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Yasuhiro Kudo
Saga university, Saga, Japan, 15575007@edu.cc.saga-u.ac.jp

Kyosuke Nakaiso
Saga university, Saga, Japan, 14575019@edu.cc.saga-u.ac.jp

Keishi Kariya
Saga university, Saga, Japan, kariya@me.saga-u.ac.jp

Akio Miyara
Saga university, Saga, Japan, miyara@me.saga-u.ac.jp

Experimental Study on boiling and condensation heat transfer in a horizontal mini channel

Yasuhiro KUDO[1], Kyosuke NAKAISO[1], Keishi KARIYA[2]* and Akio MIYARA[2]

[1] Graduate School of Science and Engineering, Saga University, 1 Honjomachi, Saga, 840-8502, Japan

[2]Department of Mechanical Engineering, Saga University, 1 Honjomachi, Saga, 840-8502, Japan
Phone: (81) 952 28 8623, E-mail : kariya@me.saga-u.ac.jp

ABSTRACT

A study of two-phase flow and heat transfer in a mini tube has been conducted experimentally. R134a has been used as the working fluid. This paper presents experimental results on boiling and condensation heat transfer coefficient in rectangular and rectangular grooved mini channels which hydraulic diameters of 0.81 and 0.64 mm, respectively. The boiling and condensation heat transfer coefficients were measured for saturated condition of R134a flowing in the horizontally placed mini channel tube. Length of the mini channel tube is 852 mm and it was heated or cooled by the water that flows through the upper and lower side of the mini channel tube. Wall temperature was measured by 24 thermocouples embedded in the tube surface. The boiling and condensation heat transfer in the mini channel has been measured over mass flux of 50,100,200 kg/m^{2}s respectively. The measured heat flux is 2 to 10 kw/m^2 for boiling and 1.5 to 9 kw/m^2 for condensation test.

1. INTRODUCTION

Nowadays, mini/micro channels are present in many applications ranging from different heat exchangers in the process industry, automotive, electronics and home applications. The ability of mini channel to provide high heat transfer coefficients, high efficiencies and system compactness are among the major advantages. However, two phase flow characteristic in the mini channels is different from conventional tube.

Enoki *et al.* (2013) conducted an experiment of vertical up flow and down flow in micro circular, rectangular, and triangle tube with about 1mm equivalent diameters by using R410a.They reported that the effect of the surface tension becomes dominant due to the diameter reduction, non-circularity of the heat transfer tube.

The characteristics and mechanism of boiling in mini channel are not completely understood yet. According to the published results, boiling heat transfer could be control by nucleate boiling, due to nearly exclusively dependency on heat flux Tran *et al.* (1996) or by convective boiling, with the dependence of mass flux and vapor quality Qu and Mudawar (2003) or by both, depending on vapor quality range, Yan and Lin (1998). Lee and Lee (2001) demonstrated that the boiling heat transfer coefficient grows with the heat flux and vapor quality, but the effect of heat flux on boiling heat transfer coefficient is small.

Jeong Seob Shin and Moo Hwan Kim (2005) experimentally studied flow condensation heat transfer inside circular and rectangular mini channel. They revealed that the influence of mass flux and vapor quality for all the test sections. The condensation Nusselt numbers increased with increasing vapor quality and mass flux due to the increase of higher vapor shear force. Also, the Nusselt number became more sensitive to the mass flux as the average vapor quality increased. Wang and Rose (2005) developed the heat transfer theory model during condensation in a horizontal micro channel with rectangular and triangular cross section. They revealed that the behavior of the general flow patterns, local average heat transfer coefficient, and the change in vapor quality.

The present paper aims to provide boiling and condensation heat transfer phenomena in rectangular and rectangular grooved mini channels. Moreover, the experimental results were compared with established mini channel correlations.

2. EXPERIMENTAL APPARATUS AND THE DATA REDUCTION METHOD

2.1 Experimental Facility and Instrumentation

An experimental facility was developed to investigate boiling and condensation heat transfer in horizontal mini channel. A schematic diagram of the test facility is shown in Fig. 1. The experimental system consists of a forced circulation loop. The main part of the loop has a pump, a pre heater, condenser and a test section. The refrigerant was circulated by the pump. Flow rate through the channel was controlled precisely by using a flow controller. The refrigerant then entered a pre heater and flowed through the test sections where the refrigerant heated or cooled. Preheater 1 is used for condensation and boiling test and preheater 2 is used for as well as condensation test. After passing though the accumulator it was directed to the cooler. Finally, refrigerant returned to the pump. Refrigerant temperature and pressure were measured at inlet of the preheater and at the inlet and outlet of the test section. Refrigerant R-134a were tested, Physical property of that refrigerant were determined by REFPROP Ver9.13. The test facility is fully automated through a PC using LabVIEW.

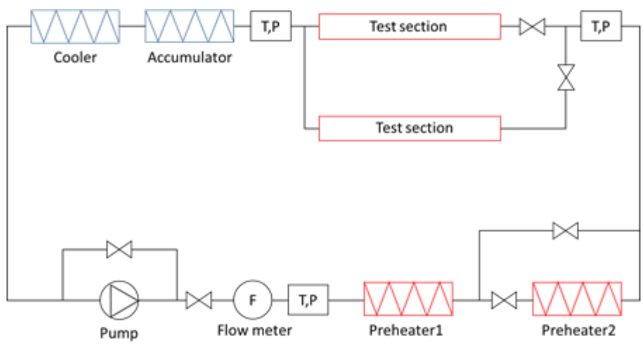


Fig.1 Experimental setup

2.2 Test Section

Fig.2 shows a schematic diagram of the test section. It consists basically of three sub sections. The wall temperature on the tube external surface is measured by twenty four T-type thermocouples. The thermocouples used for wall temperature measurement are directly fixed on the tube external wall with thermal conductive paste. The thermocouple is distributed in the tube wall as follows; the first thermocouple is located at 0.075m from the water outlet header, and all other thermocouples are located at 0.125, 0.175, 0.225, 0.325, 0.375, 0.425, 0.475, 0.575 m respectively. The test section is well shielded by using foam and tape.

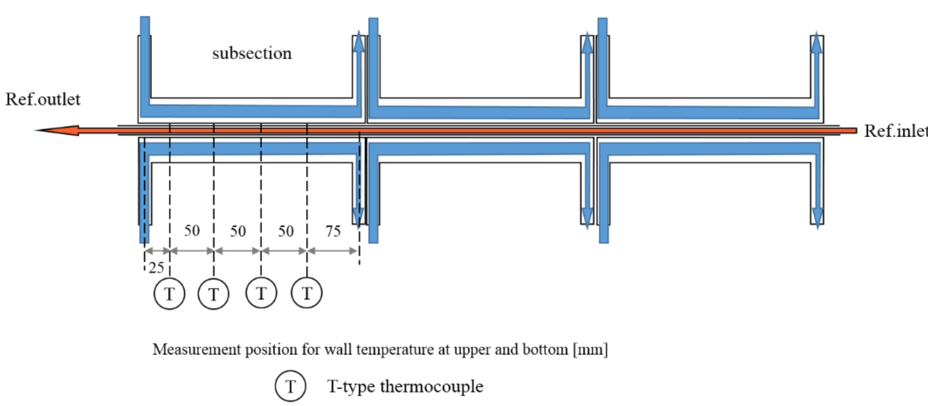


Fig.2 Test section

The test section is made from one piece of multiport tube with 20 channels as shown in Fig.3.The multiport rectangular and grooved tubes studied in this work have hydraulic diameters of 0.81 and 0.64 mm, respectively. In addition, the tube length of both test section is 0.75 m.

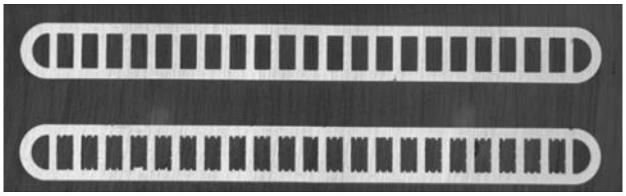


Fig.3 Mini-channels cross-sectional view
(Up: Rectangular tube Down: Grooved tube)

2.3 Data Reduction Method

Results from experimental data, including vapor quality, internal wall temperature, saturation temperature and the heat transfer coefficient, were calculated from measured data of refrigerant temperatures, wall temperatures in the test section, pressure and flow rate. The local heat transfer coefficient is defined as the following equation.

$$\alpha = \frac{q_{ref}}{T_{wall,x} - T_{sat}} \tag{1}$$

Where $T_{wall,x}$ is the tube internal wall temperature. It was calculated from the one dimensional heat conduction equation from the measured tube outer wall temperature. T_{sat} is the saturation temperature of the refrigerant; it was calculated from the temperature and pressure in the test section inlet or outlet mixing chamber. Local heat flux of the refrigerant q_{ref} was determined by the following equation.

$$q_{ref} = \frac{Q_{wat}\,HB}{LZ_x} \tag{2}$$

Where L is the wetted perimeter length and Z_x is the effective heat transfer length of the sub section. The quantity of heat exchange Q_{wat} and heat balance HB was determined by the following equation.

$$Q_{wat} = \dot{m}_{wat} \Delta h_{wat} \tag{3}$$

$$HB = Q_{ref} / Q_{wat} \tag{4}$$

Where, $\dot{m}_{wat}$ is the heat source water flow rate. Δh_{wat} is the variation in enthalpy of the heat source water side, was calculated based on the heat source water temperature difference between the atmospheric pressure. Q_{ref} is the amount of heat exchange in refrigerant in the test section which was calculated by test section inlet and outlet enthalpy difference. Q_{ref} is calculated as follows:

$$Q_{ref} = \dot{m}_{ref} \Delta h_{ref} \tag{5}$$

$\dot{m}_{ref}$ refers to the refrigerant flow rate.

The vapor quality in the test section was calculated from the following equation:

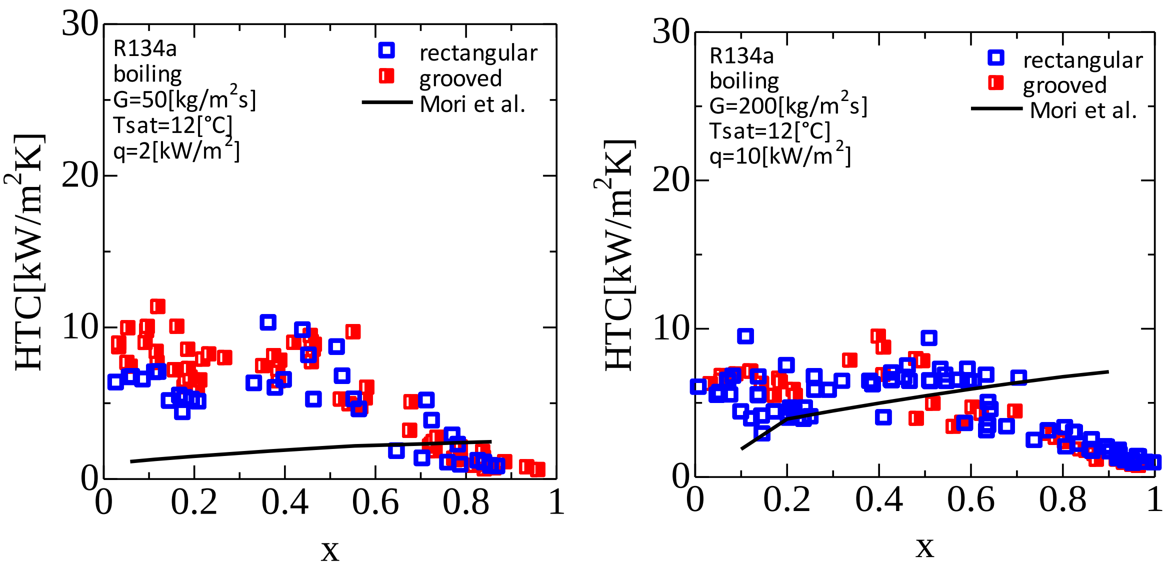
$$x = \frac{h_x - h_{liq}}{h_{vap} - h_{liq}} \tag{6}$$

The above expression for vapor quality h_x is calculated by the heat balance of the tube inlet or outlet enthalpy. h_{liq} is the value of the specific property of refrigerant saturated liquid state.

3. RESULT AND DISCUSSION

3.1 Boiling Test Result

Fig.4 shows the boiling test result of rectangular and grooved tube. It illustrates the heat transfer coefficient as a function of vapor quality at a fixed saturation temperature of 12°C.



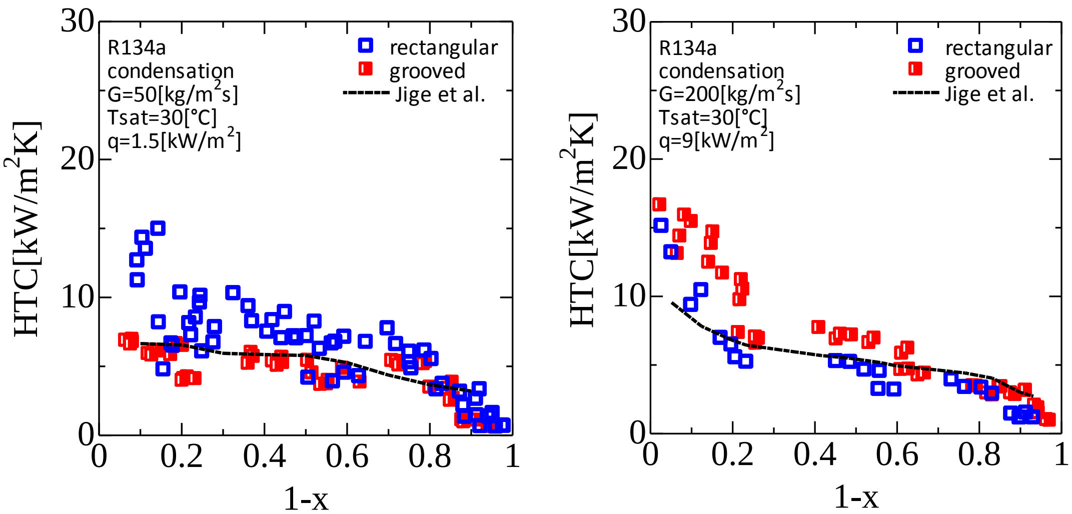
(a)Mass velocity G=50[kg/m²s] (b) Mass velocity G=200[kg/m²s]

Fig.4 HTC of boiling flow at saturation temperature 12°C

In Fig. 4(a), for the lower mass velocity (G=50 kg/m^{2}s) and heat flux (q=2 kw/m^2) the heat transfer coefficient in the rectangular grooved tube is found to be higher up to x=0.3, compared with rectangular tube. In the case of higher mass velocity (G=200kg/m^{2}s) and heat flux (q=10kw/m^{2}s), Fig.4 (b) shows that, the heat transfer coefficient in the rectangular and grooved tube almost same in all vapor quality ranges. The heat transfer coefficient data obtained from the present study are compared with one well-known [Mori *et al.*] heat transfer correlation. This result agrees with Mori *et al.* (2013) who conducted an experiment of a circular smooth tube with tube diameter d=8.4 mm. At G=50kg/m^{2}s and low quality, we can find out that the heat transfer coefficient of the grooved tube is higher than rectangular tube one and the heat transfer enhancement effect of the groove appears. Beside, in both tubes, we can confirm that the heat transfer coefficient decreased due to dry out from quality around 0.6, and the heat transfer coefficient showed same value in high quality range. On the other hand, in all quality range, the heat transfer coefficient showed same value at G=200kg/m^{2}s. The quality of the starting time of dry out is slightly higher in G=200kg/m^{2}s. In addition, in the comparison with Mori *et al.* correlation, heat transfer properties of the experimental value and the calculated value are totally different at both mass velocity conditions. The heat transfer coefficient of the experimental value shows almost constant regardless of changing quality before starting dry out, however, the heat transfer coefficient of calculated increased with increased quality. It means that we can consider the heat transfer properties in the non-circular min channels appear with the test heat transfer tubes.

3.2 Condensation Test Result

Fig.5 shows the heat transfer coefficient against the vapor quality of the condensation flow at saturation temperature of 30°C. In Fig.5 (a) and (b), shows the measured data on mass velocities of 50 [kg/m^{2}s] and 200 [kg/m^{2}s] , respectively.



(a)Mass velocity G=50[kg/m2s]   (b) Mass velocity G=200[kg/m2s]

Fig.5 HTC of condensation flow at saturation temperature 30°C

For low mass velocity G=50 [kg /m^{2}s], the heat transfer coefficient of rectangular tube is higher than that of grooved tube at lower vapor quality, merely 0.7 or less. However, in case of high mass velocity, G= 200 [kg/m^{2}s] the heat transfer coefficient of grooved tube is higher than that of rectangular tube at lower vapor qualities, merely 0.7 or less. The solid black lines in the both Figures HTC calculated using the correlations of Jige *et al* (2012). In less than wetness 0.7 at G=50kg/m2s, we can confirm that the heat transfer coefficient of the rectangular tube is higher than grooved one, and then we can find out that there are few effects of the groove at the low mass velocity and low wetness. However, with the high mass velocity of G=200kg/m^{2}s, the heat transfer coefficient of the grooved tube shows higher than rectangular one in less than wetness 0.7. we can consider that this is because liquid film is held by the corner part, and a liquid film is easy to be maintained in corner region of the groove due to the influence of groove by a surface tension with increased wetness. the heat transfer coefficient of the grooved tube shows the same value of rectangular tube one due to fill groove completely with refrigerant liquid at more than wetness 0.8. In addition, in the comparison with the correlations of Jige *et al*, the heat transfer coefficient of the grooved tube showed that it was higher than the calculated value at G=200 kg/m^{2}s and the range of low wetness. But, I assume the modification of the data reduction method is problem because the heat transfer coefficient was calculated by using a real cross area in this experiment.

4. CONCLUSION

In this study, we experimented boiling and condensation test by using rectangular and rectangular grooved mini channels with hydraulic diameters of 0.81 and 0.64 mm, respectively. Then we considered the heat transfer characteristic in these tubes from that results and then the followings are the results.

I. In boiling test, the heat transfer coefficient of the rectangular and rectangular grooved showed a same value in area with more than of quality 0.5 and same mass velocity.
The difference of cross sectional shape remarkably shows in low mass velocity.

II. In boiling test, compared with the Mori et al. equation, the heat transfer coefficient in mini channel tube is very different from circular tube one.

III. In condensation test, the heat transfer coefficient of rectangular grooved shows that high mass velocity one is higher than low mass velocity.
It means that the difference of the cross section influenced i.e. this is because the heat transfer enhancement effect with the groove is high in high mass velocity.

IV. In condensation test, compared with the Jige et al. equation, this equation can predict heat transfer coefficient well after low wetness area at G=200kg/m^{2}s

NOMENCLATURE

α	heat transfer coefficient	(kW/m^{2}K)
T	temperature	(K)
q	heat flux	(kW/m^2)
Q	heat exchanger rate	(kW)
HB	heat balance	(–)
L	wetted perimeter lenghth	(m)
Z	effective heat transfer length	(m)
$\dot{m}$	flow rate	(kg/s)
Δh	variation in enthalpy	(kJ/kg)
h	enthalpy	(kJ/kg)
G	mass velocity	(kg/m^{2}s)

Subscript

liq	liquid
ref	refrigerant
sat	saturation
vap	vapor
wat	water
wall	wall

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