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AN EXPERIMENTAL STUDY ON EVAPORATIVE HEAT TRANSFER IN A MICROTUBE WITH R-134A

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ABSTRACT

This study reports the experimental results on the two-phase flow heat transfer of R-134a in a microtube. The test section is made of a stainless steel tube with an inner diameter of 792 μm and length of 10 cm which was heated directly by electricity. Local heat transfer coefficients are measured at two locations along the test section. The saturation temperature of refrigerant was calculated from the measured saturation pressure. Inner wall temperature was calculated from measured outer wall temperature, accounting for heat generation in the tube and heat conduction through the tube wall. Mass quality of refrigerant was calculated by considering the energy balance in the preheater and the test section. The heat transfer coefficient has been measured for the mass fluxes of 750, 950 and 1250 kg/m^{2}s, and heat fluxes of 19, 35 and 50 kW/m^2. It was found that the heat flux has a strong effect on the heat transfer in the low quality region, while the effect of mass flux is minor. Measured data were compared with Shah's correlation and Gungor and Winterton's correlation, which were higher than the predicted values by 9% and 25%, respectively.

NOMENCLATURE

Bo	Boiling number	q''	Heat flux
Co	Convection number	P	Pressure
Fr_{le}	Froude number for liquid phase	T	Temperature
G	Mass flux	x	Quality
H	Heat transfer coefficient		

INTRODUCTION

In recent years, MEMS (Micro-Electro-Mechanical Systems) technique has been attracting an attention and applied to many industrial fields, for example, biotechnology, medical implements, aerospace industry, electronic circuit design and so on. In addition to these, size and weight reduction of heat exchanger is possible in refrigeration and air-conditioning systems. One of the widely used components in thermal systems is a heat exchanger, and if size of the heat exchanger is reduced or the weight of the heat exchanger is lightened, there are several merits even in cost point of view. By using small diameter tubes in heat changers, relative surface area per unit volume is increasing, although the pressure drop increases and the heat change with secondary fluid is not always good. In spite of these, a new design of small heat exchangers is important as the small scale devices are constantly being developed.

Compact heat exchangers, which are composed of a bundle of small-diameter tubes in a general way, can have a higher surface area-to-volume ratio than conventional heat exchangers. Consequently, they have a merit to increase the heat exchanging efficiency as well. Shah (1986) defined a compact heat exchanger as a heat exchanger with the surface area density ratio of over 700 m^2/m^3.

So far, there have been many studies on evaporative heat transfer and two-phase flow in small diameter tubes. Yan and Lin (1998) measured the heat transfer coefficient and pressure drop for the evaporation of R-134a flowing in a small pipe of 2 mm in diameter. Their results are quite different from those for bigger pipes. They noted that to some degree the evaporation heat transfer in a small pipe is more effective than that in bigger pipes. Fletcher et al. (2000) performed an experiment on evaporative heat transfer in a circular tube with a diameter of 1.95 mm. They assert that heat transfer coefficient in a small-diameter tube is strongly affected by heat flux and system saturation pressure and correlations for large-diameter tube cannot properly predict heat transfer coefficients in a small-diameter tube. Cornwell et al. (2001) executed an experiment on flow evaporative heat transfer in a tube of 1.1 mm diameter. As a common result from many researchers, it can be said that heat transfer in a small-diameter tube is strongly influenced by heat flux. At very high heat flux over 60 kW/m^2, the heat transfer coefficient decreases as quality increases in this range, and it is suggested that nucleate boiling is dominant at these high heat flux condition. Peng and Wang (1993) studied on flow boiling heat transfer for liquid flowing through micro channel with a cross-sectional area of 0.6×0.7 mm^2. They concluded that the nucleate boiling is greatly intensified, and the wall superheat for boiling may be much smaller than that of conventional tubes.

Many studies on evaporative heat transfer were executed in a tube with the diameter of 1 mm or greater. In this study, heat transfer characteristics for the tube with the diameter less than 1mm are investigated. A stainless steel tube with the inner diameter of 792 μm was chosen as a test section and an experiment on evaporative heat transfer inside this tube has been carried out to provide basic data for performance estimation and design information for a compact heat exchanger.

EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental apparatus is schematically shown in Fig. 1. It has two loops of a refrigerant loop and a water-glycol loop. A refrigerant loop consists of a circulating pump for refrigerant, a mass flow meter, a preheater, a test section, a liquid reservoir, a main heat exchanger and an auxiliary heat exchanger. A water-glycol loop consists of a chiller, a constant temperature bath, a circulating pump for cooling fluid and a rotameter.

A pump used for circulating refrigerant is driven by a magnetic gear, which has an advantage to eliminate the effect of refrigerant oil on heat transfer. A by-pass line is located at the outlet of a magnetic gear pump, and the refrigerant, which comes from the outlet of a magnetic gear pump and flows through the by-pass line installed for the control of the flow rate in the main loop, goes to the inlet of the main heat exchanger. Two needle valves are installed at the upstream of the mass flow meter and on the by-pass line. The flow rate is delicately controlled with these two needle valves. A thermal mass flow meter was used in this study because the mass flow rate is so small ranging from about 0.3 to 0.6 g/s. The refrigerant from a mass flow meter goes through an auxiliary heat exchanger and goes into a preheater. The pre-heater is installed to adjust the inlet quality of the refrigerant to the desired value. Heat exchanger is made of double pipes of outer diameters of 9.5 mm and 6.3 mm, where refrigerant flows through the inner tube, and the water-glycol mixture is flowing counter currently through the annulus.

The test section used in the present study is schematically shown in Fig. 2. Stainless steel tube with the inner diameter of 792 μm and outer diameter of 1597 μm is selected as a test tube, which is 10 cm long for the test section. Both ends of the test section are connected to an AC power supply, which provides a heat generation inside the tube. This direct heating method has several merits to supply uniform heat flux in the test section and to vary experimental conditions. Inner surface roughness of the test tube measures 0.341 μm.

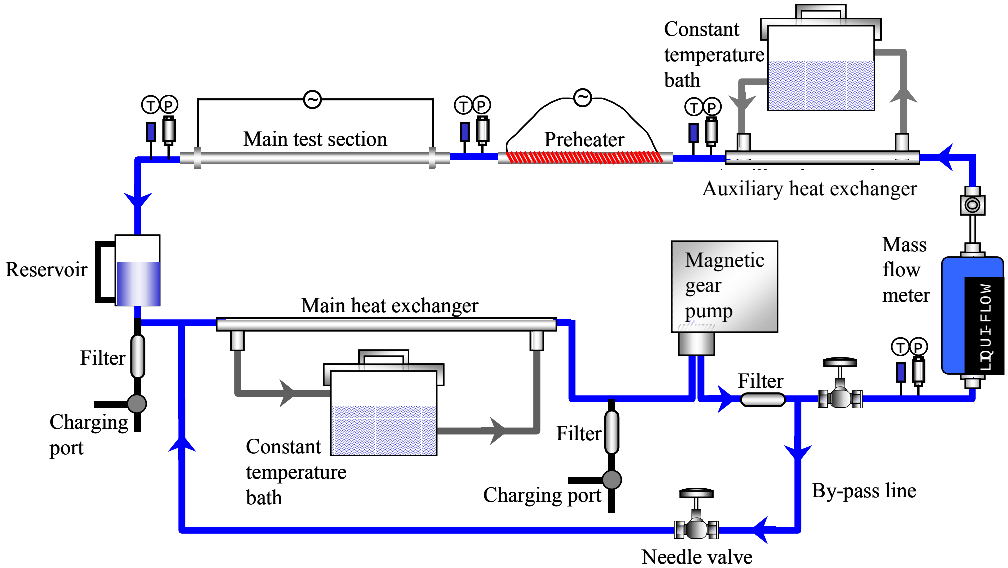


Figure 1: Schematic diagram of experimental apparatus for evaporative heat transfer test with a microtube.

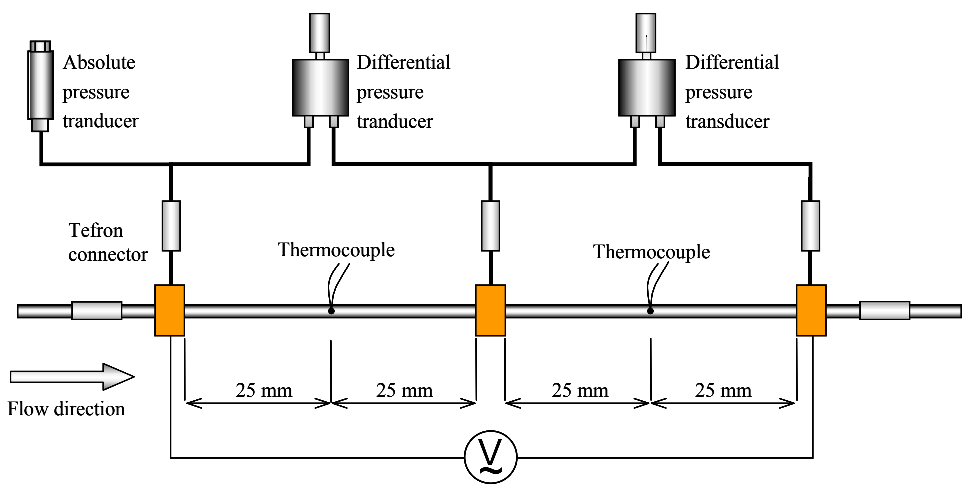


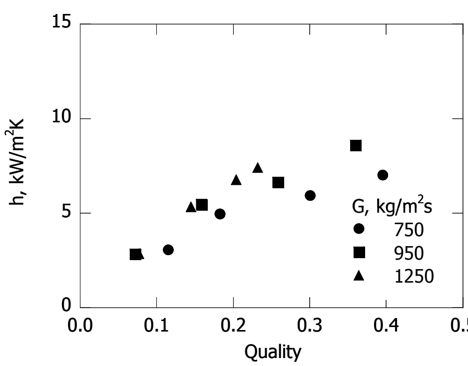
Figure 2: Representation of the measuring points in the test section.

To measure the heat transfer coefficient of refrigerant, saturation temperature of refrigerant, inner wall temperature and heat flux supplied to test tube should be known. For the measurement of outer wall temperature, the thermocouple is completely mounted to the outer wall of the test tube. With measured outer wall temperature, inner wall temperature is calculated by one dimensional heat conduction equation. Thermocouples used in this study are T type (copper-constantan) thermocouples. To obtain saturation temperature of refrigerant, saturated pressure of test tube was measured at three locations; inlet, outlet, and the middle position of the test section, and then the saturation temperature of refrigerant was calculated by REPPROP (McLinden et al., 1998). An AC power meter measures supplied heat from the preheater and AC power supply of test tube. Signals of temperature, pressure and mass flow rate are logged on data acquisition system and transferred to personal computer by GPIB communication.

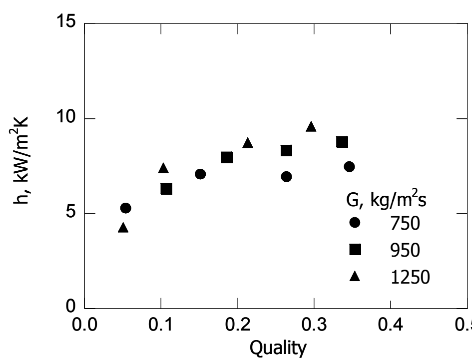
RESULTS AND DISCUSSION

Figure 3 shows that measured heat transfer coefficients at the mass fluxes of 750, 950, and 1250 kg/m^2 and at the different heat fluxes of 19, 35, and 50 kW/m^2. In the case of heat fluxes of 19 and 35 kW/m^2, heat transfer coefficient has a tendency to increase with increasing mass quality. This tendency is somewhat different from that for traditional tubes with the diameter of a few millimeters or greater. In a large diameter tube, heat transfer coefficient is so high at very low quality region due to the nucleation of bubbles. As the quality increases, the liquid film is formed and nucleation is suppressed, therefore the heat transfer coefficient decreases. When the quality increases more, mean velocity of refrigerant is increased and convective boiling heat transfer becomes dominant, resulting in the increase in heat transfer coefficient. But in microtubes, bubbles are confined by tube wall and the relative dimension of the bubble is comparable to the tube dimensions. Confined bubble is considered to make a liquid film circumferentially uniform. This phenomenon can be confirmed by the visualization test of Fukano and Kariyasaki (1993). As the refrigerant evaporates more, velocity of liquid film is accelerated due to increasing vapor ratio, resulting in an increase in heat transfer coefficients. In all cases of several heat fluxes of 19, 35 and 50 kW/m^2, heat transfer coefficients for increased mass fluxes are generally higher. This is because of the increased vapor speed and the thinner liquid film at high qualities. In the case of 50 kW/m^2, however, it can be observed that heat transfer coefficient is relatively independent of quality. This trend is because nucleation due to relatively high heat flux dominates over convection motion in microtubes.

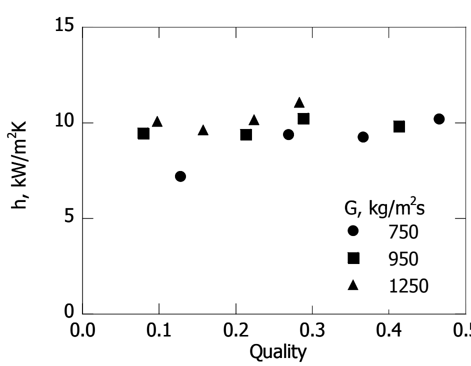
In Fig. 4, heat transfer coefficients are plotted as a function of quality for the mass flux values of 750, 950 and 1250 kg/m^2, respectively. It should be noted that the difference of heat transfer coefficients for each heat flux is great at the early stage of evaporation, but the difference is reduced as quality increases. This is because the nucleation is quite dominant at low quality region under high heat flux but as quality increases nucleation becomes suppressed and convective motion grows active.



(a) q¡È= 19 kW/m^2

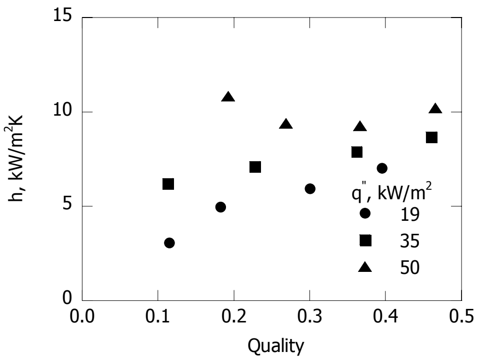


(b) q¡È= 35 kW/m^2

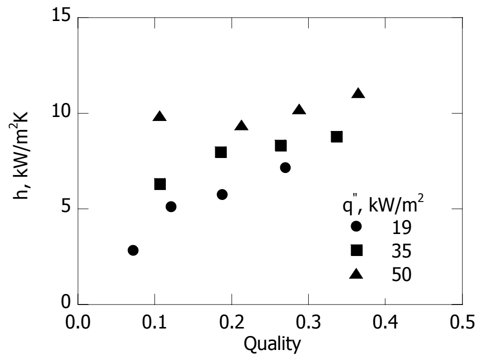


(c) q¡È= 50 kW/m^2

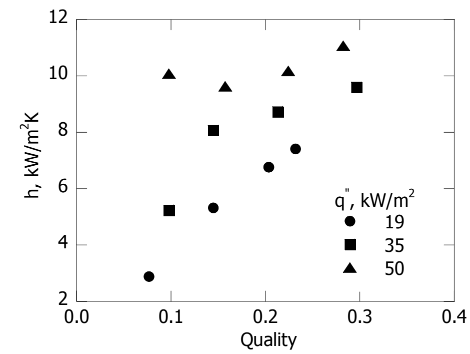
Figure 3: Variation of heat transfer coefficient with respect to mass flux at saturation pressure of 700 kPa.



(a) G = 750 kg/m²s



(b) G = 950 kg/m²s



(c) G = 1250 kg/m²s

Figure 4: Variation of heat transfer coefficient with respect to heat flux at saturation pressure of 700 kPa

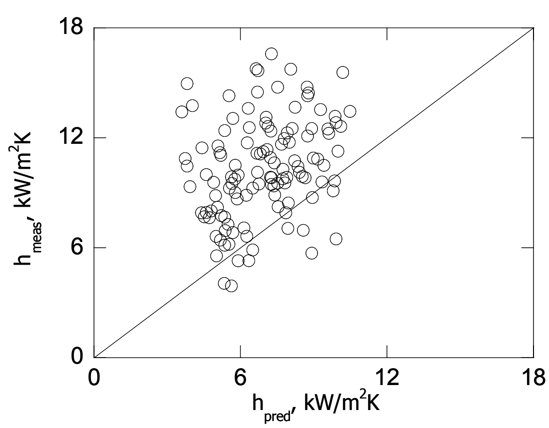


Figure 5: Comparison of measured heat transfer coefficients with Gungor and Winterton's correlation (1987).

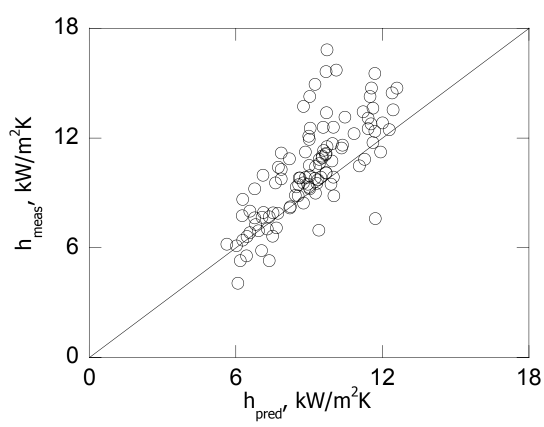


Figure 6: Comparison of measured heat transfer coefficients with Shah's correlation (1976)

Experimental data were compared with two evaporative heat transfer correlations. They are Shah's correlation and Gungor and Winterton's correlaion. In Shah's correlation, the ratio of evaporative heat transfer coefficient and single phase flow heat transfer coefficient is expressed as a function of convection number (Co), boiling number (Bo) and Froude number for liquid phase only (Fr_{le}). Single phase flow heat transfer coefficient is calculated by using Dittus-Boelter's correlation. Gungor and Winterton's correlation consists of enhancement factor (E) representing the effect of convective boiling increase and single phase heat transfer coefficient calculated by the Dittus-Boelter's correlation.

Figures 5 and 6 show the comparison results of measured data and predicted values by Gungor and Winterton's correlation and Shah's correlation. It was found that measured heat transfer coefficients were higher than the predicted values by 9% and 25%, respectively. As most of flow regimes are confined-bubble flow or annular-slug flow, liquid film at the tube-wall gets thinner and heat transfer is enhanced than that for large diameter tubes.

CONCLUSIONS

This study gives basic evaporative heat transfer information when the fluid is flowing and evaporating through the microtube. A seamless stainless steel tube with a diameter of 792 μm was used as a test section, and R-134a was used as a working fluid. The test section is 10 cm long and was heated directly by electricity. Local heat transfer coefficients are measured at two points along the tube. The heat transfer coefficient has been measured for mass fluxes of 750, 950 and 1250 kg/m^2s and heat fluxes of 19, 35 and 50 kW/m^2. Measured data were compared with two evaporative heat transfer correlations of Shah's correlation and Gungor and Winterton's correlation. It was found that measured heat transfer coefficients were higher than the predicted values by 9% and 25%, respectively. Evaporative heat transfer in a microtube is strongly affected by nucleate boiling at the early stage of evaporation under high heat flux, but as evaporation goes on, the effect of nucleate boiling is reduced. Evaporative heat transfer coefficient under the high heat flux of 50 kW/m^2 is relatively independent of quality due to strong effect of nucleation. In a microtube, bubbles generating from the tube wall make the liquid layer on the wall circumferentially uniform, and the vapor phase occupies most of the cross section. Therefore, the heat transfer is enhanced compared with that in conventional tubes. In designing a compact heat exchanger, it should be considered that the trend of evaporative heat transfer is somewhat different from the conventional tubes and evaporative heat transfer coefficient in microtube is strongly affected by heat flux.

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