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Visualization on Flow Patterns during Condensation of R410A in a Vertical Rectangular Channel

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The visualization experiments on HFC R410A condensation in a vertical rectangular channel (14.34mm hydraulic diameter, 160mm length) were investigated. The flow patterns and heat transfer coefficients of condensation in the inlet region were presented in this paper. Better heat transfer performance can be obtained in the inlet region, and flow regime transition in other regions of the channel was also observed. Condensation experiments were carried out at different mass fluxes (from 1.6 kg/h to 5.2 kg/h) and at saturation temperature 28^o C. It was found that the flow patterns were mainly dominated by gravity at low mass fluxes. The effects of interfacial shear stress on condensate fluctuation are significant for the film condensation at higher mass flux in vertical flow, and consequently, the condensation heat transfer coefficient increases with the mass flux in the experimental conditions. The drop formation and growth process of condensation were also observed at considerably low refrigerant vapor flow rate.

Keywords: flow pattern, condensation, vertical channel, heat transfer, R410A

Introduction

Flow pattern and heat transfer performance of refrigerants in a horizontal tube have been widely investigated in literatures. The study of condensation in a vertical tube or channel has been received comparatively little attention. Vertical downward gas-liquid flow is mainly dominated by gravity and vapor shear stress. Condensate film in inlet region is thin and heat transfer performance is better than that in other region. Condensate film increases gradually along the tube length. The wall is even fully covered by condensate in the downstream of the tube, as a result condensation heat transfer rate is much less than that in inlet region. For improving condensation heat transfer, the usage of micro-fin tubes [1-2] has increased the heat transfer performance of tubes, and numerous experimental studies have been done to measure heat transfer rate in mini and micro channel tubes, e.g. Wang and Du [3] and so on. On the other hand, reducing the length of tube can considerably improve the heat transfer performance, and it is also a reduction of the size and material of equipment. R410 being a kind of substitution of traditional refrigerants, particularly of R22, has been receiving support from the refrigeration and air-condensation industry, because of zero ODP (ozone depleting potential)

Wang and Du [3] presented a theoretical and experimental study on laminar film-wise condensation for vapor flow in small diameter tubes. They concluded the inclination angle affects condensation heat transfer

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Nomenclature

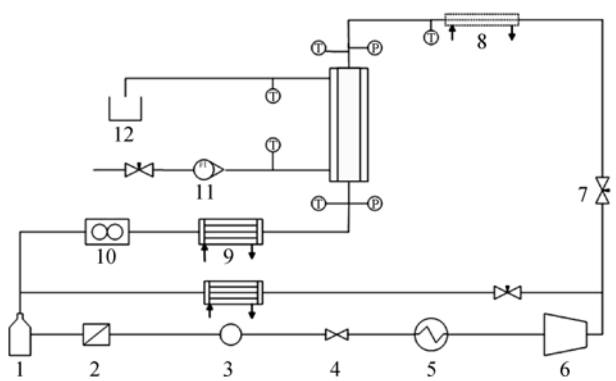
A	area (m^2)	**Greek letters**	
C	specific heat capacity	Δ	temperature difference
D	diameter(mm)	λ	thermal conductivity ($W/m^{-1}K$)
G	mass fluxes (kgh^{-1})	δ	thickness (m)
H	Heat transfer coefficient(W/m^2K^{-1})	**Subscripts**	
I	enthalpy (J/kg)	b	Between
M	water mass flow rate (kgs^{-1})	in	Inlet
Q	Heat transfer rate (W)	out	Outlet
T	temperature (℃)	re	Refrigerant
		w	Water

mainly by distributing gravity in stratifying the fluids and thinning the liquid film. In larger tubes, the former may become the main reason for heat transfer enhancement, especially in high vapor quality zone and at low inlet vapor Reynolds number. Lips and Meyer [4] conducted an experimental study of convective condensation of R134a. They found that the distribution of the fluids in the tube is the result of a balance between gravitational, shear and capillary forces, and inclination angle has a strong effect on the liquid distribution in the tube and thus affects the flow pattern. Dalkilic et al. [5] experimentally investigated the laminar film condensation and heat transfer coefficients of R134a in a smooth vertical tube. The results show that the condensation rate reaches the highest value at the pipe entrance where the highest local heat transfer coefficients exist, and the interfacial shear stress affects the condensation process of R134a in vertical tube. Dobson and Chato [6] studied the heat transfer and flow regimes during condensation of refrigerants in horizontal tubes. They focused on prediction of condensing heat transfer coefficients and the relationship between heat transfer coefficients in two-phase flow regimes. The dominant heat transfer mode was laminar film condensation in the top of the tube. This regime was characterized by heat transfer coefficients depending on the wall-to-refrigerant temperature difference, but nearly independent of mass flux. In the shear-dominated flow regime, forced-convective condensation was the dominant heat transfer mechanism. Olivier et al. [7] conducted a flow region observations in three different tubes and results showed that the micro-fin (helical micro-fin and herringbone) delays the transition from annular flow to intermittent flow, which enhances the heat transfer relative to the smooth tube. They explained that the intermittent flow regime is neither uniform nor steady. Investigation of refrigeration condensation heat transfer in channel has much less attention. Lahm et al. [8] investigated heat transfer of reflux condensation in a narrow tube (7mm inner diameter, 0.5m length) and a rectangular channel (7mm hydraulic diameter, 0.5 m length). The results showed that gas side heat and mass transfer resistances and the resistances of liquid film influence the condensation process.

Experimental apparatus and data reduction

Test system

The schematic diagram of the test apparatus is showed in Fig.1. The loops of the system were a refrigerant loop and a water cooling loop. The refrigerant loop was mainly consisted of an evaporator, compressor, a test section, condensers, and two condensation lines: the test line and the bypass line. Each of them had a control valve to control the refrigerant mass flux in the test lines and a condenser. Refrigerant vapor through the bypass line would be fully condensed into liquid by controlling the cooling water flow rate of the condenser. In the test line, the refrigerant vapor flowed through the pre-condenser, test section, condenser and then a Coriolis mass flow meter. The pre-condenser was used to control the inlet vapor temperature. A condenser was used to ensure the



1-Reservoir tank	2- Solenoid valve	3- Sight glass
4-Expansion valve	5-Evaporator	6-Compressor
7-Control valve	8-Pre condenser	9-Condenser
10-Coriolis mass flow meter	11-Rotameter	12- Water tank

Fig. 1 Schematic diagram of the test apparatus

vapor being fully condensed. The refrigerant mass fluxes through the test section were measured with Coriolis mass flow meter. Then test line combines with bypass line, and enters a reserve tank. Water cooling loop was used to cool the vapor through the test section. Two T-type thermocouples were used to measure the temperature of cooling water at inlet and outlet. In order to control and measure the water flow rate, a valve and a rotameter were positioned at the inlet of the test section.

The test section was a vertical copper channel heat exchanger (Fig. 2) with refrigerant vapor flows in a channel on one side, and cooling water flows also in a channel on the other side. The thickness of wall between the two channels is 3mm. The refrigerant vapor channel was covered with a piece of high borosilicate glass as the sight glass. Its hydraulic diameter is 14.34mm. A band-type heater was wrapped around the side of the sight glass to compensate heat losses due to heat dissipation. A high speed camera was used to record the flow regime of condensate along the channel.

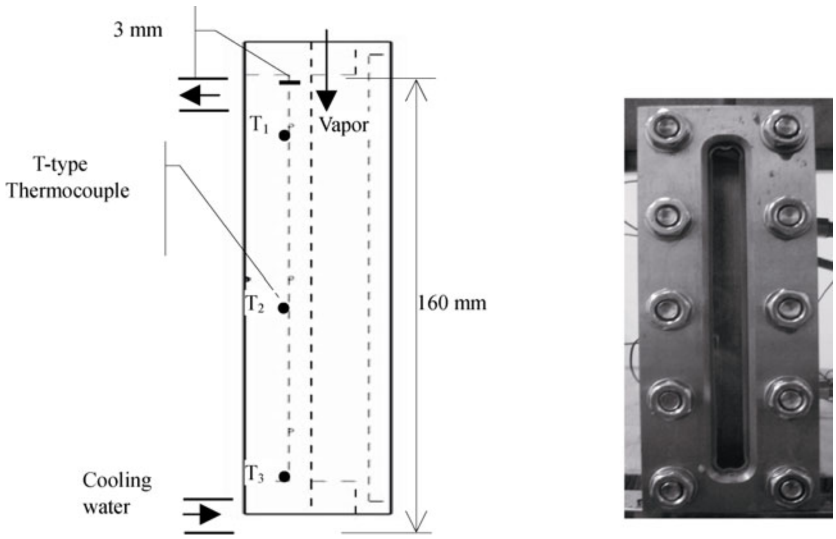


Fig. 2 Schematic diagram of the vertical channel test section

The effective length of vertical channel heat exchanger is 160mm. The T-type thermocouples and pressure transducer were installed at the inlet and outlet with capillary piping to measure the temperature and pressure of vapor in inlet and outlet, respectively. The wall between two channels is equally drilled three holes whose depth and diameter are both 1mm. In each hole, a thermocouple was placed to measure the wall temperature. All the thermocouples were calibrated.

Data reduction

The experimental data were collected after the condensation heat transfer test was in steady state. That means the inlet pressure and temperature, mass flow rate of refrigerant and water mass flow rate and temperature keep in stationary state. Each run took about 2 minutes at least.

The average condensation heat transfer coefficient in the test section was obtained as:

$$h = \mathrm{Q}_{\mathrm{re}} / (A_{re} \Delta T_{re,\ln}) \tag{1}$$

Where A_{re} is the heat transfer area of refrigerant channel and $\Delta T_{re,ln}$ is the logarithmic mean temperature difference between the vapor and the inner wall of channel. The heat transfer rate Q_{re} was determined from the mass flow rate and the change in enthalpy between the inlet supersaturated vapor and the outlet liquid. $i_{re,in}$ is the enthalpy of the supersaturated refrigerant vapor at the inlet and $i_{re,out}$ is the enthalpy of the condensate at outlet of test section.

$$\mathrm{Q}_{\mathrm{re}} = \bar{m}_{re}(i_{re,in} - i_{re,out}) \tag{2}$$

$$\Delta T_{re,ln} = \frac{(T_{re,in} - T_{wall}) - (T_{re,out} - T_{wall})}{ln\left[(T_{re,in} - T_{wall})/(T_{re,out} - T_{wall})\right]} \tag{3}$$

$$T_{wall} = T_b + Q_{re}\delta / \lambda \tag{4}$$

T_{wall} was arithmetic mean temperature of the inner wall calculated by the data of wall temperature between the refrigerant channel and cooling water channel.

The condensation heat transfer rate Q_w on the water side was obtained by the mass flow rates and the temperature difference of the cooling water:

$$\mathrm{Q}_{\mathrm{w}} = \bar{\mathrm{m}}_w c_{p,w}(T_{w,out} - T_{w,in}) \tag{5}$$

However, the heat transfer rate from cooling water should be equal to the refrigerant condensation heat transfer rate in the test section. The difference between Eq.(2) and Eq.(5) was below 20% due to the heat losses to the surroundings.

$$\Delta \mathrm{e} = \left|\mathrm{Q}_{\mathrm{re}} - \mathrm{Q}_{\mathrm{w}}\right| / \mathrm{Q}_{\mathrm{re}} \tag{6}$$

Measure uncertainty

The accuracy for sensors and parameters were summarized in Table 1. The experimental uncertainties of the heat transfer coefficient were calculated with the method of propagation of the errors applied to Eq. (1). The main part of the uncertainty came from the temperature measurement on the water side for the calculation of heat transfer rate and on the inlet refrigerant superheated vapor temperature. Uncertainty induced by vapor quality was negligible because superheated vapor was measured in the majority of experimental condensations. The relative uncertainties of condensation heat transfer coefficients were calculated to be less than 15％, depending on the experimental conditions.

Table 1 Accuracy for sensors and parameters

Refrigerant mass flow rate	±1%
Temperature difference	±0.2℃
Cooling water flow rate	±1.5%
Saturation temperature	±1℃

Results and discussion

For each experimental condition, the heat transfer coefficient was obtained and the flow pattern of R410A during condensation in a vertical channel was visualized by a high speed camera.

The dropwise condensation of R410A in the vertical channel

The processes of condensation drop formation and growth were observed during condensation at considerably low refrigerant vapor flow rate, shown in Fig.3.

Dry and clean copper channel inner wall can be seen at initial time. When refrigerant vapor flowed through the channel, a part of vapor condensed into fog in the channel, and some of the tiny drops appeared on the inner wall. About 2 minutes later, the inner wall of channel gradually became clear. The fog disappeared in the channel with heat transfer between vapor and wall, and surroundings was heated, shown in Fig.4. The temperature differences between T_1 and T_2 and between T_2 and T_3 decrease gradually at about 5 minutes. At very low refrigerant vapor flow rate, the shear stress plays a less important role and almost was negligible. The condensation drops did not line up in the vertical direction and with a variety of shapes. In the beginning, the condensation drop was very tiny which nearly covered entire inner wall, but the number of them decreased gradually with the drop growing (including drops merge and their own growth). At about 51 minutes, when the drop was large enough, tiny drop formed again on the exposed inner wall as shown in Fig.3. Fig.4 showed that the temperature difference between T_1 and T_2 was always apparently larger than the temperature difference between T_2 and T_3. It can be inferred that heat transfer performance at the inlet region is better than the other region, and that also showed us one of the ways to improve the condensing heat transfer performance of vertical channel.

Effect of mass flux on flow pattern

Not only gravity has an effect on condensation heat transfer in the vertical tube, but also interfacial shear stress affects the condensate film. The interfacial shear stress causes film fluctuation so that heat transfer is enhanced when vapor velocity increases above a certain value [9]. Lips and Meyer [4] summarized different types of flow patterns in their study for whole range of inclination angles and at large mass fluxes. Fig.5 showed some photos of the flow at different small mass fluxes.

At mass flux of 1.6kg/h, some of the drops quickly appeared and slipped on the inner wall along the perpendicular direction. Tadpole flow occurred with vapor con-

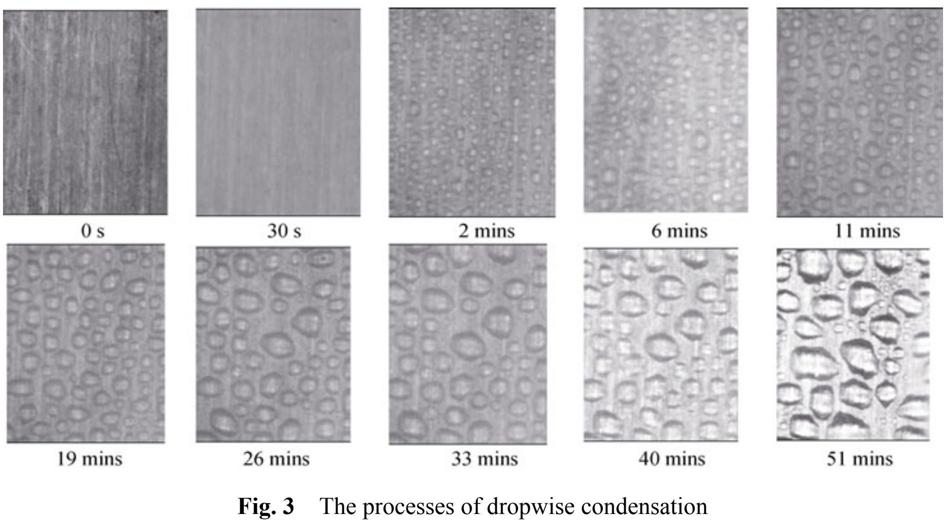


Fig. 3 The processes of dropwise condensation

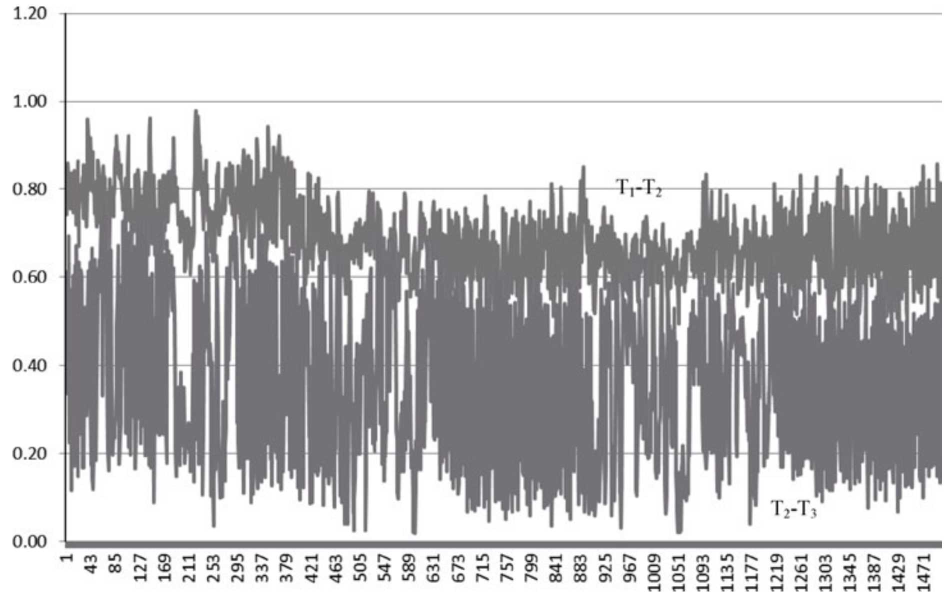


Fig. 4 Temperature difference between $T_1$and T_2 and between T_2 and T_3 during drop-wise condensation

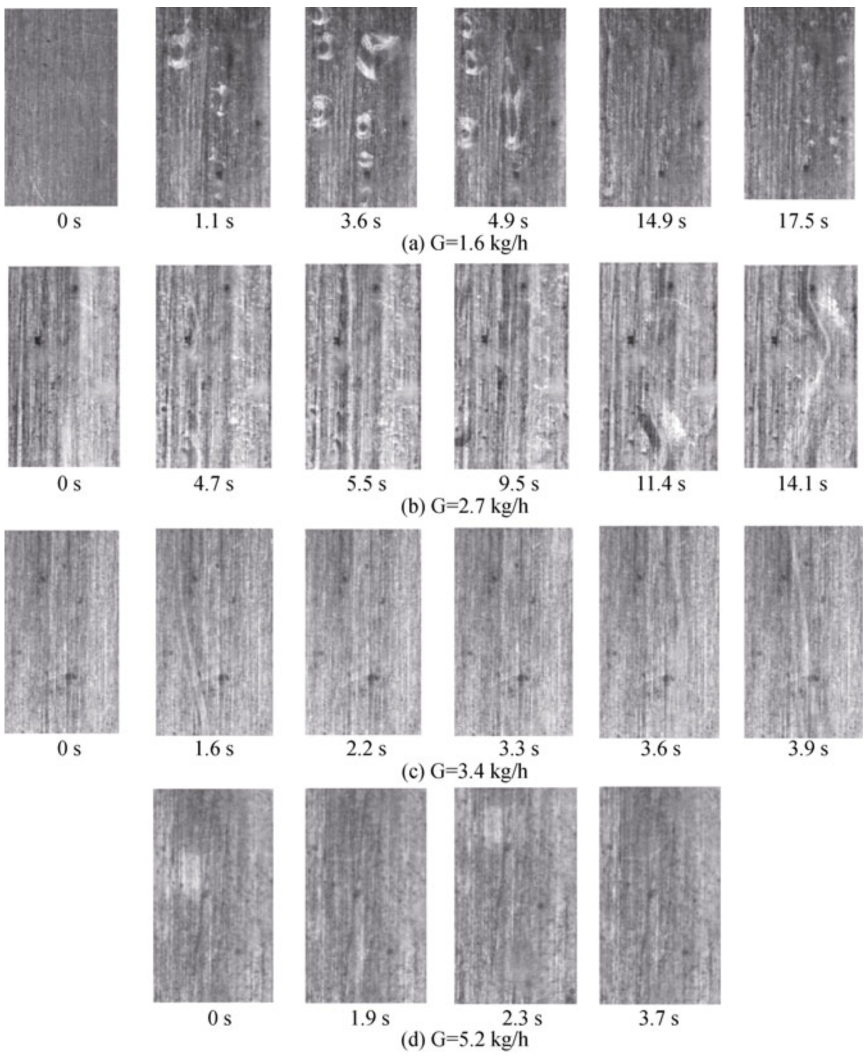


Fig. 5 Flow patterns at different mass fluxes (R410A at saturation 28℃)

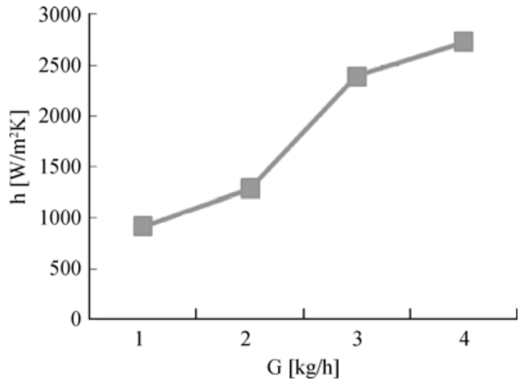


Fig. 6 Heat transfer coefficients for different mass fluxes

densation on the drop surface and drops merged in different region. Drops appeared once again after about 18 seconds. For the mass flux of 2.7kg/h, drops still can be periodically seen in the two sides of channel and formed more quickly. Rivulet flow played a main role in the central region, and flowed in the shape of snake affected by condensate in other region in a period. When mass flux increased to 3.4kg/h, slightly horizontal ripple appeared and the entire inner wall was covered by a thin liquid film all the time. At higher mass flux of 5.2kg/h, the wave flow occurred when the thickness of the liquid film increased: irregular waves continuously presented at the surface of liquid film. On the other hand, the time for the relative stability achievement of flow pattern cycle became shorter and shorter.

Heat transfer results

The mean condensing heat transfer coefficients of R410A were plotted over mass fluxes, as seen in Fig.6. Condensing heat transfer coefficient increased with mass fluxes. At low mass velocity (below or equal to 2.7kg/h), the heat transfer coefficient was independent of vapor shear force. As the mass velocity increasing, at first waves appeared in the flow pattern and it was suggested that vapor shear stress became significant. At mass flux of 5.2kg/h, condensate fluid layer became thicker which caused a lower rate of increase of heat transfer coefficient in spite of the apparent waves.

Conclusions

Condensation heat transfer experiment in a vertical rectangular channel at a relatively small mass flux was investigated. R410A at a saturation temperature of 28℃ was used. Flow patterns and heat transfer coefficients during condensation in the inlet region were observed. Some results were obtained as follows:

1. The distribution of the fluids in the channel was the result of a function of gravity and vapor shear stress. At low mass fluxes, the flow patterns were mainly dominated by gravity.

2. The effects of interfacial shear stress on condensate fluctuation were momentous for the film condensation with the increase of mass flux in vertical flow.

3. The condensation coefficients increased with the mass fluxes in the experimental conditions but at a lower rate which was the result of thicker fluid layer.

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