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Article

Heat Transfer Coefficient Characteristic Study of Natural Refrigerant with Substitute for R-134a

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Abstract. The wide-spread use of halocarbon refrigerants are making negative impact on Earth. Natural Refrigerant, such as hydrocarbon, is one alternative of several option to use. Mixing hydrocarbon are develop to improve the heat transfer characteristic. For example is, Musicool-134 (MC-134) is a mixture with two major substance of propane and iso-butane. The experimental apparatus is using a microchannel with a diameter of 0.5 mm and length of 0.5 m. The evaporative process was conducted in the experiment. The result of the experiment is that if the high coefficient value then the heat flux value is also high.

Keywords: Heat transfer coeffecient, natural refrigerant.

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Nomenclature

Bo	boiling number (-)
C	Chishlom parameter (-)
C_{sf}	Rohsenow correlation (-)
$C_{p,l}$	heat capacity (J/kg.K)
Co	confinement number (-)
d	diameter of pipe (m)
d_h	hydraulic diameter (m)
F	multiplier factor (-)
f_l	friction factor of liquid
f_v	friction factor of vapour
F	correction factor (-)
g	specific gravity (m/s^2)
G	mass flux (kg/m^2.s)
h	heat transfer coefficient (W/m^2.^{o}C)
h_{fg}	latent heat (J/kg)
h_{tp}	two-phase heat transfer coefficient (W/m^2.^{o}C)
h_{nb}	nucleate boiling heat transfer coefficient (W/m^2.^{o}C)
h_f	convective heat transfer coefficient (W/m^2.^{o}C)
i_f	enthalpy of saturation liquid (kJ/kg)
i_{fi}	enthalpy of liquid inlet (kJ/kg)
Δi	increasing of enthalpy (kJ/kg)
k	thermal conductivity (W/m.K)
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt number (-)
Pr	Prandtl number (-)
$\ddot{q}$	heat flux (W/m^2)
q_{rev}	heat received (W)
Re_{TP}	Reynolds number (-)
S	suppression factor (-)
T_{in}	outlet temperature (^{o}C)
T_{out}	inlet temperature (^{o}C)
X	Martinelli parameter (-)
x	mass vapor quality (-)
x_o	out let mass vapor quality (-)
$(1-x)$	mass liquid quality (-)
Z_{sc}	sub-cooled length (m)

Greek symbols

ρ_v	vapor density of refrigerant (kg/m^3)
ρ_l	liquid density of refrigerant (kg/m^3)
μ_l	liquid viscosity of refrigerant (Pa.s)
μ_g	gas viscosity of refrigerant (Pa.s)
σ	surface tension (N/m)
ϕ_f	two-phase frictional multiplier gradients

1. Introduction

Nowadays, peoples are using cooling system for everyday use. The refrigerant is use as a working fluid in a cooling system. However, the use of refrigerant such as CFC, HFC, and HCFC are proven to damage the earth, caused by the high number of Global Warming Potential (GWP) and Ozone Depletion Potential (ODP). Therefore, it is necessary to have an alternative of halocarbon refrigerant. One of the alternative is the use of natural refrigerant.

Natural Refrigerant are proven to have the lowest GWP and ODP. Bolaji & Huan (2013) reported that due to the impact of CFC and HCFCs, using natural refrigerant more suitable since it is already circulating in the biosphere and known to be harmless [1]. In addition, natural refrigerants cannot undergo reaction with water since the refrigerant is not containing any chlorine or fluorine atoms. Thus, the refrigerant cannot form any strong acids that can lead to premature system failure.

Hydrocarbons are one of the alternate refrigerant, such as propane, pentane and butane. Hydrocarbons are suitable to be a refrigerant because of their energy efficiency, critical point, solubility, transport and heat transfer properties [1]. Researchers are developing natural refrigerants based on hydrocarbon mixtures. Austin, Kumar, & Kanthavelkumaran (2012) researched about propane-butane as mixed refrigerant [2]. Jwo, Ting, & Wang (2009) also reported similar result, with a propane (R-290) and isobutane (R-600a) mixture [3]. Both of the experiments showed that the mixture have better performance when compared to HFC-134a.

Fang et al. (2019) review some literature of saturated flow boiling heat transfer coefficients [4]. The correlation of heat transfer coefficients may be classified as seven categories. The first model is enhancement factor type model. Heat transfer coefficient of flow boiling in this model may be reduce to

$$h_{tp} = \psi h_l \tag{1}$$

The enhancement factor (ψ) elucidate for heat transfer enhancement due to boiling.

The second model is nucleate boiling model. Heat transfer coefficient of flow boiling in this model is dominated by nucleate boiling mechanisms. Hamdar et al. (2010) reviewed nucleate boiling model in microchannel with inner diameter 1 mm length and R -152a as working fluids [5]. Nucleate boiling heat transfer coefficient as function of dimensionless number of Boiling number and Weber number. Sun and Mishima (2009) reviewed nucleate boiling model with 0.21 mm to 6.05 mm tube diameter with eleven refrigerants [6]. The correlation used Reynolds number, Boiling number and Weber number as function of dimensionless number. Lazarek and Black (1982, Tran et al. (1996), and Yu et al. (2002) also used dimensionless number for proposed nucleate heat transfer correlation [7, 8, 9]. Malek and Colin, (1983) and Stephan (1992) used NH_3 as working fluids in experiment [10, 11]. They proposed nucleate boiling heat transfer correlation too. Some literature suggested correlation for application of nucleate heat transfer prediction [12].

The third model is superposition model. At the first time, the formulation of superposition model of two-phase flow heat transfer was done by Chen (1966) [13]. A superposition model is also called Chen-type model. Two-phase heat transfer can be calculated from convective heat transfer and nucleate boiling heat transfer. As addition of correlation, Chen entered suppression factor (*S*) on nucleate boiling heat transfer and correction factor (*F*) on convective heat transfer.

In additional to Chen, some researcher has reviewed the superposition model. Gungor and Winterton (1986) adopted Chen concept and add the new correlation of suppression factor and correction factor based on database of 4300 data point, including data for R-11, R-12, R-113, R-114, ethylene glycol and water [14]. Bertsch et al. (2009) used 3899 data point from 11 fluids: R-134a, R-236fa, R-245fa, R-410A, R-141b, R-123, R-113, R-11, FC-77, nitrogen, and water [15]. Bertsch et al. (2009 developed convective heat transfer as in part of superposition model [15]. Zhang et al. (2004) modified the Chen correlation by replacing suppression factor (*S*) as function of liquid Reynolds number, and correction factor (*F*) as function of two-phase multiplier [16]. Convective heat transfer coefficient from Zhang depend of vertical or horizontal orientation of tube. The experimental data has been taken from 1023 data point of water, R-11, R-12 and R-113 on inner diameter test section 0.78 – 6 mm.

Jung et al. (1989) modified suppression factor (*S*) and correction factor (*F*) correlation as function boiling number and Martinelli parameter (X_{tt}) [17]. Two-phase heat transfer coefficient was calculated from convective heat transfer from Dittus-boelter and nucleate boiling heat transfer from Jung et al correlation. Saitoh et al. (2007) considered two-phase heat transfer for pre-dryout and post dryout separately [18]. The experiment was conducted from microchannel (d = 0.51 mm) to macrochannel (d = 10.92 mm) with R-134a.

The fourth model is asymptotic model. The asymptotic model correlation was expressed as follows:

$$h_{tp} = \left[(S.h_{nb})^n + \left(F.h_f\right)^n\right]^{1/n} \quad (2)$$

Liu and Winterton (1991) proposed suppression factor (*S*) and correction factor (*F*) as function of Reynolds number and Pradtl number for two-phase heat transfer coefficient with n value is 2 [19]. Nucleate boiling heat transfer and convective heat transfer are calculated with Cooper and Dittus-boelter correlation. Kim and Mudawar (2013) conducted research on mini/microchannel with big data. Based on the database containing 10,805 pre-dryout data for flow boiling heat transfer, the correlation of suppression factor (*S*) and correction factor (*F*) has been developed as function of dimensionless number of Boiling and Weber number [20].

Wattelet (1994) researched two-phase heat transfer coefficient in conventional tube with inner diameter 7.04 mm and using R-12, R-134a and a mixture flowing [21]. Two-phase heat transfer coefficient used nucleate boiling heat transfer without suppression factor (*S*), but Wattelet added R parameter at convective heat transfer and correction factor (*F*) as function of Martinelli parameter (X). The value of n that used to asymptotic model is 2.5.

Steiner and Taborek (1992) still proposed calculating of two-phase heat transfer with asymptotic model [22]. The correlation used nucleate boiling heat transfer from Gorenflo (1993) [23] correlation and convective heat transfer from Gnielinski (1976) [24]. Based on experimental data of natural matter, refrigerants and cryogens, the value of n that used to asymptotic model is 3.

The fifth model is largest mechanism predominant model. In this model, at first step calculate flow boiling heat transfer coefficient for each mechanism. The result is found from the largest value of calculating. The equation of this model can be written as follows:

$$h_{tp} = max(h_{nb}, h_f) \quad (3)$$

The researchers that used to largest mechanism predominant model include Ducoulombier et al. (2011), Kandlikar (1990), and Kandlikar and Balasubramanian, (2004) [25, 26, 27]. In additional to author above, Shah, (1982) used this model with four correlation of two-phase heat transfer coefficient [28]. Shah (1982) studied experiment with conventional tube and fluids of water, R-11, R-12, R-22, R-113 and cyclohexane [28].

The sixth model is flow pattern based model. This model is interesting approach for predicting two-phase heat transfer. There are some researchers such as Thome and El Hajal (2004), Cheng et al. (2008), Thome et al., (2004) and Wang et al., (2010) that developed prediction of two-phase heat transfer by flow pattern [29, 30, 31, 32].

The seventh model is hybrid model. Yoon et al. (2004) combined an asymptotic model for $x < x_{crit}$ and the flow pattern model for $x \geq x_{crit.}$ [33].

PT. Pertamina develops a new refrigerant called Musicool-134 (MC-134). This refrigerant is a natural refrigerant made to replace R-134a. In this study, the experiment was carried out on this refrigerant to see the characteristics of heat transfer coefficient under certain pressure and heat flux conditions.

2. Experimental Apparatus

The experimental apparatus was built in refrigeration laboratory, Universitas Indonesia. The experiment was conducted for research of flow boiling heat transfer on microchannel. Figure 1 showed the experimental apparatus. The test section used stainless steel microchannel with 0.5 mm diameter and 0.5 meters length.

The working fluid MC-134 enter the test section as liquid phase and came out from the test section as a two-phase flow. The thermocouples divided into 5 sections that located along the top and bottom of the microchannel with an equal distance between each sections. Additional thermocouples are immersed at inlet and outlet to measure temperature difference. The pressure transmitters are also placed at inlet and outlet to measure pressure difference of working fluid. Working fluid that came out from the test section area, will be condensed in the condenser. After that, the fluid in liquid phase are moved and circulated using magnetic pump. Cooling bath is placed after the magnetic pump to maintain working fluid low temperature.

The coriolis flow meter measured the flow rate of working fluid. Preheater is placed before the inlet of test section to adjust the working fluid temperature. Two sight glass are placed at inlet and outlet of the test section as visualization of working fluid phase.

The following Table 1 is the characteristic value from natural refrigerant (MC-134) using NIST Standard Reference Database 32. Version 8.0 [34].

Table 1. Characteristic of Musicool – 134.

Characteristic	Value
Critical Temperature	116.02 ^{0}C
Molar Mass	49.329 kg/kmol
Critical Pressure	42.67 bar
Boiling Temperature	-35.072 ^{0}C

Natural refrigerant as given from the safety data sheet (PT. Pertamina, 2017) [35], have the characteristics. Table 2 show the data.

Table 2. Composition of Natural Refrigerant.

Chemical Substance	Concentration
Ethane	<0.2 %w/w
Propane	52-59.5 %w/w
i-Butane	38.9 - 45.5 %w/w
n-Butane	<2.5 %w/w
Pentane	<100 ppm
n-Hexane	<0.03 %w/w

Uncertainty has been calculated from data experiment. Table 3 showed uncertainty of temperature, flow rate and heat measurement.

Table 3. Measurement Uncertainty.

Variable	Uncertainty
Average temperature (°C)	0.46
Mass flow (%)	0.05
Heat (%)	1

2.1. Data reduction

Experimental diameter is classified as microchannel by Kew and Cornwell (1997) with *Co* number more than 0.5 [36]. *Co* number is represented ratio of capillary length and hydraulic diameter. The correlation of *Co* number is written as follows:

$$Co = \frac{\left[\frac{\sigma}{g(\rho_l - \rho_g)}\right]^{1/2}}{d_h} \tag{4}$$

Evaporation was processed on the test section by electrical heating. Heat is given to working fluids and calculated with heat received equation as follows:

$$q_{rev} = \dot{m}\, C_p (T_{out} - T_{in}) \tag{5}$$

The regime of flow studying consist of single phase flow and two phase flow. The length of initial mass vapor quality (x = 0) is defined as Z sub-cooled. The two phase flow occurred nearly the length of Z sub-cooled. The equation of Z sub-cooled is written as follows:

$$Z_{sc} = \frac{i_f - i_{fi}}{\Delta i} \tag{6}$$

The equation of outlet mass quality is written as follows:

$$x_o = \frac{\Delta i + i_{fi} - i_f}{i_{fg}} \tag{7}$$

3. Theoretical Analysis

The value of heat transfer coefficient on the flowing fluid in the pipe can be known from the heat temperature given per unit area ($\ddot{q}$) of the pipe and also different temperatures that occur between the temperature of pipe surface and the saturation temperature of the flowing fluid. This is as shown in Eq. (8)

$$\mathrm{h} = \frac{\ddot{q}}{T_{wall} - T_{sat}} \tag{8}$$

In addition to these equations, the value of heat transfer coefficient can be predicted through calculations. One method of calculation is using the superposition method. The superposition method is expressed in Eq. (9) [13]

$$\mathrm{h_{TP}} = \mathrm{h_{NB}}\, S + \mathrm{h_f}\, F \tag{9}$$

The heat transfer coefficient of two phase (h_{TP}) is dependent to suppression factor (S) and heat transfer coefficient for nucleate boiling (h_{NB}). In addition, the heat transfer coefficient of two phase is also determined with the F parameter and convective heat transfer coefficient (h_f).

The heat transfer coefficient for nucleate boiling are calculate with the Rohsenow (1951) correlation [37]. The Rohsenow correlation equation are expressed in Eq. (10)

$$\mathrm{h_{nb}} = \frac{1}{\mathrm{C_{sf}}} \frac{\mathrm{C_{p,l}}}{(\mathrm{h_{fg}})^{0.67}} \left[\frac{1}{\mu_l} \sqrt{\frac{\sigma}{\mathrm{g}(\rho_l - \rho_v)}}\right]^{-0.33} (\ddot{\mathrm{q}})^{0.67} (\mathrm{Pr_l})^{-1.7} \tag{10}$$

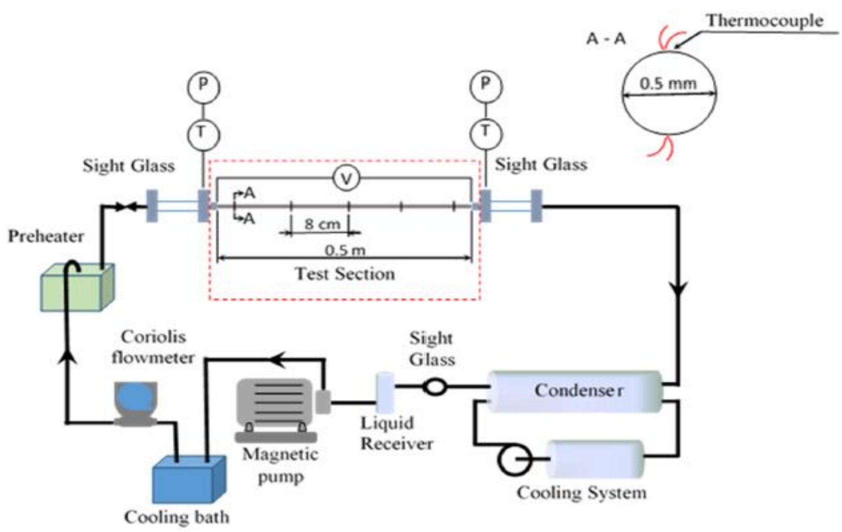

Fig.1. Experimental apparatus.

The surface/liquid parameter of the Rohsenow correlation (C_{sf}) is a coefficient between the refrigerants and the pipe surfaces. The heat capacity, $C_{p,l}$ (J/kg.K); the latent heat, h_{fg} (J/kg); the liquid viscosity, μ_l (Ns/m^2); surface tension, σ (N/m); ρ_l, ρ_v liquid and vapor densities (kg/m^3); g, specific gravity (9.8 m/s^2); and ($\ddot{q}$) heat flux (W/m^2).

The value of convective heat transfer coefficient can be determined from the value of the Nusselt number in the fluid flow, the diameter of the pipe used, and the thermal conductivity of fluid.

$$h_f = \mathrm{Nu}\frac{k}{\bar{d}} \quad (11)$$

The value of the Nusselt number is influenced by the type of fluid flow. Based on Cengel (2003), if the fluid flow is laminar (Re < 2300), the Nusselt number is 4.36 under condition it has constant heat flux on the surface of the pipe [38]. If the fluid flow is turbulent then the Nu value can be determined by the following Dittus and Boelter (1930) equation [39].

$$\mathrm{Nu} = 0.023\,\mathrm{Re_f}^{0.8}\mathrm{Pr_f}^{0.4} \quad (12)$$

The suppression factor and *F* parameter can be determined from the equations that have been made by previous studies. The following Table 4 are the equations to get the suppression factor and *F* parameter from some researchers.

Table 4. Equation of *S* factor and *F* Parameter.

Researcher	Equation for Suppression Factor (*S*) and *F* Parameter
Correlation of Chen (1966) [13]	$S = \dfrac{1}{1 + 2.53 \times 10^{-6}\mathrm{Re_{TP}}^{1.17}}$ $F = (\phi_f^{\,2})^{0.44}$
Correlation of Zhang et al. (2004) [16]	$S = \dfrac{1}{1 + 2.53 \times 10^{-6}\mathrm{Re_{TP}}^{1.17}}$ $F = \mathrm{MAX}[0.64\,(\phi_f), 1]$
Correlation of Oh et al. (2011) [40]	$S = 0.279(\phi_f^{\,2})^{-0.029}\mathrm{Bo}^{-0.098}$ $F = \mathrm{MAX}\left[(0.023\,\phi_f^{\,2.2} + 0.76), 1\right]$

Chen correlation and Zhang et al. correlation have the same equation in determining the value of the suppression factor [41]. The equation uses the value of the two-phase Reynolds number ($\mathrm{Re_{tp}}$). While Oh et al. correlation use values from boiling number (Bo) and two-phase frictional multiplier gradients and differences ($\phi_f^{\,2}$). $\mathrm{Re_{tp}}$, Bo, and $\phi_f^{\,2}$ are determined with Eq. (13), Eq. (14), and Eq. (15).

$$\mathrm{Re_{TP}} = \frac{\mathrm{G}d}{\bar{\mu}} \quad (13)$$

$$\mathrm{Bo} = \frac{\ddot{q}}{\mathrm{G}\,h_{fg}} \quad (14)$$

$$\phi_f^{\,2} = 1 + \frac{C}{X} + \frac{1}{X^2} \quad (15)$$

The two-phase Reynolds number value are dependent with the mass flux (G), pipe diameter (D), and dynamic

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viscosity ($\bar{\mu}$). The dynamic viscosity used in the equation is the two-phase dynamic viscosity. Cicchitti et al. (1960) expressed equation for average value of two phase viscosity, expressed in Eq. (16) [42].

$$\bar{\mu} = x\mu_g + (1 - x)\mu_l \tag{16}$$

F parameter in the Chen correlation, the Zhang et al. correlation, And Oh et al. correlation use the same variable, that is the two-phase frictional multiplier gradients and differences (ϕ_f^2). The value ϕ_f^2 can be calculated using Eq. (15), where X is not the quality of vapor but rather the Martinelli parameter which can be calculated through Eq. (17). and C is the Chisholm parameter [43]. The values of the Chisholm parameters for the liquid-vapor flow conditions of turbulent-turbulent (tt), laminar-turbulent (vt), turbulent-laminar (tv) and laminar-laminar (vt) are respectively 20, 12, 10, and 5.

$$X = \left(\frac{f_l}{f_v}\right)^{0.5} \left(\frac{1-x}{x}\right) \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \tag{17}$$

4. Result and Discussion

The experiment was held on 9.1 bar and 8.5 bar. From the experiments, the heat flux value 311 W/m², 221 W/m², 184 W/m² respectively on the 8.5 bar, the heat flux value 308 W/m², 739 W/m², and 276 W/m² respectively on the 9.1 bar. The heat transfer coefficients on the 8.5 bar are 783 W/m².C, 754 W/m².C, and 751 W/m².C respectively; then the heat transfer coefficients on the 9.1 bar are 784 W/m².C, 911 W/m².C, 785 W/m².C, respectively.

According to the experiment, the value of heat flux and vapor quality affecting the heat transfer coefficient value. According to Karayiannis et al. (2012), it was reported that the heat transfer coefficient increases with heat flux with little dependence on local vapor quality [44]. The heat transfer coefficient of a refrigerant is dependent on the heat flux and vapor quality. Oh et al. (2011) also mention that heat flux had a significant effect on the heat transfer coefficient at the low quality region [40].

According to Fig. 2, the experiment 1 have a higher heat flux compared with the experiment 2 and 3. The experiment 2 and 3 have a significantly low heat flux when compared to the experiment 1. Therefore, the heat transfer coefficient at the experiment 1 on 9.1 bar is higher than the experiment 2 and 3.

Yang et al. (2017) reported the result of experiment that heat transfer coefficients increase with heat flux [45]. In low heat flux, the coefficient of heat transfer rises significantly with the mass vapour quality and this trend condition is more obvious in high mass flux.

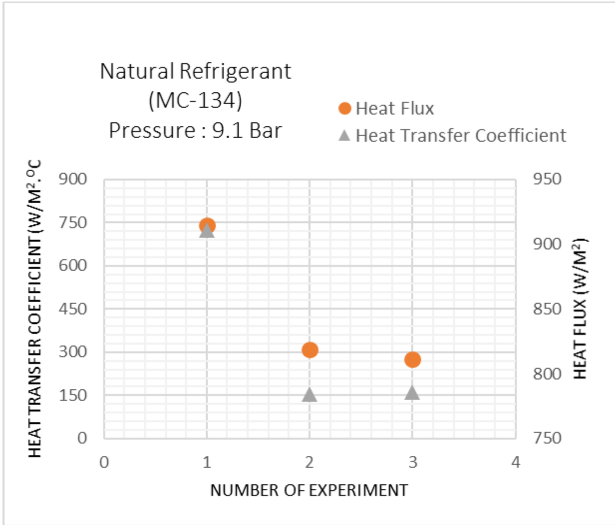

Fig. 2. The heat transfer characteristic of MC-134 at 9.1 bar.

Shin et al. (1997) reported that in low mass vapour quality, the heat transfer process is dominated by nucleate boiling in the initial stage [46].

Yang et al. (2018) studied flow boiling heat transfer characteristic of environment-friendly refrigerant mixture [47].

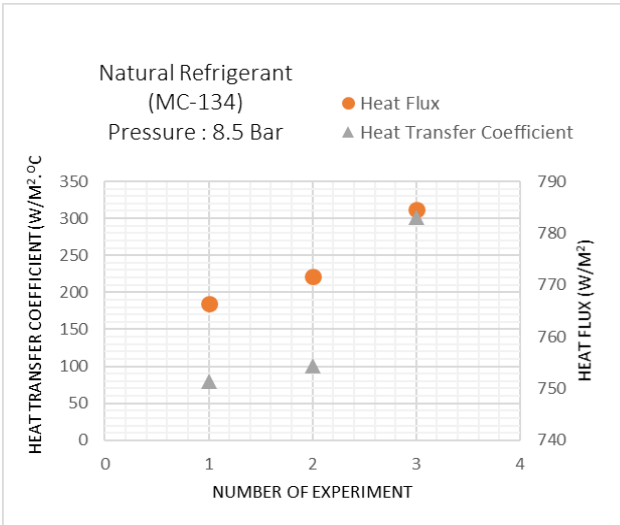

Fig. 3. The heat transfer characteristic of MC-134 at 8.5 bar.

Yang et al. (2018) reported heat transfer coefficients increase significantly with the heat flux in the low quality region [47]. In addition, the effect of heat flux on nucleate boiling heat transfer is significant. In low quality region, more nucleation site are activated with increasing of heat flux which leads to the acceleration of bubble growth and departure.

On Fig. 3, the experiment data on 8.5 bar shows similarities with the experiment on the 9.1 bar. The result on this experiment shows that the high heat flux value will have high heat transfer coefficient value.

A comparison between the value of heat transfer coefficient in this study with the value of the heat transfer coefficient prediction that calculated through several research correlations is needed to find out whether this study was successful. The research correlations that will be used as a comparison are the Oh et al. correlation, Chen correlation, and Zhang et al. correlation.

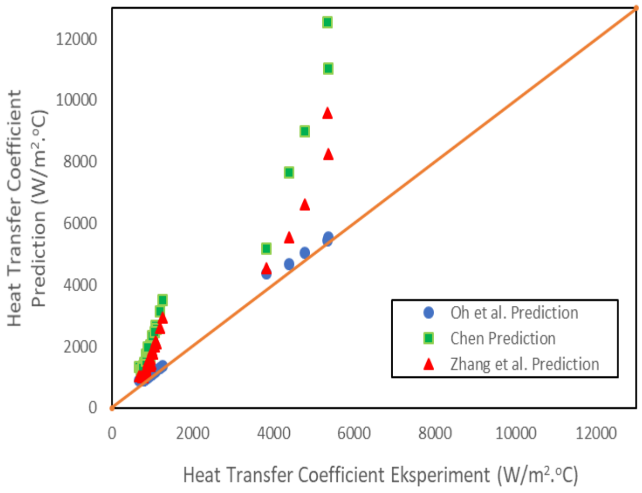

Fig. 4. Comparison of the results of experiments with the predictions.

Based on Fig. 4, it can be seen that the closest correlation to the results of the study is the correlation of Oh et al. with error deviation value 12.74%. While the Chen correlation and the correlation of Zhang et al. has predictions of heat transfer coefficient that are quite far from the results of the study. The error deviation values for both were 95.89% and 54.23% respectively.

5. Conclusion

Natural refrigerant (MC-134) is consist of two major hydrocarbon, propane and iso-butane. Both of this mixture has good characteristic on heat transfer coefficients. As reported on Austin et al. (2012) [2] and Jwo et al. (2009) [3] the mixture between propane and butane, results on better performance of heat transfer characteristic.

The result of this study shows that the heat transfer coefficient value is dependent on the heat flux value. This dependence occurs at a pressure of 9.1 bar and 8.5 bar. In that pressure is show if the heat flux value is high then the heat transfer coefficient value will be high too, and if the heat flux value is low then the heat transfer coefficient value will be low too.

The value of heat transfer coefficient from this study is almost the same as the calculation of the heat transfer coefficient prediction on the correlation of Oh et al. (2011) [40]. But the value is quite far when compared with the correlation of Chen (1966) [13] and the correlation of Zhang et al. (2004) [16].

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