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EFFECTS OF OIL ON A TRANSCRITICAL CARBON DIOXIDE AIR CONDITIONING SYSTEM
- some experiences -

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ABSTRACT

This paper presents some experience related to oil effects when running a prototype transcritical air conditioning system with R744 as refrigerant. System is described and the effects of oil and oil separator quantified. Experiences with different orientation of the sampling cylinder presented, and are related to miscibility for the PAG oil used.

1. INTRODUCTION

There are several reasons why oil is in air conditioning systems but one is buy far the most important: lubrication of the moving parts within the compressor, thus increasing its longevity. In some cases, oil is also used to lubricate valves throughout the system. Oil also helps to seal internal leakages within the compressor; especially for two-stage compressor designs.

Oil plays a necessary role in lubricating the moving parts of the compressor of a transcritical carbon dioxide air conditioning system, but, as in any other air conditioning system oil is a detriment to other components. In order to study the effects of oil, the oil concentration in the circulating charge must first be measured. Once this has been done, the effect of oil on system components can be studied. Oil increases the pressure drop of the refrigerant due to increased viscosity and due to the oil coating the interior of the pipes, thereby decreasing the effective flow area. Oil also decreases the effectiveness of heat exchangers by forming an insulating layer between the refrigerant and the heat exchanger material. The effect of oil on all of the components has an effect on overall system performance, and can be quantified using capacity and the coefficient of performance.

Oil reaches other components by being drug along by the refrigerant as it flows through the system (Cremaschi). Among other candidates for R744 systems is Polyaklylene glycol (PAG) oil which is only miscible with carbon dioxide in certain conditions. When miscible, the oil gets easily carried around, but in regions of immiscibility the oil separates from the refrigerant. Such is the case in the evaporator, where the liquid oil is deposited on the walls of the tube as the refrigerant turns into a vapor.

2. SYSTEM DESCRIPTION

The system used in this experiment was a prototype split system air-conditioner which utilized R744 (carbon dioxide) as the refrigerant. The real-life purpose of such a system is for residential cooling. As is common with carbon dioxide air conditioning systems, the system operates in a transcritical cycle in most operating conditions; the high side pressure being greater than the critical pressure of 7480 kPa. Since in the supercritical region pressure and temperature are independent of one another, controlling the high side pressure by opening or closing an expansion valve can be an effective control strategy. Figure 1 shows a P-h diagram of the cycle used in this experiment.

For this experiment, the tests were conducted with outdoor and indoor temperatures of 35˚C and 25˚C, respectively. The indoor environment was dry enough so that no condensation would take place. The combined air flow rate over the gas cooler and intercooler was 0.41 m^3/s. The air flow rate over the evaporator was 0.15 m^3/s.

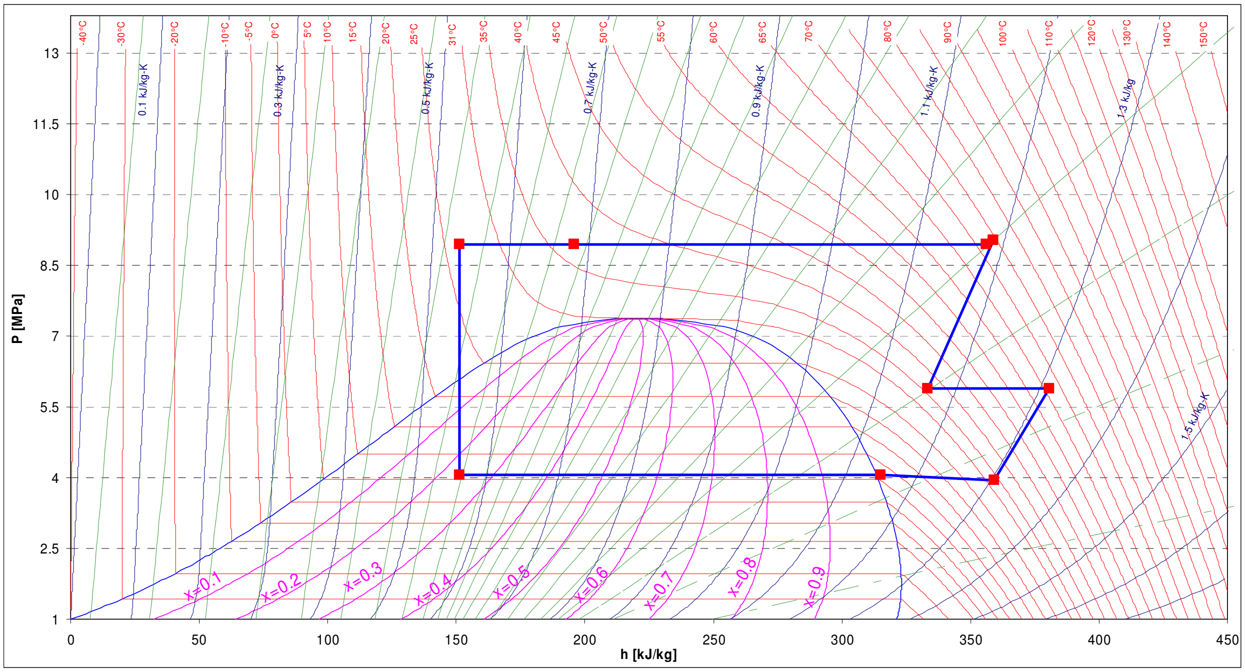

Figure 1. Pressure-enthalpy diagram of employed cycle

The compressor was a two-stage rotary piston design which could operate between 40 and 100 Hz. A speed of 70 Hz was used for this experiment. The compressor was filled with 350 ml of 46 cst PAG oil.

The gas cooler was comprised of two heat exchangers joined in parallel, as can be seen in Figure 2. Both halves were of these serpentine fin heat exchanger were oriented in a cross-counter-flow orientation with respect to the air. The gas cooler had a fin spacing of seven fins per centimeter and the combined face area was 2900 cm^2. The intercooler was a microchannel design consisting of four slabs mounted in a cross-counter-flow orientation. Both the gas cooler and intercooler tubes utilized microchannel tubes.

The suction line heat exchanger was made by sandwiching three microchannel layers together. The high temperature refrigerant flowed through the center layer while the lower temperature refrigerant flowed in the opposite direction in the outer layers.

The evaporator was round tube-in-fin, divided up into three sections to allow as much surface area as possible in a small volume. The evaporator can be seen in Figure 2. The 7mm outer diameter tube was organized so as to make two passes. There were seven and a half louvered fins per centimeter. The face are of the evaporator was 1820 cm^2.

Two-phase refrigerant entered the top of the accumulator where it was separated by a screen. Vapor, which filled the top of the vessel, exited via the tube which ran up from the bottom. A small hole was located at the bottom of this return pipe in order to return oil to the compressor.

An electronic expansion valve was used to for this experiment. The stepper motor in the valve allowed the orifice to be enlarged or shrank in order to control the pressure of the refrigerant as it exited the gas cooler. The high side pressure control strategy for this system was determined empirically as shown earlier by in numerous publications and for this system by O'Connor.

For some of the tests, a centripetal type oil separator was used. The flow through the separator could be varied by use of a needle valve. The flow was varied in order to maintain a level of oil in a sight glass which was installed between the separator and the needle valve. Schematically, the oil separator was located between the compressor and gas cooler. The oil which was separated flowed through a mass flow meter and back to the suction line of the compressor.

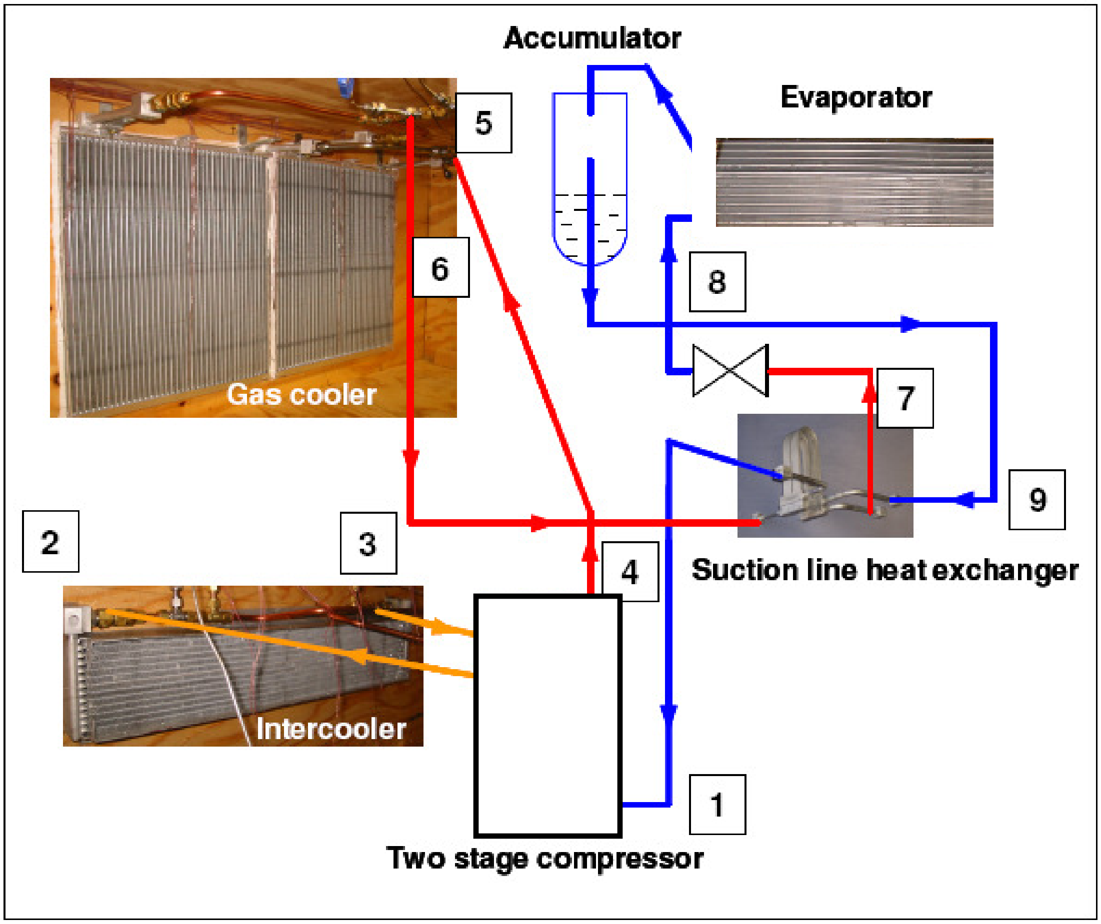

Figure 2. Components with schematic of pipe routing (O'Connor)

3. EXPERIMENTAL SETUP

In order to minimize erroneous measurements affecting system results, three independent energy balances were utilized in this experiment. If there is close agreement between the three energy balances, then the data can be viewed as valid. All tests were conducted at steady state with the prescribed conditions.

The first of these balances is the chamber side energy balance. For the experiment, there are two calorimetric chambers which are used to create the indoor and outdoor test conditions. The compressor, gas cooler, intercooler, and suction-line heat exchanger are all located within the outdoor chamber. The evaporator is located in the indoor chamber. Between the two chambers, the accumulator and an electronic expansion valve have been placed. A basic diagram of this test setup can be seen in Figure 3. The chamber is treated as a thermodynamic control volume, where the energy flows through the control volume are measured. The chambers can be forced to maintain a constant temperature by adding and removing heat. Heat is added via PID controlled electric resistance heaters and removed by a R404A refrigerant chiller system. All electric power being used within the chamber is measured by watt transducers. The chamber walls, which are constructed of foam insulation, have Type-T thermocouples attached to the interior and exterior surfaces. The conductivity of the walls has been previously determined. Knowing the conductivity allows the heat transmission to be calculated. Summing all of the energy flows through the chamber walls allows for the calculation of heat exchanger capacity.

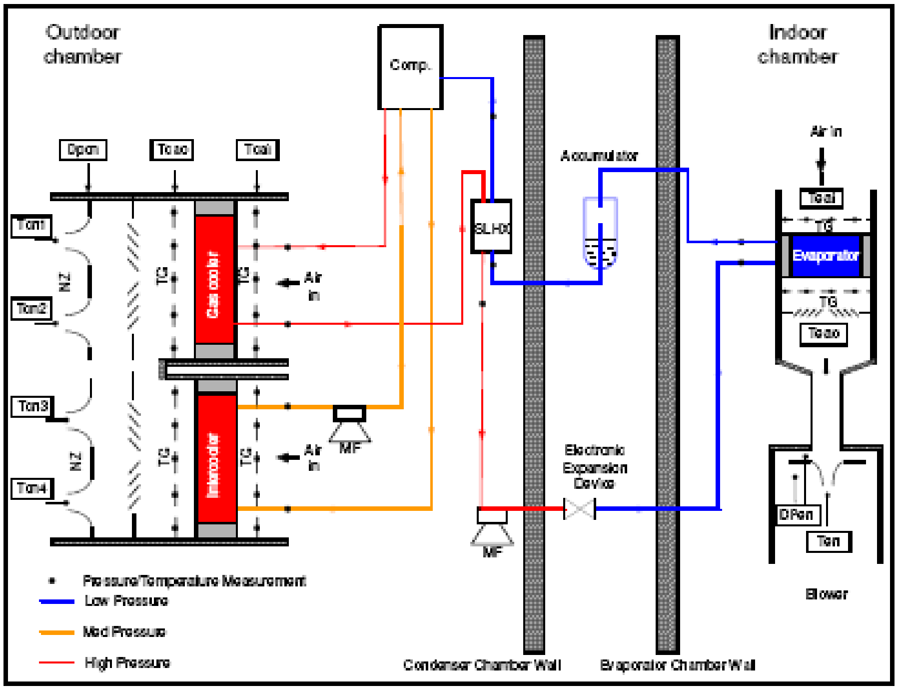

Figure 3. Diagram of experiment setup

The second energy balance is the air side energy balance, by which the change in the air's energy as it flows across the heat exchangers is measured. The air flow is directed through wind tunnels where it first passes through a grid of Type-T thermocouples. Next the air passes through the heat exchanger. Another thermocouple grid is located directly after the heat exchanger. These two grids allow for the calculation of the air's change in specific enthalpy. The mass flow rate of the air is calculated by measuring the differential static pressure across nozzles which are installed further downstream. The throat diameters of the nozzles were selected according to ASHRAE Standard 41.2-1987. Another set of thermocouples is located in the throat of the nozzles. The air flow rate is maintained at the test conditions by use of blowers that are controlled by variable frequency drives. The electric heaters which are used to maintain air temperature are located in a duct downstream of the blower. For the indoor duct, chilled mirror dew point sensors are located before and after the evaporator, which allows the change in the air's moisture content to be calculated. Both wind tunnels were calibrated in order to determine the conductance through the wind tunnel walls. All of these measurements allow the calculation of heat exchanger capacity from only air measurements.

The third, and final, energy balance is the refrigerant side energy balance. One coriolis type mass flow meter is located in the high pressure line after the suction line heat exchanger. By this placement, any pressure drop can be seen as a continuation of the pressure drop which is experienced by the throttling in the expansion valve. Another mass flow meter was placed in the intermediate pressure line after the intercooler which allowed the determination of leakages internal to the compressor. Absolute and differential pressure sensors were used to determine the static pressure entering and leaving every major component. Type-T thermocouples were placed in the center of the refrigerant stream before and after each component. The locations for the pressure and temperature measurements are denoted by the dots in Figures 1 and 3. Multiplying the refrigerant mass flow rate by the enthalpy difference across the heat exchangers yields the refrigerant side energy balance. In absence of enthalpy data for the R744 – PAG oil mixture, the effects of oil were taken in account in two ways: First by reducing measured flow rate by thefor oil contribution based on concentration measurement, and second by taking into account sensible heat contribution of oil flow. Heat of mixing was not taken in account.

Many tests were conducted with this experimental setup mostly by O'Connor, besides those tests which yielded the findings presented in this paper. Figure 4 shows the relationship between the chamber, air, and refrigerant energy balances versus the average of the three independent balances for each data point. As can be seen, all three energy measurements for the evaporator were accurate to within 10%, most within 5%. This energy transfer is also known as the cooling capacity. The coefficient of performance (COP), is this value divided by the electrical power supplied to the compressor, which is measured by a watt transducer.

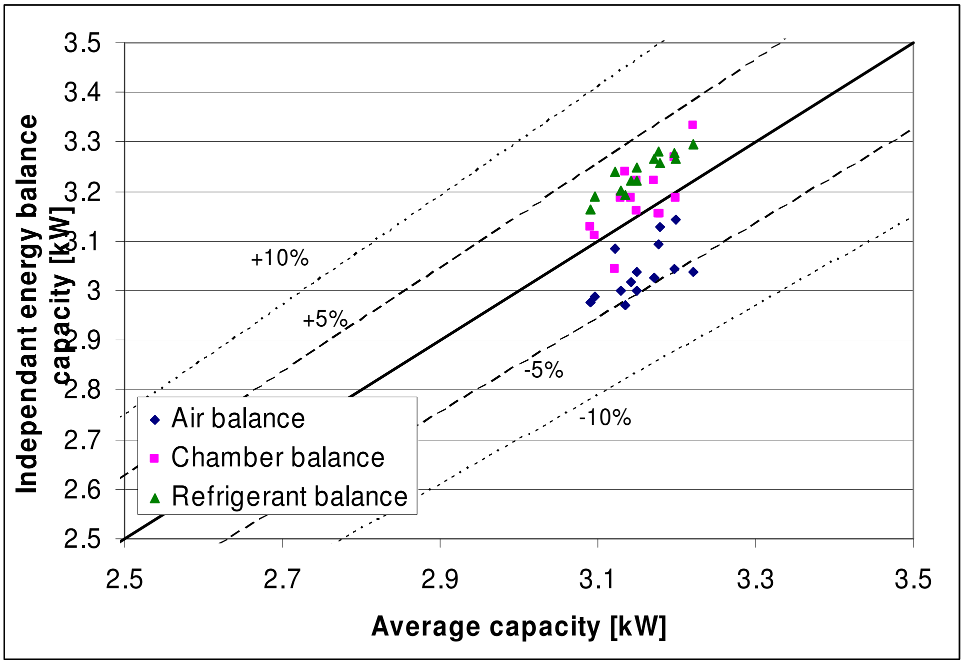

Figure 4. Evaporator energy balances

A great deal of effort was put into making the best possible measurement of the oil circulation rate (OCR). Some techniques have involved creating capacitance or density correlations (Cremaschi and DeAngelis), but due to time constraints, it was not possible to take enough data points to form a correlation. ASHRAE Standard 41.4-1996 gives a standardized method for measuring OCR for oils which are miscible with the refrigerant. This standard does not apply directly to the R744 with PAG oil because the combination is immiscible in many operating conditions. However, the ASHRAE standard does give some insight as to how a mixture can be sampled in order to determine OCR.

In order to measure OCR, an aluminum sample cylinder with ball valves at both ends was used. Before installing the sample cylinder for each test, it was cleaned with acetone and evacuated with a vacuum pump. The cleaned and evacuated sample cylinder was weighed. The sample cylinder was then attached to the system in one of three different orientations. The first orientation, inclined downward flow, can be seen in Figure 5. For this setup, all of valves were positioned so as to force all flow to go through the cylinder. After steady state measurements had been taken, the 3 way valve was turned to bypass the sample cylinder. Immediately, the other three valves were closed. Next, the cylinder was removed and the entire sample cylinder was weighed on an electronic balance. After weighing, the cylinder was oriented vertically, and the upper ball valve was opened slightly in order to allow only the refrigerant to escape through a paper filter of known weight. After the sample cylinder reached atmospheric pressure, the paper filter was removed and weighed, to check to make sure no oil had escaped. A vacuum pump was then attached to the upper valve for fifteen minutes to pull the remaining dissolved refrigerant from the oil and to remove the gasses from the cylinder. The valve was then closed, and the sample cylinder was reweighed. By knowing the weight of the empty cylinder, the cylinder with mixture, and the cylinder with just oil, the OCR could be determined. The quantity of oil removed in the sample cylinder was replaced to the system by adding fresh oil.

Besides the inclined downward orientation, other orientations were also tried as presented in Figure 6. The next orientation was similar except the refrigerant oil mixture flowed vertically downward. The third, and final, method was different from the others in that the flow did not pass through the cylinder. Instead, the cylinder with its ball valves positioned vertically above valve 3. Valve 3 was left open, and valve 2 was opened for 3 seconds. Because the mixture in the system was higher in pressure than the evacuated cylinder, the refrigerant oil mixture filled the cylinder. Both valves were then closed and the cylinder removed.

Another method of measuring the compressor's OCR was also employed. When the oil separator was installed for some tests, the mass flow of the separated oil could be divided by the total mass flow from the compressor. With this technique, the oil separator is often assumed to be completely effective at separating oil and refrigerant.

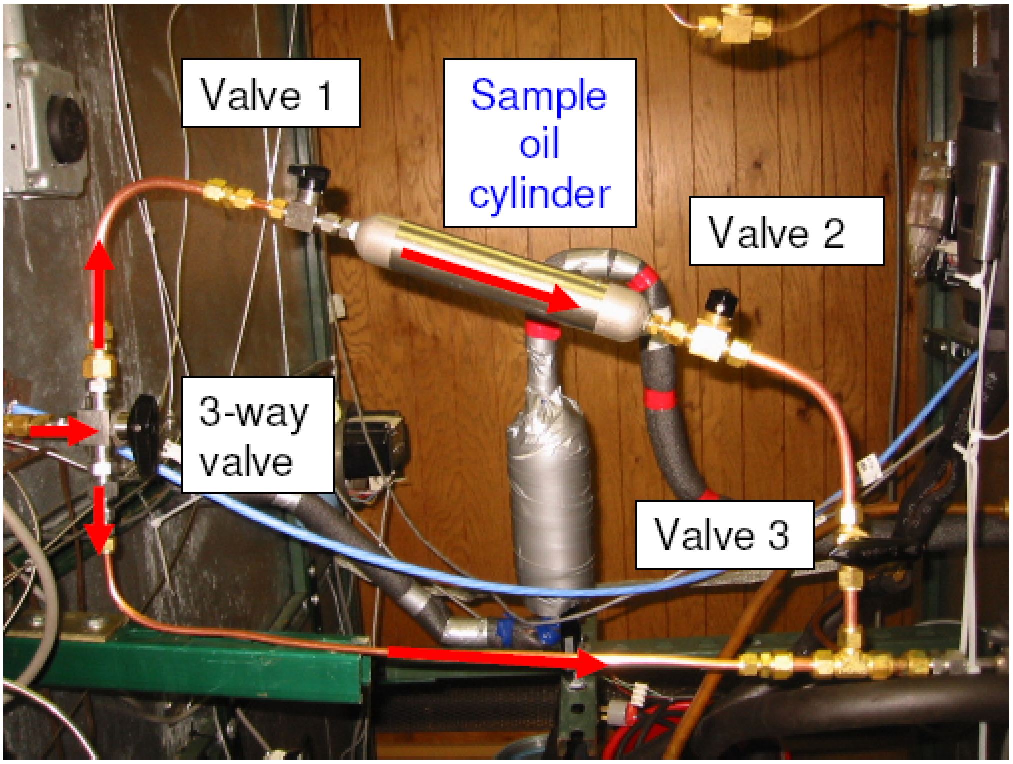

Figure 5. Inclined downward sample cylinder orientation, with valves (O'Connor)

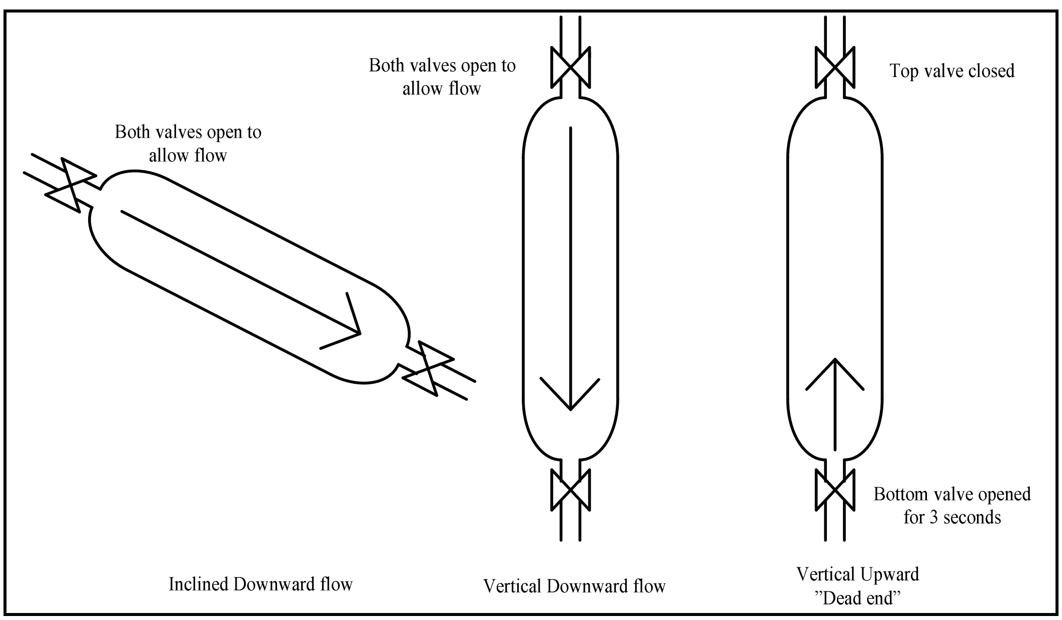

Figure 6. Orientations of the sample cylinder with valves

4. EXPERIMENTAL RESULTS

The technique used to measure OCR had a sizable effect. A table showing the data collected from the three sample cylinder techniques can be seen in Table 1. Since all points were run at the same test condition with the same oil and refrigerant charges, it is believed that the actual OCR did not change; the difference was only from the experimental technique. A plausible explanation for the differences is in immiscible combination of R744 and PAG in the operating regime. As can be seen, the results with flow through the cylinder are close to one another; however there is a large standard deviation for both of these measurements. The upward sample cylinder which was only opened at one end measured a larger OCR, but with a much smaller standard deviation. Since this result gave a far smaller deviation, it is believed that this technique gives the most repeatable results. Therefore, this technique was the only one which was used when the oil separator was installed. All test setups have an unknown offset error.

Table 1. Results of varying sample cylinder orientation on determined OCR

Experimental Setup	Average OCR [%]	Standard Deviation
Sloped downward flow	3.75	0.72
Vertical downward flow	3.53	0.89
Vertical upward, “dead end”	5.30	0.33

After placing the oil separator in the system, the vertical upward sampling technique yielded an OCR of 1.51% with a standard deviation of 0.62%. The separator efficiency was found to be 72%. This is the amount of oil which the separator “short circuits” back to the compressor. Conversely 28% of the oil still went throughout the system.

With the oil separator installed it was also possible to approximate the OCR produced by the compressor. The mass flow rate of the saturated oil, which was separated, was found to be 2.51% with a standard deviation of 0.20%. However, if this separation efficiency is taken into account, the OCR of the system without the oil separator would have been 3.5%, which is in line with the data found in Table 1. This technique was the simplest, but requires sample cylinder testing in order to determine the separation efficiency. This efficiency also may change as the difference between the refrigerant and the oil densities changes. This is due to the physics involved within the separator mechanism.

Using the energy balances described in the experimental setup, it was possible to measure the effect of reducing the oil circulation rate on the performance of various system components. The approach temperature of the gas cooler, which is the difference between the air temperature entering the gas cooler and the refrigerant temperature leaving the gas cooler was reduced from 1.46˚C to 1.10˚C. A lower approach temperature is preferred because it means that the gas cooler is allowing the refrigerant exiting the gas cooler to approach the theoretical minimum. A reason for this decrease may be that the oil layer which forms on the walls of the condenser is thinner, causing a smaller thermal resistance, thus allowing more heat transfer.

Similarly, the refrigerant pressure drop in both the evaporator and the gas cooler was reduced. In the evaporator the pressure drop was reduced by 23%, from 43 kPa to 33 kPa, by installing the oil separator. In the gas cooler, the pressure drop was reduced from 75 kPa to 62 kPa, or 18%. There are a couple reasons why this may have occurred. Again, a thinner oil film inside the tubes allows less restricted flow of the refrigerant. Secondly, the viscosity of the refrigerant-oil mixture is reduced. A reduced pressure drop is preferred because it reduces the amount of work which must be done by the compressor. Also, a reduced pressure drop is preferred in the gas cooler, because it allows for a greater decrease in refrigerant enthalpy when the approach temperature is constant.

The effect of OCR on system performance can be seen by studying the energy balances. The evaporator showed only a 20 W increase in cooling capacity, which is about 1% improvement. There are several reasons why even though the component performance. One is that the oil separator lowers the pressure of the refrigerant in the gas cooler. Also, the oil separator reduces the mass flow through the heat exchangers by about 2%. Finally, the compressor is forced to work harder in order to overcome the pressure drop in the oil separator. This can be seen by a decrease in the coefficient of performance (COP) by 6.4%.

CONCLUSION

The technique used to determine the oil circulation rate (OCR) had a large impact on results. Using the same sample cylinder but in different orientations affected the measured OCR by as much as 50% at the low OCR found in this experiment. It was found that the most consistent reading was found when the sample cylinder was mounted vertically upward into a “dead end”. This orientation also gave the largest OCR result. Having the mixture flow through the sample cylinder was shown to be less repeatable and comprised the lower limit. If an oil separator with a known separation efficiency is used, mass flow meters may be utilized in order to show the OCR that would have been present had no mass flow meter been used.

Decreasing the oil circulation rate was shown to increase component performance. Pressure drop in the heat exchangers was greatly reduced. The approach temperature in the gas cooler was also reduced substantially. However, oil effects on the system showed little or adverse effects when an oil separator was installed. The cooling capacity of the system increased a hair while the coefficient of performance was slightly worse. Therefore, it is not advantageous to add an external oil separator to the system. However, because measures of component performance generally increased with lower OCR, it would be advantageous to design an oil separator with lower pressure drop, or else produce a compressor with the smallest possible portion of oil allowed to flow through the system.

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