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Two-Phase Flow Boiling Heat Transfer Coefficient with R290 in Horizontal 3 mm Diameter Mini Channel

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ABSTRACT

Various experiments about the heat transfer coefficient of two-phase flow boiling in a mini channel tube have been carried out. In addition to obtaining data on heat transfer coefficients experimentally, many researchers compare their experimental data and even add the data from other researchers with existing correlations. This study aims to obtain experimental data and the characteristics of the heat transfer coefficient of two-phase flow boiling of refrigerant R290 in a horizontal mini channel tube. The mass flux is varied from 50 kg/m2s - 180 kg/m2s and heat flux of 5 kW/m2 - 20 kW/m2. The test section is a stainless-steel mini channel tube with a 3 mm diameter and a length of 2 m. The experimental data obtained are then predicted with various correlations of the available heat transfer coefficients. The correlation that will be used in this study is based on the asymptotic flow model, where this model is a combination of nucleate and convective flow boiling mechanism. This paper will show the example of the comparison between the heat transfer coefficient experimental and the prediction of some correlation with the condition of mass flux 100 kg/m2s and 150 kg/m2s, and heat flux 10 kW/m2 and 15 kW/m2. The result of the heat transfer coefficient experimental data depends on the mass quality and heat flux. The value increases with increasing mass quality and decreases drastically as mass quality approaches 1. The correlation of Aizuddin et al. is the most accurate, with a deviation of 10.006% in the experiment with the condition of mass flux 100 kg/m2s and heat flux 10 kW/m2. While in the experiment with the condition of mass flux 150 kg/m2s and heat flux 15 kW/m2 has a deviation of 9.969%.

Keywords:
Heat transfer coefficient; asymptotic; R290; two-phase flow boiling; mini channel

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1. Introduction

Evaporative heat transfer in conventional or mini channel tube is a theme for researchers to research energy models or processes of a system such as an evaporator, condenser, cooling tower, refrigerator, and heat exchanger. A part of the evaporative heat transfer is the heat transfer coefficient that the value can affect the heat transfer quality itself. The results of published studies conclude different things about the characteristics of the heat transfer coefficient. Lima *et al.,* [1], Ducoulombier *et al.,* [2], and Grauso *et al.*,[3] concluded that the value of the heat transfer coefficient increases with increasing mass quality until the flow enters the dry out regime then decreases

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drastically in the mist regime where the vapor began to spread in the form of bubbles, which caused a decrease in the quality of heat transfer. Meanwhile, Hamdar *et al.,* [4], Saisorn *et al.,* [5], and Anwar *et al.,* [6] revealed that the value of the heat transfer coefficient is independent of the mass quality. Then Shiferaw *et al.,* [7] showed that the value of the heat transfer coefficient peaks at the lower mass quality (usually at nucleate boiling), then decreases with increasing mass quality.

Many environmentally friendly refrigerants are used to replace refrigerants with high ODP (Ozone Depletion Potential) and GWP (Global Warming Potential) values, which harm the atmosphere. Propane or R290 is an environmentally friendly refrigerants and also a natural refrigerant with zero ODP and GWP [8]. R290 does not damage the atmosphere and environment and has a higher cooling capacity than R22 [9]. Padalkar *et al.,* [10] stated that the use of R290 resulted in high efficiency and heat transfer quality. The asymptotic model is one of the heat transfer coefficient correlation approaches. Liu and Winerton [11] and Steiner and Taborek [12] stated that the asymptotic model is a combination of nucleate and convective boiling mechanisms. The nucleate boiling mechanism occurs when the heated tube's temperature is higher than the fluid's saturation temperature. Meanwhile, convective boiling usually occurs in annular regimes, characterized by forming a liquid layer on the edge of the tube with vapor flowing in the core [13]. As the vapor increases, the convective boiling mechanism will slowly suppress the nucleate boiling, and it makes a complete boiling flow phenomenon form. This model correlates as in Eq. (1) [14].

$$h_{tp} = [(h_{nb})^n + (h_{cb})^n]^{\frac{1}{n}} \quad (1)$$

2. Methodology

2.1 Experimental Set Up

The experimental apparatus is shown in Figure 1. The components consist of a condenser, sub-cooler, receiver, refrigerant pump, mass flow meter, heater, and a test section. A needle valve controls the refrigerant's mass flow rate, and it is measured by a Coriolis-type mass flow meter.

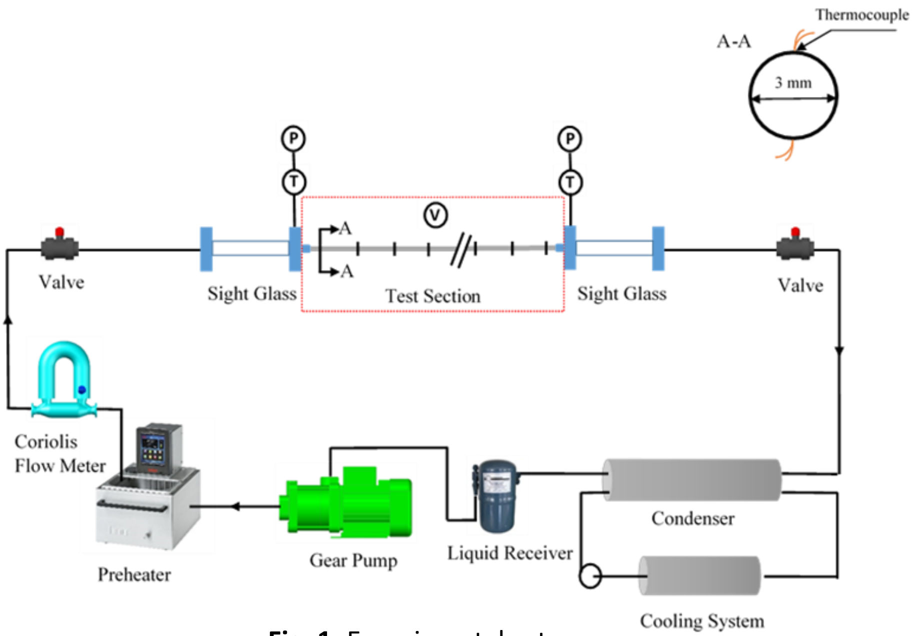

Fig. 1. Experimental set up

A preheater or cooler is installed to control the refrigerant's mass quality by heating or condensing it before entering the test section. The test section is a tube made of stainless steel with a smooth surface with a 3 mm diameter and a length of 2 m insulated to minimize heat loss to the environment. The mass flux is varied from 50 kg/m^{2}s - 180 kg/m^{2}s and heat flux of 5 kW/m^2 - 20 kW/m^2. The refrigerant's saturation pressure is used to determine the saturation temperature, measured using a pressure gauge at the inlet and outlet of the test section. Sight glass is attached to the inlet and outlet for flow visualization.

2.1 Data Reduction

The saturation pressure measured at the inlet and outlet test section is used to determine the refrigerant's physical properties. Mass quality throughout the test section calculate using Eq. (2) [13].

$$x = \frac{i - i_f}{i_{fg}} \tag{2}$$

Where i is the enthalpy in kJ/kg, f is the liquid condition at saturation temperature, and g is the vapor condition at saturation temperature. The subcooled length Z_{sc} is used to determine the starting point of saturation, calculated using Eq. (3) [13].

$$Z_{sc} = L \frac{i_f - i_{f,in}}{(Q/W)} \tag{3}$$

Where L is the pipe length in m, i is the enthalpy in kJ/kg, f is the liquid condition at saturation temperature, and g is the vapor condition at saturation temperature. Q is electric power in kW, and W is mass flow rate in kg/s. The experimental heat transfer coefficient h along the test section is obtained using Eq. (4) [13].

$$h = \frac{q}{T_{wi} - T_{sat}} \tag{4}$$

Where q is the heat flux in kW/m^2, T is the temperature in ^{o}C, wi is the test section's inner wall, and sat is the saturation condition.

2.3 Correlation Analysis

The data which is obtained from the experimental set up is processed to obtain experimental data of the heat transfer coefficient. Several correlations from the five other authors will be used to predict the experimental heat transfer coefficient data. The correlation used is based on the asymptotic model with the value of variable n is different for each author. Aizuddin *et al.,* [9], Kim and Mudawar [16], Tapia and Ribatski [17], and Zou *et al.,* [18] used n = 2 in their asymptotic model correlation. Meanwhile, Turgut and Asker [15] used n = 3 in their correlation.

All correlations used are the modification of other correlations adjusted for each author’s research data. Aizuddin *et al.*, Turgut and Asker, and Kim and Mudawar modify the nucleate and convective heat transfer coefficient correlation by adding several empirical constants. Tapia and Ribatski, and Zou *et al.,* do the same thing, but they add some variables from the superposition model to the fundamental correlation of the asymptotic model.

The research conditions and refrigerants used by each author also varied. Starting from the mass flux that varies between 19 -1608 kg/m^{2}s, the heat flux is between 0.5 - 145 kW/m^2, and the refrigerants used are varied from R290 to CO_2. However, the test section used from 0.19 - 9.52 mm, which is confirmed to be in the range of mini channels to small channels tube. The correlations of the heat transfer coefficient are used in this study is shown in Table 1.

Table 1
The heat transfer coefficient correlation used in this study

Author	Equation	Condition
Aizuddin *et al.*, [9]	$h_{tp} = [(h_{nb})^2 + (h_{cb})^2]^{\frac{1}{2}}$ $h_{nb} = \left[(a_1)\left(Bo\frac{P_H}{P_F}\right)^{(a_2)} P_r^{(a_3)}(1 - x)^{(a_4)}\right]\left(0.023Re_f^{0.8}Pr_f^{0.4}\frac{k_f}{D_h}\right)$ $h_{cb} = \left[(a_5)\left(Bo\frac{P_H}{P_F}\right)^{(a_6)} We_{fo}^{(a_7)} + (a_8)\left(\frac{1}{X_{tt}}\right)^{(a_9)}\left(\frac{\rho_g}{\rho_f}\right)^{(a_{10})}\right]\left(0.023Re_f^{0.8}Pr_f^{0.4}\frac{k_f}{D_h}\right)$	Refrigerant : R290 D : 3 mm G : 100 – 200 kg/m^{2}s q : 5 – 15 kW/m^2 T_{sat} : 10 °C
Turgut and Asker [15]	$h_{tp} = [(h_{nb})^3 + (h_{cb})^3]^{\frac{1}{3}}$ $h_{cb} = \left(0.023Re_f^{0.8}Pr_f^{0.4}\frac{k_f}{D_h}\right)\left(A_1\left(\frac{1}{Co}\right)^{A_2}\right)$ $h_{nb} = A_3 h_{Cooper}^{A_4} P_r^{A_5}(1 - x)^{A_6}$ $h_{Cooper} = 55P_r^{0.12-0.087ln\varepsilon}(-0.4343lnP_r)^{-0.55} M^{-0.5} q^{0.67}$	Refrigerant : R744 (CO_2) D_h : 0.529-9.52 mm G : 100 – 1400 kg/m^{2}s q : 5 – 45 kW/m^2 T_{sat} : -35 - 20 °C
Kim and Mudawar [16]	$h_{tp} = [(h_{nb})^2 + (h_{cb})^2]^{\frac{1}{2}}$ $h_{nb} = \left[2345\left(Bo\frac{P_H}{P_F}\right)^{0.70} P_r^{0.38}(1 - x)^{-0.51}\right]\left(0.023\,Re_f^{0.8}Pr_f^{0.4}\frac{k_f}{D_h}\right)$ $h_{cb} = \left[5.2\left(Bo\frac{P_H}{P_F}\right)^{0.08} We_{fo}^{-0.54} + 3.5\left(\frac{1}{X_{tt}}\right)^{0.94}\left(\frac{\rho_g}{\rho_f}\right)^{0.25}\right]\left(0.023\,Re_f^{0.8}Pr_f^{0.4}\frac{k_f}{D_h}\right)$	Refrigerant : 18 different refrigerant D_h : 0.19 - 6.5 mm G : 19 – 1608 kg/m^{2}s q : 0.5 – 5 kW/m^2
Tapia and Ribatski [17]	$h_{tp} = [(S.h_{nb})^2 + (F.h_{cb})^2]^{\frac{1}{2}}$ $h_{nb} = 207\frac{k_f}{D}\left(\frac{qD}{k_f T_{sat}}\right)^{0.745}\left(\frac{\rho_g}{\rho_f}\right)^{0.581}\left(\frac{\mu_f}{\rho_f}\frac{\rho_f c_{pf}}{k_f}\right)^{0.533}$ $h_{cb} = \left(0.023\,\frac{k_f}{D}\,Re_f^{0.8}Pr_f^{\frac{1}{3}}\right)$ $S = \frac{c_{s,1}Bd^{c_{s,2}}}{1+c_{s,3}\left(10^{-4}Re_f F^{1.25}\right)^{c_{s,4}}}$ $F = 1 + \frac{c_{f,1}X_{tt}^{c_{f,2}}}{\left(1+We_{ug}^{c_{f,3}}\right)}$	Refrigerant : R134a, R1234ze(E), R1234yf, and R600a D : 1 – 2.6 mm G : 200 – 800 kg/m^{2}s q : 15 – 145 kW/m^2 T_{sat} : -31 - 41 °C
Zou *et al.*, [18]	$h_{tp} = [(K.S.h_{nb})^2 + (F.h_{cb})^2]^{\frac{1}{2}}$ $h_{nb} = 55P_r^{0.12}(-lnP_r)^{-0.55} M^{-0.5} q^{0.67}$ $S = \left(1 + 0.55F^{0.1}Re_f^{0.16}\right)^{-1}$ $F = \left[1 + x_{ave}Pr_f\left(\frac{\rho_f}{\rho_g} - 1\right)\right]^{0.35}$ $K = \frac{h_m}{h_i} = \frac{1}{1 + \frac{\Delta T_{bp}}{\Delta T_{id}}\lvert y-x\rvert^{C_1}\left(\frac{P}{10^5}\right)^{C_2}\left[1+C_3 exp\left(-\frac{q}{3x10^5}\right)\right]}$	Refrigerant : a mix between R170 and R290 D_h : 8 mm G : 63.6 – 102.5 kg/m^{2}s q : 13.1 – 65.5 kW/m^2

3. Results

The experiments were carried out at the condition of mass flux varied from 50 kg/m^{2}s - 180 kg/m^{2}s and heat flux of 5 kW/m^2 - 20 kW/m^2. Based on the experimental results, the heat transfer coefficient and heat flux value are influenced by mass quality. Yang *et al.,* [19] and Karayiannis *et al.,* [20] stated that the value of the heat transfer coefficient increases with heat flux as mass quality increases. The heat transfer coefficient will increase to a specific mass quality, then it will decrease drastically. This event is due to the increasing temperature of the test section will cause a decrease in the heat transfer coefficient rate. Figure 2 shows the characteristic of the experimental heat transfer coefficient compared with the condition of heat flux from 5 kW/m^2 - 15 kW/m^2 and mass flux 100 kg/m^{2}s. The figure shows that the heat transfer coefficient value will increase if the heat flux from the experiment increased.

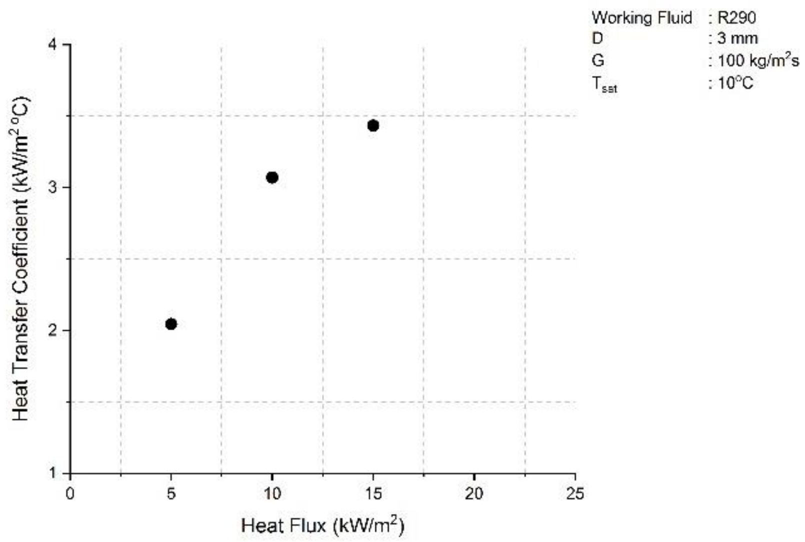

Fig. 2. Characteristics of the R290 heat transfer coefficient at the condition of mass flux 100 kg/m^{2}s

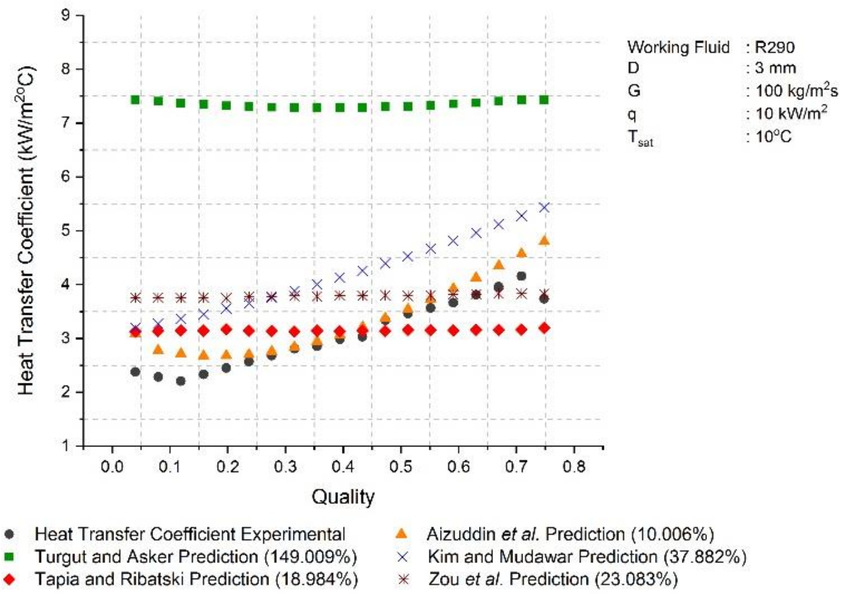

Fig. 3. Comparison of the asymptotic model correlation with the experimental results of the R290 heat transfer coefficient at the condition of mass flux 100 kg/m^{2}s and heat flux 10 kW/m^2

The correlation of the asymptotic model in predicting experimental results is needed to determine the quality of the experimental data and to know the correlation with the best accuracy in predicting the data in this study. Figure 3 and Figure 4 show the example result between the asymptotic model and the experimental data of the heat transfer coefficient at the condition of mass flux 100 kg/m^{2}s and heat flux 10 kW/m^2, and the condition of mass flux 150 kg/m^{2}s and heat flux 15 kW/m^2. In experiment with the condition of mass flux 100 kg/m^{2}s and heat flux 10 kW/m^2, the heat transfer coefficient occurs up to the mass quality of 0.748. Then in the experiment with the condition of mass flux 150 kg/m^{2}s and heat flux 15 kW/m^2, the heat transfer coefficient can occur until the mass quality up to 0.709.

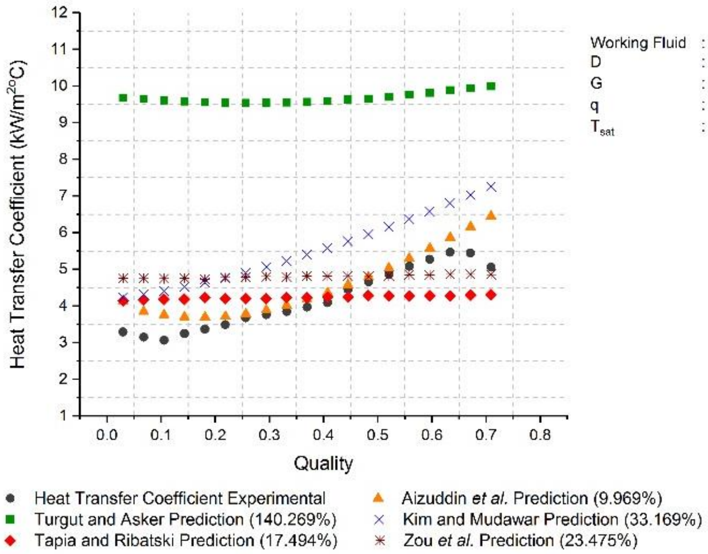

Fig. 4. Comparison of the asymptotic model correlation with the experimental results of the R290 heat transfer coefficient at the condition of mass flux 150 kg/m^{2}s and heat flux 15 kW/m^2

All of the asymptotic model correlations used in this study have a similar trend to the experimental data, but the correlation of Turgut and Asker [15] is considered less accurate in predicting this experimental data. The deviation is 149.009% in the experiment with the condition of mass flux 100 kg/m^{2}s and heat flux 10 kW/m^2. Then the deviation of 140.269% in the experiment with the condition of mass flux 150 kg/m^{2}s and heat flux 15 kW/m^2. Turgut and Asker stated that the characteristic of CO_2 could not be clearly identified due to the different thermal behavior from other refrigerants.

In the experiment with the condition of mass flux 100 kg/m^{2}s and heat flux 10 kW/m^2, the correlation of Aizuddin *et al.,* [9] is the most accurate with a deviation of 10.006%, followed by Tapia and Ribatski [17] with a deviation of 18.984%, Zou *et al.,* [18] with a deviation of 23.083%, and Kim and Mudawar [16] with a deviation of 37.882%.

Meanwhile, in the experiment with the condition of mass flux 150 kg/m^{2}s and heat flux 15 kW/m^2, the correlation of Aizuddin *et al.,* [9] is the most accurate with a deviation of 9.969%, followed by Tapia and Ribatski [17] with a deviation of 17.494%, Zou *et al.,* [18] with a deviation of 23.475%, and Kim and Mudawar [16] with a deviation of 33.169%.

4. Conclusions

Propane or R290 is a natural refrigerant with zero ODP and GWP that does not damage the environment and has high efficiency and heat transfer qualities. The experimental data shows that the heat flux influences the heat transfer coefficient. If the heat flux in the experiment is increased, the heat transfer coefficient will also increase. The heat transfer coefficient will increase to a specific mass quality. After that, it will decrease drastically. This event is due to the increasing temperature of the test section will cause a decrease in the heat transfer coefficient rate.

All of the asymptotic model correlations used in this study have a similar trend to experimental data, but the correlation of Turgut and Asker is considered less accurate in predicting this experimental data due to the use of CO_2 in developing the correlation. The characteristic of CO_2 cannot be identified in detail because the thermal behavior is different from other refrigerants, causing a huge deviation. The deviation is 149.009% in the experiment with the condition of mass flux 100 kg/m^{2}s and heat flux 10 kW/m^2 and 140.269% in the experiment with the condition of mass flux 150 kg/m^{2}s and heat flux 15 kW/m^2. The correlation of Aizuddin *et al.* is the most accurate, with a deviation of 10.006% in the experiment with the condition of mass flux 100 kg/m^{2}s and heat flux 10 kW/m^2. While in the experiment with the condition of mass flux 150 kg/m^{2}s and heat flux 15 kW/m^2 has a deviation of 9.969%.

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