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THE PRESSURE DROP AND HEAT TRANSFER OF FLOW IN ZIGZAG MICRO-CHANNEL WITH MICRO-ORIFICE

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ABSTRACT:

In this paper, the experimental investigation on friction factor and heat transfer of single phase liquid flow and two-phase boiling flow in single zigzag micro-channel with micro-orifice at inlet has been conducted. The dimension of the micro-orifice is 1mm×0.227mm×0.25mm. The experiment was conducted in copper rectangle zigzag micro-channel with the hydraulic diameter of 0.321mm and the length of 29 mm. The experimental results of friction factor and heat transfer coefficient were provided and the effect of the micro-orifice was discussed. It was found that the friction factor of flow in zigzag micro-channel deviated largely from the predictive value in the laminar flow, while it coincided well with the correlation for turbulent flow. In addition, the variation of local heat transfer coefficient showed that the inlet restrictor has significant effect on heat transfer of boiling flow in micro-channel.

INTRODUCTION

During the past few years, the investigation of flow in micro-channel has drawn worldwide attention for micro-channel heat sink features high efficiency and decreased size. The focus is on the hydro/thermodynamics performance of flow in micro-channel with a wide range of diameters. The highlight of this paper is to study the performance of pressure drop and heat transfer in zigzag micro-channel with micro-orifice entrance.

The utilization of zigzag micro-channel in heat sink for heat transfer enhancement of single-phase flow has been reported recently[1], while few papers in the public literatures were published concerning pressure drop and heat transfer performance of flow in zigzag micro-channel up to date. In fact, the zigzag channel at large scale was widely used in heat exchanger for heat transfer enhancement, for improving structure compactness. Many theoretical and experimental studies are available on pressure and heat transfer characteristics of flow in large sinusoidal wavy channels and zigzag channels. They demonstrated that the modification of channel surface could enlarge the heat transfer area and advance the production of the vortex by flow breaking and destabilizing in the twisting and turning of ducts with ribs, thereby intensifying the heat transfer[2]. In that case, it was reported that as much as 2-4 times higher heat transfer was achieved compared to a traditional channel[3].

The zigzag shape contributing to heat transfer enhancement in conventional channel triggered our curiosity to know the performance of zigzag surface of the micro-channel. However, the published research[2] showed that the vortex produced at the twists and turnings of channels could give rise to the temperature non-uniformity and flow oscillation along the large-scale channel. It may be more serious in the case of boiling flow in zigzag micro-channel for pressure and temperature oscillation induced by frequent generation and annihilation of the bubbles in constricted passage, which is deleterious and fatal to the boiling flow heat transfer in heat sink. At present, the fabrication of micro-orifice at inlet of the micro-channel is recommended as the effective measure to repress the instability of boiling flow at present. Currently, in the public literatures, most investigations on flow and heat transfer in micro-channel with micro-orifice studied the flow pattern on the basis of visualization technology, with the micro-orifice utilized as the cavitation trigger[4-6]. Rare data can be acquired with regard to the flow and heat transfer performance of flow through micro-orifice.

In this paper, our focus was on the pressure drop and heat transfer performance of flow in zigzag micro-channel with orifice ingrained at inlet. The working fluid was R134a. The flow in zigzag micro-channel was largely ranged in the turbulent region. The friction factor and heat transfer coefficient in zigzag micro-channel were achieved at high mass flux. And the effect of micro-orifice was discussed.

DEVICE FABRICATION AND EXPERIMENTAL RIG

Micro-channel Experimental Device Design

The height of the zigzag micro-channel was 0.25mm, which was machined in the middle of the copper plate with the thickness of 2mm. The single micro-channel was covered by another copper plate which sacrificed the convenience of observing the flow pattern in the micro-channel, but returned that it was more conforming to our research purpose. The flow in the micro-channel heat sink involves in heat transfer with walls of four sides while if it was covered by the glass, only three sides get involved. It was more close to the reality of heat exchange application. In current experiment, single micro-channel was fabricated. In that case, the flow mal-distribution which drastically influence the flow and heat transfer characteristics in the multiple micro-channels was avoided in current research.

On the back of the channel plate, several grooves have been fabricated to fix the thermocouples to investigate the wall surface temperature along the micro-channel. The thermocouple detecting heads were attached to the wall surface tightly aligning to the single micro-channel in the channel-plate. All the constantan-copper thermocouples were strictly calibrated. The inner wall temperature of the micro-channel was attained by correcting the data detected by the thermocouples. The schematic diagram of the zigzag micro-channel and the positions of thermocouples placed are shown in Fig.1. The whole micro-channel comprised of three parts: channel reservoir, inlet restrictor and zigzag micro-channel. The strategies to construct channel reservoir started from two reasons: one was it could be employed as the pre-heater to adjust the sub-cooling degree of flow in micro-channel; and another important factor was that it could be used as the pressure balancer, by which the dimensions were adjusted to balance the pressure gradients of different micro-channels, improving the flow mal-distribution in heat sink which is the serious case with a large number of parallel channels. The geometric dimensions of the zigzag channel are shown in Fig. 2.

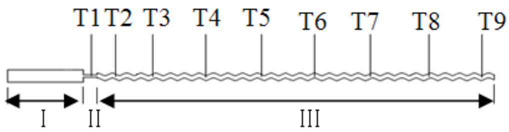

Fig. 1 Schematic diagram of zigzag micro-channel and the locations the thermocouples positioned

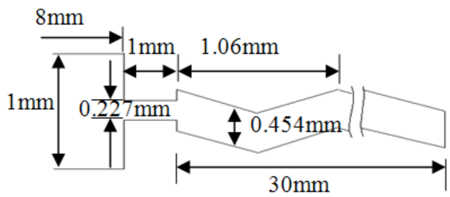

Fig. 2 Zigzag micro-channel size

Micro-channel Device Package Description

The micro-channel experimental device connecting two PTFE chambers which guided the fluid to flow through the micro-channel were assembled as the micro-channel device package, shown in Fig.3. Temperature detection port and pressure detection port placed in the front of each chamber were employed to install thermocouple and pressure sensor. The pressure and temperature detecting heads were located in the constricted passages ahead of and after the micro-channel to measure the fluid inlet and outlet temperature and pressure.

The heat source was simulated by plate heater powered by electricity which was attached below the channel plate. In order to minimize the thermal contact resistance, indium slice was unfolded placing between the heater plate and channel plate. As much as possible to reduce the heat dissipation, PVC-NBR was applied to wrap the entire micro-channel assembly for heat insulation. The heat loss was determined by the following method: firstly, the test liquid ran in the micro-channel. The heat loss was obtained by subtracting the heat absorbed by the test liquid from the total electricity input. Then, the heat loss versus temperature differences between the average surface temperature of the micro-channel and the ambient temperature could be correlated as Eq.(1). Then this correlation could be used to estimate the heat loss that the temperature differences between the average surface temperature of the micro-channel and the ambient temperature were known.

$$Q_{loss} = f(\overline{T} - T_{amb}) \tag{1}$$

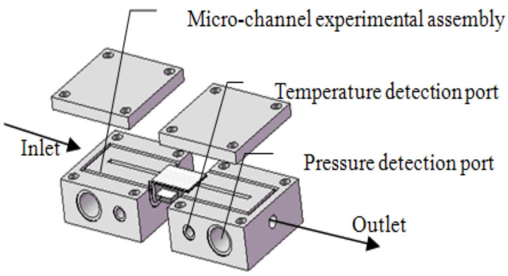

Fig. 3 Micro-channel device package

Experimental Rig

A schematic diagram of the experimental rig and apparatuses is shown in Fig.4. It was composed of the test circulation, a chilled water system, power provision devices and the measuring instrumentations.

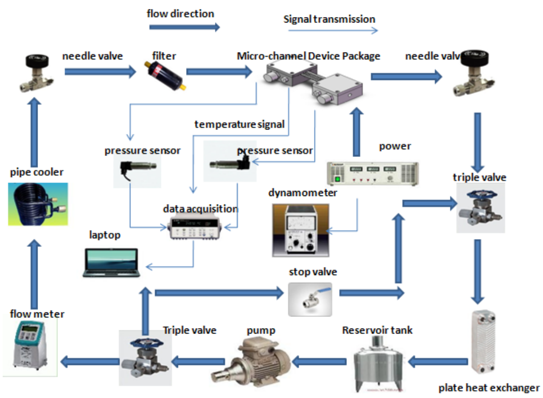

Fig. 4 Experimental rig and apparatus

The test circulation contains a double-pipe cooler, a filter, a micro-channel device package, plate heat exchanger, reservoir tank, some valves, Siemens Coriolis flow meter which is capable of measuring the small flow rate and magnetic gear pump for generating low fluid rate flow. The combined control of by-pass flow and needle valve is advantageous in flow adjustment. All cooling water was supplied by the chiller. Temperature, pressure and flow rate signals are acquired by data acquisition and are sent to the spreadsheet file for data reduction.

DATA REDUCTION AND UNCERTAINTIES ANALYSIS

Friction factor of zigzag micro-channel

The friction factor of zigzag micro-channel is one of the crucial parameter. It is expected to be obtained by the following method. A series of test runs were performed in single phase liquid flow. The pressure drop along the entire micro-channel can be presented in the form of three components.

$$\Delta p = \Delta p_1 + \Delta p_2 + \Delta p_3 \qquad (2)$$

where, Δp_1, Δp_2, Δp_3 is the pressure drop in micro-channel reservoir, inlet restrictor and zigzag micro-channel, respectively. For each component of pressure drop can be expressed in Eq.(3):

$$\Delta p_j = [k_{in} + k_{out} + f \cdot \frac{L}{D}]_j (\frac{1}{2\rho} G^2)_j$$

$$j=1,2,3 \qquad (3)$$

where k_{in} is the inlet pressure loss coefficient, k_{out} is the outlet pressure loss coefficient, and f is the Darcy friction factor.

For R134a in the micro-channel ran as the sub-cooling flow, the pressure loss due to flow contraction at inlet is given as follows[7]:

$$\Delta p = (1 - (\frac{A_2}{A_1})^2 + K_1) \cdot \frac{1}{2\rho} G_2^2 \qquad (4)$$

where K_1 is the non-recoverable loss coefficient given by

$$K_1 = 0.0088\alpha^2 - 0.1785\alpha + 1.6027 \qquad (5)$$

For two phase flow, it is calculated as[8]

$$\Delta p = \frac{G^2}{2\rho_L}[(\frac{1}{C_c} - 1)^2 + (1 - R_A^2)] \cdot [1 + (\frac{\rho_L}{\rho_v} - 1) \cdot x] \qquad (6)$$

where Cc is the contraction coefficient, determined from the following equation proposed by Chisholm [9].

$$C_c = \frac{1}{0.639 \cdot (1 - R_A)^{0.5} + 1} \qquad (7)$$

For single phase flow, the exit pressure loss due to flow expansion at the exit is given by[7]

$$\Delta p = -[2 \cdot 1.33 \frac{A_1}{A_2}(1 - (\frac{A_1}{A_2})] \cdot \frac{1}{2\rho} G_1^2 \qquad (8)$$

For two-phase flow, it is estimated by[8]

$$\Delta p = -\frac{G^2}{\rho_L} \cdot R_A \cdot (1 - R_A) \cdot [1 + (\frac{\rho_L}{\rho_v} - 1) \cdot x] \qquad (9)$$

therefore the pressure drop in zigzag micro-channel can be obtained by

$$\Delta p_3 = \Delta p - \sum_{j=1}^{2} (k_{in} + k_{out} + f_c \cdot \frac{L}{D})_j (\frac{1}{2\rho} G^2)_j \qquad (10)$$

As can be seen in Eq.(10), if the friction factor of plain micro-channel is known, the pressure drop in zigzag micro-channel can be obtained by subtracting each component of pressure drop from the total pressure drop. At present, many predictive methods are available to calculate the friction factor *of* plain micro-channel for a wide range of operating condition. And among them, Blasius correlation, showed in Eq. (11), was extensively applied and also verified by application of R134a flow in micro-channel. In this paper, it was employed to predict the friction factor in micro-channel reservoir and the inlet restrictor. Then the pressure drop in zigzag micro-channel is available and the friction factor in zigzag micro-channel can be deduced by the pressure drop.

$$\begin{aligned} &f_c = 96(1-1.3553\alpha + 1.9467\times\alpha^2 \\ &-1.7012\times\alpha^3 + 0.9564\times\alpha^4 - 0.2537\times\alpha^5)/Re, Re < 2000 \\ &f_c = 0.316/Re^{0.25}, 2000 \le Re < 20000 \\ &f_c = 0.184/Re^{0.2}, 20000 \le Re \end{aligned} \tag{11}$$

HEAT TRANSFER COEFFICIENT FOR ZIGZAG MICRO-CHANNEL

The sub-cooling flow entered in the micro-channel and was incepted to boil, consisting of two sections: sub-cooling section and boiling flow section. Single phase heat transfer is the principal heat exchanging form in sub-cooling section and Two-phase boiling heat transfer is the main form in boiling flow section.

Single phase heat transfer

The local heat transfer coefficient for single phase flow can be obtained by the following equation:

$$h = \frac{q}{T_{wi,z} - T_{l,z}} \tag{12}$$

and $T_{wi,,z}$ is micro-channel inner wall temperature of the place the thermocouple positioned. $T_{l,z}$ is the liquid temperature of the corresponding place which can be calculated by the heat balance Eq. (13).

$$T_{l,z} = T_{l,in} + \frac{\Phi_p q L_z}{m C_p} \tag{13}$$

Length of sub-cooling section

It has been mentioned that the R134a flow in the front section of the micro-channel is sub-cooling flow, followed with the boiling flow. The length of sub-cooling section which is determined by single phase pressure drop Eq. (14) and the saturated temperature Eq. (15) can be obtained from the following heat balance Eq. (16).

$$p_{sat} = p_{in} - \left(k_{in} + f\cdot\frac{L_{sub}}{D_h}\right)\cdot\frac{G^2}{2\rho} \tag{14}$$

$$T_{sat}=f(p_{sat}) \tag{15}$$

$$L_{sub} = \frac{GA\overline{C_{pl}}(T_{sat} - T_{in})}{qL_c} \tag{16}$$

The saturated temperature Eq. (15) is the relationship between saturated pressure and saturated temperature of R134a. It can be fitted by the existing pressure values and corresponding saturated temperature values.

The length of sub-cooling section can be attained by iterating these three Equations with the temperature error less than 10^{-6}.

Two-phase boiling heat transfer

In this study, the temperature was measured along the external surface of the zigzag micro-channel under the conditions of various electrical input power and flow flux. The electrical input power is determined by

$$P=U\times I \tag{17}$$

and U, I is input voltage and electric current, respectively. The inner wall temperature along the micro-channel can be calculated according to the temperature detected by the thermocouples:

$$T_{wi,z} = T_{wo,z} - \frac{(P - Q_{loss})\Delta x}{k_c A_c} \tag{18}$$

The temperature of the R134a in the boiling flow section of the micro-channel can be deduced by the Eq. (15).

In combination of solving Eq. (12)-(18), local heat transfer coefficient h of boiling flow can be obtained.

Uncertainties Analysis

The uncertainties of the experimental facilities installed to measure the temperature, pressure difference and mass flow rate are shown in Table 1. The uncertainties of the derived parameters including the heat flux, mass flux, Reynolds number, and heat transfer coefficient can be calculated using propagation analysis proposed by Moffat[10].

RESULTS AND DISCUSSION

Friction factor in zigzag micro-channel

The pressure drop in inlet and outlet manifolds of multiple micro-channel heat sink is of significant influence to the flow and heat transfer performance, and unreasonable fabrication will result in flow mal-distribution. In our research, the inlet reservoir was fabricated properly for the purpose of strictly uniform flow distribution. The dimension of the inlet reservoir was 8mm×1mm×0.25mm. The pressure drop in it could be estimated by Eq. (3) and Eq. (11). In the inlet restrictor, the pressure drop gradient goes sharp down slope due to the sudden contraction in flow area. The pressure loss and recover could not be ignored and they were calculated by equations (3)-(11). In the experiments, the measured pressure data were corrected by subtracting the pressure loss and recovery due to the area contraction and expansion as well as pressure drop in micro-channel reservoir and restrictor, then the pressure drop in zigzag micro-channel was acquired.

The friction factor values versus Reynolds numbers in the zigzag micro-channel are plotted in Fig. 5. The Reynolds number is varying from 1405 to 9952. It can be seen that with the increase of the Reynolds number, the friction factor goes down almost with linear tendency, which indicates that the flow is in the laminar region. The following trend varies moderately with further increase of Reynolds number in the turbulent flow. According to the Eq. (11), the theoretical data is also plotted for comparison. It is obvious that the bendings and turnings of zigzag micro-channel exert larger effect on pressure drop in laminar flow compared with that in turbulent flow. Specifically the friction factor of laminar flow in zigzag micro-channel is nearly three times larger than predictive value, while in the turbulent region, the experimental value can match well with the theoretical value by large. It is suggested that the production of vortexes in the laminar flow disturb the flow uniformity in the turnings and enhance the local friction factor. And the effects of the vortexes on the turbulent flow become weaker with the increase of the Reynolds number. All of them can be reflected from the curve in Fig.5.

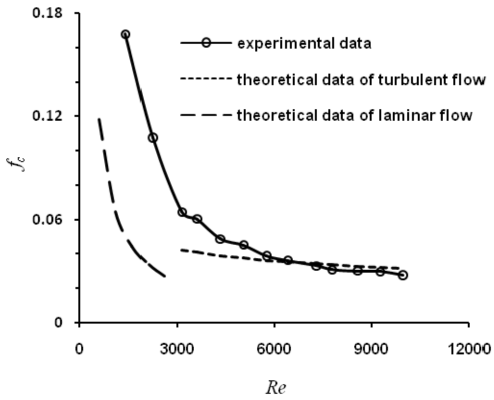

Fig.5 The fricton factor versus Reynolds number

Single phase heat transfer

It is generally recognized that the configuration of restrictor at the inlet of the micro-channel is beneficial to the stability of boiling flow. It's quite necessary to investigate the heat transfer characteristics of single phase flow through the narrow orifice due to it is widely existed in real application, while the data in the public literatures is rare. In current research, the single phase heat transfer coefficient was gained along the axial positions of the zigzag micro-channel and it could be explained by comparison with the visualization investigation. The experimental results are shown in Fig. 6. It can be seen that the heat transfer coefficient varies up and down mainly controlled by the micro-orifice structure.

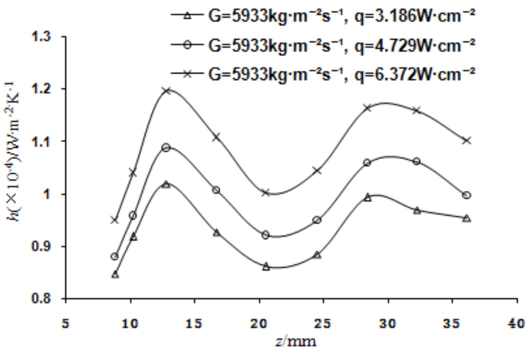

Fig.6. Heat transfer coefficient of single phase

The pressure goes down sharply in the restrictor due to the area contraction, and the flow velocity in it rises up, correspondingly. The lost pressure energy is mainly consumed by local viscous dissipation and local resistance. The static pressure recovers downstream of the orifice as the submerged jet expands and reattaches with the channel flow boundaries. The velocity increases and returns to normal correspondingly with the area contraction and expansion. Their tendency is shown in Fig.7. The first thermocouple was placed on the back surface of the restrictor. The flow velocity is high there; however, the heat transfer coefficient is estimated to be the lowest, which results from the formation of twin gas pocket. The liquid detaches from the wall there, shown in Fig.8, and therefore the local temperature soars, which has been verified by experimental investigations[11] and simulation studies[12]. The velocity at the site (point 2), shown in Fig.7, rises to its maximum and the fluid static pressure correspondingly drops to its minimum. If the pressure is sufficiently low, the bubble nuclei are incepted in the area of Ⅱ(Fig. 9). The vapor bubbles are emerging[5] and the heat transfer is enhanced initially. As the pressure turns back to normal, the vapor bubbles disappears gradually, however the submerged jet stretches downstream of the micro-channel, therefore the heat transfer is abated attributing to the separation of liquid from the channel wall in the Ⅲ section. After the liquid jet shooting reattaches the wall in the area of Ⅳ, the heat transfer coefficient is raised, which goes down later because of the development of the boundary layer. The variation tendency is reflected by the heat transfer coefficient in Fig.6. Compared with the three lines with the same mass flux, it's worth noting that with the increase of the heat flux, the heat transfer was intensified though the surface temperature along the micro-channel rose. This result is the same with that from Schneider[13], which was explained with the angle of the reduction of the cavitation number for flow in the entire region. To our perspective, the bubble nuclei became unstable with the increase of the heat flux, more of which were triggered, and addition nuclei were incepted, therefore heat transfer enhancement was acquired.

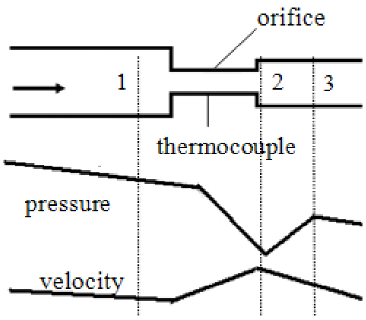

Fig. 7 Pressure and velocity variation tendency

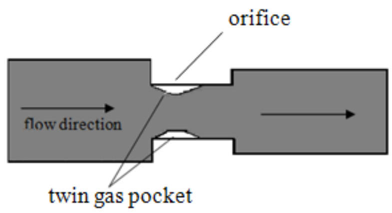

Fig. 8 The formation of twin gas pocket[5]

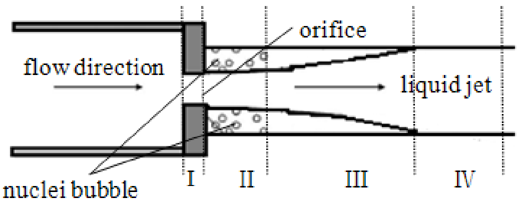

Fig. 9 The liquid jet and incepted bubble

Heat transfer of boiling flow

The heat transfer of boiling flow in micro-channel with orifice has been investigated based on visualization by many researches. But, none of them have studied the local heat transfer. The heat transfer coefficients along the zigzag micro-channel are plotted in Fig.10. It is quite easy to distinguish the boiling inception position and the existence of cavitation effect which has been explained in the literatures[4-6]. For the curve of mass flux is 5967 kg·m^{-2}·s^{-1}and heat flux is 15.93W·cm^{-2}, the boiling occurs near the position 5. Before it, the flow runs as the single phase. The heat transfer is strengthened at position 3 owing to the presence of the stream nuclei and incepted bubbles. For heat flux is 19.12W·cm^{-2}, the boiling occurs after the position 3 and it occurs before the position 3 for heat flux is 22.3W·cm^{-2}. At the position 3, the heat transfer coefficient reaches the largest. It can be explained by the following sketch of flow pattern which was described in the literature[13]. If the heat flux is high enough, the liquid ejected from the inlet restrictor is evaporated and gradually becomes vapor[13]. Meanwhile the nuclei are incepted under the low pressure condition. Therefore the heat transfer is augmented by these dual effects. The generated vapor is shot separately into twin vapor bubbles and elongated cavity which covers on the zigzag micro-channel wall and results in the weakened heat transfer. The heat transfer coefficient rises up at the position 7, shown Fig.10, which suggests that the elongated cavity collapses and the vapor slug sheds into the small bubbles downstream of the zigzag micro-channel.

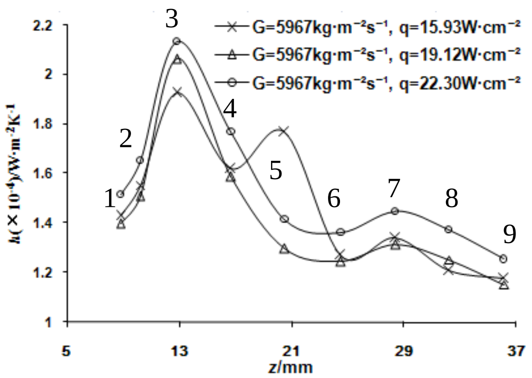

Fig.10 Heat transfer of boiling flow in zigzag micro-channel

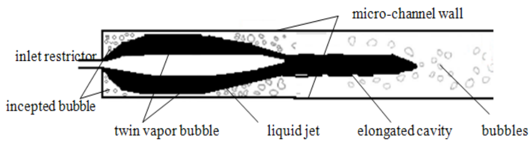

Fig.11 The schematic diagram of flow pattern[13]

The heat transfer mechanism of R134a boiling flow in micro-channel has been investigated by many researches. They found that different boiling mechanism, nucleate boiling, forced convective evaporation, or both of them dominate the heat transfer. Among of their investigations, the mass flux was small and flow in micro-channel was laminar. In current experiment, the mass flux was large and the flow in zigzag micro-channel was in the turbulent region. Compared the curve (q=15.93W·cm^{-2}) with another curve (q=19.12W·cm^{-2}), it can be seen that the heat transfer is weakly dependent on heat flux, which means nucleate boiling suppression appears earlier and the convective heat transfer dominates the boiling region. This is identical with the results from former researches[5, 14]. Which convective mode, boiling convective or single-phase convection is responsible for the heat transfer enhancement is difficult to identify, since the heat transfer enhancement was derived from the phase change. That the heat transfer enhanced by the cavitation under low pressure or boiling evaporation dominates the heat transfer is indistinct.

convective mode is ambiguous and difficult to determine.

CONCLUSION

The present experiments on flow of R134a in zigzag micro-channel with the orifice upstream have been conducted. The friction factor and local heat transfer coefficient have been attained. The results were compared with the visualization investigation. The following conclusions can be made.

(1) In the laminar region, as the result of the vortex generated at the turnings, the friction factor of flow in zigzag micro-channel was larger than the predictive values. While in the turbulent region, the vortex exerted insignificant disturbance on the turbulent flow. The regular correlation can be used to predict the friction factor of flow in zigzag micro-channel.

(2) The inlet restrictor exerted importance effect on the single phase and boiling flow heat transfer. The existence of the orifice in the single phase flow contributed to the bubble inception and the evaporation boost in the boiling flow. The heat transfer is enhanced to some extent as the result of the configuration of inlet restrictor at inlet.

(3) The local heat transfer coefficient of flow in zigzag micro-channel is attained and it showed a well tendency with (or in accordance with) the visualization results. For the size of the micro-channel was so small, it was difficult to machine the same size zigzag micro-channel and normal plain micro-channel, therefore it was hard to draw any quantitative conclusions regarding the capabilities of zigzag element in this paper. Further investigation has to be conducted and more experiments have to be done to confirm the heat transfer enhancement of flow in zigzag micro-channel.

(4) Nucleate boiling was rare to be observed in current experiments and convective heat transfer dominated the entire region, while the

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NOMENCLATURE

A	cross section area/m ^{2}
C_c	contraction coefficient
C_p	specific heat/J·kg^{-1}·K^{-1}
D_h	hydraulic diameter/m
f_c	Darcy friction factor
G	mass flux/kg·m^{-2}·s^{-1}
h	heat transfer coefficient /W·m$^{-2\cdot}$ K^{-1}
h_{fg}	latent heat of vaporization/J·kg^{-1}
I	electrical current /A
L	length/m
L_c	micro-channel wetted perimeter/m
K	non-recoverable loss coefficient
k	loss coefficient
k_c	copper heat conductivity/W·m^{-1}·K^{-1}
P	electrical power/W
p	pressure/Pa
Δp	pressure difference/Pa
Q_{loss}	heat loss/W
R_A	cross sectional area ratio
Re	Reynolds number
T	temperature/K
$\overline{T}$	average temperature/K
T_w	wall temperature/K
U	voltage/V
z	axial length along the micro-channel/m

Greek symbols

a	channel aspect ratio
μ	viscosity/kg·m^{-1}·s^{-1}
ρ	density/kg·m-^{3}

Subscripts

a	acceleration pressure drop
amb	ambient
c	critical point
cs	copper plate surface
i	inner surface
exp	experimental
in	micro-channel inlet
G	gas
j	number
l	liquid
out	micro-channel outlet
pre	predicted
r	ratio
sub	sub-cooling flow
sat	saturated
TP	two-phase
w	wall

APPENDIX

Table 1 Summary of uncertainty

Parameter	Uncertainty	Parameter	Uncertainty
micro-channel geometry			
width, thickness, length	0.5%	Area	0.7%
The measured parameters			
Temperature	±0.1°C	Pressure difference	±0.35%
Mass flow rate	±0.25%	Heater power	±5%
The derived parameters			
heat transfer coefficient	±5.1%	heat flux	±13.87%
friction pressure factor	±1.64%	Mass flux	±0.74%