

Transient Refrigerant Migration and Oil Distribution of an R134a Automotive A/C System

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ABSTRACT

Automotive fixed orifice tube (FOT) systems are especially prone to cycling losses due to their clutch cycling operation. Therefore, it is important to better understand the dynamics of the refrigerant and oil migration during transient events such as cycling and start-up.

To measure the refrigerant mass and oil distribution of an automotive R134a FOT breadboard system, two ball valves around each component are added. By simultaneously closing the valves, the refrigerant and oil is trapped in different sections of the system and can be measured. The transient refrigerant migration during a stop-start transient as well as the refrigerant mass distribution as a function of system charge at steady state operation is presented. A transparent accumulator and transparent tubes at the inlet and outlet of the accumulator are used to visualize the flow of the refrigerant. High speed video snapshots are presented for the first seconds after the start-up. The oil distribution at steady state as a function of total refrigerant charge is presented. In addition, the entrainment ratio of the liquid refrigerant and oil mixture through the oil bleeding hole of the accumulator is determined.

INTRODUCTION

Refrigerant migration, redistribution and oil solubility have been identified as factors affecting the transient and cycling performance of all refrigeration and air conditioning equipment [1]. A key component of an automotive FOT system is the accumulator. Its primary function is to store excess refrigerant so that the system can operate under a variety of ambient conditions. The

typical design of an accumulator for automotive applications consists of a vessel which has an inlet at the top and U-tube in the center with an opening close to the top of the accumulator. The U-tube dips down close to the bottom of the accumulator and then goes up and exits the accumulator. The idea behind this design is that liquid and vapor refrigerant phases are separated inside the vessel and that only the refrigerant vapor exits the accumulator. This prevents that liquid phase refrigerant can enter the compressor and potentially cause damage. However, since there is oil in circulation, an additional feature is necessary to guarantee that oil is returned to the compressor. For that reason the U-tube has a small opening at the bottom which allows entrainment of oil into the exiting vapor flow. For miscible refrigerant oil combinations, however, this entrained liquid will always be a mixture of oil and liquid phase refrigerant. It is therefore important to understand the relationship between the stored liquid refrigerant and oil in the accumulator and the entrainment since this is the mechanism by which oil is returned to the compressor

EXPERIMENTAL FACILITY

The experimental facility used for all experiments mentioned in this paper, consists of two environmental chambers as shown in Figure 1. The compressor is installed between the two chambers and is driven by a speed controlled electrical motor. A torque transducer in combination with a tachometer is used to determine the compressor power. The condenser is installed in an open-loop wind tunnel inside the condenser chamber. In the evaporator chamber the HVAC module containing the evaporator is attached to the wind tunnel and can therefore be considered part of the open-loop wind

tunnel. The HVAC module flaps were set to fresh air intake for all experiments.

Before and after each heat exchanger, thermocouple grids consisting of welded Type-T thermocouples are used to determine the dry-bulb air temperatures. Air flow rates are determined by flow nozzles. The indoor and outdoor blowers exhaust the air directly into the chambers. An external R404A chiller system with the evaporator mounted on the ceiling inside the condenser chamber compensates for the heat rejected by the condenser. PID-controlled electric heaters are installed in both chambers to heat the room to the specified test conditions.

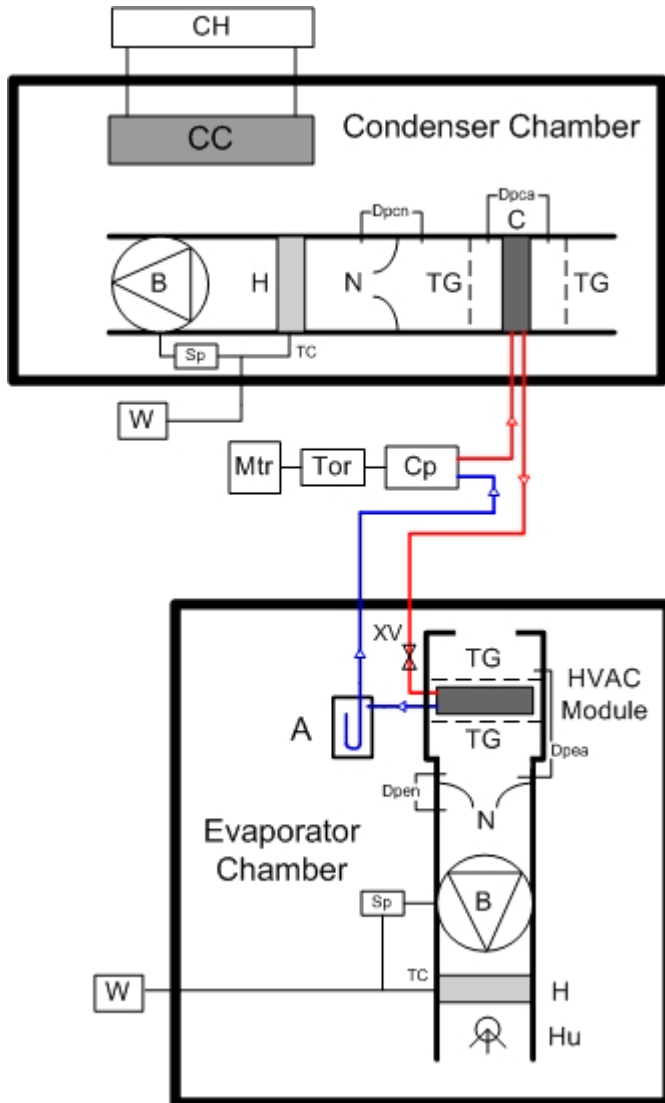


Figure 1. Experimental facility

A – Accumulator, **B** – Blower, **C** – Condenser, **CC** – Cooling Coil, **CH** – R404A Chiller, **Cp** – Compressor, **Dp..** – Differential Air Pressure Transducer, **H** – Heater, **Hu** – Humidifier, **Mtr** – Motor, **N** – Nozzle, **Sp** – Speed Controller, **TC** – Temperature Controller, **TG** – Thermocouple Grid, **Tor** – Torque Transducer and Tachometer, **W** – Watt Transducer, **XV** – Expansion Device, Indices: **a** – Air, **c** – condenser, **e** – evaporator, **n** – Nozzle

The precision of the test facility is $\pm 5\%$ for the cooling capacity and the coefficient of performance (COP) for most steady state measurements [2].

SYSTEM DESCRIPTION

The system investigated in this study is a state of the art R134a automotive A/C system with a standard 0.072" internal diameter fixed orifice tube. The components were installed into the experimental facility with the same difference in vertical height as in the vehicle. In addition, ball valves were installed around each component. By closing these ball valves simultaneously, the refrigerant and oil masses were trapped in five sections: compressor, condenser, liquid tube, evaporator and accumulator. Ball valves were chosen because of their very low flow resistance. The ball valves were installed as close as possible around the components. 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 of the real vehicle system.

REFRIGERANT MASS MEASUREMENT METHOD

By closing the ball valves around each section simultaneously the refrigerant and oil masses are trapped in each section. 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 transferred from the sampling cylinder 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. A comparison 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 less than the filled in refrigerant. This loss is primarily due 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 than 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 1 shows the result of the three tests in units of percent of total refrigerant mass for a target system charge of 1250g when the ball valves were closed at steady state operating condition (see Table 4).

Table 1: Repeatability of refrigerant mass measurement method

	Exp. #1	Exp. #2	Exp. #3
Compressor	5.9%	5.0%	5.2%
Condenser	19.2%	18.9%	18.7%
Liquid Tube	27.5%	28.5%	27.1%
Evaporator	14.8%	15.8%	15.4%
Accumulator	32.6%	31.8%	33.6%

STEADY STATE DISTRIBUTION OF REFRIGERANT MASS

The refrigerant charge distribution across the five sections of the R134a FOT breadboard system was measured for different system charges at the operating condition (see Table 4). Figure 2 below shows the air side cooling capacity of the evaporator, the coefficient of performance (COP) which does not account for the fan power, the subcooling and superheat at the condenser and evaporator as a temperature difference. The temperature differences were calculated using the pressure and temperature measurements and comparing them to pure R134a property data at the corresponding saturation state.

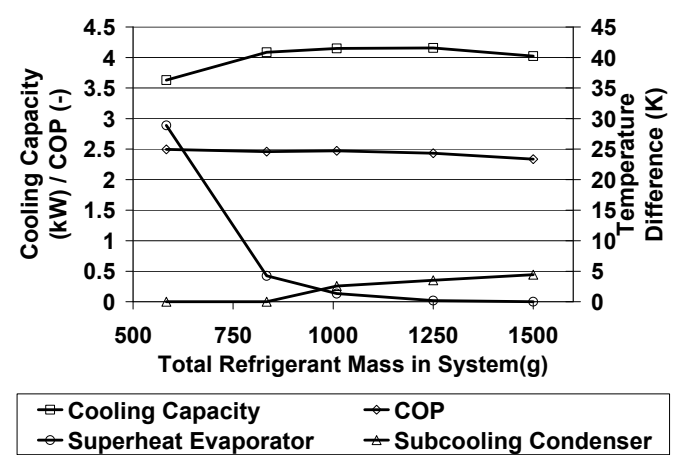


Figure 2: Cooling capacity, COP, subcooling and superheat for different system charges

As Figure 2 shows, there is an optimum in cooling capacity for refrigerant charges between 1000g and 1250g. For this 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 3, which shows the refrigerant charge distribution across the components, further insight can be gained.

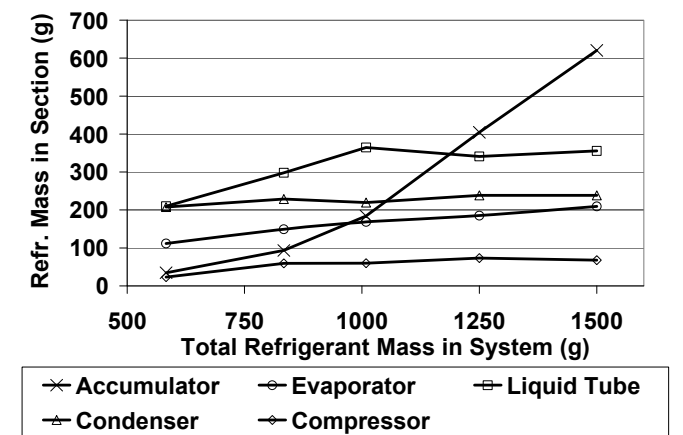


Figure 3: Refrigerant mass distribution across sections for different system charges

The first thing to notice in Figure 3 is that 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 system charge of 1000g, the refrigerant mass hold up in the liquid tube increases constantly. Then, for system charges greater than 1000g, the refrigerant mass hold-up in the liquid tube stays constant. This is because the liquid tube is filled with liquid refrigerant. Figure 2 shows that at 1000g of total refrigerant charge subcooling is observed at the condenser exit for the first time. This changes the refrigerant charge distribution of the system significantly. After subcooling occurs in the liquid tube, additional refrigerant mass can not be stored in the liquid tube because of the high density of the liquid refrigerant. In addition, the refrigerant mass hold up in the condenser is constant. For these reasons the refrigerant hold-up in the accumulator increases considerably. In absolute numbers, the refrigerant mass 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 almost entirely stored in the accumulator. For the R134a FOT breadboard system 1000g is considered the optimum charge. This confirms the manufactures optimum charge of 700g for the vehicle system. The higher charge for the breadboard system results from the longer liquid tube.

TRANSIENT REFRIGERANT MASS MIGRATION

The refrigerant mass measurement method introduced in the previous section was used to determine the refrigerant mass migration for the R134a FOT breadboard system during a transient event. This event consists of an off cycle period in which the compressor is turned off for 3 minutes but the airflow rates at the heat exchangers are maintained, followed by a start-up when the compressor is turned on again. Due to the nature of the refrigerant mass measurement method the system needs to be recharged with refrigerant after each data point is taken. To ensure the system is at an identical steady state, the system is run for several hours at the conditions shown in Table 4 before the transient event is initiated. A total refrigerant charge of 1000g was used for all transient experiments.

The refrigerant charge distribution was measured at the following times in seconds during the transient event: 0 (compressor turned off), 5, 10, 20, 60, 180 (start-up begins), 185, 190, 200, 220, 240, 360 and 480. The pressure measurements and all temperature measurements were taken every 1.5 seconds.

TRANSIENT REFRIGERANT MIGRATION DURING OFF CYCLE

Figure 4 shows the measured refrigerant mass in each section of the system during the three minutes when the compressor is turned off. The pressure ratio is also presented. Figure 5 shows the measured temperatures during the same off cycle period.

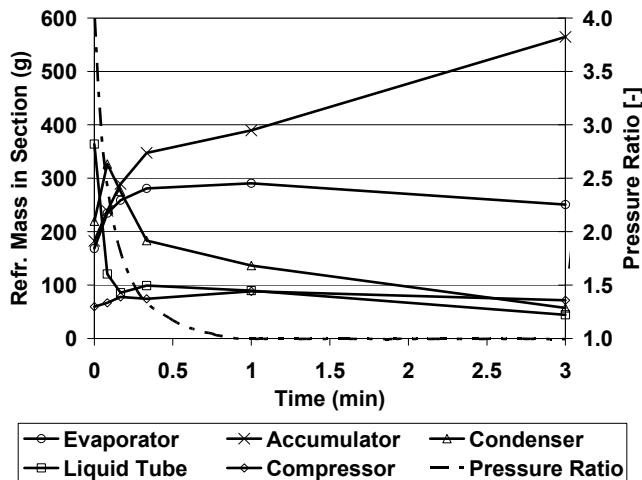


Figure 4: Refrigerant migration during off cycle

By inspecting the refrigerant mass migration in Figure 4 the conclusion is that the refrigerant migrates from the high-pressure components to the low-pressure components. At steady state, 58% of the total refrigerant mass in the system is found in the condenser and liquid tube sections. Three minutes after the compressor is stopped only 11% of the systems' refrigerant mass is found in those sections. The majority of the refrigerant

mass migrates to the accumulator section, which holds 56% after three minutes compared to 18% of the total refrigerant mass at steady state.

The major driving force for mass migration during the first minute after the compressor is stopped is the pressure difference. The pressure in the condenser and liquid tube sections is much higher than the pressure in the evaporator and accumulator sections during the first seconds. In addition, at steady state, the refrigerant inside the liquid tube section is subcooled and therefore in the liquid phase. This leads to a significant refrigerant migration from the liquid tube through the fixed orifice tube into the evaporator and subsequently to the accumulator. However, Figure 4 shows that after 5 seconds, the refrigerant mass in the condenser increased. Since the refrigerant mass is increasing in every section during the first 5 seconds except in the liquid tube section, it can be argued that liquid refrigerant is flowing back from the liquid tube section to the condenser for a short period.

After one minute, the majority of the refrigerant mass migration has taken place. However, Figure 4 shows that between minute one and three refrigerant migrates from all components to the accumulator. Since the airflow rates are maintained at the heat exchangers, the lowest temperature in the whole system can be found in the accumulator during the off cycle. This leads to condensation of refrigerant vapor inside the accumulator and therefore to mass migration. It should be noted that if the system is turned off for a longer period of time, the compressor eventually will have the lowest temperature in a real system. This, of course, will lead to a redistribution of refrigerant charge.

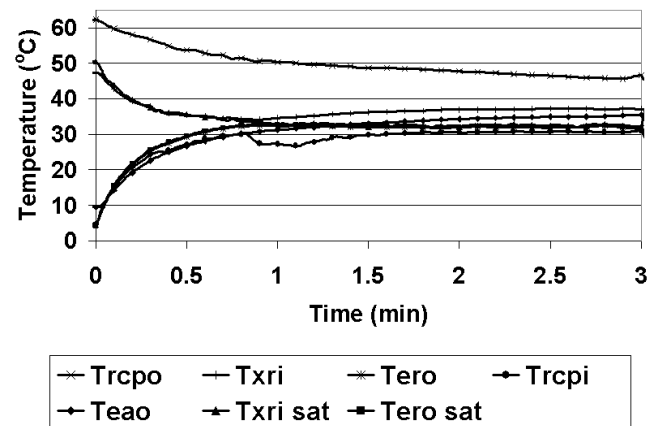


Figure 5: Temperatures during off cycle

Although the temperature of the refrigerant inside the accumulator was not measured, Figure 5 gives an indication that the temperature inside the accumulator is lower than the saturation temperature based on the measured pressure. After 50 seconds, the thermocouple reading at the suction side of the compressor inlet (Trcpi) drops by 4°C for a period of 20 seconds and then gradually increases again. From Figure 4 it can be seen