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ABSTRACT

A comparison of the start-up cooling capacities for an automotive R744 and an automotive R134a air conditioning system is presented. It is shown that development of the cooling capacities during start-up can be described with one correlation for both systems. The comparison of the cooling capacities and the redistribution of refrigerant mass during start-up suggest a strong relationship between the two. The transient refrigerant migration of a capillary tube throttled R134a automotive air conditioning system during start-up is presented. It was found that 28% of compressor energy could be saved by preventing the migration of refrigerant before start-up. The measurement of oil hold-up before start-up and at steady state operation conditions indicates that the majority of oil migration happens between the compressor and accumulator.

1. INTRODUCTION

One factor that influences the performance of an air conditioning system during start-up is the refrigerant mass migration and the resulting redistribution of the refrigerant mass across the components. Experimental data regarding refrigerant mass migration during start-up can be found in the open literature, e.g. (Tanaka et al., 1982, Mulroy and Didion, 1985, Belth et al., 1988, Rubas and Bullard, 1994). Although they each used different systems, refrigerants and methods to determine the refrigerant mass distribution, all show that during off condition, the refrigerant mass migrates from the high-pressure components to the low-pressure components. Therefore, during start-up, refrigerant mass has to migrate from the low-pressure components to the high-pressure components. (Wang and Wu, 1990) showed that preventing the migration of refrigerant mass during the shut-down period can reduce energy losses during start-up.

Another factor that influences the start-up performance is the oil solubility. For soluble refrigerant oil mixtures (Rubas and Bullard, 1994) conclude that refrigerant released from or dissolved into the oil during start-up influences the refrigerant mass distribution and therefore the system performance.

This paper presents an empirical study focused on the refrigerant mass migration during start-up and presents preliminary results regarding oil migration during start-up.

2. EXPERIMENTAL FACILITY

The experimental facility used for all experiments mentioned in this paper, consists of two environmental chambers as shown in Figure 1. The precision of the test facility is ±5% for the cooling capacity and the coefficient of performance (COP) for steady state measurements. A detailed description of the experimental facility can be found in (Peuker, 2006). The following abbreviations are used in Figure 1.
A – Accumulator, **B** – Blower, **C** – Compressor, **CC** – Cooling Coil, **CH** – R404A Chiller, **Dp..** – Differential Air Pressure Transducer, **Evap** – Evaporator, **GC** – Gas Cooler or Condenser, **H** – Heater, **Hu** – Humidifier,
IHX – Internal Heat Exchanger, **Mtr** – Motor, **N** – Nozzle, **RH** – Relative Humidity Probe, **Sc** – Condensate Scale,
Sp –Speed Controller, **TC** – Temperature Controller, **TG** – Thermocouple Grid,
Tor – Torque Transducer and Tachometer, **W** – Watt Transducer, **XV** – Expansion Device
Indices: **a** – Air, **c** – Gas Cooler, **e** – Evaporator, **n** – Nozzle

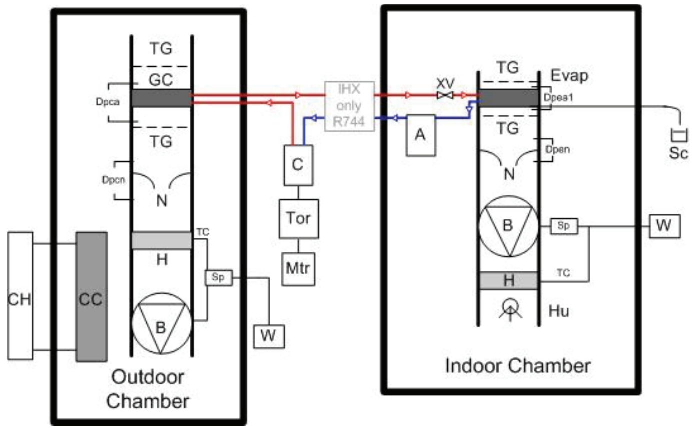

Figure 1: Experimental facility

3. DESCRIPTION OF R744 AND R134a SYSTEM

The R744 breadboard system originated from a prototype two-evaporator automotive system used in an US Army HMMWV (Peuker and Hrnjak, 2006). For this investigation, only one evaporator was used. This system had an internal heat exchanger, and a prototype accumulator with sight glasses was used, which allows monitoring the refrigerant liquid level inside the accumulator. This allows determining the refrigerant mass in the accumulator as described in the next sub-section.

The components of the R134a system, compressor, condenser, fixed orifice expansion device, evaporator and accumulator all originated from an automotive A/C system. The components were installed into the experimental facility with the same difference in vertical height as in the car. In addition, ball valves are installed around each component. By closing those ball valves simultaneously, the refrigerant mass was trapped in five sections: compressor, condenser, liquid tube, evaporator and accumulator. The pipe lengths of the breadboard system were comparable to the original vehicle system with the exception of the liquid tube section. Due to the arrangement of the calorimetric chambers, the liquid tube for the breadboard system is 3.7 times longer than the liquid tube in the real system.

The following Table 1 shows the operating conditions for the two different systems.

Table 1: Operation conditions for breadboard systems

	Compressor speed (RPM)	Air flow gas cooler, condenser (m^3/h)	Air flow evaporator (m^3/h)	Air temperature gas cooler, condenser (°C)	Air temperature evaporator (°C)	Humidity
R744	810	2880	288	35°C	35°C	Dew point temperature below evaporation temperature
R134a	900	1650	489	35°C	35°C	

3.1 Refrigerant Mass Measurement Method

For the R744 system, only the refrigerant charge inside the accumulator could be determined. This was accomplished by using a prototype accumulator which had a sight glass with a scale in units of volume. During operation, the refrigerant liquid level was recorded by a video camera. The images were later analyzed to determine the liquid height of the refrigerant at each point in time. Using the pressure, temperature and internal volume, the refrigerant mass inside the accumulator was calculated. This method has two uncertainties: The amount of oil inside the accumulator and the determination of the liquid refrigerant height by visual inspection due to the mixing of vapor bubbles in the liquid-phase refrigerant. Since the refrigerant-oil combination used is immiscible and due to the design of the accumulator (no dryer, vapor outlet on top, liquid outlet on bottom), the oil hold-up is assumed to be not much greater than the oil in circulation rate. The accumulator is box-shaped and the volume is 1.5 liter, which reduces the turbulence. However, the mixing of vapor bubbles in the liquid-phase refrigerant cannot be determined from visual inspection and is therefore neglected.

For the R134a system, as mentioned above, the refrigerant and oil are trapped in each section by closing the ball valves simultaneously. To extract the refrigerant mass an evacuated sampling cylinder with known weight is connected to the section. After opening the valves between the sampling cylinder and the section, liquid nitrogen is used to cool the sampling cylinder. The chilled sampling cylinder stays connected to the section for 20 minutes, after which it is disconnected and heated to room temperature to avoid condensation of water vapor on its outside shell. The weight of the captured refrigerant is determined and the refrigerant is flushed from the sampling cylinder and recovered into a recovery cylinder. To determine if there was oil extracted from the section, the sampling cylinder is evacuated and weighed again, the weight this time being the extracted oil. This method is repeated twice for each section with exception of the accumulator section where this method is repeated three times. Comparing of the amount of refrigerant mass filled into the system to the amount of refrigerant mass recovered, shows that for 20 tests the difference is 8.5g on average, always less refrigerant is recovered than filled in. This loss is due primarily to the refrigerant left in the hose after filling the system and minor amounts of refrigerant left in the hose when the refrigerant is recovered from the sampling cylinders. For a system charge greater then 850g, the uncertainty of how much refrigerant mass was actually in the system is less than 1%.

To ensure that the simultaneous closing of the ball valves was repeatable, three tests at steady state condition were performed. At the steady state condition, the refrigerant mass flow rate was 35g/s indicating that if the valves would be closed with a one-second delay, up to 35g of refrigerant could enter or leave each section. Table 2 shows the result of the three tests in units of percent of total refrigerant mass for a target system charge of 1250g.

Table 2: Repeatability of refrigerant mass determination method

Test	Evaporator	Accumulator	Condenser	Liquid Line	Compressor
#1	14.8%	32.5%	19.2%	27.5%	5.9%
#2	15.7%	31.8%	18.9%	28.5%	5.0%
#3	15.4%	33.6%	18.7%	27.2%	5.2%

3.2 Refrigerant Mass Distribution for R134a Automotive System at Steady State

A charge optimization test was conducted for the R134a system. This is necessary since the breadboard system has a longer liquid tube then the vehicle system due to the arrangement of the calorimetric chambers. Usually the optimum amount of refrigerant charge is found by comparing the cooling capacity with the coefficient of performance (COP) at the design condition for various refrigerant charges at steady state. In addition, to determine if the expansion device (here a fixed orifice tube) is properly sized the superheat at the evaporator outlet and the subcooling at the condenser outlet is also considered. These results and the measured refrigerant mass distribution across the different components are shown in Figure 2(a) and 2(b).

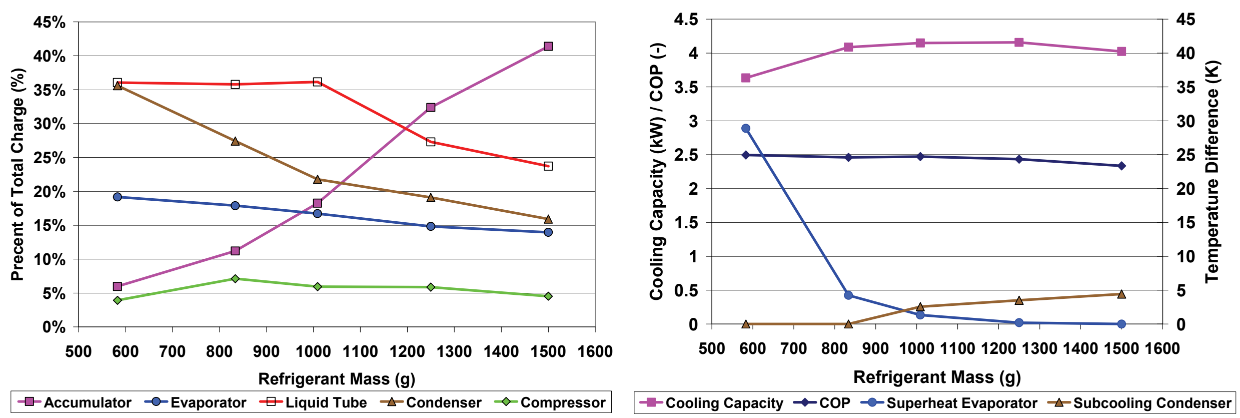

Figure 2: (a) Refrigerant mass distribution and (b) cooling capacity, COP, subcooling and superheat for charge test

Figure 2(b) shows that there is an optimum in cooling capacity for refrigerant charges between 1000g and 1250g. For the same range the COP is almost constant, the subcooling increases and the superheat vanishes. From just considering these values, any refrigerant charge between 1000g and 1250g could be used. However, by considering

Figure 2(a), which shows the refrigerant charge distribution across the components, further insight can be gained why 1250g is not a good choice. The first thing to notice is that only the slope of the accumulator is constantly positive. That means that by increasing the total charge, more and more charge will be held up in the accumulator. This behavior is actually desired since the role of an accumulator should be to hold excess charge, which is not needed based on the operation condition. The hold-up of refrigerant charge in the accumulator increases especially in the range from 1000g to 1250g. This behavior can be explained by looking at the data points for the liquid tube. Until a charge of 1000g, the percent of the total charge, which is in the liquid tube, is constant. However, since the total absolute refrigerant charge increases, the absolute refrigerant charge stored in the liquid tube increases. Then, for charges greater 1000g, the total percentage is declining. Actually, it is found that the absolute refrigerant mass hold-up in the liquid tube stays constant for refrigerant charges higher than 1000g. This is because the liquid tube is filled with liquid refrigerant, which can be deducted from Figure 2(b) since at 1000g of total refrigerant charge subcooling is observed at the condenser exit for the first time. This changes the behavior of the refrigerant charge distribution of the system significantly. After subcooling occurs in the liquid tube, not much more refrigerant mass can be stored there because of the already high density of the liquid refrigerant. This is the reason that the refrigerant hold-up in the accumulator increases considerably. In absolute numbers the refrigerant hold-up in the accumulator increases by 220g when the total charge is increased from 1000g to 1250g. This indicates that additional charge above 1000g is mostly stored in the accumulator.

In Section 5, it is shown that the refrigerant mass held up in the accumulator plays a significant role regarding the dynamic behavior of the system during start-up. Therefore, it is beneficial to choose a total refrigerant charge that reduces the refrigerant hold-up in the accumulator as much as possible without sacrificing a reduction in cooling capacity or COP. Consequently, a refrigerant charge of 1000g was chosen for this system.

4. COMPARISON OF START-UP COOLING CAPACITIES

In most start-up situations, it is desired to reach the target cooling capacity as quickly as possible. In automotive applications, a quick cool-down increases passenger comfort. For cycling operation of a refrigeration system, reaching a target cooling capacity as quickly as possible minimizes the cycling losses.

To compare the transient cooling capacity of the investigated R744 and R134a systems, the following approach was used. Each system was run for several hours to reach a steady state operation condition as defined in Table 1. Then the compressor was shut off for 3 minutes (2 minutes for the R744 system) while the blowers at both air-to-refrigerant heat exchangers were left running. The air inlet temperatures to the heat exchangers were held within ±2K during the off period. After the defined off time the compressor was switched on again.

The steady state cooling capacities were 3.2kW and 4.1kW for the R744 and R134a system, respectively. Since each system had a different steady state cooling capacity, it is difficult to compare the absolute values of the transient cooling capacities, so the relative cooling capacities were formed for comparison between the systems. At each point in time, the measured airside cooling capacity (Q) was divided by the steady state value of the airside cooling capacity (Q_{SS}). The results are shown in Figure 3(a) where t=0 is the time when the compressor was switched on.

(Kim and Bullard, 2001) present a correlation of the start-up cooling capacity as function of time for a R410A residential air-conditioning split system. An equation similar to equation (3) from (Kim and Bullard, 2001) is used in this work to represent the non-dimensional air side cooling capacity with t as time in minutes, two time constants S_1 and S_2 and a constant A:

$$\frac{Q}{Q_{SS}} = \left(1 - \left.\frac{Q}{Q_{SS}}\right|_{t=0}\right)\left[\frac{1-\exp\left(-t/S_1\right)}{1+A\exp\left(-t/S_2\right)}\right] + \left.\frac{Q}{Q_{SS}}\right|_{t=0} \qquad (1)$$

For the investigated start-up scenario the airside cooling capacity at start-up (t=0) is not 0 as shown in Figure 3(a). To account for this offset at t=0, the two terms left and right of the square bracket in equation (1) are introduced. Table 3 shows the constants used in equation (1) to correlate the transient airside cooling capacities for both systems.

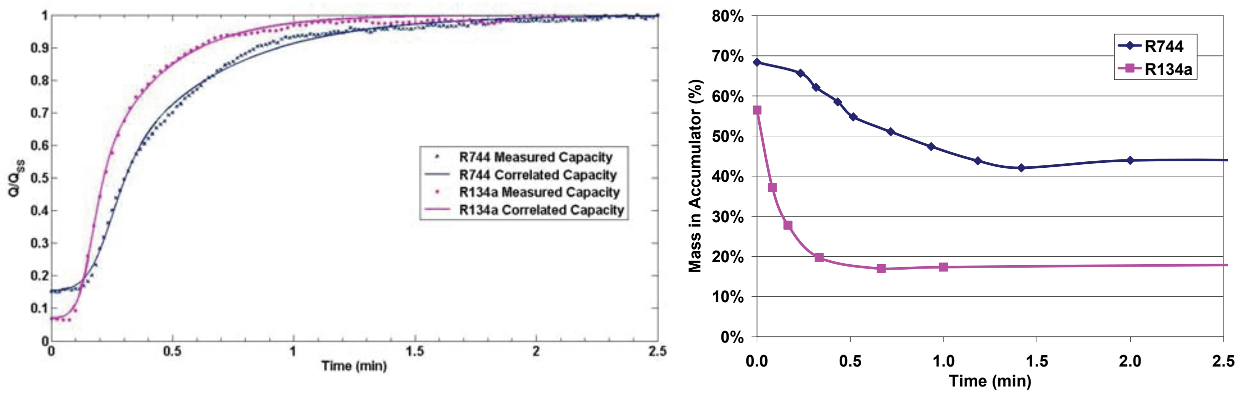

Figure 3: (a) Comparison of start-up cooling capacities and (b) change of refrigerant mass inside accumulators

Table 3: Coefficients used in equation (1)

	Measurement data sampling rate [s]	Q/Q_{SS} [-]	S_1 [min]	S_2 [min]	A [-]	R-square value
R744	1.0	0.154	0.4399	0.05476	58.63	0.998
R134a	1.5	0.070	0.2754	0.03425	96.05	0.997

Figure 3(a) shows that equation (1) can be used to express the transient airside cooling capacity during start-up for both, R744 and R134a automotive air conditioning systems. The R134a system shows a faster increase in cooling capacity after start-up. Figure 3(b) shows the change of refrigerant mass in the accumulators during start-up in percent of the total system refrigerant mass. It can be seen that the trend is the same for both accumulators. The refrigerant mass decreases quickly until it reaches a minimum. Then it slightly increases to reach a steady state value. However, the time it takes for the refrigerant masses to migrate from the accumulators into the systems are different. For the R744 system, it takes longer to equilibrate the refrigerant mass distribution. Comparing this result to the transient cooling capacity shown in Figure 3(a) suggests that there is a strong relationship between refrigerant migration and development of cooling capacity during start-up. A faster redistribution of refrigerant mass during start-up might therefore enhance the start-up performance.

5. REFRIGRANT MASS MIGRATION DURING START-UP

The refrigerant mass measurement method introduced in Section 3 is used to determine the refrigerant mass migration for an R134a automotive breadboard system during start-up transient. First, the system was run for several hours to reach a steady state based on the operation condition shown in Table 1. Then the compressor was stopped but the airflow rates were maintained. The compressor was then turned on again after 3 minutes. The refrigerant charge distribution was measured at the following times in seconds during the start-up: 0, 5, 10, 20, 40, 60 and 300. After each measurement, the refrigerant was returned and the system was run again for several hours for the next measurement. The pressure measurements and all temperature measurements were taken every 1.5 seconds. Figure 4(a) shows the refrigerant mass distribution and the pressure ratio, which is the pressure measurement at the compressor outlet divided by the pressure measurement at the compressor inlet. Figure 4(b) shows the measurements of several temperatures during the start-up transient event.

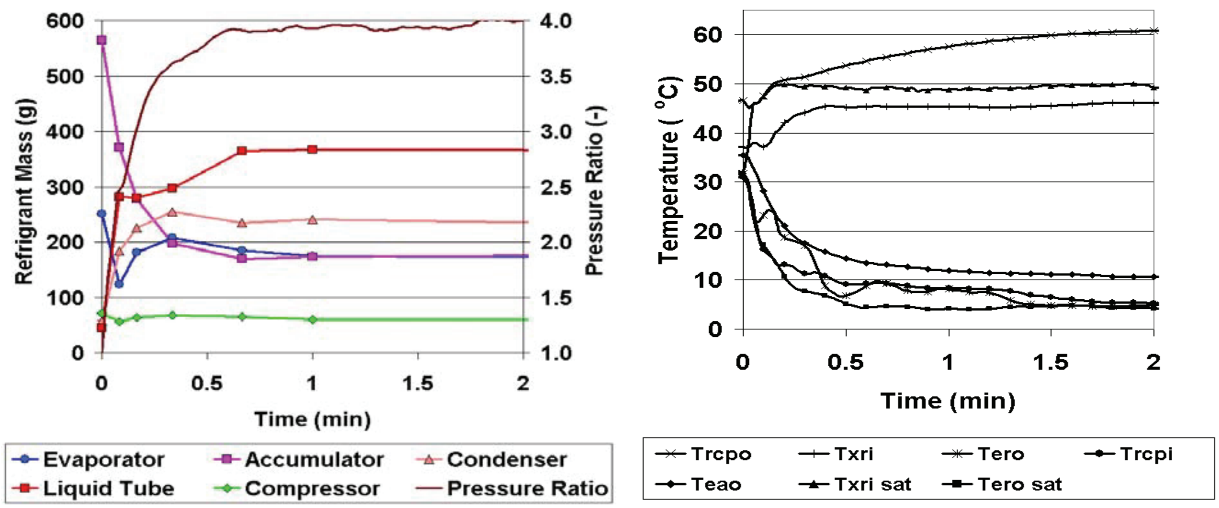

Figure 4: (a) Refrigerant mass distribution and (b) temperatures during start-up transient

The refrigerant mass migration during the start-up, as shown in Figure 4(a), demonstrates that refrigerant mass has to migrate from the low-pressure components to the high-pressure components. Prior to start-up, 81% of the total refrigerant mass of the system is located in the low-pressure components, of which 56% is in the accumulator. The migration of refrigerant during start-up is almost completely achieved after one minute, at which time, 98% of the steady state airside cooling capacity is reached, according to the data presented in Figure 3(a).

At the beginning of the start-up, the refrigerant mass in the evaporator section is decreased. This leads to a deficit of refrigerant mass inside the evaporator. The evaporator refrigerant exit temperature measurement (Tero) confirms this by showing a temperature higher than the saturation temperature (Tero sat) 5 seconds after start-up. The comparison between the temperature at the expansion valve inlet (Txri) and the saturation temperature based on the pressure measurement (Txri sat), indicates that after 5 seconds the refrigerant at the inlet of the orifice tube is subcooled. The refrigerant mass in the evaporator increases after 5 seconds but the temperature measurement still indicates superheat at the evaporator outlet. The amount of refrigerant mass in the evaporator peaks 20 seconds after start-up, and at 30 seconds after start-up the refrigerant exit temperature almost reaches saturation temperature before it increases again and eventually reaches saturation temperature 80 seconds after start-up. The air exit temperature of the evaporator (Teao) is not affected by this unsteady behavior and shows a smooth decline during the start-up.

Figure 4(a) shows that during the first 20 seconds 350g of refrigerant (that is 35% of the total refrigerant mass in the system) leaves the accumulator. Figure 4(b) shows that between 5 and 20 seconds after start-up, the inlet temperature to the compressor (Trcpi) is lower than the refrigerant temperature at the evaporator exit (Tero) indicating that a significant amount of two-phase refrigerant is entering the compressor during this period. This leads to a high refrigerant density at the compressor inlet and explains why it is possible for the compressor to provide a high enough refrigerant mass flow rate to shift significant amounts of refrigerant mass within the first 20 seconds of the start-up. The assumption of two-phase refrigerant at the inlet of the compressor is supported by two additional observations. First, the refrigerant temperature at the compressor outlet (Trcpo) is at the saturation temperature based on the pressure measurement at the expansion valve inlet (Txri sat) between 3 and 10 seconds after start-up. Secondly, although the data are not shown here, the coriolis mass flow meter installed at the compressor outlet did not read the mass flow rate during this period. This indicates that two-phase refrigerant is present because this type of mass flow meter can only determine mass flow rates when there is single-phase refrigerant present.

To see if compressor energy can be saved during start-up the following experiment was conducted. After the system reached steady state, the compressor was stopped and simultaneously the valve directly upstream of the orifice tube was closed to prevent liquid refrigerant migration from the liquid tube to the evaporator. The valve was opened again after 3 minutes when the compressor was turned on again. Comparing the compressor energy during the first 20 seconds after start-up between this case and the case where the valve was not closed, shows that 28% of compressor energy can be saved. The airside cooling capacity showed the same value after 20 seconds for both cases. This demonstrates that by preventing refrigerant charge migration during off-cycle condition, the same cooling capacity can be achieved and 28% of compressor power can be saved. It should be pointed out, however, that because, as mentioned in Section 3, the liquid tube is longer in the breadboard system, less refrigerant mass migration can be expected in the real vehicle system and therefore the compressor energy savings are expected to be less. Nonetheless, the presented experimental investigation shows that refrigerant migration plays a significant role during start-up transients and that there is potential to improve the start-up performance.

6. OIL MIGRATION

The measurement of oil migration during start-up transient is an ongoing research effort and therefore data regarding the transient migration of oil within the system are not available at this time. However, preliminary data will be presented.

Two measurements regarding the distribution of oil throughout the five sections of the automotive R134a breadboard system are accomplished. One measurement was taken at steady state and one measurement after the compressor has been stopped for 3 minutes. Those two measurements can be seen as the two extremes of the oil distribution: before the start-up and after the system reaches a steady state. To determine the amount of oil in a section the refrigerant mass is first recovered by the method described in Section 3. Then the section is flushed with liquid R134a using a flushing device which separates the oil from the recovered refrigerant after each flushing cycle. This was done for the evaporator, accumulator, condenser and liquid tube section. The amount of oil in the compressor section was than determined by induction. The following Table 4 shows the results of the two experiments.

Table 4: Oil distribution

Oil in g	Evaporator	Accumulator	Condenser	Liquid Tube	Compressor
At start-up (t=0)	20	100	9	4	97
Steady state	19	126	12	9	64

The major reasons why there is a significant amount of oil hold-up in the accumulator are the miscibility between refrigerant and oil and the, for automotive typical, U-tube design of the accumulator. The majority of the refrigerant mass leaving the accumulator is refrigerant vapor entering the top of the U-tube. A small hole on the bottom of the U-tube provides entrainment of the liquid refrigerant-oil mixture to ensure oil returns to the compressor. At steady state, the entrainment of the liquid refrigerant-oil mixture through the hole in the bottom of the U-tube must be enough to fulfill the requirement that the oil in circulation rate (OCR) has to be constant. From Figure 4(a) it can be seen that the amount of refrigerant in the liquid tube at steady state is 365g and from Figure 4(b) that the refrigerant is subcooled. By dividing the amount of oil by the amount of refrigerant mass found in this section, it is possible to determine the OCR. This assumes that all oil is dissolved in the liquid refrigerant phase and that the dissolved oil travels with the same velocity as the refrigerant. Based on this assumption the OCR is calculated to be 2.5%. The refrigerant mass in the accumulator at steady state is 181g and considering the temperature, pressure and inlet quality the liquid refrigerant mass can be calculated to be 160g. This results in an oil to liquid refrigerant ratio of 79% at steady state.

The results shown in Table 4 indicate that 33g of oil is migrating from the compressor to primarily the accumulator during start-up. This implies that overall less oil enters then is leaving the compressor during the start-up transient. The reverse is implied for the accumulator. It is difficult without additional measurements to determine how this migration of oil takes place. Figure 4(a) shows that most of the refrigerant migration is over after 40 seconds implying that the majority of oil migration probably takes place in the same time frame. In addition, the oil to liquid refrigerant ratio in the accumulator before start-up is only 19% because of the amount of liquid refrigerant hold-up in the accumulator. This indicates that at the beginning, although a lot of refrigerant mass is leaving the accumulator, the amount of oil leaving with this refrigerant mass might be lower than the amount of oil leaving the compressor. As mentioned at the beginning of the section, the migration of oil during start-up is an ongoing research effort and more measurements at different times during start-up are expected soon.

6. CONCLUSIONS

A measurement method to determine the refrigerant mass distribution during start-up transient is introduced. This method is based on simultaneously closing valves to trap refrigerant in different section of the system. It is demonstrated that the uncertainty of this method regarding the total refrigerant mass in the system is less than 1%. The uncertainty regarding the determination of how much refrigerant is in one section is less then 2% in terms of total refrigerant mass in the system.

Applying this method to an automotive R134a breadboard system reveals that during start-up a significant amount of refrigerant is migrating from the low-pressure components to the high-pressure components. For the presented case study, 81% of the total refrigerant mass is located in the accumulator and evaporator section prior to start-up. At steady state, only 35% of the total refrigerant charge is located in those sections. It was demonstrated that 28% less compressor energy is used when the liquid refrigerant migration from high-pressure to low-pressure components was prevented compared to the typical usage which allows free migration during the off cycle period.

Although the refrigerant mass during start-up transient for the automotive R744 breadboard system is only determined in the accumulator, a comparison to the refrigerant mass migration of the automotive R134a system shows the same qualitatively development. The slower migration of refrigerant mass from the R744 accumulator corresponds to a slower development of the airside cooling capacity compared to the R134a system. It is shown that one equation can be used to correlate the transient airside cooling capacities for both systems during start-up. Preliminary measurements of the oil hold-up in an automotive R134a breadboard system indicate that oil is primarily migrating from the compressor to the accumulator during start-up. At steady state, the accumulator holds more then 50% of the total amount of oil in the system. The miscibility between refrigerant and oil and the design of the accumulator are identified as the major reasons why there is a significant amount of oil hold-up in the accumulator.

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