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Observation of R600a flow at subcooled temperature conditions in a vapor-compression refrigeration system

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ABSTRACT

Vapor compression refrigeration system is usually designed such that refrigerant exits a condenser at subcooled liquid state in order not to cause possible damage to expansion devices. Even in small refrigeration systems incorporating capillary tube as an expansion device, the refrigerant is supposed to be subcooled before entering the expansion device.

In this study, experimental apparatus equipped with several thermocouples, pressure transducers and visualization device was constructed in order to observe the behavior of refrigerant flow at condenser outlet of a vapor compression refrigeration system. Visual observation as well as temperature and pressure readings indicate the presence of two-phase flow at highly subcooled temperature. An alternative equation was suggested to calculate the enthalpy of the two-phase refrigerant in subcooled region and verification test was conducted.

1. INTRODUCTION

Domestic refrigerators are significant energy consuming appliances (Radermacher et al., 1996). Household refrigeration systems are typically constructed using vapor compression refrigeration cycle systems, and these systems are equipped with a compressor, a condenser, an expansion device, and an evaporator. The condenser changes the state of the refrigerant from a superheated vapor phase to a subcooled liquid phase through rejecting the heat into the ambient atmosphere.

For numerous reasons, refrigeration systems are designed such that the state of refrigerant at the inlet of an expansion device is in a subcooled liquid state. Previous studies that investigated the characteristics of capillary tubes and condensers considered the refrigerant state at the inlet of the expansion device as a single-phase liquid (Melo et al., 2002; Bansal and Yang, 2005; Chingulpitak et al., 2012; Kaew-on et al., 2012; Dabas et al., 2014). However, some studies have reported two-phase flows of refrigerant at the expansion device inlet despite the subcooled condition. Boeng and Melo (2012) visually observed the refrigerant at the inlet of the capillary tube with various operating conditions. They claimed that a two-phase flow could be observed at the capillary tube inlet even at a subcooling temperature of 5 K. Hartmann and Melo (2013) performed an experimental study to determine the cause of noise in a R134a household refrigerator. They used an accelerometer to measure the vibration at the inlet of the capillary tube. Visualization devices were used to observe the refrigerant flow behavior in the same area. They confirmed the presence of vapor bubbles in the capillary tube and concluded that these were the source of the noise.

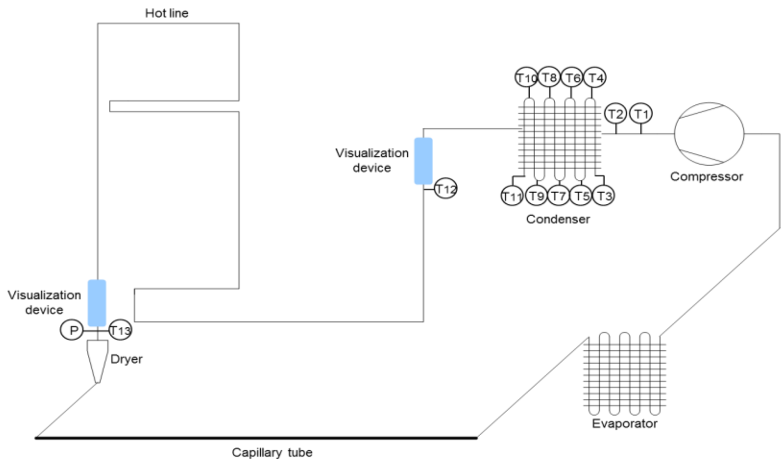


Figure 1: Schematic of the refrigerator experimental apparatus

Martinez et al. (2014) performed visualization experiments to assess the actual conditions at the capillary tube inlet. They observed the two-phase flow at the capillary tube inlet in subcooled conditions. In the studies mentioned above, however, sound proof, such as a direct measurement of refrigerant vapor temperature and energy balance, has not been provided to demonstrate the two-phase flow of the refrigerant in subcooled temperature conditions.

If the refrigerant flows through a refrigeration system as a non-equilibrium two-phase flow even at highly subcooled temperature conditions, the actual behavior of the refrigeration system will deviate from the expected behavior, which is determined based on thermodynamic equilibrium assumptions. In this regard, more detailed experimental work is necessary in order to investigate the presence of the non-equilibrium two-phase flow in the line between the condenser and capillary tube. The primary objective of this study is to clarify the existence of the two-phase flow in the line between the condenser and capillary tube in subcooled temperature conditions and to investigate the thermodynamic state of the refrigerant flow. The experimental equipment was designed to visualize and analyze the phase behavior of the refrigerant in the subcooled region.

2. TWO-PHASE FLOW AT SUBCOOLED TEMPERATURES IN A REFRIGERATOR

A domestic refrigerator that is commercially available was modified in order to visually observe the refrigerant flow. Figure 1 presents a schematic of the refrigerator used in this investigation. This refrigerator was originally designed for R600a refrigerant with 80 g charge and has a storage capacity of 820 liters. This refrigerator consisted of a linear compressor, a wire-tube condenser, a capillary tube, and a fin-tube evaporator. A pressure transducer with

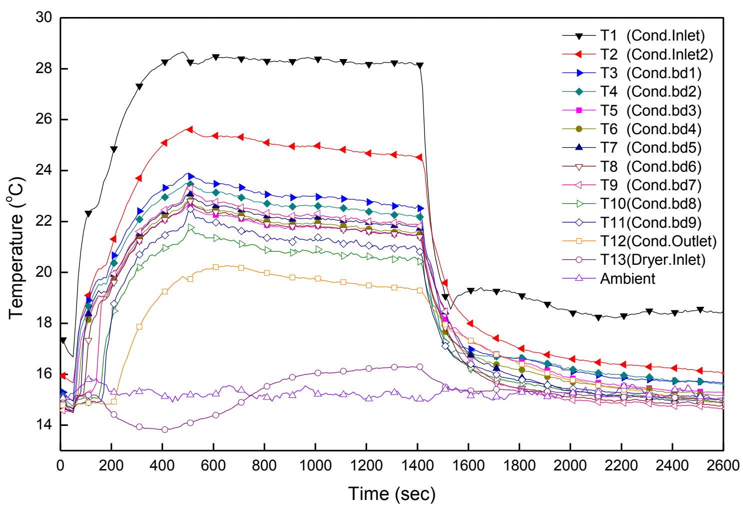
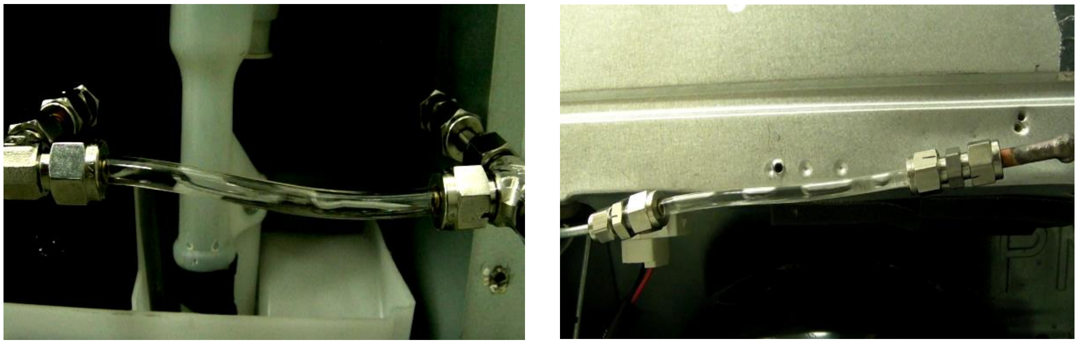


Figure 2: Temperature profiles at the condenser, dryer, and ambient during the ON/OFF cycle



(a) (b)

Figure 3: Phase behavior in the subcooled region: (a) condenser outlet and (b) dryer inlet

an uncertainty of ±0.4 kPa was installed upstream of the dryer in order to measure the condensing pressure of the refrigerator. The temperatures were measured using thermocouples (type T) with an uncertainty of ±0.3 ℃. The thermocouples were attached to the tube surface upstream of the dryer, the inlet and outlet of the condenser, and along the heat transfer tube of the condenser. Transparent tubes were installed upstream of the dryer and at the condenser outlet in order to visually observe the flow behavior of the refrigerant. This refrigerator was placed in a temperature and humidity controlled chamber. The experiments were conducted in ambient conditions of 15 ℃ and 40% relative humidity. In this refrigerator, 80 g of R600a was charged.

The temperature variations at each location during one cycle of ON/OFF operation are presented in Fig. 2. The temperatures tended to remain steady until 1400 seconds and then it dropped to the ambient temperature after 1400 seconds because the compressor was turned off. The condensing pressure immediately prior to the compressor being turned off was 330.88 kPa, and the corresponding saturation temperature was 23.0 ℃. Compared with this saturation temperature, the refrigerant was speculated to be superheated at the condenser inlet (T_1, T_2) and saturated in the condenser (T_3–T_{11}). The refrigerant at the condenser outlet (T_{12}) and dryer inlet (T_{13}) appeared to be subcooled at 3 ℃ and 6 ℃, respectively. In this condition, photographs of the refrigerant flow through the transparent sections were taken. Figure 3 presents the photographs taken at the condenser outlet and dry upstream. Figures 2 and 3 demonstrate that R600a might flow in a two-phase flow state even though the refrigerant is in subcooled conditions.

3. NON-EQUILIBRIUM IN THE REFRIGERATION SYSTEM

In the previous experiments with a refrigerator, the two-phase flow of refrigerant was observed despite the subcooled temperature conditions. In order to investigate the thermodynamic state of R600a in this condition, the experimental apparatus of refrigeration system was constructed and its schematic is illustrated in Fig. 4. This

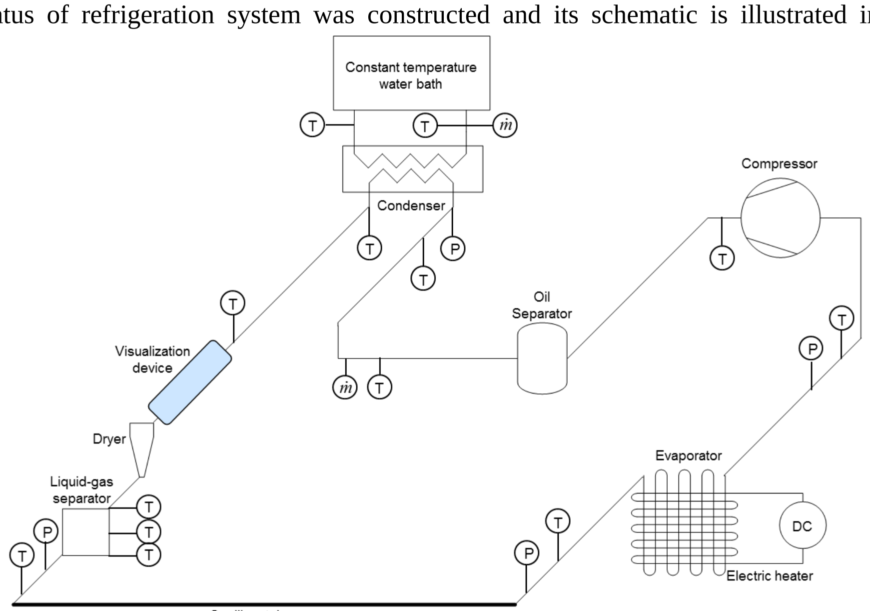


Figure 4: Schematic of the experimental apparatus

refrigeration system consisted of a linear compressor for R600a, a condenser of a plate-type heat exchanger, a capillary tube, and a fin-tube evaporator. In order to avoid the effect of oil circulation, an oil separator was installed between the compressor and condenser. A 15 cm long transparent tube was fitted in the refrigerant line at the upstream dryer in order to visually observe the behavior of the refrigerant flow in accordance with the test conditions. Figure 5 illustrates the liquid-gas separator incorporated into the refrigerant line at the inlet of the capillary tube. The refrigerant entered the liquid-gas separator near the top of the separator and separated into a gas phase and a liquid phase due to gravity. The liquid phase refrigerant exited the separator near the bottom of the separator. Three thermocouples were installed in the separator to measure the refrigerant temperature. Figure 5 illustrates that the upper and middle thermocouples measured the vapor phase temperature, while the lower thermocouple measured the liquid phase temperature. The uncertainties of all measurements are listed in Table 1. Table 2 describes the test conditions of the refrigeration system. In this system, 89 g of R600a was charged.

The pressure at the condenser inlet was 400.1 kPa and the corresponding saturation temperature was 29.6 ℃. The temperature reading upstream of the dryer was 12.7 ℃, which was 16.9 ℃ lower than the saturation temperature. If evaluated based on these temperature and pressure measurements, the refrigerant at the upstream dryer should be in a subcooled state. However, the two-phase mixture flow of refrigerant was observed upstream of the dryer as seen in Fig. 6. The two-phase refrigerant flowed through the dryer and entered the liquid-gas separator. The temperature measurements for the vapor phase according to the upper and middle thermocouples was 11.8 ℃ and 11.7 ℃, respectively. The temperature readings for the liquid phase according to the lower thermocouple was 11.5 ℃. These levels for the three temperature readings were approximately 18 ℃ lower than the saturation temperature. These temperature readings demonstrate that both the vapor phase and the liquid phase were simultaneously present in the subcooled temperature level. Therefore, this two-phase flow of refrigerant in the subcooled temperatures is believed to be in a thermodynamic non-equilibrium state where a subcooled liquid and a subcooled vapor coexist. It is interesting to observe that the refrigerant existed in the vapor phase even at highly subcooled temperatures.

4. ENTHALPY OF NON-EQUILIBRIUM TWO-PHASE REFRIGERANT

The enthalpy of a subcooled liquid-subcooled gas mixture, i.e. $h_{ref,sub}$, can be expressed as follows:

$$h_{ref,sub} = (1-x)\cdot h_{ref,sub.liq}(T_{ref,sub}, P_{ref,sub}) + x\cdot h_{ref,sub.vap}(T_{ref,sub}, P_{ref,sub}) \tag{1}$$

where x represents the quality of the mixture. The enthalpy of a subcooled liquid, i.e. $h_{ref,sub.liq}$, can be determined if the pressure and temperature are known. Because the effect of pressure on the enthalpy of a subcooled liquid is negligible, the enthalpy can be obtained with only the temperature, as follows:

$$h_{ref,sub.liq} = h_{ref,sub.liq}(T_{ref,sub}, P_{ref,sub}) \cong h_{f,Tref} \tag{2}$$

where $h_{f,Tref}$ represents the enthalpy of a saturated liquid at the refrigerant temperature. The enthalpy of a

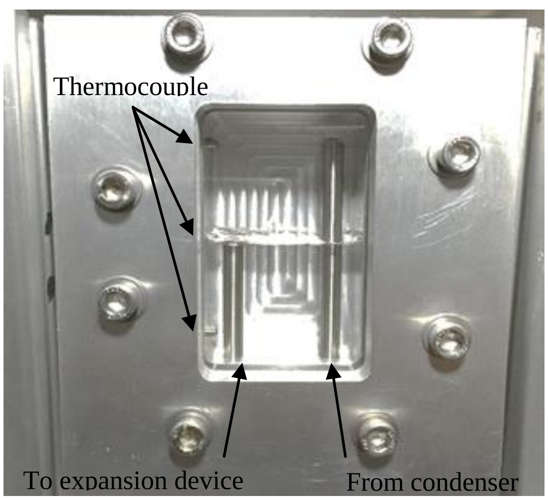


Figure 5: Liquid-gas separator and thermocouples used to measure the phase temperature

Table 1: Uncertainties of the measurements for the refrigeration system

Measured parameter	Uncertainty
Temperature	$\pm$ 0.3 ℃
Pressure	$\pm$ 0.4 kPa
Mass flow rate (Refrigerant)	$\pm$ 0.023 kg h^{-1}
Mass flow rate (Water)	$\pm$ 0.3 kg h^{-1}

subcooled vapor, i.e. $h_{ref,sub.vap}$, in Eq. (1) can be determined as the enthalpy of a saturated vapor at refrigerant temperature, i.e. $h_{g,Tref}$, subtracted by the enthalpy reduction due to the temperature drop, as follows:

$$h_{ref,sub.vap}(T_{ref,sub}, P_{ref,sub}) \cong h_{g,Tref} - C_p(T_{ref,sat} - T_{ref,sub}) \tag{3}$$

where C_P, $T_{ref,sat}$, and $T_{ref,sub}$ represent the specific heat, saturation temperature corresponding to the refrigerant pressure, and temperature of a subcooled refrigerant mixture, respectively. Substituting Eq. (2) and (3) into Eq. (1) gives the following relationship:

$$h_{ref,sub} = h_{f,Tref} + x \cdot (h_{g,Treft} - h_{f,Tref} - C_p(T_{ref,sat} - T_{ref,sub})) \tag{4}$$

Except for quality (x), all properties on the right side of Eq. (4) are obtainable from the TPT. The quality can be expressed using the density of each phase, void fraction (α), and slip ratio (S), as follows:

$$x = \frac{1}{\left(\frac{\rho_f}{\rho_g}\right)\left(\frac{1-\alpha}{\alpha}\right)\frac{1}{S} + 1} \tag{5}$$

The slip ratio is the ratio of the vapor phase velocity to the liquid phase velocity. If it is assumed that the refrigerant vapor can be treated as an ideal gas, the density of the subcooled vapor phase can be obtained using the following relationship:

$$\rho_{sub} = \frac{T_{sat}}{T_{sub}} \rho_{sat} \tag{6}$$

In order to obtain the quality using Eq. (5), the void fraction should be known. The void fraction can be obtained through analyzing digital video images of the two-phase refrigerant flow through the visualization device constructed from a transparent tube. The two-phase refrigerant in the horizontal circular tube flows in a slug flow pattern as depicted in Fig. 6. Several assumptions were used to calculate their volumes. For the spherical part of the bubble, the radius was assumed to be equal in all directions. The cap-shaped part was assumed to be a half ellipsoid. For the frustum of the cone-shaped part, the radius was assumed to linearly change along the flow direction. For the

Table 2: Test conditions at the condenser of the refrigeration system

Parameter	Value
Refrigerant side condenser inlet pressure (kPa)	400
Refrigerant side mass flow rate (g s^{-1})	0.311
Refrigerant side inlet temperature (℃)	31.8
Refrigerant side outlet temperature (℃)	9.3
Water mass flow rate (g s^{-1})	4.970
Water side inlet temperature (℃)	9.1
Water side outlet temperature (℃)	13.8

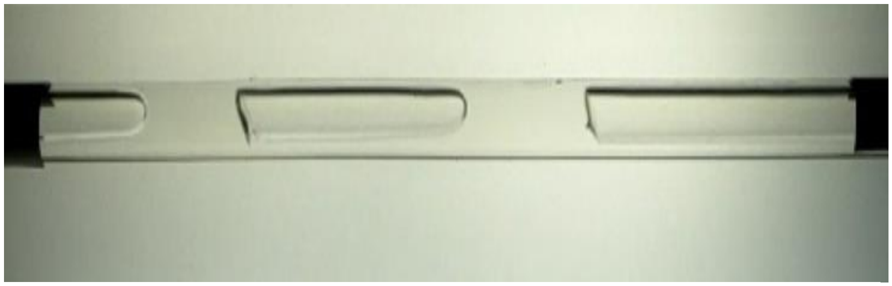


Figure 6: Phase behavior at the dryer inlet of the refrigeration cycle

cylindrical part of the bubble, the radius was assumed to be equal along the flow direction. With these assumptions, the volumes of each part of the bubble were calculated.

A simple experimental apparatus was constructed in order to verify the void fraction calculation method explained above. The mass flow rates of water and nitrogen gas were controlled using the gear pump and the mass flow controller (MFC), respectively. A pressure transducer and a thermocouple were installed in order to determine the thermodynamic properties of each fluid. Based on the mass flow rate and specific volume of each phase, the provided void fraction was obtained. The provided void fraction is expressed as follows:

$$\langle\overline{\alpha}\rangle = \frac{1}{\left(\frac{\rho_g}{\rho_f}\right)\left(\frac{\dot{m}_f}{\dot{m}_g}\right)S+1} \tag{7}$$

The specific enthalpy of the refrigerant at the inlet and the specific enthalpy of the water at the inlet and outlet of water-side were obtained from the TPT. The mass flow rates of the water and refrigerant were measured. Therefore, the specific enthalpy value at the condenser outlet of the refrigerant side was calculated.
The specific enthalpy of the refrigerant at the condenser outlet can be obtained from the TPT. As discussed earlier, this value is for an equilibrium state at a specified temperature and pressure. In order to assess whether the TPT can be used for a non-equilibrium condition, the specific enthalpy from the TPT at the measured temperature and pressure was compared with the values obtained from the experimental heat balance in Table 3. The specific enthalpies obtained from experimental heat balance and TPT were 239.6 kJ kg^{-1} and 227.3 kJ kg^{-1}, respectively. The specific enthalpy obtained from the TPT was 5.1% smaller than the value from the experimental heat balance. This result demonstrates that the TPT does not provide appropriate specific enthalpy values when the refrigerant is in a non-equilibrium condition.

Table 3 also compares the calculated specific enthalpy using Eq. (4). This calculation method was proposed in order to estimate the specific enthalpy of the refrigerant in a non-equilibrium two-phase state. The values and properties associated with the calculation of the specific enthalpy are listed in Table 4. The specific enthalpy calculated using Eq. (4) was 241.3 kJ kg^{-1}. The difference between the specific enthalpy calculated using the experimental heat balance and the calculated enthalpy was 0.7%. This result indicates that the calculated enthalpy proposed in this work can determine the specific enthalpy with a slight error in non-equilibrium conditions.

Table 3: Comparisons of the specific enthalpy values and error

Specific enthalpy	Value	% of error
Heat balance (kJ kg^{-1})	239.6	-
TPT method (kJ kg^{-1})	227.3	5.1
Calculated (kJ kg^{-1})	241.3	0.7

Table 4: Refrigerant properties for the calculated enthalpy

Property	Value
Void fraction	0.725
Quality	0.042
Liquid phase specific density (kg m^{-3})	566.89
Gas phase specific density (kg m^{-3})	9.45
Specific heat at constant pressure (kJ kg^{-1} K)	1.79
Specific enthalpy of saturated gas (kJ kg^{-1})	587.4
Specific enthalpy of saturated liquid (kJ kg^{-1})	227.2
Saturation temperature (℃)	24.7

5. CONCLUSION

In this paper, an experimental study on the two-phase behavior phenomenon in a subcooled region in a refrigeration system has been described. The results demonstrate that a two-phase refrigerant flow exists in the subcooled region in both commercial refrigerator-freezer and domestic refrigeration systems. The measured temperatures of the gas phase and liquid phase in the subcooled region were 11.8 ℃ and 11.5 ℃, respectively, but the saturation temperature was 29.6 ℃. These results indicate that a thermodynamic non-equilibrium flow coexists with the subcooled gas and the subcooled liquid exists in the subcooled region despite the subcooled condition.

An enthalpy calculation method that considers the subcooled gas-subcooled liquid mixture was proposed. Compared with the heat balance method, the calculated enthalpy predicted the specific enthalpy value well with an error of 0.7%. However, the TPT method predicted it with an error of 5.1% because this state was not in the equilibrium condition. Therefore, the calculated enthalpy method could define the specific enthalpy value with a small error in the non-equilibrium condition. In addition, the calculated enthalpy method can be applied to investigate the characteristics of refrigeration systems if a non-equilibrium phenomenon exists in the system.

NOMENCLATURE

C	specific heat	(kJ kg^{-1} K)
h	specific enthalpy	(kJ kg^{-1})
$\dot{m}$	mass flow rate	(kg h^{-1})
P	pressure	(kPa)
S	slip ratio	
T	temperature	(K)
x	refrigerant quality	

Greeks

α	void fraction	
ρ	density	(kg m^{-3})

Subscripts

f	saturated liquid
g	saturated vapor
p	constant pressure
ref	refrigerant
sat	saturation
sub	subcooled state
vap	vapor phase

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