A Numerical Study on Effects of Microchannel Shape to Condensation of Steam

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Abstract

This paper studied the effects of microchannel shape to condensation of steam by the numerical method. There are three microchannel condensers with different channel shapes were simulated. The numerical results are in good agreement with the literature reviews. From the numerical results, the condenser T1 has the highest heat efficiency because it has the smallest heat transfer thickness. At the same the hydraulic diameter, the condensation efficiency of T3 is better than that obtained from T2; this result is due to the heat transfer thickness of T3 is smaller than that of T2. In summary from results, the condenser T1 is the best for condensation efficiency and fabrication. These results are essential to use for the numerical simulation of microchannel condenser.

Keywords: *Numerical Simulation, Heat Transfer, Condensation, Microchannel, Steam.*

1. Introduction

In recent years, many investigations on microchannel heat transfer have already been presented. Regarding to heat transfer in two-phase flow, a modified k- ε turbulence model was developed by Hu and Zhang [1] to simulate the gas–liquid two-phase flow and the heat transfer in the steam surface condensers, using a quasi-three-dimensional algorithm. The results obtained from the proposed model agree well with the experimental results. In addition, the results also show an obvious improvement in the prediction accuracy comparing with previous results. Fossa [2] built a simple model for evaluating the heat transfer and flow characteristics in annular two-phase flow. The model was applied with the various flow conditions including thermal non equilibrium and variable cross section ducts. Lim and Yu [3] investigated the numerical simulations on the heat transfer characteristics of the axis symmetric air - water two-phase flow in microtubes with inner diameter of 300 and 500 mm. The two-phase flow was achieved by injecting nitrogen gas coaxially from the centrally positioned tube to the continuous liquid phase flow. Numerical results showed that the Nusselt number enhancement could be as high as 200% while the two phase frictional pressure loss for the bubbly flow was about 20% higher than that of the liquid flow alone. The results also showed that the heat transfer performance varies with the bubble size, frictional pressure drop and Reynolds number. Numerical simulation of an air and water two-phase flow in a 20 μm ID tube was carried out by Fukagata et al. [4]. They studied the flow and heat transfer characteristics in bubble-train flows. The finite difference method was used to solve the governing equations, while the level set method was adopted for capturing the interface of gas and liquid. Regardless of the flow conditions, the gas-phase velocity was found approximately 1.2 times higher than the liquid-phase velocity. This was in accordance with the Armand correlation for two-phase flows in macro-sized tubes. The computed wall temperature distribution was qualitatively similar to that observed from experiments in a mini channel. In this study, the local Nusselt number beneath the bubble was higher than that of the single-phase flow. However, the results in [1-4] did not simulate the microchannel dimensions of the condenser.

Xia et al. [5] studied the heat transfer in heat exchangers using micro channel. In this study, a simulation model was established using computational fluid dynamics (CFD) and a new design of heat exchanger was proposed based on these extra micro-effects. The results showed that as the inlet area decreases, the heat transfer rate increases and the pressure loss decreases. Yu et al. [6] focused on the hydraulic and thermal characteristics of fractal tree-like microchannels by numerical and experimental methods. The experimental results showed that the fractal tree-like microchannels had a higher the heat transfer coefficient than that of the straight microchannels at the cost of a higher pump power. Ling et al. [7] presented a three dimensional direct simulation on boiling flow in a rectangular microchannel. Growth and merger of the bubbles were simulated and the impact of the bubbles' merger on the heat transfer was analyzed. In this study, the merger can produce a temporal growth in heat flux, while the thin liquid film between the bubble and the wall issues the main reason for the high heat flux in microchannel boiling flow. The condensation flow of the refrigerant FC-72 in a rectangular microchannel with a hydraulic diameter of 1 mm was numerically studied by Chen et al. [8], using the volume of fluid (VOF) model. In this study, the vapor phase formed a continuous column with decreasing the downstream diameter. Slugs were periodically generated at the head of the column.

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Decreasing the wall cooling heat flux and increasing the flow mass flux increased the vapor column length. Hasan et al. [9] evaluated the effect of channel geometry on the heat transfer and fluid flow behaviors of a counter-flow microchannel heat exchanger by using numerical simulation. In the study, heat exchanger effectiveness and performance index (as functions of the relative size of channel, Reynolds number, and thermal conductivity ratio) were presented. Chein and Chen [10] presented a numerical study of the inlet/outlet arrangement effect on microchannel heat sink performance. Six types of heat sink were studied with the best performance being the V-type. Foli et al. [11] performed numerical study on heat flux, heat transfer rate, and pressure drop in each channel with a variety of aspect ratios. However, results in [9-11] mentioned for the single phase. Doan et al. [12] numerically simulated on phase change of steam in a microchannel condenser. However, the results in this study only simulated for one condenser which did not mention the effect of geometry.

From the literature review above, studies on the condensation process in microchannel are not yet well established, particularly in numerical simulations for the channel geometry. So it is essential to investigate effects of the microchannel shape to condensation of steam. Following sections, three microchannel condensers with different channel shapes will be discussed in this study.

2. Methodology

2.1 Design and Numerical Simulation

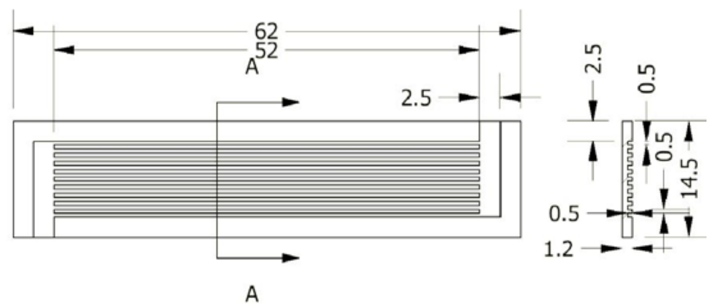
Based on several literature reviews above, there are three microchannel condensers were designed. The heat transfer process of these devices is carried out between the water vapor (steam) and the cooling water. Each condenser has two PMMA (Polymethyl methacrylate) plates which were bonded on a substrate. The material for the substrate is Aluminum.

Table 1: Comparison on dimensions of the condensers

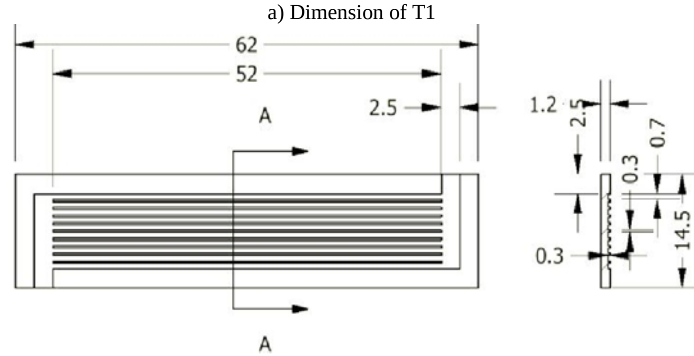
Type	Substrate, mm			Channels of vapor, μm		
	L	W	T	W_c	D_c	D_h
T1	62	14.5	1.2	500	500	500
T2	62	14.5	1.2	700	300	420
T3	62	14.5	1.2	550	400	420

(where L is substrate length, W is substrate width, T is substrate thickness, W_c is channel width, D_c is channel depth, and D_h is hydraulic diameter of channel)

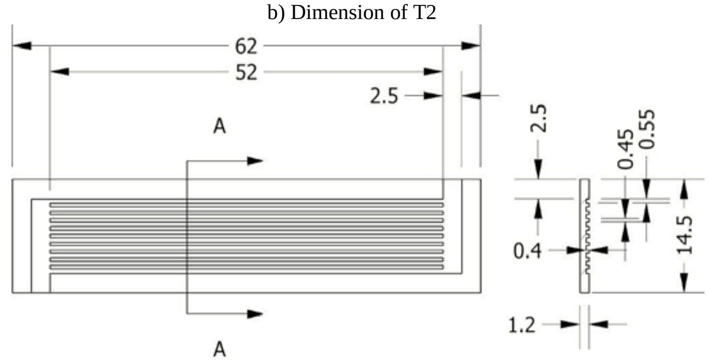
For the vapor side, 10 microchanels were made on substrate, while a large channel was made on PMMA for the water side. The distance between microchannels is 500 μm. The dimensions of the channel for the water side are fixed for all condensers (the depth of 0.5 mm and the width of 9.5 mm). Dimensions of the substrate and the microchannels of vapor are listed in Table 1. Three condensers have the same overall dimensions of substrates but have the different dimensions of vapor channels. In addition, the condenser T1 has the same dimensions of substrate and microchannels in [12], except the condenser in [12] has the substrate thickness of 700 μm, as shown in Fig. 1. The dimensions of substrate and the hydraulic diameter of microchannel of T2 are the same those obtained from T3; however, they have a difference of microchannel width and depth. The manifolds of the condensers have a rectangular cross-section with a width of 2.5 mm, a length of 12 mm, and a depth of 0.5 mm.



a) Dimension of T1



b) Dimension of T2



c) Dimension of T3

Fig. 1. Dimensions of the condensers

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2.2 Mathematical Model

To analyze condensation in the three microchannel condensers, there are several consumptions: continuum fluid and no radiation. The governing equations in this system can be expressed by [12-14].

Continuity equation

$$\frac{\partial\rho}{\partial t}+u\frac{\partial\rho}{\partial x}+v\frac{\partial\rho}{\partial y}+w\frac{\partial\rho}{\partial z}+\rho\left[\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right]=0 \qquad (1)$$

Momentum equations

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\frac{\mu}{\rho}\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \qquad (2a)$$

$$\frac{\partial v}{\partial t}+u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\frac{\mu}{\rho}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \qquad (2b)$$

$$\frac{\partial w}{\partial t}+u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial z}+\frac{\mu}{\rho}\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \qquad (2c)$$

For steady-state conditions: $(\partial\rho/\partial t)=0$, $(\partial u/\partial t)=0$, $(\partial v/\partial t)=0$, and $(\partial w/\partial t)=0$

where, μ is dynamic viscosity, ρ is density, u is velocity in the x-direction, v is velocity in the y-direction, w is velocity in the z-direction, p is pressure, and p_0 is initial pressure.

Besides, the heat transfer formulas were applied as [12-14]. Fourier's law for heat conduction

$$q = -k.\nabla T \qquad (3)$$

Heat transfer in solid

$$\rho C_p\left(\frac{\partial T}{\partial t}+u_{trans}.\nabla T\right)+\nabla.(q+q_r)=-\alpha T:\frac{dS}{dt}+Q \qquad (4)$$

Heat transfer in liquid

$$\rho C_p\left(\frac{\partial T}{\partial t}+u.\nabla T\right)+\nabla.(q+q_r)=\alpha_p T\left(\frac{\partial p}{\partial t}+u.\nabla p\right)+\tau:\nabla u+Q \qquad (5)$$

Heat transfer in phase change from vapor to liquid can be determined by $\rho=\theta\rho_{ph1}+(1-\theta)\rho_{ph2}$ with θ is vapor quality and heat transfer coefficient $\alpha_m=\frac{1}{2}\frac{(1-\theta)\rho_{ph2}-\theta\rho_{ph1}}{\rho}$ (the mass fraction), where ρ_{ph1} and ρ_{ph2} are density of vapor and liquid, respectively.

To solve these equations, the COMSOL Multiphysics – version 5.2a was applied with the Time Dependent solver. These models were set up and generated mesh with the Free Tetrahedral type and it included 43843 domain elements, 13134 boundary elements, and 3262 edge elements. The meshing results of the condenser T1 are shown in Fig. 2.

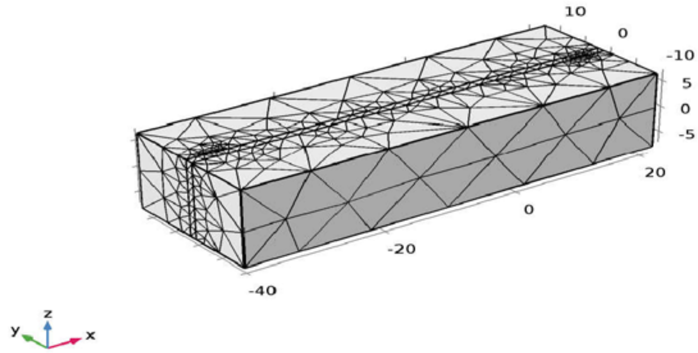

Fig. 2. Meshing results of the condenser T1

3. Results and Discussion

Throughout the study, the parameters of the cooling water were fixed at the inlet temperature of 29 °C and the mass flow rate of 3 g/s. The model was solved under the ambient temperature of 31 °C. The vapor flow rate was varying from 0.01 to 0.1 g/s. In addition, the inlet temperature of vapor was also fixed at 105 °C. For numerical simulations, the mathematical model and the solver were the same with [12]. The Figure 3 shows the vapor quality profile for the phase change of the condenser T1. The numerical results of T1 are in good agreement with those obtained from the experimental data in [15].

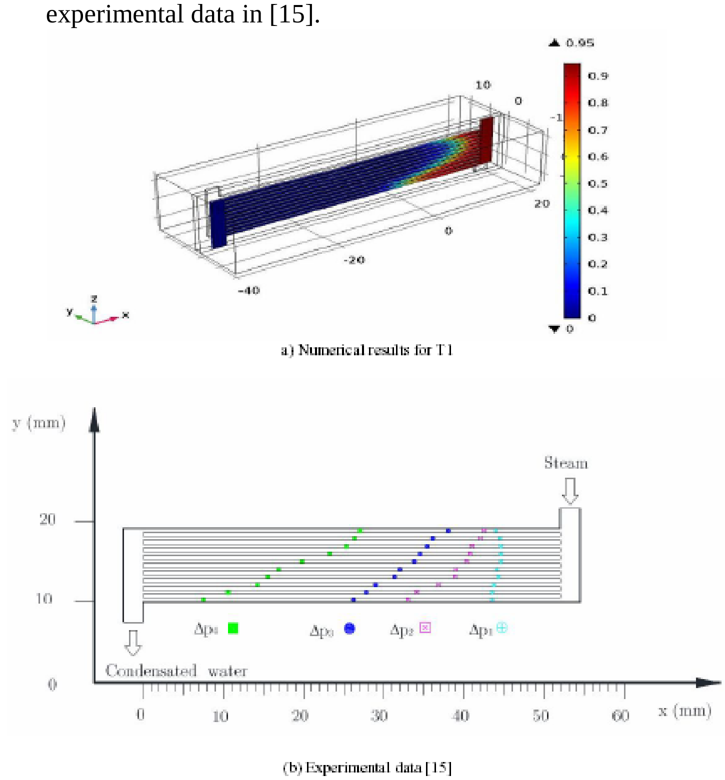

Fig. 3. Profiles for the phase change of the condensers

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To compare the effect of substrate thickness to condensation, the condenser T1 was compared with the condenser in [12] at the same conditions. Figure 4 shows a comparison between the condensed water temperature of the condenser T1 and the condenser in [12].

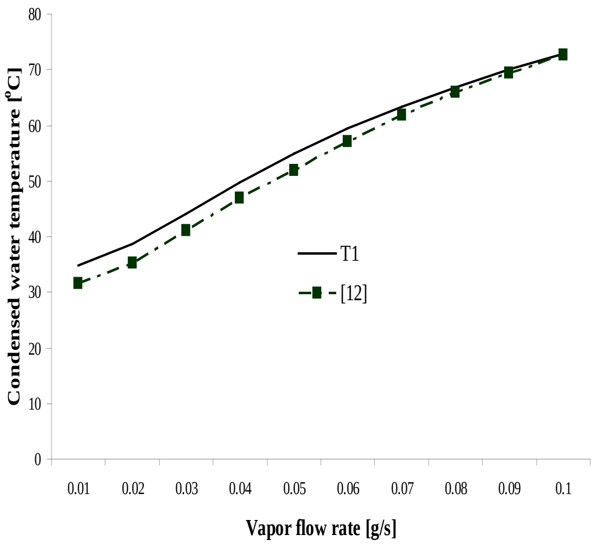

Fig. 4. Comparison between the condenser T1 and the condenser in [12]

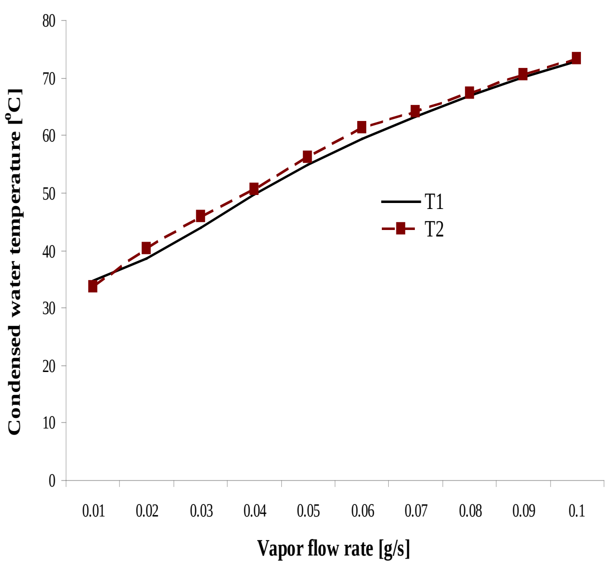

Fig. 5. Comparison between the condenser T1 and the condenser T2

It is observed that the condensed water temperature of T1 is higher than that obtained from [12]. It is due to the substrate thickness of the condenser in [12] is thin to compare with the condenser T1 (0.7 mm to compare with 1.2 mm). It means that the condenser in [12] has higher the heat transfer efficiency than that of the condenser T1. However, the condenser in [12] is easy to meet several troubles when manufacturing because of its thickness. The figure is also indicated that the difference between two condensers is negligible at high vapor flow rate. When the vapor is varying from 0.01 to 0.1 g/s, the difference of two condensed water temperatures reduces from 3.3 to 0.3 °C.

Figure 5 shows a comparison the condensed water temperature between T1 and T2. The results show that the condensed water temperature of T1 is lower than that obtained from T2, leading to the condensation efficiency of T1 is better than that of T2. It is noted that the hydraulic diameter of microchannel in T1 is 500 μm and the hydraulic diameters of microchannel in T2 and T3 are the same (420μm), as listed in Table 1.

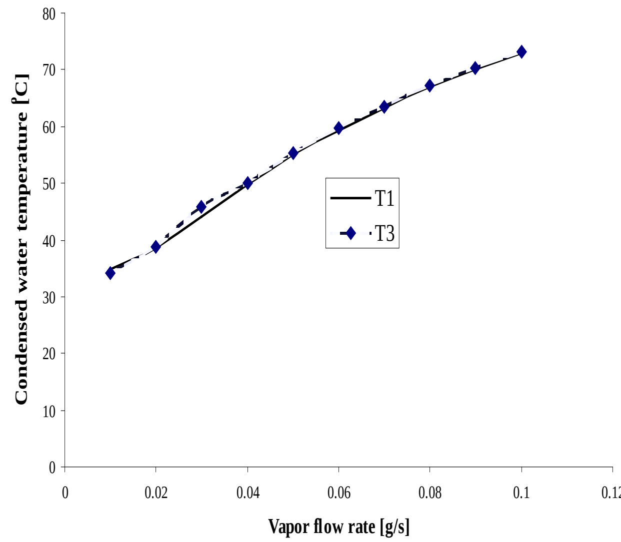

Fig. 6. Comparison between the condenser T1 and the condenser T3

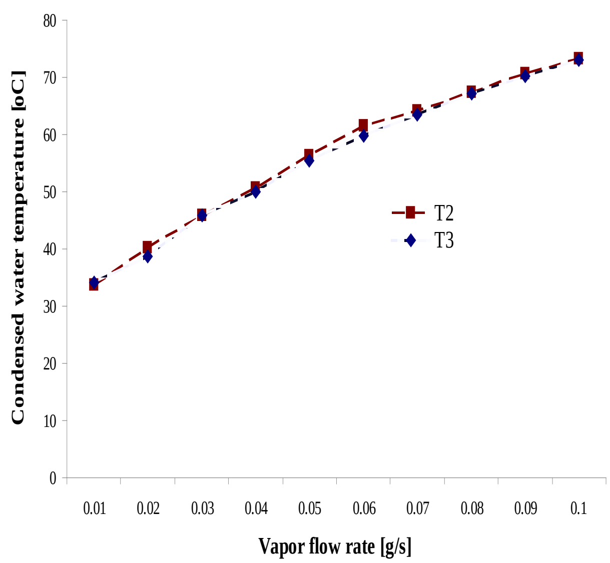

Fig. 7. Comparison between the condenser T2 and the condenser T3

The condensation efficiency of T1 is also higher than that of T3, as shown in Fig. 6. From Figs. 5 and 6, it is indicated that the T1 has the highest heat efficiency, it is due to T1 has the smallest heat transfer thickness. At the same the hydraulic diameter, the condensed water temperature of T3 is lower than that of T2, leading to the condensation efficiency of T3 is better than that obtained from T2. It is due to the heat transfer thickness of T3 is smaller than that of T2, as shown in Fig. 7. From Figs. 4-7, it is also indicated that the condenser T1 is the best for condensation efficiency and fabrication. However, the different efficiency of three condensers is not strong. These results complement for studies on the numerical simulation of condensation in microchannels.

4. Conclusions

A numerical simulation on the effects of microchannel shape to condensation of steam was done in this study. To solve this problem, three microchannel condensers with different channel shapes were simulated, using the COMSOL Multiphysics – version 5.2a.

The numerical results in this study were compared with the results from literature reviews. The comparison indicated that the numerical results are in good agreement with the reviews.

The condenser T1 has the smallest heat transfer thickness, leading to the T1 has the highest heat transfer efficiency (the condensed water temperature of T1 is lowest). At the same the hydraulic diameter, the condensation efficiency of T3 is better than that obtained from T2; it is due to the heat transfer thickness of T3 is smaller than that of T2.

In this study, the condenser T1 is the best for condensation efficiency and fabrication. However, the difference on condensation efficiency of three condensers is not strong. These results are useful for the numerical analysis of condensation, especially in microchannels.

Acknowledgments

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