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Boiling Heat Transfer Coefficient, Pressure Drop and Flow Pattern of a Two-Phase Flow in a Horizontal Mini Channel with R290

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ABSTRACT

Many researchers have done various studies on the correlation development to predict the heat transfer coefficient and pressure drop. It is of great urgency to accurately predict the correlation to avoid a discrepancy when designing a heat exchanger. This study is aimed at determining the characteristics of a heat transfer coefficient and a pressure drop of refrigerant R290 by employing experimental data. Moreover, it is also aimed at making several comparisons in order to obtain the best correlations used to predict the experimental data. The experimental data are the two-phase flow boiling in a horizontal tube with the inner diameter of 3 mm, the mass flux of 50 kg/m^{2}s to 180 kg/m^{2}s, the heat flux of 5 kW/m^2 to 20 kW/m^2, the saturation temperature of 0^{o}C to 11^{o}C, and the vapor quality of up to 1. This study shows the effects of the two-phase Reynolds Number on the experimental heat transfer coefficient and the pressure drop. Aizuddin *et al.*,'s [1] heat transfer coefficient correlation and Sun and Mishima's pressure drop correlation showed the best prediction, namely mean absolute deviation of 14.07% and 27.64%, respectively. Wojtan *et al.*,'s [2] flow pattern map showed better flow pattern prediction with the experimental data.

1. Introduction

The higher fossil resources depletion rate and the problems such as pollution, global warming, and ozone depletion provides the indication that fossil resources will not support the future energy demand [3]. Many environmentally-friendly refrigerants have been used to replace various refrigerants with a high ozone depletion potential (ODP) and Global Warming Potential (GWP), which may result in an ozone depletion and increase the global warming effect [4]. R290, or propane, is a natural refrigerant with zero ODP and GWP, so it will not damage the environment. Moreover, it has

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a higher cooling capacity than R22 [1,5]. Based on the results of a study on various heat pump trials at WPZ Toss Switzerland [6], it was found out that R290 was the second most common refrigerant used in many heat pumps after R407C, accounting for 12% of all heat pumps tested in 2002. Based on the results of various published studies, several conclusions on the characteristics of the heat transfer coefficient and pressure drop, depending on their test conditions have been drawn. da Silva *et al.,* [7], Ducoulombier *et al.,* [8], Ribatski [9], and Markal *et al.,* [10] concluded that the heat transfer coefficient was dependent on vapor quality, mass flux, and heat flux.

The heat transfer coefficient reaches a higher value at greater mass flux and heat flux. Then at high vapor quality, the heat transfer coefficient would enter the dry out regime until it decreased so sharply in the mist regime that the vapor began to diffuse in the form of bubbles, thus reducing the heat transfer. While Hamdar *et al.,* [11] and Saisorn *et al.,* [12] stated that the heat transfer coefficient was independent from the vapor quality. Meanwhile, in regards to the pressure drop, Ju *et al.,* [13] and Bashar *et al.,* [14] pointed out that, since the friction occurred at a certain mass flux, the characteristics of the pressure drop would be better-enhanced in a small diameter tube. Then, an increased saturation temperature would reduce the pressure drop. Qu *et al.,* [15], Padilla *et al.,* [16], and Umar *et al.,* [17] concluded that the pressure drop increased slightly and linearly in various mass fluxes in a vapor with low vapor quality. In terms of the correlation, the heat transfer coefficient and pressure drop are divided into several methods; so, the experimental data would be able to be analyzed.

Turgut and Mustafa [18] stated that the coefficient correlation of the heat transfer was divided into five models, all of which were the combinations of various nucleate and their convective boiling mechanisms. The models were the enhancement factor, the superposition model, the nucleate boiling model, the asymptotic model, and the largest mechanism predominant model. Moreover, the correlation of the pressure drop was divided into the homogeneous flow model and the separated flow model [19]. Based on those previously mentioned models, the prediction process would be carried out with some correlations on the heat transfer coefficient and the pressure drop of R290 data flowing in a horizontal mini-channel tube. This study is aimed at determining the characteristics of the heat transfer coefficient and the pressure drop with refrigerant R290 and at revealing which best correlation can predict the data. Then, the result also will be plotted to the flow pattern maps of Wojtan *et al.,* [2] and Zhuang *et al.,* [20].

2. Methodology

2.1 Experimental Set Up

This study employed the experimental data from the study done by Pamitran *et al.,* [21] as Figure 1 showed. The components mainly consisted of a condenser, cooling system, liquid receiver, refrigerant pump, preheater, mass flow meter, and test section. The needle valve adjusted the mass flow rate of the refrigerant, and the flow rate was measured with a mass flow meter. A preheater was installed to control the vapor quality of the refrigerant, so it was heated before it entered the test section. The test section was a stainless-steel mini-channel tube with a smooth surface, a 3-mm diameter, and a 2-m length. It was insulated to resist any heat loss to the environment. The outer part of the test section, especially at the top, middle, and bottom sides, were fitted with several thermocouples with a 100-mm interval. The mass flux varied from 50 kg/m^{2}s to 180 kg/m^{2}s and the heat flux varied from 5 kW/m^2 to 20 kW/m^2. The saturation pressure of the refrigerant was used to determine the saturation temperature, and it was measured by using a pressure gauge at the inlet and outlet of the test section. The sight glass was fitted at the inlet and outlet to visualize the flow.

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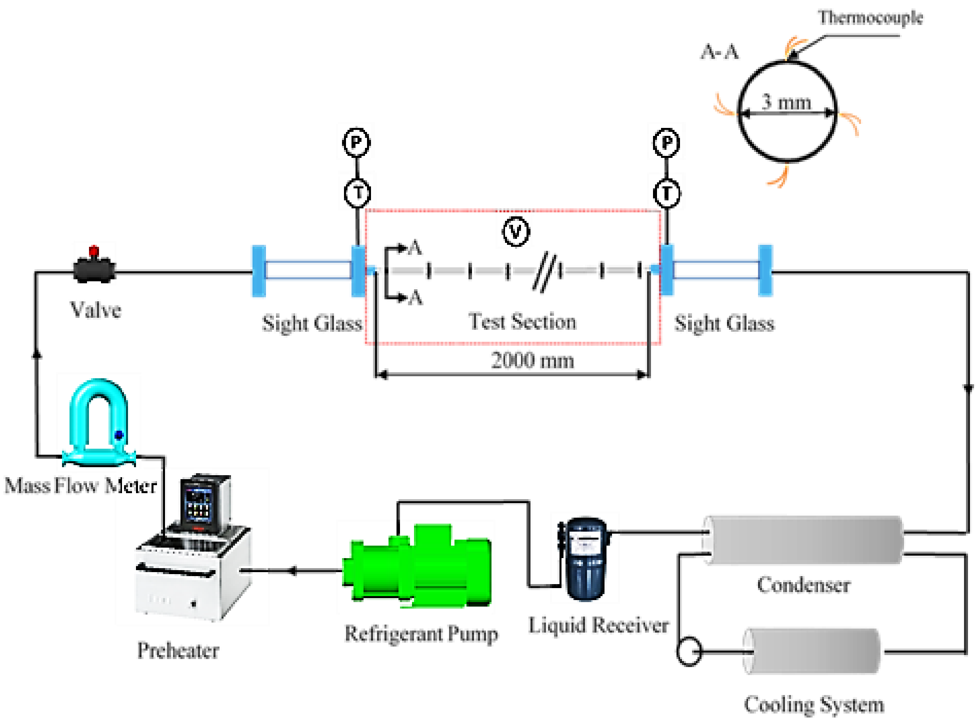

Fig. 1. Experimental set up [21]

The temperature and the pressure data were recorded with the data acquisition. Table 1 from study done by Pamitran *et al.,* [21] shows the parameter's uncertainties consisting of the temperature, pressure, mass flux, heat flux, and vapor quality. The uncertainties were obtained using both random and systematic errors, which varied according to the conditions of the working fluid flow. Therefore, the minimum and maximum error ranges were also displayed.

Table 1
Parameter uncertainties [21]

Parameters	Uncertainties
Temperature $\boldsymbol{T}$ (%)	$\pm$ 0.18 - $\pm$ 5.58
Pressure $\boldsymbol{P}$ (kPa)	$\pm$ 2.5
Mass flux $\boldsymbol{G}$ (%)	$\pm$ 1.85 - $\pm$ 9.78
Heat flux $\boldsymbol{q}$ (%)	$\pm$ 1.67 - $\pm$ 3.58
Vapor quality $\boldsymbol{x}$ (%)	$\pm$ 1.79 - $\pm$ 9.82

2.2 Data Reduction

The vapor quality x in the test section was calculated by using Eq. (1).

$$x = \frac{i - i_l}{i_{lv}} \tag{1}$$

where i is the enthalpy, the subscript l and v is the condition of the liquid and vapor at the saturation temperature, respectively. The subcooled length z_{sc} in m is used to determine the starting point of the saturation calculated by using Eq. (2).

$$z_{sc} = L\frac{i_l - i_{l,in}}{(Q/W)} \tag{2}$$

where L is the tube length, Q is the electricity power, and W is the mass flow rate. The experimental heat transfer coefficient h along the test section shall be obtained by using Eq. (3).

$$h = \frac{q}{T_{wi} - T_{sat}} \tag{3}$$

where q in kW/m^2 is the heat flux, T is the temperature in K, the subscript *wi* is the inner wall of the test section, and *sat* is the saturation condition. Meanwhile, the experimental pressure drop was obtained by employing the pressure difference between the pressure gauge at the inlet and that of the outlet of the test section.

2.3 Correlation Analysis

Several correlations from the other 20 authors would be used to predict the experimental data. All the employed correlations were modified and adjusted from the data of their studies. They were based on each of the heat transfer coefficients and their pressure drop correlation model. The conditions and the refrigerants of the studies used by those authors were different, too. The mass flux varied from 12.4 kg/m^{2}s to 11071 kg/m^{2}s. The heat flux varied from 0.1 kW/m^2 to 2620 kW/m^2, and the refrigerant varied from R290 to CO_2. The employed test section varied from 0.1 mm to 32 mm, and it was confirmed that the shape of the test section ranged from a microchannel to a conventional tube. The all-previous cited correlation was developed with variation of mass flux and heat flux. Many refrigerants have been used, including synthetics and natural refrigerants, such as R290. Table 2 from the previous studies [1,18,22-29] and Table 3 from studies [30-39] show the correlation of the heat transfer coefficient and the pressure drop used in this study.

Table 2
The heat transfer coefficient correlations used in this study

Authors	Equations
Shah [22] (Enhancement factor)	$h_{tp} = F \cdot h_{cb}$ $h_{cb} = 0.023 . Re_l^{0.8} . Pr_l^{0.4} \frac{k_l}{D}$ $F = fn(Bo, N)$ and $N = fn(Co, Fr_l)$ R11, R12, R22, R502 D_{in}: 0.1-26.4 mm G: 50-11071 kg/m^{2}s q: 1- 12 kW/m^2
Cooper [40] (Nucleate boiling model)	$h_{nb} = 55 {p_r}^{0.12} (-ln p_r)^{-0.55} M^{-0.5} q^{0.67}$ R22, R21, R13, R12, R11 D_{in}: 5.02- 15.8 mm q: 0.1-1000 kW/m^2
Kandlikar [24] (The Largest Mechanism Predominant model)	$h_{tp} = max\ (h_{nb}, h_{cb})$ $h_{nb} = 0.6683 . Co^{-0.2} . (25 . Fr_l)^{0.3} + 1058 . Bo^{0.7} . 0.023 . Re_l^{0.8} . Pr_l^{0.4} \frac{k_l}{D}$ $h_{cb} = 1.136 . Co^{-0.9} . (25 . Fr_l)^{0.3} + 667.2 . Bo^{0.7} . 0.023 . Re_l^{0.8} . Pr_l^{0.4} \frac{k_l}{D}$ 8 different refrigerants D_{in}: 3-20.5 mm G: 13-8179 kg/m^{2}s q: 0.3-2280 kW/m^2
Liu and Winterton [25] (Asymptotic model)	$h_{tp} = [(S \cdot h_{nb})^2 + (F \cdot h_{cb})^2]^{\frac{1}{2}}$ $h_{nb} = h_{Cooper}$ $h_{cb} = 0.023 . Re_{lo}^{0.8} . Pr_l^{0.4} \frac{k_l}{D}$ $S = fn(F, Re_{lo})$ and $F = fn(x, Pr_l, \rho_l, \rho_v)$

	9 different refrigerants D_{in}: 2.95-32 mm G: 12.4-8179.3 kg/m^{2}s q: 0.38-2620 kW/m^2
Zhang *et al.,* [26] (Superposition model)	$h_{tp} = S\,.\,h_{nb} + F\,.\,h_{cb}$ $h_{nb} = 0.00122\left[\frac{k_l^{0.79}.\,C_{pl}^{0.45}.\,\rho_l^{0.49}}{\sigma^{0.5}.\,\mu_l^{0.29}.\,i_{lv}^{0.24}.\,\rho_v^{0.24}}\right].\Delta T_{sat}^{0.24}.\Delta P_{sat}^{0.75}$ If $Re_l < 2300$, so: $h_{cb} = \frac{k_l}{D_h}.max\left(Nu'_{sp,v}, Nu_{sp,t}\right)$ If $Re_l \geq 2300$, so: $h_{cb} = \frac{k_l}{D_h}.Nu_{sp,t}$ $S = fn(Re_l)$ and $F = fn(C, X)$ Water, R11, R12, R113 D_{in}: 0.78-6 mm G: 23.4-2939 kg/m^{2}s q: 2.95-2511 kW/m^2
Bertsch *et al.,* [27] (Superposition model)	$h_{tp} = S\,.\,h_{nb} + F\,.\,h_{cb}$ $h_{nb} = h_{Cooper}$ $h_{cb} = h_{conv,l}.(1-x) + h_{conv,v}.x$ $S = fn(x)$ and $F = fn(x)$ 12 different refrigerants D_{in}: 0.16-3 mm G: 20-3000 kg/m^{2}s q: 0.4-115 kW/m^2
Kim and Issam [28] (Asymptotic model)	$h_{tp} = [(h_{nb})^2 + (h_{cb})^2]^{\frac{1}{2}}$ $h_{nb} = \left[2345.\left(Bo.\frac{P_H}{P_F}\right)^{0.70}.p_r^{\,0.38}(1-x)^{-0.51}\right].0.023.Re_l^{0.8}.Pr_l^{0.4}\frac{k_l}{D_h}$ $h_{cb} = \left[5.2.\left(Bo.\frac{P_H}{P_F}\right)^{0.08}.We_{lo}^{-0.54} +\right.$ $\left.3.5.\left(\frac{1}{X_{tt}}\right)^{0.94}\left(\frac{\rho_v}{\rho_l}\right)^{0.25}\right].0.023.Re_l^{0.8}.Pr_l^{0.4}\frac{k_l}{D_h}$ 18 different refrigerants D_{in}: 0.19-6.5 mm G: 19-1608 kg/m^{2}s q: 5-50 kW/m^2
Turgut and Mustafa [29] (Asymptotic model)	$h_{tp} = [(h_{nb})^3 + (h_{cb})^3]^{\frac{1}{3}}$ $h_{nb} = 0.5740708489546.h_{Cooper}^{1.0097850898120}.p_r^{\,-0.673980526542}.(1-$ $x)^{0.1260865942051}$ $h_{cb} = \left[1.9246533128433.\left(\frac{1}{Co}\right)^{0.9518395312523}\right].0.023.Re_l^{0.8}.Pr_l^{0.4}\frac{k_l}{D_h}$ R744 (CO_2) D_{in}: 0.5-9.52 mm G: 50-1400 kg/m^{2}s q: 5-40 kW/m^2
Aizuddin *et al.,* [1] (Asymptotic model)	$h_{tp} = [(h_{nb})^2 + (h_{cb})^2]^{\frac{1}{2}}$ $h_{nb} = \left[2215.\left(Bo.\frac{P_H}{P_F}\right)^{0.88}.p_r^{\,-0.43}(1-x)\right].0.023.Re_l^{0.8}.Pr_l^{0.4}\frac{k_l}{D_h}$ $h_{cb} = \left[0.31.\left(Bo.\frac{P_H}{P_F}\right)^{-0.03}.We_{lo}^{-0.85} +\right.$ $\left.8.92.\left(\frac{1}{X_{tt}}\right)^{0.775}\left(\frac{\rho_v}{\rho_l}\right)^{0.33}\right].0.023.Re_l^{0.8}.Pr_l^{0.4}\frac{k_l}{D_h}$ R290 D_{in}: 3 mm G: 100, 150, and 200 kg/m^{2}s q: 5, 10, and 15 kW/m^2

Turgut and Mustafa [18] (Asymptotic model)	$h_{tp} = [(h_{nb})^{2.695516415880346} + (h_{cb})^{2.695516415880346}]^{2.695516415880346}$ $h_{nb} = 66.636181187049520 p_r^{1.244926529779103}$ $(-ln p_r)^{0.258952076070707} M^{0.399836377153093} q^{0.505546027893485}$ $h_{cb} = [1 + 1.499118607477336. X_{tt}^{-1.040878186584161}]$ $0.551669381417827. Re_l^{0.185318440184329}. Pr_l^{0.354519104766204} \frac{k_l}{D_h}$ R290 D_{in}: 0.3-7.7 mm G: 50-600 kg/m^{2}s q: 2.5-227 kW/m^2

Table 3
The pressure drop correlations used in this study

Authors	Equations
Beattie and Whalley [30] (Homogeneous flow model)	$\left(\frac{dp}{dz}\right) = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_z + \left(\frac{dp}{dz}\right)_a$ $\beta = \frac{\rho_l x}{\rho_l x + \rho_g (1-x)}$ $\mu_{tp} = \mu_l (1-\beta)(1+2.5\beta) + \mu_g \beta$
Tran *et al.,* [31] (Separated flow model)	$\left(\frac{dp}{dz}\right) = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_z + \left(\frac{dp}{dz}\right)_a$ $\emptyset_{lo}^2 = 1 + (4.3X^2 - 1)[N_{conf} X^{0.875}(1-x)^{0.875} + x^{1.75}]$ $\Delta p_l = \Delta p_{lo} . \emptyset_{lo}^2$ R134a, R12, R113 D_{in}: 2.46 and 2.92 mm G: 33-832 kg/m^{2}s q: 2.2-129 kW/m^2
Zhang and Ralph [32] (Separated flow model)	$\left(\frac{dp}{dz}\right) = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_z + \left(\frac{dp}{dz}\right)_a$ If $x = 0$, so: $\emptyset_{lo}^2 = 1$ If $x > 0$, so: $\emptyset_{lo}^2 = (1-x)^2 + 2.87x^2 \left(\frac{P}{P_c}\right)^{-1} + 1.68x^{0,8}(1-x)^{0.25}\left(\frac{P}{P_c}\right)^{-1.64}$ R134a, R22, R404a D_{in}: 3.25 and 6.25 mm G: 200-1000 kg/m^{2}s
Miyara *et al.,* [33] (Separated flow model)	$\left(\frac{dp}{dz}\right) = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_z + \left(\frac{dp}{dz}\right)_a$ $C = 21\{1 - exp(-0.28Bd^{0.5})\}\{1 - 0.9\exp(-0.02Fr^{1.5})\}$ $n = 1 - 0.7\exp(-0.08Fr)$ $\emptyset_v^2 = 1 + CX_{tt}^n + X_{tt}^2$ Water, R744 (CO_2) D_{in}: 1-4 mm G: 50-200 kg/m^{2}s
Hwang and Min [34] (Separated flow model)	$\left(\frac{dp}{dz}\right) = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_z + \left(\frac{dp}{dz}\right)_a$ $C = 0.227 Re_{lo}^{0.452} X^{-0.32} N_{conf}^{-0.82}$ R134a, R22, R404a D_{in}: 0.24, 0.43, and 0.79 mm G: 140-930 kg/m^{2}s
Park and Hrnjak [35] (Separated flow model)	$\left(\frac{dp}{dz}\right) = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_z + \left(\frac{dp}{dz}\right)_a$ $\left(\frac{dp}{dz}\right)_{lo} = 0.079\left(\frac{\mu_l}{GD}\right)^{0.25}\left(\frac{2G^2}{D\rho_l}\right)$ $\left(\frac{dp}{dz}\right)_{vo} = 0.079\left(\frac{\mu_v}{GD}\right)^{0.25}\left(\frac{2G^2}{D\rho_v}\right)$ $\left(\frac{dp}{dz}\right)_F = \left(\frac{dp}{dz}\right)_{lo}(1-2x)(1-x)^{1/3} + \left(\frac{dp}{dz}\right)_{vo}[2x(1-x)^{1/3} + x^3]$ R744 (CO_2), R410A, R22

	D_{in}: 6.1 mm G: 100-400 kg/m^{2}s q: 5-15 kW/m^2
Sun and Kaichiro [36] (Separated flow model)	$\left(\frac{dp}{dz}\right) = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_z + \left(\frac{dp}{dz}\right)_a$ If $Re_{lo} < 2000$, so: $C = 26(1 + \frac{Re_l}{1000})\left[1 - exp\left(\frac{-0.153}{0.27La+0.8}\right)\right]$ $\emptyset_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$ If $Re_{lo}\ and\ Re_{vo} > 2000$, so: $C = 1.79\left(\frac{Re_v}{Re_l}\right)^{0.4}\left(\frac{1-x}{x}\right)^{0.5}$ $\emptyset_l^2 = 1 + \frac{C}{X^{1.19}} + \frac{1}{X^2}$ 12 different refrigerants D_{in}: 0.5-6 mm G: 198-1080 kg/m^{2}s q: 10.6-17 kW/m^2
Li and Zan [37] (Separated flow model)	$\left(\frac{dp}{dz}\right) = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_z + \left(\frac{dp}{dz}\right)_a$ If $Bd \leq 1.5$ *, so:* $C = 11.9Bd^{0.45}$ If $1.5 \leq Bd \leq 11$ *, so:* $C = 109.4(Bd.Re_l^{0.5})^{-0.6}$ 12 different refrigerants D_{in}: 0.14-3 mm G: 50-2000 kg/m^{2}s
Li and Takashi [38] (Separated flow model)	$\left(\frac{dp}{dz}\right) = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_z + \left(\frac{dp}{dz}\right)_a$ If $Re_l\ and\ Re_v < 2000\ (turbulent - turbulent)$, so: $C = 6.28N\mu_{tp}^{0.78}Re_{tp}^{0.67}x^{0.32}$ If $Re_l > 2000\ and\ Re_v < 1000\ (turbulent - laminar)$, so : $C = 1.54N\mu_{tp}^{0.14}Re_{tp}^{0.52}x^{0.42}$ If $Re_l < 1000\ and\ Re_v > 2000\ (laminar - turbulent)$, so : $C = 245.5N\mu_{tp}^{0.75}Re_{tp}^{0.35}x^{0.54}$ If $Re_l\ and\ Re_v < 1000\ (laminar - laminar)$, so: $C = 41.7N\mu_{tp}^{0.66}Re_{tp}^{0.42}x^{0.21}$ 10 different refrigerants D_{in}: 0.1-3 mm G: 50-2000 kg/m^{2}s q: 5- 500 kW/m^2
Maher *et al.,* [39] (Homogeneous flow model)	$\left(\frac{dp}{dz}\right) = \left(\frac{dp}{dz}\right)_F + \left(\frac{dp}{dz}\right)_z + \left(\frac{dp}{dz}\right)_a$ $\mu_{tp} = \left[(1-x)\mu_f + x\mu_g\right]^{0.94}\left(\frac{1-x}{\mu_l} + \frac{x}{\mu_g}\right)^{-0.94}$ $f_{tp} = \left(0.79Re_{tp}^{-0.25}\right)^{1.4} + \left\{0.17\left\{0.69ln\left(Re_{tp}\right) - 2.2\right\}^{-1.5}\right\}^{1/0.7}$ $\left(\frac{dp}{dz}\right)_{tp} = \frac{G_{tp}^2 \cdot f_{tp}}{2.D_{in} \cdot \rho_{tp}}$ 7 different refrigerants D_{in}: 0.52-8 mm G: 35.5- 1600 kg/m^{2}s

3. Result and Discussion

The experimental data showed that the heat transfer coefficient and the pressure drop were affected by the two-phase Reynolds Number. Figure 2 shows the effects of the two-phase Reynolds Number on the experimental heat transfer coefficient data with heat flux amounting to 15 kW/m^2.

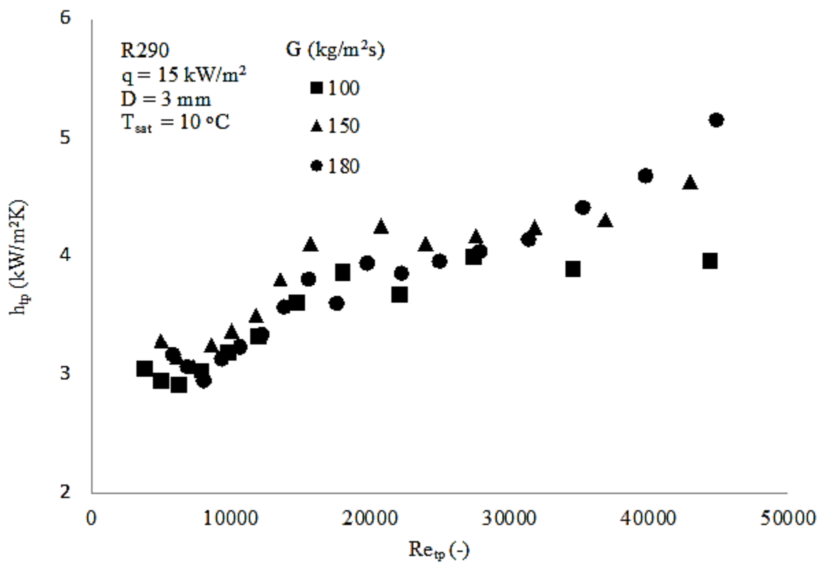

Fig. 2. Effect of two-phase Reynolds number on the heat transfer coefficient

The higher the two-phase Reynolds number was, the higher the heat transfer coefficient would be. However, at a lower two-phase Reynolds number, the effects of the mass flux were not too significant. The nucleate boiling mechanisms dominated the heat transfer process, often occurring in a mini-channel tube [41]. Figure 3 shows the effects of the two-phase Reynolds number on the pressure drop with the heat flux amounting to 15 kW/m^2. The pressure drop would become higher when a high two-phase Reynolds number occurred. The effects of the mass flux were also insignificant at a lower two-phase Reynolds Number. Then, the pressure drop would increase as the mass flux increased.

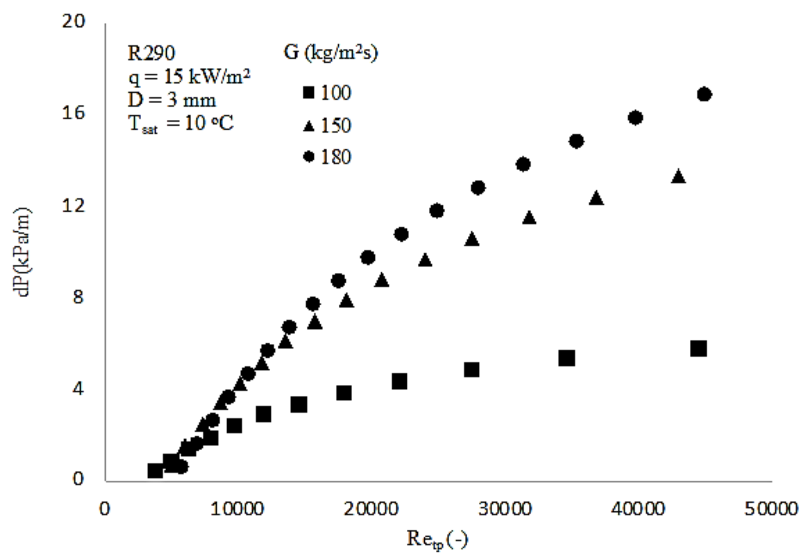

Fig. 3. Effect of two-phase Reynolds number on the pressure drop

In this study, Table 4 from the correlations of [1,18,22,24-29] shows the deviation of the heat transfer coefficient prediction. Moreover, Figure 4 shows the selected comparison between the heat transfer coefficient and the existing correlation. Overall, Aizuddin *et al.,*'s [1] correlation gave the best prediction. The correlation was an asymptotic model that developed with R290, mass flux of 100 up to 200 kg/m^{2}s, and heat flux of 5 up to 15 kW/m^2. Those experimental conditions were very close with those of the present study. Therefore, this correlation was good to predict the heat transfer coefficient with the present data. Shah's correlation [22] was the second-best heat transfer coefficient correlation in an Enhancement Factor model. This correlation was a further development of his previous study where he predicted the heat transfer coefficient by using a graph adjusted to simplify the calculation process [42]. Shah's basic equation divided the Enhancement Factor F into many conditions based on the nucleate and convective boiling mechanism. With this division, this correlation could predict the heat transfer coefficient well at various qualities of the vapor.

Table 4
Deviation of the heat transfer coefficient correlations

Correlations	Mean absolute deviation (%)
Aizuddin *et al.,* [1]	14.07
Shah [22]	14.72
Bertsch *et al.,* [27]	19.29
Kim and Mudawar [28]	25.06
Zhang *et al.,* [26]	27.89
Turgut and Coban [17]	40.32
Kandlikar [24]	43.82
Liu and Winterton [25]	55.27
Turgut and Asker [29]	77.51

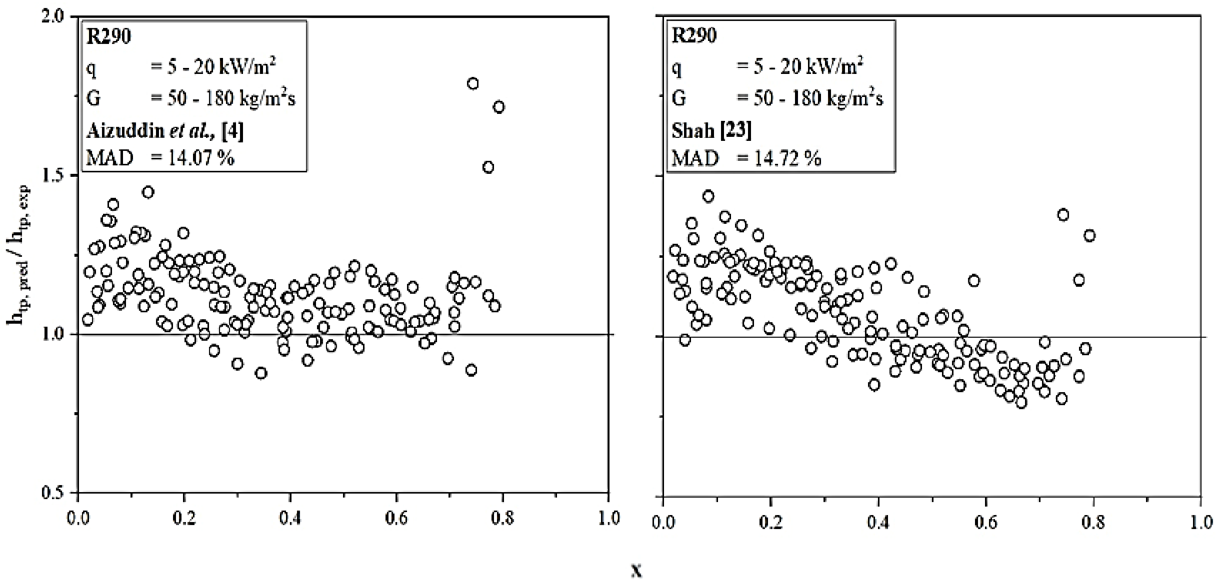

Fig. 4. Selected comparison between heat transfer coefficient with existing correlations

Table 5 from the correlations of [30-33,35-39] shows the deviation in the prediction of the pressure drop. Moreover, Figure 5 shows the selected comparison between the pressure drop and the existing correlation. Sun & Mishima's correlation [36] provided the best prediction. This correlation was a separated flow model developed with 12 different refrigerants on condition that the mass flux ranged from 198 kg/m^{2}s to 1080 kg/m^{2}s, by modifying Chisholm's Parameter. With this modification, this correlation could predict the pressure drop under wider various conditions of the mass flux. Li's and Wu's correlation [37] was the second-best pressure drop correlation in the Separated Flow Model. This correlation was developed with 12 different refrigerants and based on the condition that the mass flux ranged from 500 kg/m^{2}s to 2000 kg/m^{2}s. Li and Wu modified the Chisholm Parameter, which was divided based on Bond Number's condition. With this division, this correlation could predict the pressure drop under various conditions of surface tension.

Table 5
Deviation of the heat pressure drop correlations

Correlation	Mean absolute deviation (%)
Sun and Mishima [36]	27.64
Li and Wu [37]	34.05
Miyara *et al.,* [33]	34.81
Park and Hrnjak [35]	43.68
Maher *et al.,* [39]	45.69
Li and Hibiki [38]	47.57
Beattie and Whalley [30]	57.32
Zhang and Webb [32]	57.79
Tran *et al.,* [31]	69.07

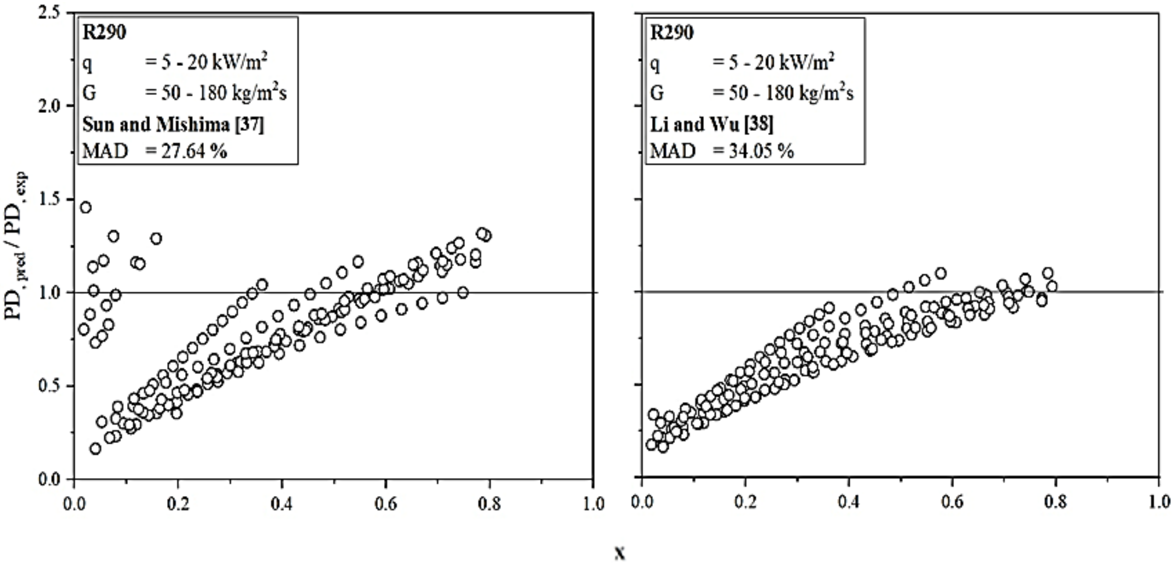

Fig. 5. Selected comparison between pressure with existing correlations

Ghiaasiaan [19] stated that the flow pattern map was a method most frequently used to predict a two-phase flow pattern. The shape of the flow pattern map is usually an empirical graph with various coordinate axes for different parameters in each version. This study used the flow pattern maps from Wojtan *et al.,* [2] and Zhuang *et al.,* [20] to predict the flow pattern in the experimental data. Figure 6 shows Wojtan et al.,'s [2] flow pattern map with the heat flux amounting to 15 kW/m^2. It was observed that mass flux influenced the flow pattern map. The flow pattern would go through the intermittent and annular regimes in the experimental data with the mass flux amounting to 100 kg/m^{2}s. Then, it would enter the stratified-wavy regime before entering the dry out regime. Meanwhile, the flow pattern only entered the intermittent and annular regimes if the max flux amounted to 150 kg/m^{2}s or 180 kg/m^{2}s.

Figure 7 shows Zhuang *et al.,*'s [20] flow pattern map with the heat flux of 15 kW/m^2. It was observed that the Weber Number influenced the flow pattern map. In the experimental data, the Weber Number was lower with the mass fluxes of 100 kg/m^{2}s and 150 kg/m^{2}s. It shows that the flow pattern would enter the transition regime before entering the wavy-annular and smooth-annular regimes. Meanwhile, the flow pattern is the wavy-annular and smooth-annular regimes for a higher Weber Number with the mass flux of 180 kg/m^{2}s. Weber Number is a function of the liquid and vapor Reynolds Number. Higher mass flux and vapor quality are directly proportional to the higher liquid and vapor Reynolds Number, causes the refrigerant flow became more turbulent in both the liquid and vapor phases.

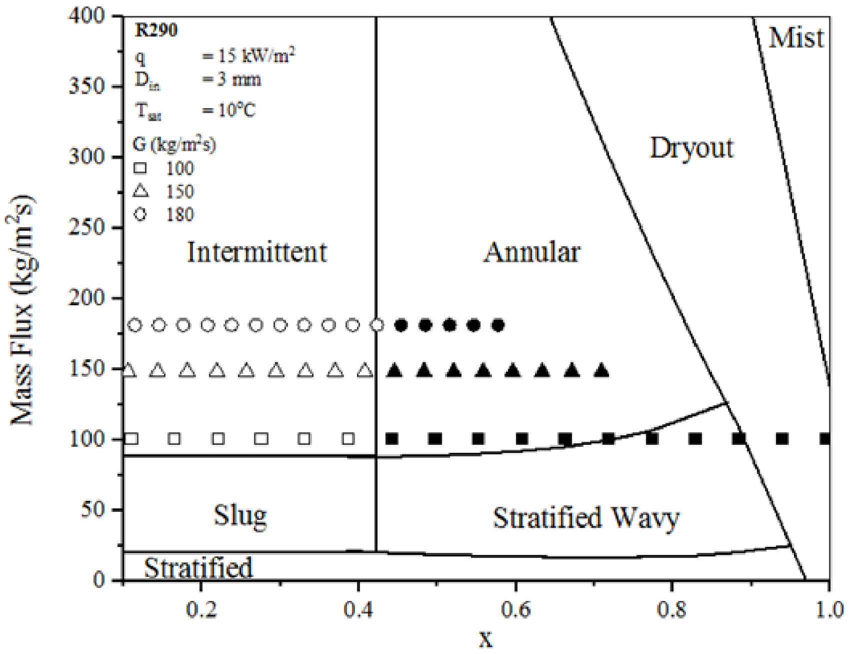

Fig. 6. Wojtan *et al.*,'s [2] flow pattern map with 15 kW/m^2 of heat flux

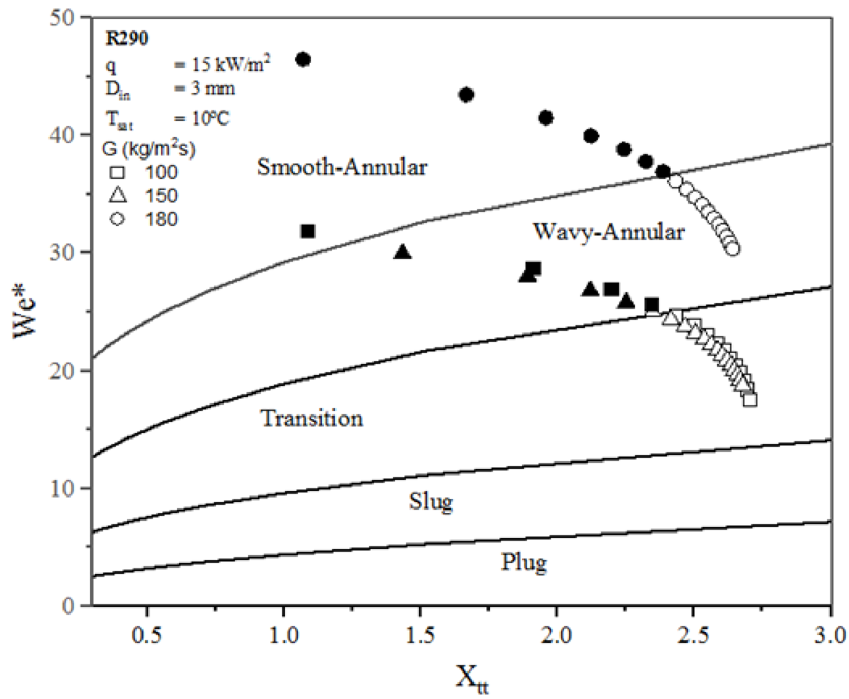

Fig. 7. Zhuang *et al.*,'s [20] flow pattern map with 15 kW/m^2 of heat flux

Based on the two flow pattern maps shown in Figures 6 and 7, it was known that higher mass flux caused a lower vapor quality at the outlet of the test section, which affected the flow pattern and its transition. Under the mass flux of 180 kg/m^2s, the flow transition only passed through the intermittent and annular regimes in the Wojtan *et al.*,'s [2] flow pattern map. In the flow pattern map of Zhuang *et al.*, [20] the flow transition is primarily only in the annular regime. The results of the flow pattern transition, which looks equally short, are predicted because of the large liquid and vapor Reynolds Numbers at a certain vapor quality, causing the fluid flow to show a more significant inertial effect. The flow pattern maps obtained from Wojtan *et al.*, [2] and Zhuang *et al.*, [20] were developed by using a larger inner diameter of the test section than that of the test section used in this study. Hence, this would result in a delay when it is used to predict the experimental data. Table 6 shows the results of the delay for each of the flow pattern maps.

Table 6
Delay result of the flow pattern maps

Mass flux	(x) Wojtan *et al.,* [2]		(X_{tt}) Zhuang *et al.,* [20]		Experimental data result	
(kg/m^{2}s	Annular	Dry out	Wavy-annular	Annular	Dry out	Wavy-annular
100	0.42	0.82	3.04	0.38	0.82	1.52
150	0.42	0.77	3.04	0.4	-	2.4
180	0.42	0.71	3.04	0.42	-	-

Wojtan *et al.,*'s [2] flow pattern map showed a slight delay and would be smaller when the mass flux reached a higher value. Then, Zhuang *et al.,*'s [20] flow pattern map had a significant delay, but the delay was lower when the mass flux reached a higher value. Overall, both of the two flow pattern maps really produced a delay in the experimental data due to the presence of a small inner diameter in the test section. However, it was deemed to be tolerable because the value was not too high.

4. Conclusions

Based on the results of the experimental data and the prediction of the R290 refrigerant heat transfer coefficient, several conclusions can be drawn. The heat transfer coefficient and the pressure drop were influenced by two-phase Reynolds number. The higher two-phase Reynolds number is, the higher the heat transfer coefficient will be. Moreover, the effects of the mass flux were insignificant at a low two-phase Reynolds number. Likewise, the higher two-phase Reynolds number is, the higher the pressure drop will be. The correlation of Aizuddin *et al.,* [1] obtained from the asymptotic model is the best correlation used to predict the heat transfer coefficient with a 14.07-% mean absolute deviation. Moreover, the correlation of Sun and Kaichiro [36] obtained from the separated flow model is the best correlation used to predict the pressure drop with a 27.64-% mean absolute deviation. The flow pattern maps of Wojtan *et al.,* [2] and Zhuang *et al.,* [20] have a good ability and suitability to predict the experimental data with a tolerable delay.

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