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ABSTRACT

Oil return from the evaporator to the compressor crank case is a critical requirement affected by the refrigerant/lubricant properties, gas velocities and piping geometry. Refrigerant R-22 has been successfully used with mineral oil in residential air-conditioning and heat pump systems (AC/HP). R-407C and R-410A are two chlorine free HFC refrigerant candidates to replace R-22. This paper describes the test methods and the results from controlled oil return experiments in a 3 ton residential split system. The HFC refrigerants were evaluated with mineral oil (MO) and polyol ester lubricants (POE) and are compared to the R-22/MO baseline. The results support the decision to use POE's with HFC refrigerants in residential split heat pump systems.

INTRODUCTION

The performance and reliability of a residential heat pump system are affected by the refrigerant and lubricant properties. The solubility and miscibility characteristics affect the oil return characteristics. The air conditioning and commercial refrigeration industries have successfully applied refrigerant/lubricant pairs with varying miscibility (i.e. CFC-12/MO systems with complete miscibility, R-22/MO systems with partial miscibility, and R-502/MO systems with poor miscibility).

The objective of this study is to evaluate the oil return characteristics of some recently introduced refrigerant/lubricant pairs. R-407C and R-410A are two HFC substitute refrigerants for R-22 in AC/HP system applications[1]. Based on the commercial refrigeration experience with HFC substitutes for R-12 and R-502, POE lubricants are the leading candidates for the HFC refrigerants[2-10]. However, POE lubricants have two major drawbacks: hygroscopicity and high cost. There is interest and need to know to what extent is it possible to replace POEs with a lower cost, but immiscible, oil such as mineral oil. It is the purpose of this study to experimentally investigate the oil return behavior of mineral oil and the HFC's most likely to replace R-22. Specifically, the following refrigerant/lubricant pairs are investigated in a residential heat pump; R-22/MO, R-407C/POE, R-407C/MO, R-410A/POE, and R-410A/MO.

TEST FACILITY

The test facility consists of primarily two components, an indoor loop and an outdoor chamber[11]. The indoor unit of the AC/HP system is located in a closed air loop simulating indoor conditions while the outdoor unit of the AC/HP system is placed in an environmental chamber that simulates outdoor conditions. The test facility meets or exceeds ASHRAE Standard 116-1983 requirements [12].

The test unit is designed for R-22 and is rated at 10.6 kW (3 tons). The indoor heat exchanger of the AC/HP system is an A-coil with four parallel passes and a cross-over pattern between the two sections of the A. The outdoor heat exchanger of the test unit is a plate-fin coil, two rows deep and has four parallel passes. The indoor and outdoor heat exchangers utilize smooth tubes. Thermostatic expansion valves are used for both the heating and cooling modes. The system is not designed for an accumulator and one is not used for the data presented here. Two scroll compressors are utilized in this study, one for the medium pressure refrigerants (R-22 and R-407C) and one for the high pressure refrigerant (R-410A). Neither of the compressors poses a crank case heater. Each of the compressors is modified by installing a sight tube and sight glasses, as shown in Figure 1. The sight tube, which is graded in millimeters, allows the liquid level in the compressor to be quantified while the sight glass allows the fluid flow to be qualified. The liquid level in the compressor case is the sum of the liquid refrigerant level and the oil level. However, during normal operation the liquid in the compressor is solely lubricant. The liquid level and the flow pattern in the compressor are observed with a video camera. The video image is used after the completion of the test to determine the numerical value of the oil level as a function of time.

EXPERIMENTAL PROCEDURE

First, a battery of tests is conducted to determine the appropriate amount of refrigerant to charge into the AC/HP system. This is called the charge optimization and is done for each refrigerant tested. The charge optimization consists of conducting several tests with varying refrigerant charge at the ASHRAE B (steady-state cooling) test conditions[12]. The charge which produces the highest COP for the refrigerant in question is used for all of the other oil circulation tests. The ASHRAE B test is chosen for the charge optimization since the SEER is most sensitive to this test.

After the amount of refrigerant to be used in the system has been determined, three tests are used to study the oil return performance of each refrigerant/lubricant pair. Each of the three tests is three hours in length during which the oil level is measured as of function time. The first test is the steady state cooling test (ASHRAE B). The ASHRAE B test is the least rigorous of oil return tests considered here. Since the most significant changes in oil level occur when the compressor is initially started, a cyclic test of five minutes on and five minutes off is used to quantify the unsteady effects of oil return phenomena. The cyclic test is performed at ASHRAE D test conditions[12]. The last test, the simulated oil pump out test, is developed to evaluate the worst case oil return scenario and is conducted at ASHRAE B test conditions[12]. In this test most of the oil is removed from the compressor and injected in the discharge line immediately after the compressor. This test reveals the time required for the lubricant to return in the event of a catastrophic oil pump out. This situation can occur to a lesser extent in the field when the compressor is in a cold location relative to the rest of the system. This would cause refrigerant to condense in the shell, possibly producing a large volume of liquid refrigerant in the shell. The liquid refrigerant would then elevate the lubricant level into the scroll set and upon start-up, the scroll would pump out all of the lubricant.

When the oil was changed from one test series to another, the system was flushed with a solvent and then charged with the new oil. The oil charge, which was 32 oz., was then replaced four to six times in between running the system for 30 minutes. With this procedure, the concentration of the oil being studied exceeded 97% as measured with a refractometer. The refrigerant in question was then added. For each series of tests a new refrigerant charge was used to ensure that no oil contamination was introduced through recycled refrigerant. Therefore, using these procedures and the three oil return tests developed here, the following refrigerant/lubricant pairs are investigated: R-22/MO, R-407C/POE, R-407C/MO, R-410A/POE, and R-410A/MO.

RESULTS

The results from the three tests used to characterize oil return can be seen in Figures 2, 3, and 4. Each of these figures is a plot of the liquid level in the case of the compressor as a function of time. It should also be kept in mind that the oil levels from the R-410A tests can not be compared in an absolute sense to the other refrigerants since a different compressor was used for R-410A.

Figure 2 is the plot of the oil level for the steady state test. Most of the refrigerant/lubricant pairs exhibit similar trends. Specifically, soon after start-up, the liquid level in the compressor dramatically drops. This occurs because the sudden foaming and out gassing of the refrigerant/lubricant pair provides a mechanism for the compressor to pump the lubricant out of the compressor shell. The sudden foaming is caused by the abrupt drop in pressure and may be enhanced by liquid refrigerant initially entering the compressor shell. For most of the refrigerant/lubricant pairs studied, the oil returns to the compressor and a steady state oil level is achieved; typically within less than fifteen minutes. Generally, the level remains essentially constant for the duration of the test. After the compressor has stopped, the oil level increases over the next five minutes to a level that is five to ten millimeters higher than the steady state level. This occurs because the lubricant which had been along the shell due to the agitation slowly migrates to the oil sump under the influence of gravity.

All of the refrigerant/lubricant pairs settle at different steady state oil levels, as seen in Figure 2. Among the medium pressure refrigerants, R-22/MO maintains the highest level of about 65 mm (2.6 in) followed closely by R-407C/POE at 60 mm (2.4 in). R-407C/MO has the lowest steady state oil level of the medium pressure refrigerants. Similar results are obtained for R-410A, where R-410A/POE significantly outperforms R-410A/MO. Furthermore, R-410/MO has the distinction of being the only refrigerant/lubricant pair that does not reach steady state. This implies that given sufficient time all of the lubricant may leave the compressor shell which would result in an eventual compressor failure. Even if the compressor did not fail, the accumulation of MO in the heat exchangers could result in a loss of capacity and efficiency due to an increased heat transfer resistance. However, for this system no negative oil accumulation effects could be measured.

The results from the cyclic test are plotted in Figure 3. As with the steady state test, most of the refrigerant/lubricant pairs exhibit similar trends. During the first 50 minutes, the liquid level changes as cyclic equilibrium is established. After the first 50 minutes, the oil levels are plotted only for the end of the on and off times since the levels were found to be reproducible. After cyclic equilibrium is reached, it is observed that for each refrigerant/lubricant pair, there is a significant change in oil level between the on-time and the off-time, with the off-time having the higher level. This occurs for the same reason that the liquid level rises at the end of the steady state test.

The liquid level at cyclic equilibrium for the refrigerant/lubricant pairs follows the same trends as the steady state test. For the medium pressure refrigerants, the refrigerant/lubricant pairs in order of increasing liquid level are: R-407C/MO, R-407C/POE, and R-22/MO. As before, POE performs better than MO for R-410A. It is interesting to note that the liquid level of R-407C/MO declines throughout the test. Hence, given sufficient time the R-407C/MO pair would probably cause a compressor failure. It is interesting to note that the cyclic test causes a continuous loss of lubricant for R-407C/MO but not for R-410A/MO, which lost lubricant continuously during the steady state test. As expected, this indicates that oil migration is not a function of miscibility alone.

Lastly, the oil pump out test results are reported in Figure 4. For this test, the immiscible refrigerant/lubricant pairs behave differently than the miscible refrigerant/lubricant pairs. Generally, the miscible refrigerant/lubricant pairs initially loose lubricant for roughly one minute when the compressor is first turned on. This phase is similar to the steady state test. Next, the lubricant in the discharge pipe is motivated through the system. As a result, the liquid level in the compressor shell starts to rise. The steady state liquid level, which is significantly greater than the initial liquid level, is reached in about 20 minutes. These trends are not exhibited by the immiscible refrigerant/lubricant pairs (R-407C/MO and R-410A/MO). R-407C/MO and R-410A/MO have a different character. Initially the liquid level in compressor fluctuates wildly. This is followed by a minimal recovery in the liquid level at about 7 minutes. After which, the liquid level approaches its steady state value within about 10 minutes. The steady state liquid level for the immiscible refrigerant/lubricant pairs does not change appreciably from the initial liquid level which indicates that the lubricant never returns to the compressor. Therefore, in the event of catastrophic oil pump out with an HFC/MO pair, the oil would never return to the compressor shell and a compressor failure would ensue.

DISCUSSION

The oil transport inside the condenser is influenced by high velocity gas at high temperatures. In the liquid line, the mechanism of oil transport is quite different between the miscible and immiscible cases. When the oil and refrigerant are miscible, there exists only single phase flow. When they are immiscible, as in the case of HFC/MO, there is two phase flow. It can be observed that the oil floats on top of the refrigerant stream in the form of droplets that at times combine to a rivulet. The droplets are pushed and rolled along the top tube wall by the faster moving liquid.

Along the evaporator, at low temperatures and pressures, the amount of liquid refrigerant decreases and the lubricant concentration in the remaining liquid refrigerant increases accordingly. Here viscosity effects are dominant. Of the five refrigerant/oil combinations studied, only R-407C/MO and R-410A/MO are not miscible and have poor solubility. Thus the predominant oil return mechanism for these fluids, shear forces that motivate the oil film, are not sufficient for reliable oil return.

In the suction line, the temperatures are slowly increasing and the vapor velocities are higher than the average evaporator vapor velocity. Here R-407C/MO and R-410/MO are almost completely immiscible and the solubility is poor. Nevertheless, the viscosity is decreasing with increasing temperatures facilitating oil return. The model of a liquid film being pushed along the tube wall is consistent with the experimental results observed in Figure 3. When the oil is very viscous, because of low temperatures and/or low refrigerant solubility, then a thicker oil film is required to allow for effective momentum transfer from the refrigerant to the oil film. Accordingly, there should be less oil in the compressor. This is the case with the HFC/MO pairs.

One can make a strong case that with agitation and low oil viscosities, there is sufficient mixing to overcome the tendency of the fluids to separate into two layers. However, even in the liquid line, the effects of immiscibility are clearly observed during operation of the system. Even in immiscible cases, the viscosity of the oil rich phase depends on the solubility of the refrigerant in the oil and a lower viscosity will most likely result into the oil film being pushed along the tube wall more effectively.

There are several parameters that will influence the oil return process. Since the force moving the oil film along the wall is a result of the momentum transfer from the refrigerant to the oil, this effect is strongly influenced by the refrigerant velocity. In addition, the pressure in the suction line controls the vapor density and mass flow rate. Thus it controls the momentum transfer from the refrigerant to the refrigerant vapor to the oil film on the tube wall. Lastly, the thickness of the oil film and whether the oil film covers the entire wall homogeneously or flows in the form of rivulets depends on the surface tension difference between the oil and the refrigerant vapor. In summary, it is expected that the interdependence of the following variables (which may not be independent of each other) needs to be studied: 1) Geometry of the system, 2) vapor velocity, 3) vapor density, 4) liquid viscosity, 5) oil/liquid refrigerant/metal surface tension difference, 6) surface structure of the tube, 7) film thickness, 8) temperature, 9) pressure and 10) transients caused by the load and environment.

CONCLUSIONS

- A comprehensive test protocol, which consists of a steady state test, a cyclic test and the simulated oil pump out test, has been developed. The most discriminating test among these three tests is the simulated oil pump out test.
- R-407C/POE and R-410A/POE have reliable oil return characteristics which are similar to R-22/MO.
- R-407C/MO and R-410A/MO have unreliable oil return characteristics. Both of these refrigerant/lubricant pairs exhibited continuous lubricant removal from the compressor which could lead to eventual compressor failure. Furthermore, the lower oil level at steady state indicates that oil is accumulating in system components. This may have negative consequences in terms of system performance and reliability.

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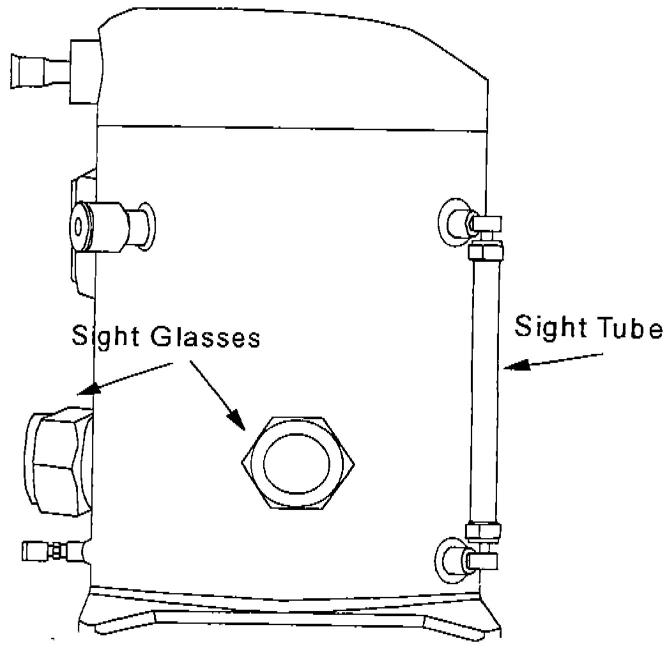

Figure 1 Compressor fitted with sight glasses and sight tube

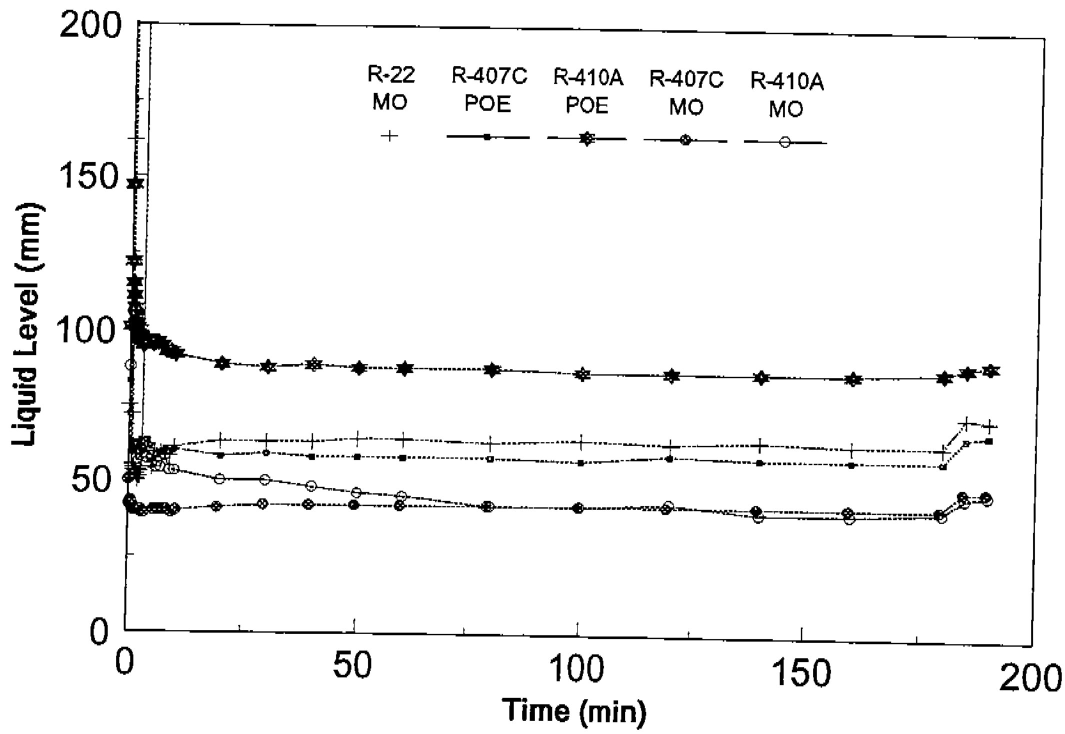

Figure 2 Steady State Test

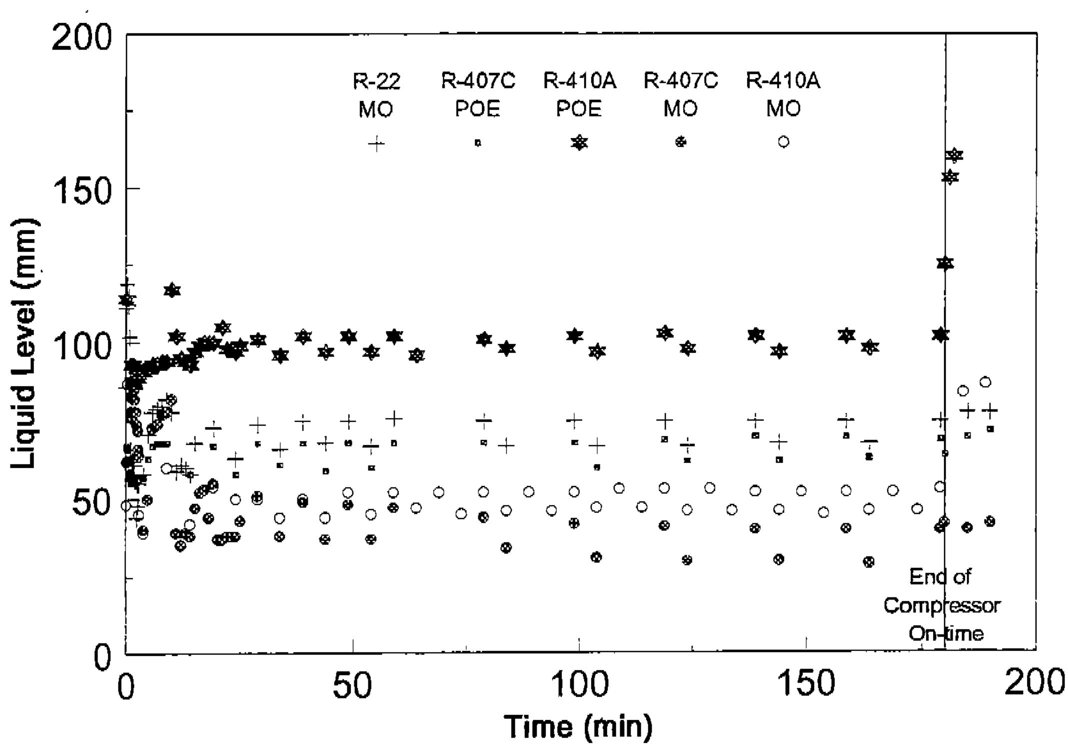

Figure 3 Cyclic Test

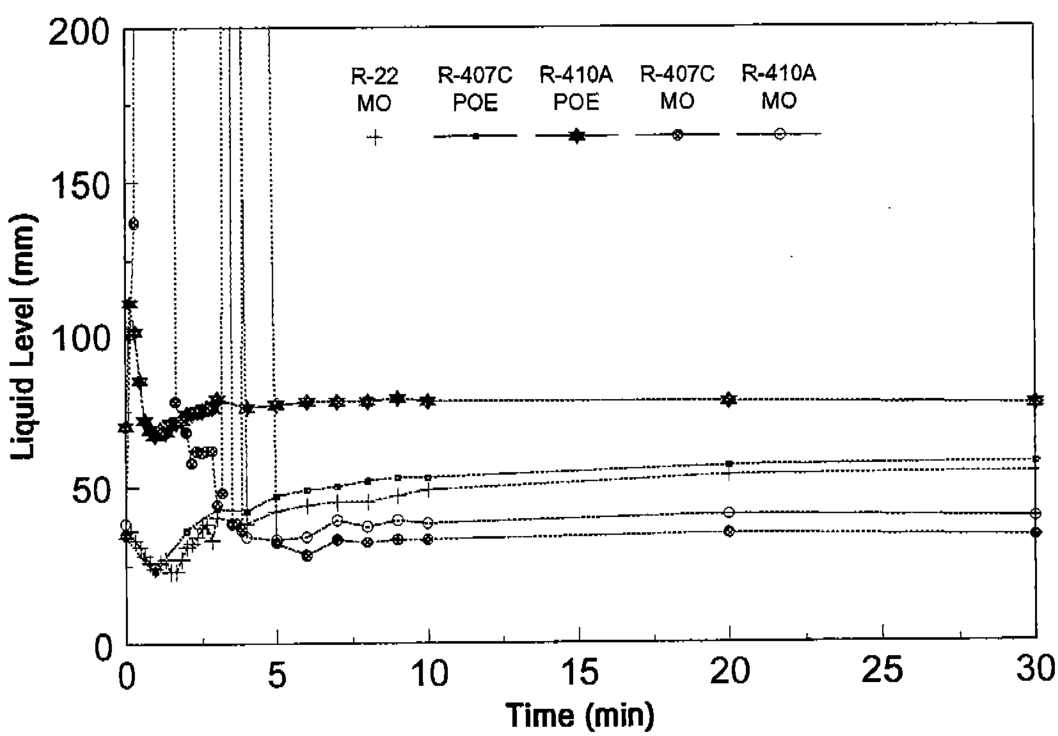

Figure 4 Oil Pump Out Test