An Experimental Investigation on Condensation Heat Transfer of Microchannel Heat Exchangers

Thanhtrung Dang and Minhhung Doan
Department of Heat and Refrigeration Technology, Ho Chi Minh City University of Technical Education, Ho Chi Minh City, Vietnam

ABSTRACT:
The present study investigated experimentally condensation heat transfer of two microchannel heat exchangers. The heat transfer rate of a microchannel heat exchanger was achieved 272.9 W for the vapor having the inlet temperature of 101 ºC and the mass flow rate of 0.123 g/s and for the cooling water having the inlet temperature of 32ºC and mass flow rate of 3.1133 g/s. It was also observed that the heat transfer rate obtained from the counter flow arrangement is always higher than that obtained from the parallel one: the value obtained from the counter flow arrangement is 1.04 to 1.05 times of that obtained from the parallel flow. The results for two phases are in good agreement with the results for single phase. In addition, the condensation heat transfer coefficient in the microchannel heat exchangers decreases as increasing the inlet cooling water temperature.

KEYWORDS: *Microchannel, condensation, configuration, heat exchanger, efficiency.*

NOMENCLATURE

h	entalpy	kJ/kg
m	mass flow rate	kg/s
Q	heat transfer rate	W
T	temperature	ºC
Subscripts		
i	*inlet*	
o	*outlet*	
v	vapor	
w	water	
Greek symbols		
η	efficiency	

I. INTRODUCTION

The use of microchannel heat exchangers can bring advantages such as enhanced heat transfer coefficient and decreased characteristic dimension. The use is found in chemical, medical areas and in micro electro-mechanical systems (MEMs). In relevant studies of microchannel heat transfer, Hu and Chao [1] studied five condensation flow patterns for water in silicon micro condenser. For a hydraulic diameter of 73 μm and the range of mass fluxes of steam from 5 to 45 kg/(m^{2}s), the heat transfer coefficients obtained were from 220 to 2400 W/(m^{2}K) and the corresponding pressure drops were from 100 to 750 kPa/m. In the study, the maximum heat flux achieved was 40 kW/m^2. The study indicated that cooling capacity and the coefficient of performance (COP) of the microchannel condenser were higher than those of the conventional condenser under the same operating condition. In addition, the amount of fluids used in the heat exchanger systems with microchannel condensers was smaller than that with the conventional one. Two R410A residential air-conditioning systems, one with a microchannel condenser and the other with a round-tube condenser, were studied by Park and Hrnjak [2]. The cooling capacity and the coefficient of performance (COP) of the system with the microchannel condenser were 3.4% and 13.1% higher, respectively, than those obtained with the round condenser. The amount of working fluids charged for the system with the microchannel condenser was 9.2% smaller than that with the round-tube condenser. The numerical model also used to calculate for the non-uniform distribution of the air and refrigerant. The numerical results did not make a significant difference for predicting the condenser capacity. Numerical model for microchannel condensers and gas coolers was investigated by Martínez-Ballester et al. [3].

The main conclusion of this study is that it is possible to take into account the heat conduction between tubes in a more fundamental way than other fin efficiency based approaches, which have to apply heat conduction terms to an approach that uses the adiabatic-fin-tip assumption, which is not satisfied in such cases. However, this study just indicated in description and validation state only. Cho et al. [4] studied the fabrication of the micro condenser tube by direct extrusion. The materials used to manufacture for this micro condenser are A1100 and A3003 aluminum alloy. Because A3003 alloy can be durable when it used in a heat exchanger that requires high working pressure, so A3003 alloy is more suitable than A1100 alloy for the condenser tube with eco-friendly refrigerant usage. A comparison of condensation heat transfer and pressure drop of CO_2 in rectangular microchannels was studied by Heo et al. [5]. The condensation heat transfer coefficients of CO_2 in rectangular microchannels with 7, 23, and 19 ports were experimentally investigated with the variation of the mass flux. The condensation heat transfer coefficient increased with the decrease in hydraulic diameter, which is dominant at condensation temperatures of 0 and -5 °C. Increasing or decreasing the heat transfer coefficient at critical vapor quality was observed for the microchannel of 23 ports. The existing models for the prediction of heat transfer coefficients over-predicted the experimental data and the model of relevant literature showed the smallest deviation of 47.7%. A study on condensation of compact heat exchanger was done by García-Cascales et al. [6]. In this study, an iterative algorithm has been suggested for the study of two-row heat exchangers. A brief state-of-the-art study has been done for condensation in micro- and mini-channels and some remarkable correlations introduced in the existing literature have been described. They have also been compared under typical operating conditions. Several heat exchangers working as a condenser or an evaporator have been studied experimentally and the results obtained have been compared with those provided by the model. Hrnjak and Litch [7] presented a study of two aluminum condensers: a microchannel condenser (a parallel tube arrangement between headers and microchannel tubes, having a hydraulic diameter D_h = 0.7 mm) and another one (with single serpentine macrochannel tube having a hydraulic diameter D_h = 4.06 mm). The ratio between amount of working fluids charged and capacity of the microchannel condenser was around 76% less than for the serpentine condenser.

Dang and Teng [8, 9] studied the effects of configurations on performance of the microchannel and minichannel heat exchangers. Effect of flow arrangement on the heat transfer related behaviors of a microchannel heat exchanger was presented by Dang and Teng [10]. For all cases done in the study, the heat flux and performance index obtained from the counter-flow arrangement are always higher than those obtained from the parallel-flow one: the values obtained from the counter-flow are 1.1 to 1.2 times of those obtained from the parallel-flow. However, in [8-10] they studied on microchannel heat transfer on single phase. Goss and Bassoc [11] studied heat transfer during the condensation of R134a inside eight parallel microchannels. The study indicated that there is no clear influence of the fluid saturation temperature and the heat flux removed on the heat transfer coefficient for the range tested. On the other hand, the heat transfer coefficient value increases with rising the mass velocity. It is also demonstrated that the consideration that all of the heat transfer resistance is due to the heat conduction through the liquid film is a good approximation. The interfacial and disjoining pressure resistances can be neglected even in the higher quality range. From the above literatures, it is important to study the efficiency of condensation heat transfer in microchannel heat exchangers. For the present study, two microchannel heat exchangers with different geometrical configurations will be discussed.

II. METHODOLOGY

The experimental system consists of four major components: the test sections (the microchannel heat exchangers), pump system, steam boiler, and overall testing loop, as shown in Fig. 1. The vapor produced in the boiler passes through the micro condenser, where it condenses by the cooling water. Two microchannel heat exchangers were designed and tested in this study. Figure 2 shows the dimensions of the test sections. The material for the heat exchangers is aluminum, used as a substrate with thermal conductivity of 237 W/(mK), density of 2,700 kg/m^3, and specific heat at constant pressure of 904 J/(kg K). At each inlet or outlet, a temperature sensor was set to record temperature values. The inlet and outlet water temperatures were measured using T-type thermocouples. These thermocouples were inserted into the tubes at the inlets and the outlets. So, there were total four temperature sensors were used to record data.

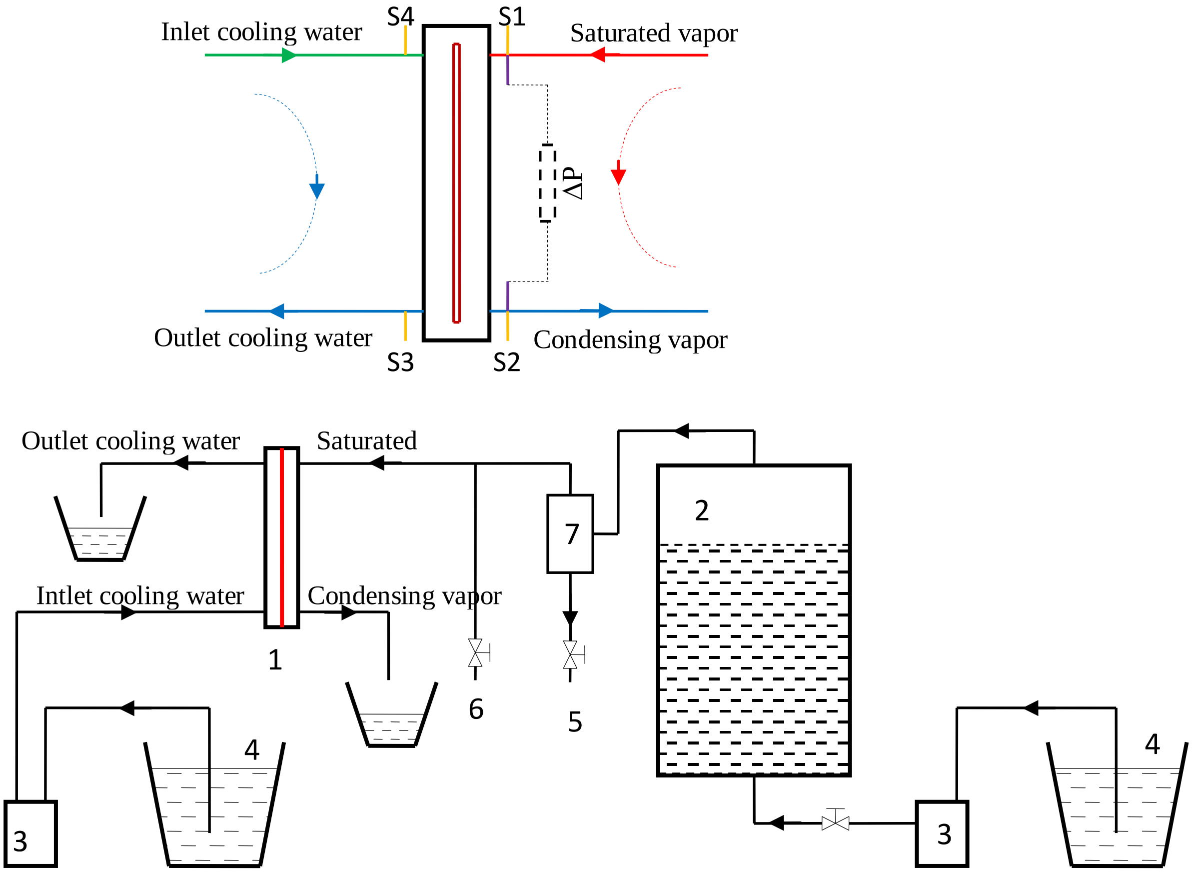

Fig. 1. Schematic of the test loop

1. Microchannel heat exchanger;
2. Electric mini Boiler;
3. Mini pump;
4. Water tank;
5. Condensing vapor valve;
6. Adjusting valve;
7. Buffer tank

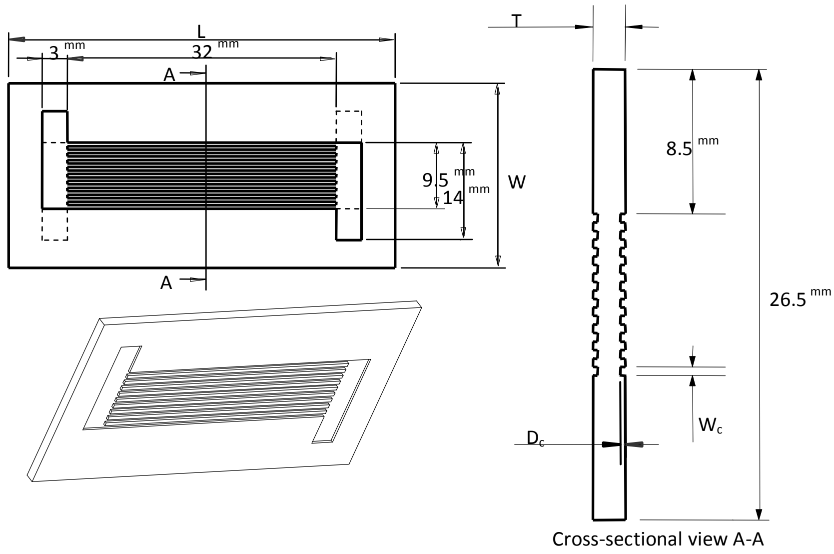

Fig. 2. Dimensions of the test section

Basically, the two microchannel heat exchangers (T1 and T2) have the same dimensions; however, the substrate thickness of T1 is 1.2 mm while T2 is 1 mm and the channel depth for the cooling water side of T1 channel is 300 μm (having hydraulic diameter of 375 μm) while T2 is 180 μm (having hydraulic diameter of 265 μm). Geometric parameters of the microchannel heat exchangers are listed in Table 1.

Table 1. Geometric parameters of the microchannel heat exchangers

No.	Dimensions of the substrate (mm)			Dimensions of the channel (μm)			
				Vapor side		Cooling water side	
	L	W	T	W_c	D_c	W_c	D_c
T1	46	26.5	1.2	500	300	500	300
T2	46	26.5	1	500	300	500	180

All channels are connected with a manifold for each inlet or outlet of vapor and cooling water, respectively. The manifolds have a rectangular shape with the width of 3 mm and the depth of 300 μm. Figure 3 shows a photo of the microchannel heat exchanger T1. These test sections of the heat exchangers were manufactured by precision micromachining [8-10]. Each inlet or outlet of the heat exchangers has cross-sectional area of 9 mm^2. The four sides of the heat exchanger were thermally insulated by the glass wool with a thickness of 5 mm. To seal the microchannels, two layers of PMMA (polymethyl methacrylate) are bonded on the top and bottom sides of the substrate, as indicated in Fig. 3. The physical properties of the PMMA and the glass wool are listed in Table 2. Accuracies and ranges of testing apparatus are listed in Table 3.

Fig. 3. A photo of the test samples

Table 2. Physical properties of PMMA and glass wool

Material	Density kg/m^3	Thermal conductivity W/(mK)
PMMA	1420	0.19
Glass wool	154	0.051

Equipments used for the experiments are listed as follows:
[1] Thermocouples, Model PT-100, made by Omega
[2] Mini boiler, Model EMB-S-9, made by ELectromn
[3] Pump, VSP-1200, made by Tokyo Rikakikai
[4] Differential pressure transducer, Model UNIK5000, made by GE Druck
[5] Micro electronic balance, Model TP - 214, made by Denver.

Table 3. Accuracies and ranges of testing apparatuses

Testing apparatus	Accuracy	Range
Thermocouples	± 0.1 °C	0 ~100 °C
Differential pressure transducer	±0.04% FS	0 ~1 bar
Precision balance	± 0.0015 g	0.0000 ~ 210g

III. RESULTS AND DISCUSSION

For the experiments carried out in this study, the condensation heat transfer of the heat exchangers is discussed as follows.

The energy balance equation is calculated by

$$Q_v \eta = Q_w \quad (1)$$

$$\text{Or } m_v(h_{vi} - h_{vo}) \eta = m_w(h_{wo} - h_{wi}) \quad (2)$$

The heat transfer rate of the heat exchanger, Q, is calculated by

$$Q = Q_w = m_w(h_{wo} - h_{wi}) \quad (3)$$

where m is mass flow rate (subscripts v and w stand for the vapor side and water side, respectively), h_{vi}, h_{vo}, h_{wi} and h_{wo} are inlet and outlet enthalpy values of the vapor and cooling water sides, respectively [12].

Experimental data obtained from the microchannel heat exchangers are under the room temperature condition of 30~32°C. In order to compare the behaviors of heat transfer between the microchannel heat exchangers, all experimental conditions for the two heat exchangers were kept the same. The experimental results for T1 and T2 were recorded by MX 100 recorder; the calculated values were taken by the mean values. For all cases done in this study, the steam outlet from the mini boiler was the dry saturated vapor having the absolute pressure of 1.4 bar corresponding with the saturated temperature of 110 ºC. Because of the heat loss from the outlet of mini boiler to the inlet of microchannel heat exchanger, the inlet temperature of microchannel heat exchangers for vapor side was kept constantly at 101 ºC.

Table 3. Experimental results were recorded by MX100 recorder

a) for microchannel heat exchanger T1

Channel	Start Data No.	End Data No.	Min.	Max.	P-P	Mean	RMS
CH00001[C]	0	661	100.7	101.3	0.6	101.0	101.0
CH00002[C]	0	661	71.3	73.0	1.7	72.2	72.2
CH00003[C]	0	661	52.3	54.1	0.8	52.7	52.7
CH00004[C]	0	661	31.7	32.7	1.0	32.1	32.1

b) for microchannel heat exchanger T2

Channel	Start Data No.	End Data No.	Min.	Max.	P-P	Mean	RMS
CH00001[C]	0	406	98.7	102.7	4.0	101.0	101.0
CH00002[C]	0	406	38.7	40.0	1.3	39.3	39.3
CH00003[C]	0	406	39.7	40.9	1.2	40.2	40.2
CH00004[C]	0	406	31.6	32.1	0.5	31.9	31.9

In Table 3, the channel CH00001 is the saturated vapor temperature, the channel CH00002 is the condensing vapor temperature, the channel CH00003 is the outlet cooling water temperature, and the channel CH00004 is the inlet cooling water temperature. The results in Table 3 recorded for the cooling water having the inlet temperature of 32ºC and mass flow rate of 3.1133 g/s and for the vapor having the inlet temperature of 101 ºC. The results shown that the outlet temperatures for both vapor and water obtained from T1 are higher than those obtained from T2. As a result, the heat transfer rate obtained from the heat exchanger T1 (272.9 W) is higher than that obtained from the heat exchanger T2 (104.6 W). The heat transfer rate of two heat exchangers was calculated and shown in Table 4. The Table 4 shows that the mass flow rate of condensing vapor decreases as reducing the hydraulic diameter of channels.

Table 4. Heat transfer rate for two models

Model	m_v g/s	m_w g/s	h_{vi} kJ/kg	h_{vo} kJ/kg	h_{wi} kJ/kg	h_{wo} kJ/kg	Q W
T1	0.123	3.1133	2677.64	306.388	134.14	221.8	272.9
T2	0.0481	3.1133	2677.64	164.64	134.14	167.7	104.6

In this study, effect of flow arrangement on the heat transfer phenomena of the heat exchangers was also investigated. For the case done in this study, the two models had the same conditions: the temperature of saturated vapor was 101°C and the temperature of cooling water was 32 °C with mass flow rate of 3.159 g/s. Both heat exchangers were tested in the horizontal position. It was observed that the heat transfer rate obtained from the counter flow arrangement is always higher than that obtained from the parallel one: the value obtained from the counter flow arrangement is 1.04 to 1.05 times of that obtained from the parallel flow. The results for two phases are in good agreement with the results for single phase [10]; however, it is shown that the effect of flow arrangement in two phases is not stronger than single phase, as shown in Table 5.

Table 5. Heat transfer rate with counter flow and parallel flow arrangements

Model	m_v g/s	m_w g/s	Heat transfer rate Q, W	
			Counter flow	Parallel flow
T1	0.123	3.159	194,63	184,32
T2	0.0481	3.159	111.52	107.197

Figure 4 shows effect of the inlet cooling water temperature to the outlet temperatures of vapor and water for the microchannel heat exchanger T1. It is observed to see that when the inlet cooling water temperature increases, the outlet cooling water temperature increases; however, the temperature difference of cooling water is increasing slowly. As a result, the temperature difference of vapor side decreases or the condensing vapor temperature increases. It means that the condensation heat transfer efficiency in the microchannel heat exchangers decreases as increasing the inlet cooling water temperature, as shown in Fig.4.

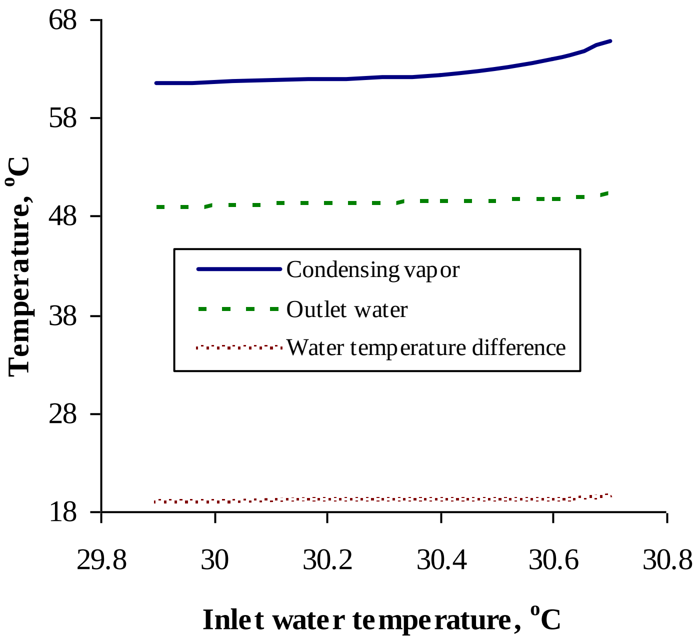

Fig. 4. Effect of the inlet cooling water temperature for heat exchanger T1

IV. CONCLUSION

Experimental work was done for the two microchannel heat exchangers with rectangular channels having hydraulic diameters of 375 µm and 265 µm to investigate condensation heat transfer. For the microchannel heat exchanger T1, the heat transfer rate of 272.9 W was achieved for the vapor having the inlet temperature of 101 ºC and the mass flow rate of 0.123 g/s and for the cooling water having the inlet temperature of 32ºC and mass flow rate of 3.1133 g/s. In this study the heat transfer rate obtained from the counter flow arrangement is always higher than that obtained from the parallel one: the value obtained from the counter flow arrangement is 1.04 to 1.05 times of that obtained from the parallel flow. The results for two phases are in good agreement with the results for single phase; however, it is shown that the effect of flow arrangement in two phases is not stronger than single phase. In addition, the condensation heat transfer efficiency of the microchannel heat exchanger decreases as increasing the inlet cooling water temperature.

ACKNOWLEDGMENT

The supports of this work by (1) the project (Project No. T2013-22TĐ /KHCN-GV) sponsored by the specific research fields at Hochiminh City University of Technical Education, Vietnam and (2) the project (Project No. 65-13/CTHT-NEPTECH) sponsored by New Product & Technology Center (NEPTECH) – Hochiminh City Department of Science and Technology of Vietnam, are deeply appreciated.

REFERENCES

[1] J.S. Hu and C.Y.H. Chao, An experimental study of the fluid flow and heat transfer characteristics in micro-condensers with slug-bubbly flow, International Journal of Refrigeration, Vol. 30, (2007), pp. 1309-1318

[2] C.Y. Park and P. Hrnjak, Experimental and numerical study on microchannel and round-tube condensers in a R410A residential air-conditioning system, International Journal of Refrigeration, Vol. 31, (2008), pp. 822-831

[3] Santiago Martínez-Ballester, José-M. Corberán, José Gonzálvez-Maciá, Numerical model for microchannel condensers and gas coolers: Part I e Model description and validation, International Journal of Refrigeration, Vol. 36, 2013, pp. 173-190

[4] H. Cho, D.Y. Hwang, B.S. Lee, and H.H Jo, Fabrication of micro condenser tube through direct extrusion, Journal of Materials Processing Technology, Vol. 187-188, (2007), pp. 645-648

[5] Jaehyeok Heo, Hanvit Park, and Rin Yun, Comparison of condensation heat transfer and pressure drop of CO_2 in rectangular microchannels, International Journal of Heat and Mass Transfer, Vol. 65, (2013), pp. 719–726

[6] J.R. García-Cascales, F. Vera-García, J. Gonzálvez-Macía, J.M. Corberán-Salvador, M.W. Johnson, and G.T. Kohler, Compact heat exchangers modeling: Condensation, International Journal of Refrigeration, Vol. 33, 2010, pp. 135-147

[7] P. Hrnjak and A.D. Litch, Microchannel heat exchangers for charge minimization in air-cooled ammonia condensers and chillers, International Journal of Refrigeration, Vol. 31, (2008), pp. 658-668

[8] T.T. Dang and J.T. Teng, Comparison on the heat transfer and pressure drop of the microchannel and minichannel heat exchangers, Heat and Mass Transfer, Vol, 47, 2011, pp. 1311-1322

[9] T.T. Dang and J.T. Teng, The effects of configurations on the performance of microchannel counter-flow heat exchangers – An experimental study, Applied Thermal Engineering, Vol. 31, Issue 17-18, 2011, pp. 3946-3955.

[10] T.T. Dang and J.T Teng, Influence of flow arrangement on the performance index for an aluminium microchannel heat exchanger, IAENG Transactions on Engineering Technologies Volume 5, the American Institute of Physics (AIP), Vol. 1285, 2010, pp. 576-590

[11] G. Goss Jr. and J.C. Passos, Heat transfer during the condensation of R134a inside eight parallel microchannels, International Journal of Heat and Mass Transfer, Vol. 59, pp. 2013, pp. 9–19

[12] Michael J. Moran, Howard N. Shapiro, Fundamentals of Engineering Thermodynamics, John Wiley & Sons Ltd, England, 2006.