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EXPERIMENTAL ANALYSIS OF AN AUTOMOTIVE AIR CONDITIONING SYSTEM WITH TWO-PHASE FLOW MEASUREMENTS

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ABSTRACT

A fully equipped automotive air-conditioning test system consists of 5 main components: a compressor, a condenser, an orifice tube, an evaporator and an accumulator. R-134a is used as refrigerant. The coefficient of performance, evaporator cooling capacity, compressor power consumption, total mass flow rate, vapor mass flow rate, liquid mass flow rate and oil in circulation, pressures and temperatures of refrigerant at each component (inlet and outlet) are measured and analyzed with the variation of the temperatures at evaporator and condenser and refrigerant charge. The systematic experimental results obtained from this real-size test system depict the relations of the above-mentioned parameters in an automotive air conditioning system, which constitute a useful source for automotive air conditioning systems design and analysis. The two-phase flow measurements realized in this work provide an extremely important tool to diagnose system performance.
Keywords: Automotive; Air conditioning; Two-phase flow; Performance; R-134a; Experimental

1. INTRODUCTION

Compared with other air conditioning (A/C) systems, automotive A/C systems have some significant characteristics. Automotive A/C systems present challenges not normally encountered in stationary A/C systems, such as those used in building A/C systems. For example, the A/C panel outlet airflow direction, volume, velocity and temperature must be adjustable over a wide range of climatic and driving conditions. The rotating speed of the compressor is usually directly related to the vehicle speed, which changes dramatically. The sun load through the windshield and side windows is much greater than that through buildings. The A/C system must be capable of cooling down a hot vehicle quickly and it must be reasonably quiet. All these aspects make the analysis of automotive A/C system more complicated than that of a stationary A/C system (Bhatti, 1999; Kargilis, 2003).

During the last two decades automotive A/C systems have experienced worldwide significant advances introduced by industry and research institutes. Experimental analysis is an effective method in analyzing automotive A/C system as well as the method of simulation (Joudi *et al.*, 2003). Ratts and Brown (2000) used experimental method to analyze automotive A/C system COP, focusing on the relationships between the COP and the compressor revolution and vehicle speed. Kiatsiriroat and Euakit (1997) analyzed the COP of an automotive A/C system with R12/R124/R152a refrigerant mixture. Jabardo *et al.* (2002) analyzed an automotive A/C system with a variable capacity compressor. Schwarz *et al.* (2002) studied the flow inside an accumulator of an automotive air-conditioning system, which is critical to system COP, lubrication and safety of the compressor. Gu *et al.* (2003) experimentally determined the two-phase flow inside an evaporator and noticed its effect on heat transfer performance of an evaporator. In this work, two-phase flow is measured using a custom-built vapor-liquid separation device and two Coriolis mass flow meters. Data analyses will quantitatively determine the two-phase state at each stage of the

system, e.g., refrigerant quality, two-phase flowrate etc, which are closely related to the performance of an automotive A/C system, but a lack of knowledge on this aspect exists.

2. DESCRIPTION OF EXPERIMENTAL SETUP

There are two main types of automotive air-conditioners: one is a Thermal Expansion Valve (TXV) system which regulates rate of refrigerant flow into evaporator as governed by evaporator outlet pipe temperatures sensed by the sensing bulb, and the other is Clutch Cycling Orifice Tube (CCOT) system which controls evaporator temperature by turning the compressor on and off with a clutch cycling switch. A CCOT automotive A/C system includes a compressor, a condenser, an expansion device, an evaporator and an accumulator as shown in Figure1.

The schematic illustration of the experimental setup used in this experimental investigation is shown in Figure 2. It is a closed refrigeration loop charged with R-134a (1,1,1,2-Tetrafluoethane) as working fluid and built to operate as a CCOT system. The main components are a compressor stand, a water-cooled condenser stand, an evaporator stand, two mass flow meters, an accumulator stand, a separator, and various process instruments. These stands enable an easy service access and changing system components.
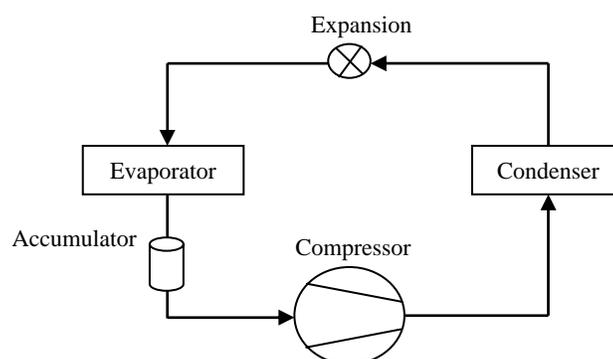

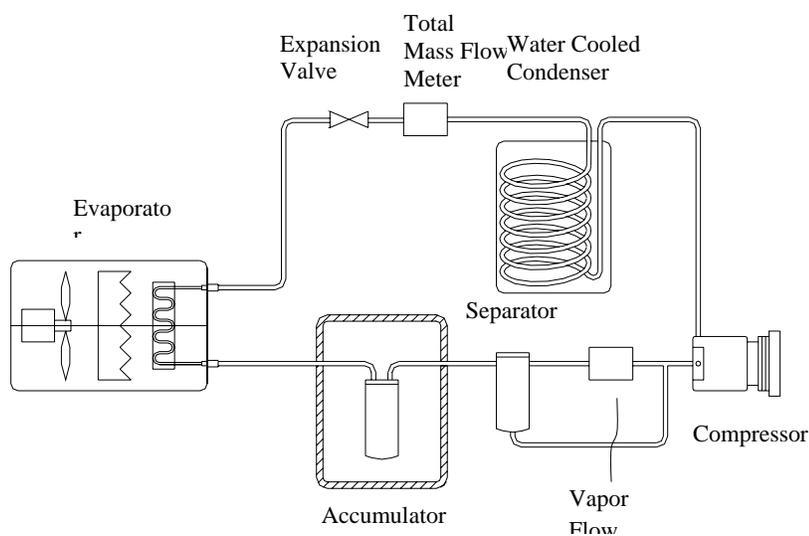Figure 1. Vapor compression refrigeration cycle

Figure 2. Experimental Setup

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The compressor stand consists of a typical automotive swashplate type compressor driven by a 10 HP variable-speed electric motor. A one-to-one pulley ratio was used to connect the electric motor to the compressor. The water-cooled condenser stand consists of a helical coil of half-inch diameter aluminum pipe as condenser, a water bath, a water pump, a valve, and temperature controller. A mass flow meter was installed at the condenser outlet to measure the total mass flow rate of the refrigerant–lubricant oil flow. The compressor is lubricated using Polyakylene-glycol (PAG). A sight glass tube is placed at the outlet of the flow meter to ensure that the flow meter chamber was filled with sub-cooled liquid refrigerant–PAG oil single phase. An orifice tube is used as expansion device to change the higher pressure sub-cooled liquid refrigerant into lower pressure and temperature vapor-liquid mixture. The evaporator stand consists of an automotive evaporator, an electric heater, an air-blower to circulate the air inside the duct to simulate environmental conditions, and a temperature controller to keep the air at a certain temperature. The evaporator stand was well insulated so that the power input from heater is considered as the same as cooling capacity. The accumulator stand consists of an accumulator, an electric heater to simulate environment temperature, a temperature controller to keep the air at a certain temperature, and an insulated box to contain these components. A sight glass tube is placed at the outlet of the accumulator for observing the refrigerant–PAG oil flow (two-phase). An aluminum pipe, 1.2 m long and 52 mm in diameter, is placed following the sight glass tube, which allows the annular flow to slow down and become separated. The flow separates completely in the liquid/vapor separator. The refrigerant vapor flows through the mass flow meter so that the vapor flow rate was measured. A sight glass tube at the outlet of mass flow meter ensures that the flow meter chamber was filled with vapor refrigerant only. Then, the vapor refrigerant and liquid refrigerant–PAG oil are mixed again at the inlet of compressor.

To conduct data acquisition processing and control the operation of A/C system, a control and measurement system was set up, which includes thermocouples, sensors, transducers, data acquisition devices, signal conditioning extensions for instrumentation devices and a computer with LabVIEW software. By this system, the temperature and pressure at each component, mass flow rate, power, compressor speed are measured as analog signals, then the analog signals are converted to digital signals and sent to the computer to display on LabVIEW windows directly or be processed to calculate other parameters. All the data can be recorded to Excel file in the interval up to once one second. This system also can control the operation of the air conditioning system, such as turning the power on or off, setting the temperature of evaporator and condenser, adjusting the compressor speed. Meters are placed at the inlet and outlet of the compressor to show the gauge pressure at these two locations.

3. EXPERIMENTAL RESULTS AND DISCUSSIONS

The experiments were carried out at different refrigerant charges, evaporator air-side temperatures, and condenser water temperatures. Several parameters were measured, charted and discussed in following paragraphs, such as the cooling capacity, the compressor power consumption, total flow rate, vapor flow rate, the quality (R-134a and PAG) at the inlet of compressor, the pressure at outlet of compressor, and the pressure at the outlet of evaporator.

3.1 Effects of Refrigerant Charge

In the automotive A/C system, loss of refrigerant through seals, elastomeric hoses, and fittings causes a charge reduction over time. The practical solution is overcharging the systems, and let the accumulators contain the spare refrigerant. It is time to recharge the system when enough refrigerant is lost so that A/C performance is degraded. The effects of refrigerant charge on system operation are shown in Figures 3-5. During this experiment, the evaporator air-side temperature is at 30 ^{o}C, condenser water temperature 35 ^{o}C and the compressor speed is 2000 rpm. Figure 3 shows that the total mass flow rate (R-134a and PAG oil) increases slightly, vapor flow rate stays at a constant value or decreases slightly and the quality at accumulator outlet decreases slightly with the increasing of refrigerant charge. The quality is the ratio of vapor flow rate to total mass flow rate, which is believed to have a major influence on the compressor. The quality is controlled by the accumulator, in which vapor R-134a, liquid R-134a and PAG oil separate and PAG oil is metered back to the compressor. Since the PAG oil and liquid R-134a are miscible there will always be liquid R-134a metered with the PAG oil. The high speed vapor R-134a flow generally also causes some liquid (R-134a and PAG oil) carryover. When the refrigerant charge increases, more refrigerant is stored in accumulator, and more liquid refrigerant and PAG oil are returned to the compressor. The mechanism of oil return will be discussed in a separate work. Figure 4 shows that cooling load and compressor power input

increase slightly, but COP decreases slightly with the increasing of refrigerant charge. COP is the ratio of cooling load to compressor power input.
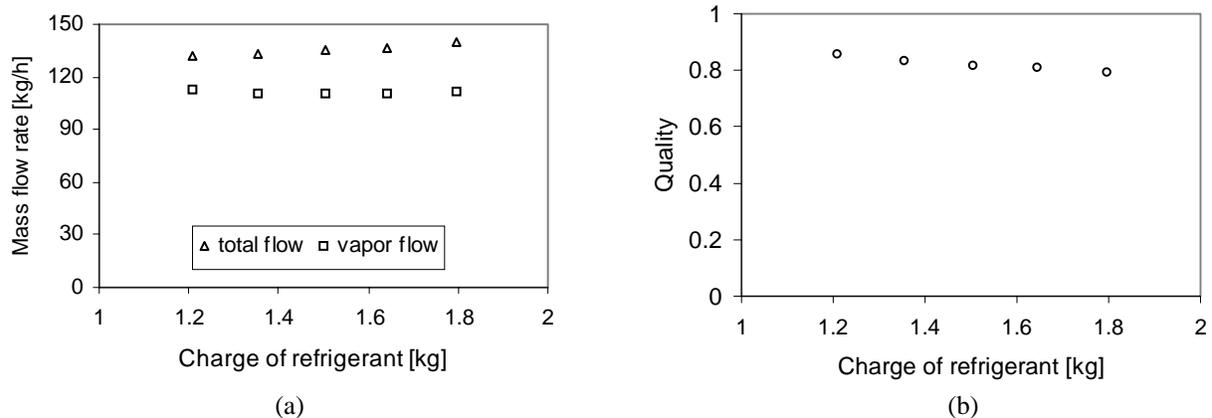

Figure 3. The effects of refrigerant charge on: (a) total mass flow rate and vapor flow rate; (b) quality
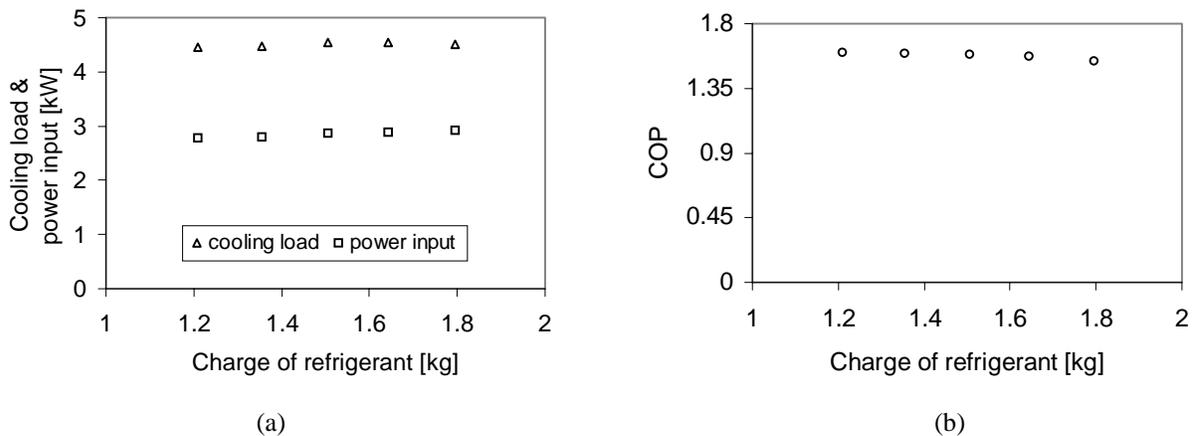

Figure 4. The effects of refrigerant charge on: (a) cooling load and compressor power input; (b) COP
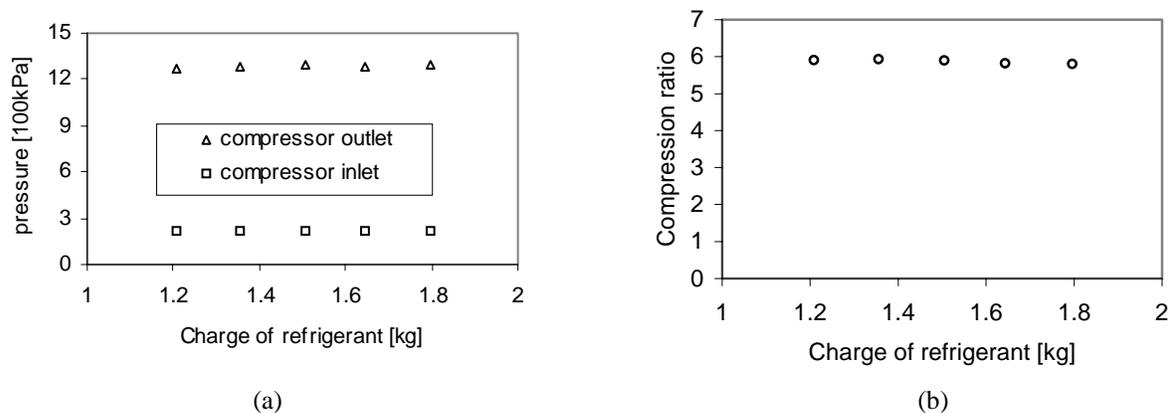

Figure 5. The effects of refrigerant charge on: (a) compressor outlet pressure and inlet pressure; (b) compression ratio

The decrease of quality, as shown in Figure 3, means relatively less vapor is compressed in the compressor, which causes the decrease of COP. Figure 5 shows the compressor outlet pressure and inlet pressure increase slightly with the increase of refrigerant charge, but the compression ratio decreases with the increase of refrigerant charge. The compression ratio is the ratio of outlet pressure to inlet pressure.

The compression ratio directly affects the volumetric efficiency of the compressor. The volumetric efficiency of the compressor is the ratio of the actual delivered gas volume to the swept volume of the cylinder. When the compression ratio increases, the actual delivered gas volume decreases. Therefore, the volumetric efficiency decreases with the increasing of the compression ratio, because of the high fraction residual of gas.

In general, all the parameters, such as quality, cooling load and COP, change slightly or keep constant, when refrigerant charge is between 1.2 kg and 1.8 kg for this specific system. When refrigerant charge is less than 1.2 kg or more than 1.8 kg, COP decreases apparently. Jabardo *et al.* (2002) obtained the similar results in their experiments. The results of this research showed one of the important functions of the accumulator, which is to store excess refrigerant (Lee *et al.* 2003). However, that is not included in this work. When refrigerant charge varies within 1.2 kg to 1.8 kg, the accumulator performs well to hold excess refrigerant with no or slight effect on the system. A refrigerant charge lower than 1.2 kg cannot meet the minimum demand of the system; the condenser cannot not get enough refrigerant vapor; the system runs at a reduced capacity. A refrigerant charge more than 1.8 kg causes excess refrigerant that is more than accumulator can hold; large amount of liquid returning back to the compressor will affect the vapor compression, so the COP decreases.

3.2 Effect of Air-side Temperature to Evaporator

The effects of evaporator air-side temperature are shown in Figures 6-8. During these experiments, the charge of R-134a is 1.8 kg, condenser water temperature at 43 ^{0}C and the compressor speed at 1500 rpm. When the evaporator air-side temperature increases the temperature difference between air and refrigerant in the evaporator increases, which improves the heat exchange efficiency of the evaporator. Figure 6(a) shows that both total mass flow rate (R-134a and PAG oil) and refrigerant vapor flow rate at compressor inlet increase with increasing evaporator air-side temperature. However, the quality decreases sharply to a point then increases slightly. When the velocity of vapor is very low, it can only carry a limited amount of liquid R-134a and PAG oil, but when the velocity approach a certain value, liquid R-134a and PAG oil could flow at high velocity because of the interfacial shear force (Huang and Gu, 1989), which dominates the liquid transport and the total quality. From thermodynamics of vapor compression cycles, the temperature differences between hot and cold reservoir are critical to COP of the cycles. Figure 7 shows that cooling load, compressor power input and COP all increase with increasing evaporator air-side temperature. Figure 8 depicts the compressor outlet pressure and inlet pressure increase with the increase of evaporator air-side temperature, but the compression ratio decreases with the increase of evaporator air-side temperature. The compressor outlet pressure is mainly determined by the ambient temperature of condenser, increasing with increasing ambient temperature of condenser. The compressor inlet pressure is mainly determined by the
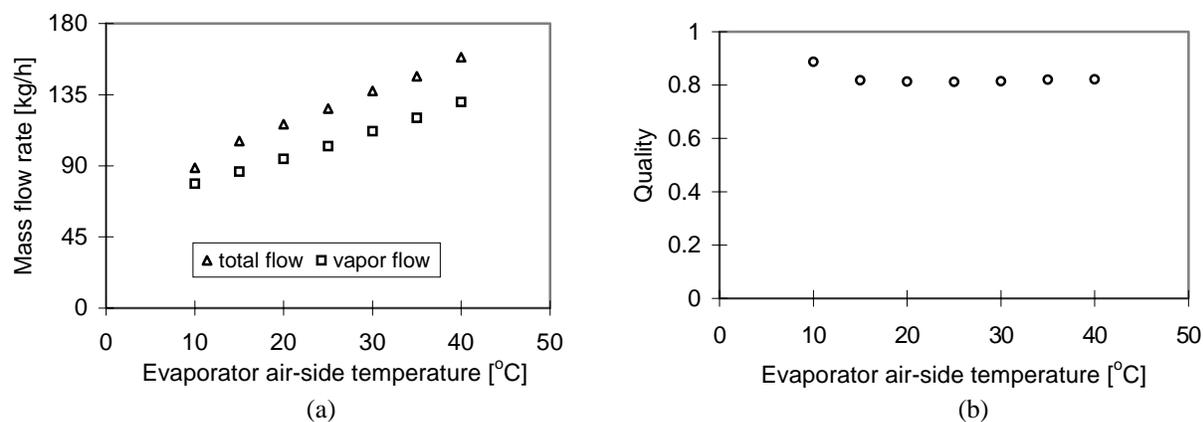

Figure 6. The effects of evaporator air-side temperature on: (a) total mass flow and vapor flow rate; (b) quality
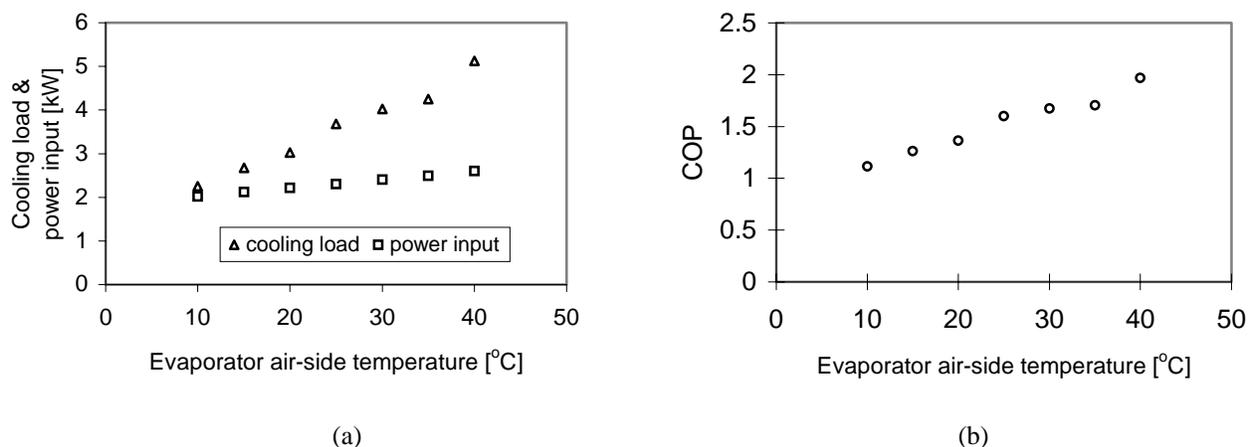
(a)

(b)

Figure 7. The effects of evaporator air-side temperature on: (a) cooling load and compressor power input; (b) COP
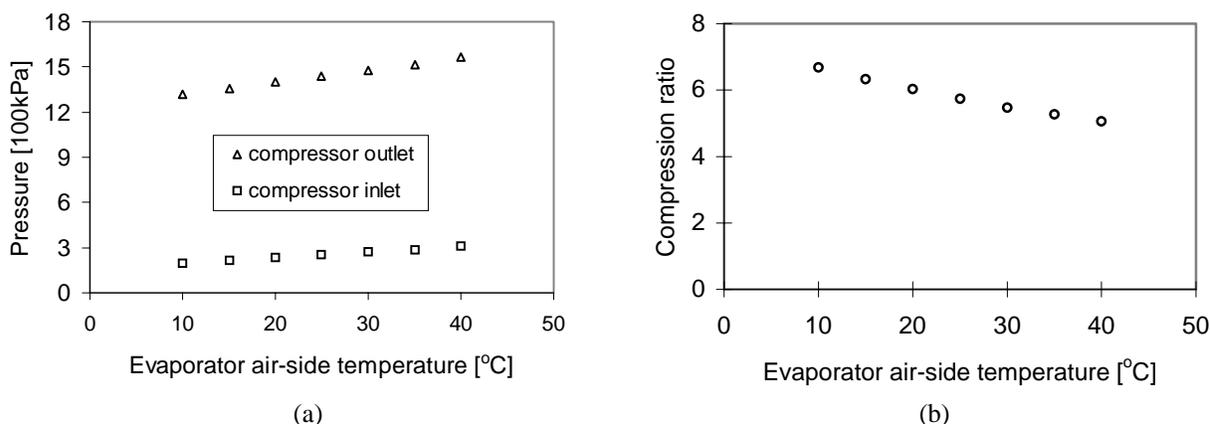

(a)

(b)

Figure 8. The effects of evaporator air-side temperature on: (a) compressor outlet pressure and inlet pressure; (b) compression ratio

ambient temperature of the evaporator, increasing with increasing ambient temperature of the evaporator. In this case, condenser water temperature is constant at 43 ^{0}C, and the evaporator air-side temperature increases, so the compression ratio decreases.

3.3 Effect of Condenser Water Temperature

The effects of condenser water temperature are shown in Figures 6-8. During this experiment, the charged R-134a is 1.8 kg, evaporator air-side temperature is at 30 ^{o}C, and the compressor speed is 1500 rpm. When the condenser water temperature increases the temperature difference of water and refrigerant decreases, which weakens the heat exchange efficiency of the condenser. Figure 9(a) shows that the total mass flow rate (R-134a and PAG oil) and vapor flow rate at the compressor inlet increase with increasing condenser water temperature. However, the quality decreases. Figure 10 shows that cooling load decreases, compressor power input increases, and COP decreases with increasing condenser water temperature. Figure 11 shows the compressor outlet pressure increases sharply and inlet pressure increases slightly with increasing condenser water temperature, because the compressor outlet pressure is mainly determined by condenser ambient temperature that increases from 25 ^{o}C to 50 ^{o}C, and the compressor inlet pressure is determined by evaporator ambient temperature that is constant at 30 ^{o}C in this experiment. The above factors result in an increasing of the compression ratio with the increasing of condenser water temperature, which causes a decrease of the volumetric efficiency of compressor.
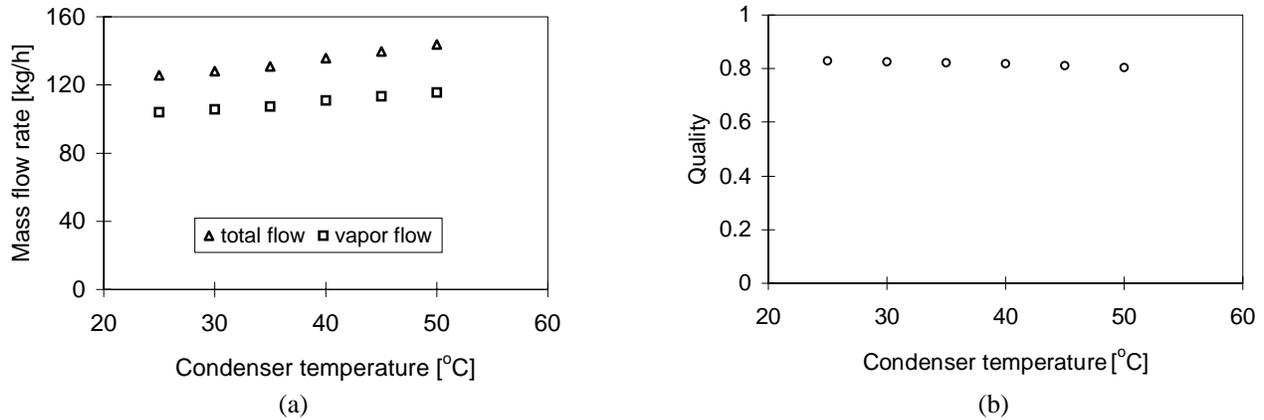
(a)

(b)

Figure 9. The effects of condenser temperature on: (a) total mass flow rate and vapor flow rate; (b) quality
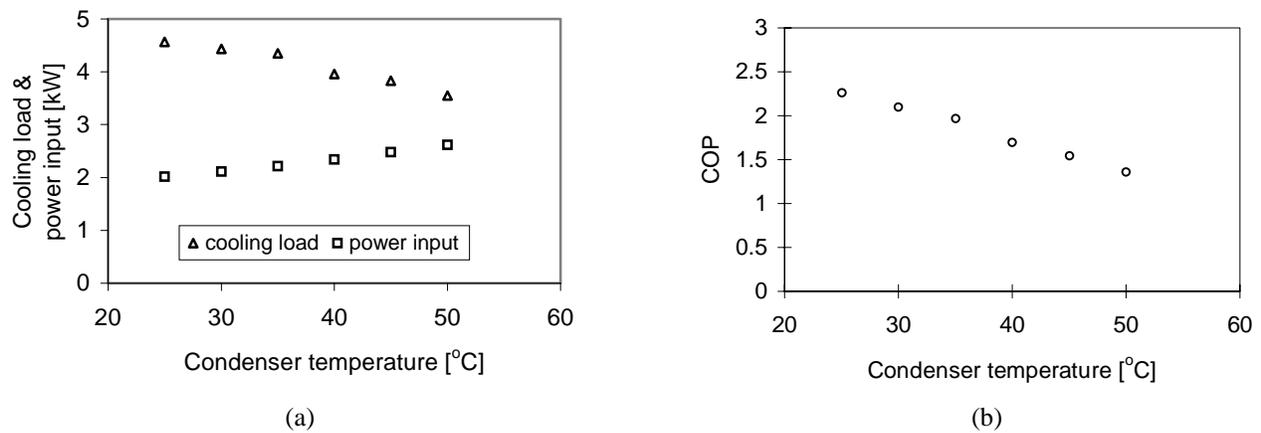

(a)

(b)

Figure 10. The effects of condenser temperature on: (a) cooling load and compressor power input; (b) COP
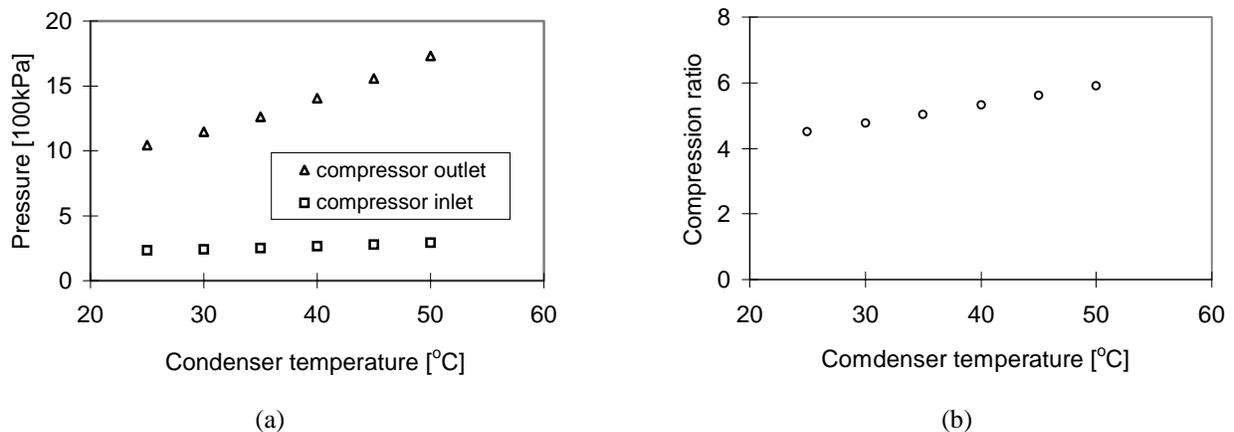

(a)

(b)

Figure 11. The effects of condenser temperature on: (a) compressor outlet and inlet pressure; (b) compression ratio

3.4 Experimental Uncertainty

The uncertainty in the temperature data collected by the data acquisition system was estimated to be ±0.2 °C; Pressure sensors, ±10 kPa; Coriolis mass flow meters, ±0.1 kg/h; Refrigerant charge, ±0.001 kg. The thermostats used could control the desired constant temperatures within ±1°C.

4. CONCLUSIONS

With the experimental set-up, some results have been obtained with two-phase flow measurements, which revealed some unknown aspects of the automotive air conditioning systems. The following remarks summarize some key performance characteristics of an automotive air-conditioning system with an accumulator. These results could be used to guide accumulator sizing and optimizing in future work.

1. The total mass flow rate of circulation in an automotive air conditioning system increases with the increase of refrigerant charge, evaporator air-side temperature and condenser water temperature. The total quality at the accumulator outlet increases with the increase of evaporator air-side temperature, and decreases with the increase of refrigerant charge and condenser water temperature.
2. The cooling capacity does not change with the variation of refrigerant charge, but increases with the increase of evaporator air-side temperature, and decreases with the increase of condenser water temperature. Coefficient of performance of the system decreases with the increase of refrigerant charge and condenser water temperature, increases with the increase of evaporator air-side temperature.
3. The compression ratio decreases with the increase of refrigerant charge and evaporator air-side temperature, increases with the increase of condenser water temperature. The change of compressor volumetric efficient is opposite to that of compression ratio.

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