of 100 ksamples/s. Due to the sharp transitions of the contact resistance data and the high bandwidth required for the dynamic pressure microphone, the data-logger was used to acquire data at 50 ksamples/s. Also, the film thickness data was acquired with this device because the bandwidth of the signal was unknown. The SoMat has its own 2Mb memory.
storage. Once data were taken, they were uploaded to a PC with TCE version 2.0.1 software. The data was then viewed and manipulated with EASE version 3 software as shown in the software manual [14].
Chapter 3: Optical Refraction Sensors

3.1 Introduction to Optical Refraction

At the temperatures and pressures present in the suction line of the system, vapor refrigerant dissolves in the oil liquid. Therefore, measurement of the concentration of dissolved refrigerant in oil entering the compressor was necessary to understand compressor lubrication. Initially, it was thought that the oil mass flow rate returning to the compressor would determine the compressor lubrication. In order to determine the mass flow of oil in circulation, it was necessary to know the concentrations of the oil and the refrigerant and the total mass flow rate. It was thought that, with real time monitoring of the percent concentration of oil possible, the state of lubrication of the compressor at any given time could be determined. An oil and refrigerant concentration sensor was developed to meet these ends.

The goal of this optical refraction technique is the measurement of the critical angle between two media when exposed to a point light source. In the ideal case, a point source of light originates from within a medium with a relatively high index of refraction. As the light reaches the edge of the high index of refraction material, it encounters a second material. If this second material has a refractive index that is higher than the first medium, then all of the incident light is refracted and passed through the boundary. On the other hand, if the second medium has a lower refractive index than the first, then some of the light is refracted and passed through to the second medium and the rest is reflected back, depending on light polarization. The angle at which the light stops passing through to the second medium and begins reflecting back is called the critical angle. Snell’s Law, shown in equation 3.1, determines the critical angle. All angles are measured from normal at the boundary of the two media. The refractive indices for PAG and R134a are $n \approx 1.5$ and 1.77, respectively. Using these indices of refraction, the critical angle determined by Snell’s Law is $\theta \approx 57.9^\circ$.

\[ n_1 \sin \Theta_2 = n_2 \sin \Theta_1 \]  

(3.1)

where the $n_1$, $\Theta_1$ and $n_2$, $\Theta_2$ represent the indices of refraction and light ray angles for the higher and lower index of refraction material, respectively.

Oil Return Sensor

The overall effect of the light’s interaction at the barrier of the two media is to create two distinct zones. The first zone in the radial direction, seen in Figure 3.1, is called the shadow region. This is the region where the emitted light is below the critical angle and passed through
to the second medium, creating a dark ring. The second region is the light ring. In this region, the total reflection of incident light begins for light rays greater than the critical angle. The distance from the point source to the beginning of the light ring will change based on the index of refraction of the second medium. The higher the index of refraction of the second medium, the farther the light ring moves away from the point source. The theoretical distance from the point source to the light ring for two given indices of refraction may be calculated using the geometry of Figure 3.1 and equation 3.1. The theoretical light ring distance is determined using equation 3.2.

Figure 3.1- Ideal Oil Return Sensor Geometry
\[ n_2 = \frac{n_1 D_r}{\sqrt{D_r^2 + 16 t^2}} \] (3.2)

where: 
- \( n_1 \) index of refraction of the window
- \( n_2 \) index of refraction of refrigerant/oil mixture
- \( D_r \) light ring diameter
- \( t \) thickness of the window

3.1.1 Film Thickness Sensor

The film thickness sensor is similar in design to the oil return line sensor, however, the intention is not to measure the index of refraction of the oil. The goal of the film thickness sensor is to measure the thickness of an oil film when in the presence of a refrigerant covering layer. Contrary to the oil return sensor, the thickness sensor window material is Pyrex glass. All incident light on the window passes through to the oil medium. After the light enters the oil, it will then encounter the refrigerant layer. R134a has a lower index of refraction than oil. Total internal reflection as described previously will occur at the oil and refrigerant boundary. At the critical angle, the incident light will be reflected back through the oil into the glass window and onto the photodiode. The photodiode measures a voltage proportional to the intensity of reflected light, and hence, the oil film thickness.

This refractive sensor was used to measure the oil film thickness in a sight glass in the suction line of the compressor. The thickness of the oil film travelling along the wall correlated with both oil traveling as fine droplets entrained in the refrigerant vapor flow and with total oil mass flow rate. The oil and refrigerant are assumed to have constant indices of refraction. This implies that the oil traveling along the tube walls has reached a steady state with respect to refrigerant in oil concentration.

Due to the complexity of 2-phase flow regimes, the sensor voltage is difficult to correlate to an exact local film thickness. As refrigerant vapor travels along the suction line the vapor shears the liquid travelling along the wall. Eventually, shearing forces on the oil tear off small droplets entrain them in the vapor flow, forming an oil mist. Also, the faster vapor flow also causes waves on the surface of the oil. This causes the thickness of the film to vary locally, however the average could be used as an indication of oil mass flow returning to the compressor. This sensor voltage will be correlated with the results from the oil return line measurement. The oil return line sensor could not be used during transient operation. Since transient operation is where most of the damage occurs, this sensor could be used as an indication of oncoming failure.
A schematic is presented in Figure 3.2. The distance the reflected light will be returned to the window surface can be calculated from the geometry of the sensor and the average thickness of the oil film.

![Figure 3.2- Ideal Film Thickness Sensor Geometry](image)

3.2 Sensor Design

3.2.1 Oil Return Sensor

In order to measure high percent concentrations of oil in refrigerant; it was necessary to use a window material with a high index of refraction. Use of normal Pyrex glass as in Newell’s sensor, $n=1.6$, was ruled out in this case because the index of refraction of the PAG oil was approximately $n=1.5$. Low concentrations of refrigerant in oil would cause the index of the mixture to be nearly equal to the index of refraction of the glass window. If this were to happen, the incident light would be reflected back, but at an angle close to $90^\circ$. At high angles of reflection such as this, a small change in reflection angle causes a large change in light ring diameter. This results in a high scattering of light and low reflected light level. For instance, the
theoretical light ring diameter for a glass window with a 5mm thickness, similar to the sensor used in this experiment, was calculated to be 53.9mm based on equation 3.2. To lower the critical angle, a higher refractive index material was needed. Sapphire was selected as the window material because of its high index of refraction, n= 1.77. Hence, the refractive index of the sapphire window was always significantly higher than the mixture. With the sapphire window, the light ring diameter remained relatively small and the reflected light level high. A sapphire window of 2mm thickness was selected for the sensor. The theoretical light ring diameter with pure PAG was calculated from equation 3.2 to be approximately 12.8mm with the sapphire window.

Approximation of a point source of light was accomplished with a red, 680 nm, high intensity LED. The domed surface of the LED was ground flat so that it would mount flush to the sapphire window in order to keep the light emitting crystal as close to the window as possible. In addition, the surface between the LED and window was covered with a thin coat of white paint to diffuse the light in all directions. This allowed the incident light to be emitted at all angles instead of having a large concentration of light in the direction normal to the window surface.

A photodiode was used to determine the distance of the reflected ring from the light source. This photodiode was mounted at a distance from the point source such that it was able to pick up the reflected light over the entire range of mixture concentrations. A photodiode will develop a voltage proportional to the intensity of light incident upon on it. Hence, the net effect was that the sensor diode developed a voltage nearly proportional to the index of refraction of the fluid, and correspondingly, the concentration of the mixture. The theoretical maximum and minimum light ring diameters based on equation 3.2 were calculated to be 12.8mm and 7.4mm for pure PAG (0% refrigerant in oil) and pure R134a (100% refrigerant in oil), respectively.

The sensor housing consists of two main parts: a brass, high-pressure filter body and an electronics and optics housing box. A small hole was drilled in the bottom of the brass filter body. The sapphire window was to be mounted flush with the bottom surface of this filter body. A dash size 8 o-ring was installed between the housing and window to create a seal and prevent leakage of the mixture. In addition, to reduce the error from the stray light reflected back from the inside walls of the housing to the photodiode, the inside of the housing was painted flat black.
An aluminum mounting plate machined with a hole for the LED and the photodiode was bolted over the sapphire window. The electronics and optics housing box was mounted over the optical window, LED, and photodiode. To condition the diode output, a simple amplifying circuit shown in Figure 3.3 was designed and built. The conditioning circuit was mounted in the electronics and optics housing box and the housing was mounted to the filter body section.

![Sensor Conditioning Circuit](image)

**Figure 3.3- Sensor Conditioning Circuit**

### 3.3 Oil Sensor Calibration and Testing

Two types of tests have been used to examine the sensor’s sensitivity and repeatability. First, a “static” tube test was used where the sensor measured a known refrigerant and oil concentration. The sensor was tested with two refrigerant and oil combinations, R22/Alkylbenzene and R134a/PAG. The second test consists of an In-Situ test performed in the mobile A/C loop. The static test was performed in a test section. The test section consisted of two lengths of copper tubing along with several fittings. This test section is shown in Figure 3.4. Known amounts of oil and refrigerant were added to the test section by mass measurement. In this manner, a calibration for the entire range of concentrations was obtained. In addition, a type T thermocouple was mounted directly onto the outer surface of the filter body. Mixture temperatures were recorded throughout the trials to determine if the sensor exhibited temperature sensitivity. During the static test, the refrigerant and oil mixture was continuously mixed while the sensor voltage and mixture temperature was monitored.
The In-Situ calibration was performed in the mobile A/C test stand. The oil separation section is shown in Figure 3.5. In the separation section, a sampling section was placed in series with the optical sensor was used to catch a sample of the circulating mixture. During the calibration, a sample of liquid oil and refrigerant mixture was trapped in the sampling section while the mobile A/C loop was running under steady state conditions. The sampling section was removed from the loop and the refrigerant in the sampling section was allowed to gradually boiled off, leaving only the oil. Concentration was determined by taking mass measurements of the sampling section before and after the boiling process.
3.4 Oil Return Line Sensor Calibration Results

3.4.1 Static Calibration

The first combination of refrigerant and oil examined with the sensor was of R-22 and alkybenzene. The results of this calibration and a best-fit line are shown in Figure 3.6. These results show a characteristic curve for the relationship between %R22 in alkybenzene and the sensor output voltage. This trend was observed for both of the refrigerant and oil combinations tested. The voltage curve is non-linear at the extremes in concentration and linear in the middle concentration range. These trends in sensor voltage may be a result of how the refractive index of the refrigerant and oil mixture changes. Other likely causes for this nonlinearity are from the sensor design. Recall the geometry of the sensor measuring the intensity of a light reflection with a photodiode. The shape of this photodiode is square and the light is reflected radially from the light source, which is circular. This shape difference may cause the edges of the shadow to have a gradual decline in light intensity. Further investigation of this phenomenon is necessary to fully understand the nonlinearity of the curve, but despite this observation, the correlation of concentration to the sensor voltage proves useful for measurement of oil and refrigerant concentration. The sensor performed well for this mixture, producing an accuracy of 1% throughout the entire range tested. In this static test, the sensor showed little temperature sensitivity with this combination of refrigerant and oil.
The static test method was also used to examine a R-134a and PAG oil mixture. The results of this test are shown in Figure 3.7. As the calibration of the sensor progressed, a leak in the sensor was noticed where a rubber gasket seal was used. Initially, the seal was made with a silicone gasket. This seal between the filter body and the sapphire window was replaced with an O-ring seal, which has subsequently provided a more reliable seal. In this test, the output of the sensor was not only a function of the concentration, but also a function of the temperature. The O-ring caused a temperature dependence in the sensor. The temperature dependence appeared to be due to thermal expansion and contraction of the O-ring. This affects the area of the sapphire exposed to the mixture and, correspondingly, the intensity of light incident on the photodiode. The temperature sensitivity of the sensor was repeatable and proved to be a linear relation in the temperature range studied.

The static calibration was performed with both pure PAG oil and a commercial grade PAG oil with additives. Throughout the full range of concentrations, little difference in the sensor output was observed. The presence of additives in the commercial grade PAG oil caused no change in sensor output.
Figure 3.7- Static Calibration: %R134a in PAG vs. Sensor Voltage at Constant Temperatures

3.4.2 In Situ Calibration

A liquid separator on the suction side of a compressor was used to remove oil from the flow in the suction line as described in Chapter 2. The refractive sensor was placed in series with a mass flow meter in the oil return line. With the combination the mass flow meter and calibrated concentration sensor, an accurate measurement of oil mass return to the compressor for lubrication measurements was possible.

In order to perform an In-Situ calibration, an oil sampling section was placed immediately downstream of the oil sensor in a vertical orientation in series with a sight glass. The sampling section had a series of valves to isolate a sample of oil and refrigerant. To minimize measuring error and meet space constraints, the section was flared out from the 6.35mm tubing to 22.23mm tubing and back down to 6.35mm to allow a 50g sample, approximately, to be captured within a small tube length. The sampling section was orientated vertically and a sight glass was installed to verify there was no refrigerant vapor being captured in the test section. The refrigeration loop was run until the oil concentration sensor reached a steady value. The sample section was then closed off. Next, the section was removed from the loop, weighed, and the refrigerant was bled out slowly. A final mass measurement of sampling section and lost oil allowed the concentration to be calculated. This In-Situ technique was developed in order to periodically check sensor
calibration over time because sensor aging and aging of oil in the system may change sensor calibration over time.

During the initial stages of the In-Situ calibration, the output of the sensor dropped out of range, and it was determined the epoxy holding the LED in place had failed. After reapplying the epoxy, it was observed the range of sensor output had been reduced. The epoxy forms an optical coupling of the LED to the sapphire window. From this experience it was determined that the sensor proved to be very sensitive to the optical coupling of the LED and window. A second In-Situ calibration of the sensor was performed. The results of this In-Situ calibration for the sensor and a best-fit line are shown in Figure 3.8.

![Figure 3.8- In-Situ Sensor Calibration Results](image)

The calibration results demonstrated sensor voltage was linearly proportional to concentration of refrigerant in oil. Contrary to what was observed in the static calibration, the results of the In-Situ calibration showed little change in sensor voltage with changes in temperature. This is mainly due to the fact that oil temperature varied only slightly between samples. Changes in evaporator exit conditions had little effect on oil temperature in the oil return loop.

During the course of the calibration, the refrigerant concentration in the return line stayed approximately between 15-20% refrigerant in oil. Larger concentrations of refrigerant in oil
could not be measured in the test setup due to the limitations of the mass flow meter. The in-situ calibration could not be performed with the evaporator exit in a 2-phase condition. The type of mass flow meter used in the test setup induced a large pressure drop in its interior. With a significant amount of refrigerant moving through the flow meter, the dissolved refrigerant had a tendency to flash, causing inaccurate concentration measurements from the sampling section.

3.5 Film Thickness Sensor Calibration and Results

Film thickness sensor data were taken during both steady state and transient compressor operation. This was done to correlate the sensor voltage reading with a corresponding oil mass flow rate. To accomplish this calibration, data were taken at several different steady state oil return rates as well as during slugging tests.

The oil surface waves showed in the diode voltage output. Peaks and valleys are seen in the raw data. During the steady state experiments when film thickness data were taken, the refrigerant exiting the evaporator was superheated. Thus, in the suction line, the refrigerant is vapor and the oil travels along the bottom and sides of the tubing. This was a 2-phase flow situation.

Film thickness was measured during several different steady oil return rates, 0.5%, 0.9%, and 1.2% concentration oil by mass. The raw data contain relatively large voltage fluctuations, so an average voltage was computed for each oil return rate. Each steady state data set was taken at 5 kilosamples/s for approximately a 6s period. Computing the average signal voltage over this period was an indication of the average oil film thickness on the window of the sightglass. In addition, the standard deviation of the signal was calculated. The standard deviation was an indication of the magnitude of sensor voltage variations, which should correlate with the wave structure at the oil and refrigerant interface. The results of this analysis appear in Figure 3.9.
As the film thickness on the sensor increased, the average voltage decreased. Due to the relative placement of the LED and photodiode, any increase in the light ring diameter should decrease the light intensity on the photodiode. So, the results of the calibration made sense because as film thickness increased the light ring diameter increased, and consequently, the sensor voltage decreased. As the film thickness increased, the calculated standard deviation also increased. The increase in standard deviation may be an indication of the presence of small, surface waves. As film thickness increases, the film should be able to maintain larger surface waves. The number of small, high frequency ripples that form on top of these larger low frequency waves also grows as the film thickness increases. These smaller waves should generate a larger variation in the voltage signal.

In addition to the steady state measurements, data were also taken during two liquid slugging tests. After the loop was running under steady state conditions for several hours, the loop was shut down and a 3.8kg refrigerant recovery tank connected to the suction line was placed in an ice bath. The valve on the tank was opened and the refrigerant was allowed to migrate into the
tank for about 30 minutes. Approximately 75% of the refrigerant charge was collected in the recovery tank. At this point, the tank valve was closed and the tank was turned upside down. Simultaneously the compressor was turned on and the tank valve was opened allowing liquid refrigerant to be pulled into the compressor. Data were taken at a rate of 50 ksamples/s in 1s windows at 10s intervals with the film thickness sensor throughout this test. A more detailed description of this test is provided in Chapter 5.

The average and standard deviation for the two slugging tests were calculated and plotted versus time. The results are presented in Figure 3.10. The average sensor voltage and deviation tends to increase during the first half of the test, but then experiences a sudden drop. After the drop, the voltage tended to decrease and the deviation to increase. Although the exact oil flow rate was unknown, these values agreed with the values seen during steady state. At this point during the test, oil return from the evaporator was reestablished and oil began to flow through the sensor.

![Figure 3.10- Average Voltage and Standard Deviation of Film Thickness Sensor During CR5 Slugging Test](image)

At 60s after the start of the test, the sensor experienced the drop in average voltage and deviation. This corresponded to visual observations that liquid refrigerant was no longer flowing through the sensor. Since the index of refraction of the refrigerant was relatively low, it was
expected that the light ring would decrease in diameter relative to steady oil flow. Therefore, the average voltage should increase while liquid refrigerant was flowing through the sightglass. At this time, the liquid refrigerant had completely cleaned the sightglass of oil. This should correspond to a true zero point for the sensor. During steady state testing with a zero oil flow condition, the sightglass was observed to retain a small puddle of oil on the sensor surface. This was unavoidable because the sightglass window had a small lip that tended to collect oil while the refrigerant in the suction line was superheated. It was seen that the average voltage during the “clean” portion of the slugging tests was approximately 4.8mv higher than the zero oil return data from steady state testing. Also, the calculated standard deviation was 5 times lower at this point in the slugging test than the value recorded during steady state testing.

To further investigate the wave structure on the oil surface, a Power Spectral Density analysis (PSD) was performed on the raw sensor data. The sensor output was shown to be band limited to approximately 100 Hz. The sensor components were the limiting factor. The photodiode output was unable to respond to light variations above 100 Hz due to its long response time. Wave structures have been shown by Jayani et al. [9] to have a maximum frequency of 20 Hz. These experiments were conducted with water as the liquid phase. Water is less viscous than oil, therefore, the frequency content in the test loop was expected to be smaller than 20 Hz. The operation of the sensor was not impeded by this limitation.

Electronic noise was determined to be present at 60, 120, and 240 Hz in the raw data. This noise was most likely induction noise from the electric drive motor. Unfortunately, this was unavoidable due to the placement of the sensor in the suction line. In order to get rid of the noise in the raw signal, the data were passed through a digital low pass filter with a cutoff frequency of 50 Hz. The power spectral densities of the film thickness data taken during tests of 0.5% concentration oil by mass both before and after filtering are shown in Figures 3.11a and b.
Figure 3.11a- PSD Analysis before Application of Filter

Figure 3.11b- PSD Analysis after Application of 50 Hz Low Pass Filter
Typical samples of the PSD analyses of 0.5%, 0.9%, and 1.2% concentration oil by mass are shown in Figures 3.12a, b, and c. The power spikes at the frequencies seen in these data moved around in an unpredictable manner from one steady state oil concentration to another. Since only 3 oil concentration values were examined in this study, several more oil concentration values would need to be examined to determine how these power spikes shift in frequency as a function of oil concentration. One important observation was that the power spectrum seemed to increase in the range from 20-50 Hz as oil concentration was increased. The increase in power for this frequency band was most likely the reason for the increase in standard deviation with oil concentration seen above in Figure 3.9. As power in this frequency band increased, the number of data points above and below the mean value increased, increasing the calculated standard deviation.

![Figure 3.12a- PSD Analysis of Filtered 0.5% Oil Concentration Film Thickness Data](image-url)
Figure 3.12b- PSD Analysis of Filtered 0.9% Oil Concentration Film Thickness Data

Figure 3.12c- PSD Analysis of Filtered 1.2% Oil Concentration Data
Chapter 4: Oil Shut off Experiments

4.1 Motivation

As described in previously by Drozdek [7], steady state operation of the system did not significantly damage the compressor. Although regular dips in contact resistance were seen during steady state operation, the compressor was determined to be undamaged. These contact resistances at the shoe and swashplate interfaces were shown to be well within the resistances associated with boundary and well-lubricated conditions. Initially, a test was performed to determine if the compressor required a specific minimum threshold level of oil mass return to maintain well-lubricated conditions at the shoe and swashplate interface. Drozdek determined that this was not the case. Even a small amount of returning oil could keep the compressor well lubricated. This was thought to be due to the long length of time oil is held up within the compressor, maintaining the film at the swashplate surface. Although the 100Ω and 1000Ω thresholds were demonstrated to vary with oil mass return rate, these resistance levels are not necessarily indications of damaging conditions. These resistance levels are indications of thinner oil films at the interface. The 10Ω and 1Ω threshold resistance levels are indications of damaging conditions. Drozdek demonstrated that the 10Ω and 1Ω resistance levels varied little with varying the oil mass return rate in the range tested. To better understand compressor failure, these steady state observations have motivated a study of possible damaging transient conditions.

In order to investigate possible failure conditions for the compressor, industry experts were interviewed. Several common compressor failure modes are commonly seen in the field. One common failure mode of mobile A/C compressors as describe by these experts was a low charge failure. This type of failure occurs when a loop begins to leak some of the refrigerant charge out into the atmosphere. When some of the refrigerant charge is leaked, refrigerant exiting the expansion valve during operation is no longer in a 2-phase condition. The oil mass flow rate is dramatically influenced by the refrigerant state in the liquid line. Oil mass flow rate through the liquid line is reduced when the refrigerant is vapor. While the refrigerant is vapor, the oil tends to form a film along the wall of the loop and is carried in the flow direction by shear at the vapor and liquid interface. On the other hand, when the refrigerant is in the liquid state, oil is returned easily. PAG oil is soluble in liquid R134a, so the oil is dissolved and carried along easily by liquid refrigerant. Since a good portion of the oil can be retained in the evaporator, drying out the loop downstream of the expansion valve will cause overall oil mass flow rate to
drop dramatically. During low refrigerant charge conditions, failure is due to lack of oil returning to the compressor.

When the exit of the expansion device is 2-phase, as oil mass is retained in the evaporator, smaller and smaller amounts of oil are returned to the compressor. The compressor will continue pushing oil out at the rate as it would in normal steady state conditions. The overall effect of this is to gradually remove oil from the interior of the compressor and put it elsewhere in the loop. As less oil is available to form a film at the shoe and swashplate interface, the contact resistance is expected to become smaller as the metallic surfaces begin to interact. This is the point during low charge conditions when damage begins to occur. If it were possible to determine when the surfaces begin to interact in a damaging manner, an appropriate feedback device could be used to prevent further compressor damage.

This first transient test attempted to replicate this type of failure in a controlled environment. The replication of a low charge failure is not exact in this case because of the refrigerant vapor return rate to the compressor. In a field low charge failure, the overall mass of refrigerant in the loop is reduced due to leakage, therefore the refrigerant mass flow rate is reduced. The presence of the additional refrigerant mass in the experiment raises the heat transfer rate inside the compressor at the critical interface. Since, scuffing failure is caused by a combination of frictional heating and material transfer, the presence of additional refrigerant may change the results slightly. PAG lubricant serves not only as a barrier between parts and protecting from wear, but it also serves to cool the sliding interface. The additional refrigerant mass flow rate in the experimental setup reduces the temperature, and hence, reduces the frictional heating between the sliding shoe and swashplate surfaces. In the field failure, the conditions are harsher and failure would occur in a shorter period of time due to lower heat transfer and higher temperatures at the shoe and swashplate interface. Failure in the laboratory setup was nearly identical, but it took a longer period of time without oil return than would be expected in a field failure.

**4.2 Test Setup**

To simulate low charge conditions, the test loop was modified as described in Chapter 1 to separate liquid from the flow exiting the evaporator. The test was run with approximately 10-20 °C of superheat at the evaporator exit in order to measure oil mass flow properly. This insured that the liquid content of the separated fluid was mostly oil with a small portion of
dissolved refrigerant. Large concentrations of liquid refrigerant in the liquid line of the separator would cause refrigerant flashing due to the pressure drop. 2-phase fluid moving through the mass flow meter causes erroneous measurements. If the refrigerant were to flash upstream of the mass flow meter, this would cause incorrect oil mass flow measurements. For all of the loop conditions tested, the equilibrium concentration of refrigerant in oil in the liquid return line was between 13-17% refrigerant by mass.

In order to perform the oil shut off test, the evaporator exit flow was forced through the separator section. The loop was started in steady state conditions with overall oil return at approximately 1.2% oil by mass. After the loop had been running with a steady state oil return rate for 45-60 minutes, the liquid return valve was closed off. Oil return to the compressor was seen thorough the return line sightglass to slowly become smaller and dwindle to nothing. This change was observed to occur within approximately a 15-minute period.

Only the refrigerant vapor was allowed to return to the compressor and the oil retained in the compressor gradually began to exit. As the test progressed, less and less oil was available to form the protective lubricating film causing more asperity interactions between shoe and swashplate. This oil run down test was performed on four separate compressors, CR1, CR3, CR4, and CR5. Two of these compressors, CR1 and CR3, were run until failure and the other two, CR4 and CR5, tested to determine if the data trends were repeatable.

4.3 Experimental Results and Discussion

4.3.1 Contact Resistance

4.3.1.1 Compressor CR1

The first compressor tested was CR1. This test was not conducted continuously until compressor failure. The test was actually conducted over a three-day interval, for 3-4 hours during each day of testing. The test loop was started at the beginning and shut down at the end of each day. The experimental components remained in the same state from one day to the next. The difference between starting cold every day and running the test continuously was that the heat generation at the shoe/swashplate surface took time to reach a steady state at the beginning of each test day. The results from testing over a three-day interval are not expected to vary significantly from a test conducted continuously because the system took only about 15 minutes to come to steady state with respect to system temperatures. Due to the short amount of time
required to come to steady state, it is expected that the results would not significantly change from a test that was conducted continuously.

Before compressor CR1 was installed into the experimental setup, the loop was run with a steady state oil return rate of approximately 5% oil by mass. This insured oil was distributed throughout the loop and the compressor would receive sufficient oil return at the start of the test. This leftover oil was then gradually removed from the loop by the liquid separator as the test progressed. In this test, the compressor was run with steady oil return for 15 minutes before the initiation of oil shut off. As the oil shut off was initiated, several different regimes were observed in the contact resistance signal. Immediately following oil shut off, the raw contact resistance was shown to drop drastically as seen in Figure 4.1. Asperity interactions appeared to be occurring at the shoe and swashplate interfaces. The resistance dropped to around 1Ω for a significant portion of the cycle. The test was stopped after 1 hour and the compressor opened to perform a visual check on the swashplate. Surprisingly, some smoothing of the swashplate was identified, but no significant damage was visible. At this point, the compressor was reinstalled into the loop and the oil shut down test was continued.

![Figure 4.1- Raw Contact Resistance at Beginning of Oil Shut off Test](image)

Figure 4.1- Raw Contact Resistance at Beginning of Oil Shut off Test
As the run-down of the first compressor progressed, an unexpected trend appeared. The contact resistances throughout the cycle became higher. This rise in contact resistance occurred between 1-2 hours after initiation of oil run down until about 30 minutes before compressor failure. The raw contact resistance signal during this portion of the test is shown in Figure 4.2. The raw contact resistance data during this portion of the test was similar to the contact resistance data taken during steady state as seen in Drozdek [7]. This rise in contact resistance may be due to either thickening of the tin oxide layer, the oil film, or smoothing of asperities on the swashplate.

![Figure 4.2- Raw Contact Resistance in Middle of Oil Shut off Test](image)

Approximately 12.5 hours after oil shut off, failure occurred. The contact resistance was shown to drop off to levels of 0.1-0.01Ω during the 15 minutes directly preceding the binding of the compressor. The raw contact resistance signal directly before failure is shown in Figure 4.3. Not much difference can be seen visually between the raw contact resistance data during failure and the data during run-in. The largest difference being that contact resistance during failure reached as low as 0.01Ω during failure.

Failure was defined to have occurred once the motor in the test setup was unable to maintain the torque required to turn the compressor. When the friction in the compressor increased as the compressor came close to failure, the motor in the test setup overheated. This
occurred twice in the testing of compressor CR1. In each case, the motor was shut off and allowed to cool for 2 hours, and then the test was resumed. 12.5 hours after the initiation of oil shut off, compressor friction became high enough to cause the magnetic clutch to slip. At this point the compressor was opened and inspected. The swashplate surfaces appear severely blackened and damaged on the highly loaded portion of the swashplate, but scuffing had not occurred. The swashplate surface that was closest to the magnetic clutch was called the top of the swashplate and the surface that was farthest from the clutch was called the bottom. The bottom surface of the swashplate can be seen in Figure 4.4a. The portion of the contact resistance data in which the lowest values were seen was the first half of the revolution. This corresponds to the downward piston compression. The bottom surface of the swashplate appeared to be the more damaged than the top. From these two observations it can be concluded that contact resistances between 0.1-0.001Ω were an indication that damage was occurring at the swashplate surface.

![Figure 4.3- Raw Contact Resistance Directly Before Failure of Oil Shut off Test](image-url)
Compressor failure was caused by the failure of one of the piston rings. One of the damaged pistons is shown in Figure 4.4b. The piston ring wore away and the metal surfaces of both piston and corresponding bore were damaged. Although the swashplate surface did not directly cause failure of the compressor, it is believed that friction at the swashplate surface did indirectly cause the failure. As the shoe rides over the swashplate surface, there is a difference in local velocities between the portion of the shoe that is closest to the center of rotation of the swashplate and the portion of the shoe that is farthest from the center of rotation. The local sliding velocity is highest the farther away one moves from the center of rotation. This shoe inner and outer portion velocity difference causes a moment on the piston, driving the piston bridge into the swashplate, and causing additional friction. At the same time, the thrust load drives the piston head into the cylinder wall, wearing the piston ring away. In addition, the oil during normal operation lubricates the piston and bore surfaces. The lack of oil at the piston and bore interface during run down conditions aggravated the situation. The more friction present at the swashplate surface, the greater the force driving the piston against the bore. This combination of the lack of oil at the swashplate surface and the piston and bore surfaces was believed to be the major cause of failure. Hence, although the tin swashplate coating does save the swashplate surface from causing failure, the high friction at the swashplate surface is indirectly the cause of failure.
In the field, the compressor is belted to a more powerful motor than is present in the test setup. Failure in a field test would occur later because of the more powerful motor used. However, the test motor was powerful enough to cause significant damage which would indicate field failure was not be far behind. Overall, during the testing of CR1, contact resistance started low, but rose as the test was continued. The contact resistance remained high throughout the majority of the test and experienced a sudden drop off right before failure.

By examining only the raw contact resistance signal, it was difficult to determine what was occurring at the shoe and swashplate interface. An alternative data processing technique was developed to observe trends over time as the test progressed. Arbitrary threshold contact resistance values were established at 1000Ω, 100Ω, 50Ω, 10Ω, and 1Ω. Above 100Ω there is not much interaction occurring between shoe and swashplate. Boundary layer lubrication begins to occur as the contact resistances fall to around 10Ω. Below 1Ω there is significant asperity interaction occurring at the surfaces. At contact resistances below 1Ω, damage is expected to occur. For this first oil shut off test, the percentage of time contact resistance was below each of these threshold levels was calculated. This calculation was performed by summing the number of points below the given threshold level and dividing by the total number of data points. Roughly, this calculation translates to the percentage of time during the approximately 1s data acquisition period that the contact resistance signal was below the threshold level of resistance. The threshold analysis for the beginning and end of the test is shown in Figures 4.5a and b.
Figure 4.5a- Threshold Analysis of Beginning of Compressor CR1 Oil Shut off Test

Figure 4.5b- Threshold Analysis of End of Compressor CR1 Oil Shut off Test
Even before oil shut off, the threshold resistance levels were higher than expected. This was seen previously in the raw contact resistance signal. Immediately following oil shut off, all threshold levels jump to higher percentages. The oil film at the shoe/swashplate interface decreased from the lack of fresh return oil. After this initial jump, the threshold levels gradually fall back down below the steady state values before the oil shut off. This trend in the data was surprising because it was expected that the contact resistance would drop and stay low as more and more swashplate damage was occurring. Correspondingly, the threshold levels would rise and remain high. This phenomenon was investigated later in subsequent compressor testing.

After 55 minutes into the test, the threshold resistance levels stayed approximately constant until approximately 10 hours after oil shut off. At this point compressor friction and torque increased drastically and the resistance threshold levels experienced a corresponding rise. After about 30 minutes at this high compressor torque condition, the test was stopped and resumed due to motor overheating. The threshold resistance levels remained high as the test was resumed. The test was stopped once more to allow the motor to cool. After this second test stoppage, the compressor torque became high enough to cause the clutch to slip and the test was concluded.

4.3.1.2 Compressor CR3

In order to clear up the differences between the expected and actual trends during the testing of compressor CR1, another oil shut down test was performed to failure on a different compressor. This was compressor CR3. Again, the oil shut off test was not continuous, but it was performed over a three-day interval. Compressor CR3 was run for about 5-8 hours per day until compressor failure occurred. Failure occurred 18.5 hours after initiation of oil shut off. The results of this test are slightly different than the oil shut off of compressor CR1 as seen in the threshold analysis of Figures 4.6a and b. Immediately following oil shut off the threshold resistance levels climb rapidly. After the initial drop in contact resistance and corresponding rise in threshold resistance levels, the threshold levels drop dramatically back down to about 0% within 35 minutes after the oil shut off. The threshold resistance levels maintained around 0% until approximately 1.75 hours before failure occurred. The threshold levels climb at that point.
Figure 4.6a- Threshold Analysis of Beginning of Compressor CR3 Oil Shut off Test

Figure 4.6b- Threshold Analysis of End of Compressor CR3 Oil Shut off Test
The overall trends of the CR3 data were similar to those of CR1, however, the magnitudes of the threshold levels were very different in the two cases. The CR1 threshold levels never approached zero as in the case of CR3. Even during the period of time directly before failure the threshold levels of CR1 are higher than CR3. The presence of a rise and drop in threshold levels immediately following the initiation of oil shut off was consistent with the previous test on CR1. However, the rise and drop occurred in a much shorter time interval. The low resistance values, seen in the CR1 test before the initiation of oil shut off, are not present in the CR3 test data. These differences in threshold resistance during the data sets from compressors CR1 and CR3 may be due to compressor run-in.

The CR1 oil shut off test was performed on a brand new compressor, directly from the factory. On the other hand, compressor CR3 was first run for approximately 6 hours under steady state oil return conditions before the oil shut off test was initiated. The different running time before failure in the testing of CR1 and CR3 could also be due to run-in. CR1 lasted 10.5 hours after oil shut off, whereas CR3 lasted 18.5 hours before failing. In order to remove these inconsistencies possibly associated with initial compressor run-in, all subsequent compressors were run under steady state conditions for 6.5-9 hours before any transient testing was performed.

4.3.1.3 Compressors CR4 and CR5

Due to the inconsistencies in the data from testing with compressors CR1 and CR3, two other oil shut off tests were performed on compressors CR4 and CR5. CR4 was tested in the same manner as CR3. CR4 was run for approximately 6.5 hours with steady state oil return before the oil shut off test to make sure that compressor run-in had occurred. During the oil shut off test, the compressor was run for 120 minutes with steady oil return before the initiation of oil return shut off. Once oil was shut off, the test was continued for 50 minutes with no oil returning to the compressor. The test was stopped at this point to ensure failure did not occur. The data from this test are presented in Figure 4.7. The data trends from the CR4 test are consistent with the CR3 test data trends. After oil shut down was initiated, the 1000Ω, 100Ω, and 50Ω threshold resistance levels increased and the 10Ω and 1Ω decreased. However, the contact resistance levels do not drop back down nearly as quickly as in the CR3 test. Even at the conclusion of this test, the 1000Ω, 100Ω, and 50Ω threshold levels have begun to drop down, but have not fallen below the steady state values before oil shut off was initiated.
Two separate oil shut off tests were performed on the final compressor, CR5. The first test was performed to verify the trend from the testing of CR4. The results of this test are shown in Figure 4.8. CR5 was run for approximately 9 hours with steady oil return before either of the oil shut off tests was performed. This insured that complete run-in had occurred before the initiation of the oil shut off so that resistance trends would not be due to compressor run-in. For the first oil shut off test, the compressor was run for about 3.5 hours in steady oil return conditions before the initiation of the oil shut off. Despite the fact that these two oil shut off tests are conducted on different compressors, the steady state values before oil shut off were consistent with one another. The 1000Ω, 100Ω, and 50Ω resistance levels rose immediately following the oil return shut off and began to drop back down after 30 minutes. As in the CR4 oil shut off test, the threshold levels in the CR5 test had not dropped below the steady state values at the conclusion of the test.
Figure 4.8- Threshold Analysis of Compressor CR5 Oil Shut off Test

For the second test performed on CR5, the oil return was shut off and turned back on. This test was performed to help determine if the threshold resistance level trends were an indication of shoe and swashplate surface run-in or if they were an indication of oil and refrigerant temperature effects at the surfaces. If low threshold resistance levels after oil shut off were an indication of compressor run-in, then the threshold levels should stay low after oil return was reestablished because these levels would be due to the thickening of the resistive oxide layer. However, if the resistance levels recovered to their original value before oil shut off, then another explanation for the data trends must be found.

After the 9 hours of run-in, CR5 was run in steady state conditions for 125 minutes, oil return was shut off, and 100 minutes later the oil return was turned back on. The results of this test are presented in Figure 4.9. The steady state values at all the threshold levels are similar to the two previous tests with CR4 and CR5. As the oil return is shut off, the threshold level trends are also similar to the previous tests. The 1000Ω, 100Ω, and 50Ω resistance levels jump up initially and then fall back below the steady values, and the 10Ω and 1Ω resistance levels drop and stay low while no oil is returned to the compressor. Once the oil return was turned back on,
the threshold levels returned to the same values as was seen in the steady state after about 20 minutes.

This test seemed to indicate that there were effects other than smoothening of the swashplate surface contributing to the trends in contact resistance. Another plausible explanation for the test data could be that the residual oil stuck on the surface of the swashplate was becoming more viscous and forming a thicker film. As the mechanical friction heats the swashplate surface with a lack of high refrigerant concentration oil returning to the compressor, the refrigerant concentration in the oil stuck on the swashplate could be decreasing. Although no data was available for the exact combination of R134a and PAG used in this study, oil viscosity tends to increase dramatically as refrigerant concentration decreases as demonstrated in the ASHRAE Refrigeration Handbook [1]. If this were the case, the shoe would have a harder time displacing oil out from underneath it, therefore causing thicker oil films and higher contact resistances.

Figure 4.9- Threshold Analysis of CR5 Oil Shut off and Turn Back On Test

4.3.2 Compressor Run-In

Due to the unexpected data trends during the testing of compressor CR1, further analysis of the swashplate was required. A 5mm by 8mm sample was cut from the damaged portion of
the swashplate of CR1. It was then analyzed by Auger spectroscopy and compared with a similar sized sample of virgin swashplate material. The amount of tin oxide present in the sample from the failed compressor was determined to be approximately 4 times the amount present in the virgin swashplate sample. One possible explanation for the rise in contact resistance seen in the tests after oil shut off could be the thickening of the tin oxide layer on the swashplate. Resistance of pure tin is much lower than the resistance of tin oxides. As the compressor was damaged, there may be an additional resistance imparted by an increasing tin oxide layer thickness. The shoe and swashplate may still be making significant contact at the time when the contact resistance measurement is high. Based only on the contact resistance measurement, with this extra resistance there appears to be an oil film present.

After the CR1 oil shut off test, surface profile measurements were performed on the virgin swashplates of the remaining compressors. Surface roughness measurements were made with a DekTak\textsuperscript{3} surface profilometer. Measurements were taken at 90° intervals both on the top and the bottom of the swashplate starting at TDC. Two traces were taken at each swashplate position. The four measurements at each position, two on the top and two on the bottom, were then averaged.

A second compressor, CR3, was tested in the same manner and for the same amount of time as CR1 and then disassembled for a second surface profile measurement. The second surface profile measurement showed that initially as the compressor was run, the surfaces became smoother. This initial smoothening of the swashplate is referred to as run-in. Surface profile measurements are summarized in Figure 4.10. This was thought to be due to the tin coating of the swashplate. The tin coating acts as a solid lubricant when the oil film is not sufficient to maintain lubrication. As the shoe and the swashplate make contact, the tin is pushed around the swashplate surface.

The initial swashplate surface was relatively rough. The average surface roughness was between 40-100 nm. Then as the surfaces interact, the swashplate becomes smoother as the microscopic peaks are knocked down and valleys are filled in. The second set of surface profile measurements taken after 3.5 hours of steady oil return showed that the average surface roughness decreased even during steady state operation. The surface profiles at this point in the test were between 35-45 nm. CR3 was installed back into the loop and run for another 3.5 hours with steady state oil return to determine if the swashplate surfaces would become any smoother. The values of surface did not decrease during the second 3.5 hours of operation. From this test it can be seen that the surface profiles came to a steady value within the first few hours of
operation. The low contact resistance measurements during the initial few minutes of the CR1 oil shut off test may be an indication of these surface interactions and smoothening of the swashplate surface.

Due to the results from the surface profile measurements of compressors CR3, CR4 and CR5, they were run for several hours under steady state oil return to insure that further contact resistance results would not be obscured by the run-in process. Surface profiles of CR4 and CR5 were taken after the steady state operation to demonstrate that these swashplates had reached a steady value of surface roughness. It was demonstrated that the swashplates reached a steady value of surface roughness after 6-8 hours of steady state operation. The initial surface profiles and profiles after 8 hours of steady operation for CR4 are shown in Figure 4.11. The initial surface roughness measurements for CR4 are less than CR3, however the average roughness measurements follow the same trend as for CR3. The initial roughness measurements were between 60-70 nm, after 8 hours of steady state operation, the average roughness measurements were between 30-45 nm. This was approximately the same as CR3.
4.3.3 Dynamic Pressure Transducer

During the failure of a compressor on another ACRC project, a loud screeching occurred that was audible to the human ear. It was thought that if the human ear could detect this screeching sound during failure, perhaps a pressure transducer could detect this noise before it became audible outside of the compressor housing. This prompted the installation of a dynamic pressure transducer (DPX) into the swashplate cavity as described by Drozdek [7]. If this sound were generated inside the compressor housing, a pressure transducer at this location should be able to detect these pressure variations as the compressor becomes severely damaged. The dynamic pressure transducer demonstrated it could detect loss of proper lubrication and imminent failure of the compressor during the oil shut off tests.

The signal from the pressure transducer during steady oil return is shown in Figure 4.12. The signature of the pressure signal may be due to many things, opening and closing of valves, sound generation from moving parts, scraping of the shoe and swashplate surfaces, or other sources. Unfortunately, due to the number of possible sources, the source of sound generation could not be located during this study. However, as long as the pressure transducer showed a signal that could be correlated to the condition of either proper lubrication or loss of lubrication, locating the exact source of the noise was unnecessary.
From Figure 4.12 it can be seen that the signal had a mean level that cycles at a rate of once per swashplate revolution. In addition to this, there are other pressure variations superimposed on the top of this signal. In order to gain more insight into the cyclic nature of the signal, a Fast Fourier Transform (FFT) was performed on the steady state data. A typical sample data taken during steady state oil return after the FFT analysis is shown in Figure 4.13.

![Figure 4.12- DPX Signal During Steady Oil Return Conditions](image)

Data taken after initiation of oil shut off showed a different pressure signature. These differences in the signal could be detected with the naked eye as soon as 15 minutes after the initiation of oil shut off. Contrary to what was seen in the contact resistance measurements, the pressure signal remained approximately the same from 15 minutes after the initiation of oil shut off to about 30 minutes before compressor failure. This signal was consistently present during all oil shut off tests. The mean pressure signal, seen in the case of steady oil return, was still apparent after oil return was shut off, but several regularly spaced, high frequency acoustic events not seen during the steady state were superimposed on this bulk signal. This is seen in the oil shut off test of CR1 shown in Figure 4.14.
Figure 4.13 - FFT Analysis of Steady Oil Return Data

Figure 4.14 - DPX Signal During Oil Shut off
The high frequency events occurred 10 times per revolution. Some acoustic events had a larger magnitude than others, but the number of events per revolution did stay constant. Between data sets, the acoustic event with the highest magnitude did not stay consistently the same event number. In some of the data sets, the highest magnitude event would be the 3rd event and in other data sets, it would be the 6th acoustic event as in Figure 4.14. Since there are 10 compressions per cycle, 5 pistons with 2 compressions per piston per cycle, this seemed to imply that the acoustic events had something to do with the pistons or valves.

A FFT analysis was also performed on this data. The results are shown in Figure 4.15. Frequency-wise there was not much of a difference between the FFT analyses of the data taken before and after oil shut off. However, the signal after oil shut off had a power spectral density magnitude that was an order of magnitude higher than the steady oil return signal. The magnitude of the spikes was different for each data set, but the spike between 11-13 Hz always had the highest magnitude within a given data set.

The large amplitude mean portion of the signal can be seen at approximately 12 Hz in both Figures 4.13 and 4.15. In addition, there are several other spikes in the power spectrum at 24, 36, and around 60 Hz. For both conditions, the power spikes occur at multiples of the compressor speed. These could be harmonics of the base compressor speed. Also, there appears to be three spikes in the power spectral density around 60 Hz. This was most likely induced electronic noise from the electric drive motor. Due to constraints of the experimental setup, the compressor was located approximately 1m away from the motor. This and other power sources in the lab setup could account for this spike in the power spectrum. The majority of the power in the signal was band limited to 100 Hz. Frequencies higher than 100 Hz were either not present in the signal or damped out.

The acoustic events observed in the raw data after oil shut off did not show up in the FFT analysis. This was due to the nature of the acoustic signals. A FFT analysis is good for showing signals at discrete frequencies that are continuous in the time domain. The acoustic events had a high frequency content, but existed for only a short time period, the bursts present in the data during periods of oil shut off were time-limited in nature. With this fact in mind, a wavelet analysis of this data is recommended. Wavelet analysis techniques are designed to better describe a time-limited type of signal. A wavelet analysis should be able to detect the differences in the pressure transducer data that were seen with the naked eye.
Figure 4.15- FFT Analysis of Oil Shut off Data
Chapter 5: Slugging and Dry Start Tests

5.1 Introduction

After the oil shut off tests were performed, two other common conditions where compressor failure might occur were examined. These conditions were slugging and dry startup. Slugging is when the refrigerant entering the compressor is almost completely liquid. As stated previously, oil dissolves easily in liquid refrigerant. If liquid refrigerant were to return to the compressor, it would dissolve the lubricating oil inside the compressor and carry it out into the rest of the loop. This can occur either during startup, or during steady operation.

A standard automotive system has a suction line accumulator. One of the purposes of this accumulator is to make sure that any 2-phase liquid exiting the evaporator is trapped and not sent directly to the compressor. The refrigerant gradually boils inside the accumulator and return to the compressor as a vapor. During normal operation, this is enough to keep the compressor from seeing any liquid refrigerant.

The primary situation in which liquid refrigerant entering the compressor causes problems is during startup. Normally during startup, the suction line accumulator can prevent liquid refrigerant from entering the compressor. In a certain case, however, the accumulator is unable to prevent liquid from reaching the compressor. This occurs when the automobile has experienced the greenhouse effect. When the automobile has been parked out in the sun, the cabin tends to warm up hotter than the rest of the automobile. This is because the solar radiation can enter through the windows of the vehicle, but the cabin environment does not exchange heat with its environment effectively. The evaporator of the A/C system is located in the cabin of the vehicle and maintains the same temperature as the cabin. On the other hand, the hood of the vehicle reflects solar radiation and components underneath the hood remain cool relative to the cabin. This temperature difference is enough to cause the refrigerant to evaporate from the hotter parts of the system and condense in the colder parts of the system. In this situation refrigerant tends to condense in the suction line of the compressor. When the A/C system starts, this liquid refrigerant is drawn into the compressor and washes out the lubricating oil. As operation continues, oil is returned from the accumulator, but there is a short period of time after startup that the compressor receives no return oil. This is when compressor damage may possibly occur.

Dry start conditions also occur due to this same greenhouse effect phenomenon. The difference is that thermal cycling must occur to dry out the compressor. When the vehicle is parked outside the refrigerant migrates to the suction line and into the compressor. Then at
night, the temperatures inside the cabin and underneath the hood equilibrate. There is no longer a driving force to keep the refrigerant inside the compressor and the refrigerant migrates back into other components in the system. This migration may or may not be a boiling process. The process is unknown at this time.

If the migration is not a boiling process, the refrigerant stays as a liquid and gradually leaks out of the compressor. Since the oil in the compressor dissolves in liquid refrigerant, a portion of the oil leaves along with the refrigerant. Not all of the oil is taken at once, it requires many days of thermal cycling to clean the surfaces of the compressor dry. However, if the migration is not a boiling process, the oil may not leave the compressor. Instead, the oil is cleaned off of the critical swashplate and shoe surfaces. When the vapor refrigerant condenses during the day, it condenses on all of the inner surfaces of the compressor. Once the refrigerant drops become large enough, they fall to the bottom of the housing, taking the oil with them. In this manner, the critical surfaces may be left dry and all of the oil is collected at the bottom of the compressor. If the system is not operated during the cycling process, the compressor can be completely dried out. Compressor damage occurs when the system is started after thermal cycling.

5.2 Slugging Test

5.2.1 Test Setup

Slugging tests were performed on two different compressors, CR4 and CR5. Special modifications were made to the loop in order to make this test possible. A refrigerant fill port was added to the suction line upstream of the sight glass as seen in Figure 5.1. Attached to the fill port was a 1m piece of flexible refrigerant hosing. The other end of hosing was attached to a 6.8kg refrigerant recovery tank.

Before the slugging test was started, the loop was run for 30 minutes with a steady oil return concentration of 3% oil by mass using the oil separation section. This insured that the compressor would receive adequate oil return after the liquid slugging. After the 30 min of steady state oil return, the oil separation section was valved out of the loop and the loop was shut down. The refrigerant recovery tank was placed in a 5gal bucket filled with ice water. The valve to the tank was opened and the loop refrigerant charge was allowed to condense in the tank. The tank was left in the ice bath for approximately 30 minutes after the loop was shut off. The slight temperature difference between the room temperature A/C system components and the 0° C
walls of the tank caused approximately 80% of the loop refrigerant charge to condense in the tank in this amount of time.

At this point, the tank valve was shut off, the tank was suspended upside down above the fill port, and the tank was heated with a heat gun. Once the tank came back to room temperature, the loop was started running again. Simultaneously, the tank valve was opened. Liquid refrigerant flowed from the tank directly into the compressor. Liquid refrigerant flow as observed in the sight glass lasted approximately 65-75s.

Due to memory constraints in the SoMat data acquisition system, measurements could not be taken continuously during the entire test period. Instead, contact resistance and pressure measurements were taken in approximately 1s windows at 10s intervals. This allowed the memory space of the SoMat to be broken up to cover the entire 65-75s when liquid refrigerant was flowing through the compressor.

5.2.2 Test Results

5.2.2.1 Contact Resistance

The contact resistance changed throughout the test. As the first slugs of liquid entered the compressor, the contact resistance raw data appeared similar to the values seen during steady
state testing. This was mainly due to the compressor loading. The refrigerant took about 20
seconds to redistribute itself throughout the loop. During the startup period, the pressure
differential from suction line to compressor outlet was small relative to its steady state value. As
described by Drozdek [7], the compressor loading at idling speed was dominated by the pressure
differential term. When the loading is small, there is not much of a driving force to push oil out
from underneath the shoe, allowing a thicker oil film. It makes sense that the contact resistance
was high during this part of the test.

Once the pressure differential built to its steady state level, the liquid refrigerant had started
to remove some of the oil from the swashplate surface. The contact resistance dropped for about
30 seconds. This appeared to be the time when damage may have occurred. Eventually, oil
began to return from the suction line and the oil film at the swashplate was reestablished. The
contact resistances came back up and approached steady state values.

A threshold analysis was performed on the test data. The threshold analysis of the slugging
test data from compressor CR4 is shown in Figure 5.2. The threshold levels start low, but
rapidly rise. All of the threshold levels peaked at 50 seconds after the start of slugging. The 1Ω
threshold level, the level at which asperity interaction is occurring, reached as high as 65%. This
peak was short-lived, however, and the threshold levels dropped back down rapidly. The 1Ω and
10Ω threshold resistance levels dropped back to their steady state values within 40 seconds from
the time when their peak values occurred. On the other hand, the 50Ω, 100Ω, and 1000Ω
thresholds took approximately 34 minutes to drop back down to the values seen during steady
state. These contact resistance levels were indications of thinner oil films, but were not
indications that damage was occurring. A steady value of oil film thickness took some time to
build back up. One possible explanation for this trend is that oil holdup in the compressor
increased. As more oil flowed into the compressor, more stuck to the swashplate and created a
thicker film. After 34 minutes had elapsed, the flow of oil entering and exiting the compressor
reached equilibrium and the threshold resistance maintained constant percentages for the rest of
the test.
Compressor CR5 was tested in the same manner as CR4 to make sure the data was repeatable. A threshold analysis was performed on the data to examine contact resistance trends as time progressed. The results of the threshold analysis of the CR5 test data are similar to the CR4 test data. Threshold resistance levels initially start low, jump up while liquid was entering the compressor, and then return to steady state levels as oil return was reestablished. The results of the threshold analysis of CR5 contact resistance data are shown in Figure 5.3. The peak in the threshold levels appeared at about the same time, 50 seconds, in this test as the previous test. The system took about 30 minutes to reach steady state values of contact resistance. This also was consistent with the data from the previous test.
5.2.2.2 Dynamic Pressure

Results from the dynamic pressure microphone in the slugging test are similar to the results from the oil shut off test. At the start of the slugging test, the pressure microphone signature was similar to the signature seen during steady state before the oil shut off test as seen in Figure 5.4a. There appears to be a mean pressure signal oscillating once per compressor revolution. The main difference between this signal and the signal seen during steady state was that one high frequency acoustic burst was present each cycle. This was probably due to differences in compressor machining.

After a few seconds, the refrigerant liquid started to enter the swashplate cavity and the pressure began to vary wildly as shown in Figure 5.4b. During both slugging tests, the high amplitude signal occurred approximately 10 seconds after the start of the test and continued for 30 seconds. This was due either to liquid impingement on the transducer surface, or by refrigerant liquid boiling. The exact pressure would be highly dependent on local frictional heat generation at the swashplate surface, liquid flow in the swashplate cavity, and transducer location. However, this high amplitude signal may be used as an indication of the presence of...
liquid in the swashplate cavity. In this manner, the dynamic pressure transducer could be used to
detect liquid slugging.

In the period after liquid flow through the compressor stopped and before adequate oil return
was reestablished, the pressure transducer showed a signature that was similar to the no oil return
condition of the oil shut off test. This occurred 50 seconds after the start of the test and
continued until about 110 seconds after the start of the test. The dynamic pressure transducer
signal during this time period is shown in Figure 5.4c. It is apparent by observation that the
pressure signal had 10 high frequency acoustic events per cycle.
Figure 5.4b- DPX Signal 20 seconds After Start of CR4 Slugging Test

Figure 5.4c- DPX Signal 110 seconds After Start of CR4 Slugging Test
5.3 Dry Start Test

5.3.1 Test Setup

The dry start test was also performed on two different compressors, CR4 and CR5. CR4 was tested first. Before installing the compressor into the test setup, the inside of the compressor was thoroughly cleaned. All of the inner surfaces, including the pistons, bores, valveplates, swashplate, and shoes, were wiped with a paper towel and then wiped again with a solvent. The solvent used was propanol. This was the most extreme possible case of dry start. During thermal cycling, only a portion of the oil in the compressor is washed out during a given thermal cycle. After several of these cycles, a significant portion of the oil is washed out, but there is still some left when the system is started. In these tests, the compressor was completely cleaned, so the damage was expected to be more severe than seen in a field test.

As in the slugging tests, the system was operated for about an hour with steady state 3% oil return before the start of the test. This was done with a different compressor than the compressor that was tested to insure that the compressor being tested was completely dry, yet would experience normal oil return immediately upon startup. Once steady state was reached and held for approximately one hour, the system was shut down and the compressors were switched. The dry compressor was placed in the loop and the other used to bring the loop to a steady state condition was removed.

Data was taken for approximately 5 seconds during startup. Then it was taken in 1 second windows at 10 second intervals, as in the slugging test. Since the inside of the compressor was cleaned, there was no lubricating oil on the shoe and swashplate surfaces, but there was oil in the suction line. Oil return to the compressor would be rapid, but there would be no oil for the first few revolutions of the swashplate until the oil could be drawn into the compressor. Data was taken in this manner because the most damage was expected to occur during the first few revolutions of the swashplate. The rest of the available memory of the data acquisition system was split into 1s windows to observe changes over time.

5.3.2 Test Results

5.3.2.1 Contact Resistance

The contact resistance changed almost as was predicted. During the first few seconds after the clutch was engaged, the contact resistance was low. The contact resistance values measured in the first data set of this test were similar to the values measured during failure in the oil shut
off test, or at the height of the peak in the slugging test. This was somewhat surprising due to the compressor loading. During the first revolutions of the swashplate, there was not much of a pressure differential from suction to discharge and, therefore, shoe loading should be relatively small, yet contact resistance was low. However, the compressor seemed to recovery from this low contact resistance condition rather rapidly. Within approximately 2 minutes after startup, the contact resistance values measured were similar to the values seen in the steady state measurements.

A threshold analysis of the contact resistance data from the testing of CR4 was performed. This is shown in Figure 5.5. The first data set had low contact resistance values. The 1Ω and 10Ω threshold contact resistance levels started high. The 1Ω threshold level started at 22% and the 10Ω threshold level started at 48%. As was evidenced by the high threshold resistances, significant asperity interaction was occurring at the shoe and swashplate interface early in the test. Within 3 seconds, the 1Ω threshold level dropped back its steady state level, and within 50 seconds, the 10Ω threshold level dropped back to its steady state level.

On the other hand, the 50Ω, 100Ω, and 1000Ω levels experience a dip, jump back up, and settle down to steady state levels 5 minutes later. Once again, these threshold resistance levels were indications of what was happening to the oil film on the swashplate. The initial burst of oil that returns to the compressor as it is turned on was at equilibrium with the refrigerant vapor in the suction line. As these initial droplets of oil enter the compressor they had a relatively high viscosity and formed a thick film. As operation continues, the oil in the suction line was no longer in equilibrium with its refrigerant surroundings and the concentration of refrigerant in the oil becomes higher. This oil had a slightly lower viscosity than the oil that had initially coated the swashplate, forming a thinner film and allowing the 50Ω, 100Ω, and 1000Ω levels to increase.
The same slugging test was repeated on a separate compressor, CR5, to make sure data were repeatable. This test was performed in the same manner as the slugging test of CR4. The results of this test were similar to the dry start of CR4. The raw contact resistance signal appeared low for the first few seconds of operation until oil return could be established. The contact resistance rose to steady state levels within 15 minutes of the start of the test.

A threshold resistance analysis was performed on this data. The results are seen in Figure 5.6. Again, the 1Ω, and 10Ω levels are high during the first several revolutions of the swashplate. The 1Ω threshold level started at 71% and the 10Ω threshold level started at 89%. These levels are higher than what seen during failure in the oil shut off tests, and during the peak of the slugging tests. These low contact resistances were localized to the first 4 seconds after startup and the 1Ω threshold level reached its steady state value within 10 seconds after startup as in the previous test.

The 50Ω, 100Ω, and 1000Ω threshold levels started high, experienced a dip, and then jumped back up as in the previous test. This dip in threshold levels was at first thought to be spurious data, but it was repeatable in both CR4 and CR5 tests. One possible explanation was
that the swashplate oil film thickness was changing due to oil viscosity changes, however, verification of this conjecture was not possible in the course of this study. The 50Ω, 100Ω, and 1000Ω threshold levels came back up to their steady state values within 15 minutes after startup.

![Graph showing contact resistance trends](image)

**Figure 5.6- Threshold Analysis of CR5 Dry Start Test Data**

Most of the contact resistance trends are similar between the two tests, but there was one major difference. The 1Ω threshold resistance level only started out at 22% in the CR4 test, yet it started as high as 71% in the CR5 test. This was probably due to the manner in which the data were taken. During the CR5 test, the data were acquired during startup, but in the CR4 test, data acquisition was started 0.5 seconds after startup. Data were not acquired during the first few revolutions after the compressor clutch was engaged. It was likely that the contact threshold value would have been this high if data acquisition was started immediately during startup because the two threshold analyses would nearly line up if the CR4 data were shifted by this amount.
5.3.2.2 Dynamic Pressure Transducer

The results of the dynamic pressure transducer measurements were similar for both CR4 and CR5 tests. Some changes in the pressure signal were observed during the dry start tests. At the start of the test, the dynamic pressure signal was similar to the signal seen during steady oil return conditions. This signal seemed to indicate that the compressor was being properly lubricated, but instead this signal was probably due to the low piston and swashplate loading at the start of the test. If the discharge line and suction line pressures were subtracted, the pressure differential across the compressor could be calculated as shown in Figure 5.7. This pressure differential, within 60 seconds after startup, acted as if it was a step response to a first order system. If pressure differential were treated as a first order system, the time constant of this system would be approximately 8 seconds. The signal from the first data set was taken while the pressure differential was ramping up. The pressure transducer signal taken about 1s after startup is seen in Figure 5.8a.

![Figure 5.7- Loop Pressures during Dry Start Test](image)

The next set of data was taken 10 seconds later. The high frequency acoustic bursts observed during the oil shut off test were also present in this data as shown in Figure 5.8b. The pressure differential at the point this data were taken was 77% of the steady state value. At this point in
the test the piston and shoe loading was close to its value during steady state, but the oil in the suction line did not yet fully lubricate the swashplate as demonstrated by the contact resistance data. This may explain why the acoustic bursts were present in the transducer data.

In both dry start tests the acoustic bursts appeared only briefly for between 10-20 seconds. After that, the pressure transducer returned to showing the same signature seen during well-lubricated conditions. The pressure transducer signal stayed the same for the remainder of the test. This was because, although the piston and shoe loading was at its steady state value, the compressor had received enough oil return to lubricate the compressor surfaces and form a film on the swashplate surface. The pressure transducer signal 40 seconds after the start of the test is shown in Figure 5.8c.

Figure 5.8a- DPX Signal at Beginning of Compressor CR4 Dry Start Test
Figure 5.8b- DPX Signal 15 seconds after Beginning of Compressor CR4 Dry Start Test

Figure 5.8c- DPX Signal 30 seconds after Beginning of Compressor CR4 Dry Start Test
Chapter 6: Conclusion and Recommendations

6.1 Conclusions

Experimental results and the previous analyses have shown:

- The oil concentration sensor was able to measure high concentrations of oil in refrigerant. This sensor showed a temperature sensitivity during the static calibration, however this temperature dependence disappeared during the In Situ calibration. A linear best fit line was found for oil concentration vs. sensor voltage for the In Situ calibration.

- The film thickness sensor was determined to have the ability to measure oil film thickness in the suction line. The average sensor voltage decreased with increasing film thickness. In addition, the filtered sensor output was determined to show larger signal variation with increasing film thickness. This corresponded to increasing the power in the frequency range 20-50Hz.

- The contact resistance was shown to decrease dramatically during the final stages of oil shut off tests during the 30 minutes before compressor failure. Also, the contact resistance thresholds, which are possible indications of compressor damage, started low, peaked, and returned to steady state levels during slugging tests. Finally, during dry start tests, high threshold resistance levels were limited to the first 1-2 seconds after startup.

- The dynamic pressure transducer demonstrated the ability to detect low and no oil return conditions during oil shut off tests. The pressure transducer showed large pressure variations while liquid refrigerant was moving through the compressor. This type of signal could be used as an indication of slugging conditions. The same signal seen during poor lubrication conditions was also seen in the slugging test, after the liquid had moving through the compressor, but before oil return was reestablished. The poor lubrication signal was seen briefly during dry start tests.

6.2 Future Work

Although the instrumentation techniques have been validated in this study, future work is recommended in several areas:

- Further examination of the oil return line sensor is needed to understand what is the cause of the temperature sensitivity of the device. Once the cause of the temperature sensitivity is found, a new sensor design could be made to eliminate this effect.
• Additional experimentation is required to examine compressor run-in and its relation to contact resistance measurements. Although a conjecture for the contact resistance data trends was put forth in this work, further experimentation is required to prove or disprove this conjecture if this technique is to be used in a production environment. Many more compressor failures are required to obtain a statistical measure of what the contact resistance measurements mean in terms of time to compressor failure.

• Supplementary work is required to determine the cause of the high frequency acoustic events seen in the dynamic pressure transducer data. Although the presence of these acoustic events has been shown to correlate to poor lubrication conditions, the exact cause for the events was not found. Another data reduction technique, wavelet analysis, should be used to determine if these events are occurring. If the automobile's computer could determine if the compressor was being poorly lubricated using a wavelet analysis, then an appropriate feedback technique could be developed to protect against compressor failure.
Bibliography


