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Flow boiling heat transfer characteristics of R245fa refrigerant in a plate heat exchanger

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Abstract. Plate heat exchangers (PHEs) are used extensively in industrial applications and, owing to their compactness, and higher thermal efficiency, they are keys component of organic Rankine cycle (ORC) application. This study presents the experimental heat transfer characteristics during flow boiling of R245fa refrigerant with a commercial working fluid used in ORCs; inside brazed plate heat exchanger with chevron angle of 65 degree. The flow boiling heat transfer characteristics were measured with varying saturation temperatures, mass flux and heat flux, which range from 55.5 °C-61.8 °C, 15.5-17.4 kg m⁻² s⁻¹, and 6400-10120 W m⁻², respectively. The experimental results showed that flow boiling heat transfer coefficient is dependent upon the heat flux and mass flux. The results showed that the heat transfer coefficient increases with the increase of heat flux, and then starts to decrease due to local dry out. However, at low mass flux, locally triggering dry out was more prominent than that high mass flux. The heat transfer coefficient showed to be sensitive to the change in the saturation temperature. Moreover, flow boiling heat transfer coefficient showed a linear relationship with mass flux of the refrigerant.

1. Introduction

Plate heat exchangers (PHEs) are used extensively for industrial applications, such as refrigeration, air conditioning, heating, cooling, etc. PHEs consist of corrugated plates which provide a large effective heat transfer surface area as well as modification of the flow field in order to promote enhanced thermal-hydraulic performance [1]. Plate heat exchangers can be gasketed, brazed, welded/semi-welded and shell and plate [1]. Among them, brazed plate heat exchangers (BPHEs) exhibits better sealing performance, higher resistant to corrosion and pressure and for these reasons are suitable for evaporation of refrigerants in small-scale organic Rankine cycle (ORC) plants [2], and therefore this heat exchanger type is considered in this study. The heat transfer characteristics of refrigerants in such BPHEs have significant impacts on the ORC design. Therefore, the research on the heat transfer performance in PHEs is fundamentally important in order to design heat exchangers for more efficient and economically viable ORC units [3].

In general, flow boiling with PHEs is a complex heat transfer process; because it is affected by many factors including mass flux, heat flux, vapour quality, saturation temperature (pressure), properties of the working fluids and the geometries of the PHEs. However, experimental results from different studies regarding the effects of these factors on the heat transfer and pressure drop are inconsistent [2]. D Han et al. [4] performed experimental heat transfer studies in a BPHE using refrigerants R410A and R22 with varying the evaporation temperature 5 °C, 10 °C and 15 °C. The experimental results showed that both the evaporation heat transfer coefficient and pressure drop increase with decreasing evaporation temperature. G A Longo [5] investigated the effect of heat flux, mass flux, saturation temperature, outlet conditions on heat transfer and pressure drop during vaporisation of R1234yf refrigerant inside a BPHE. The heat



transfer coefficient showed a great sensitivity to heat flux and outlet conditions and weak sensitivity to saturation temperature ranging from 4.8 °C to 20.2 °C. J Zhang et al. [2] performed experimental studies to investigate the effect of mass flux, heat flux, and saturation temperature, outlet conditions during evaporation of R1234yf and R1234ze, and R134a in a BPHE. The heat transfer coefficient results showed a strong dependence on the heat flux and saturation temperature ranging from 60 °C to 80 °C. The mass flux exhibited a negligible effect on the boiling heat transfer coefficient. Y Y Hsieh and T F Lin [6] studied saturated flow boiling heat transfer and pressure drop of R410A in a BPHE. The experimental results showed that both the boiling heat transfer coefficient and frictional pressure drop increase almost linearly with the imposed heat flux. Furthermore, the refrigerant mass flux exhibited a significant effect on the saturated flow boiling heat transfer coefficient only at higher imposed heat flux. They found that the refrigerant saturation temperature of 10 °C, 15 °C, and 20 °C have very slight influences on the saturated flow boiling heat transfer coefficient.

Despite of these studies, the heat transfer characteristics of ORC working fluids in such BPHEs have been studied very little as compared to conventional refrigerants used in air conditioning and refrigeration systems [3]. Therefore, the objective of this work is to study flow boiling heat transfer of R245fa refrigerant with a commercial working fluid used in ORCs; inside BPHE and its dependence on operating conditions prevailing in the evaporator. Herein, the effects of heat flux, mass flux of refrigerant, and saturation temperature on flow boiling heat transfer characteristics are investigated and discussed in the following section. Moreover, the experimental results were compared with an empirical correlation in PHEs.

2. Experimental setup

2.1. Experimental apparatus

The experimental setup designed for the present study is shown schematically in Fig. 1. It mainly consists of three loops, a refrigerant loop, a water loop to supply heat to the evaporator (test section) and to evaporate the refrigerant, and a cold water-glycol loop to condense the refrigerant. In the refrigerant loop, a gear pump is used to circulate the refrigerant as well as control the mass flow rate. The liquid refrigerant is pumped from the receiver through a filter/dryer, mass flow meter, pre-heater and evaporator. The pre-heater is installed upstream of the test section. A power meter is used to measure the electricity supply to the pre-heater. In the pre-heater, the liquid refrigerant is heated to obtain the sub-cooled liquid refrigerant (5 °C) before entering the test section. Then, the sub-cooled liquid refrigerant moves into the test section for further heating and R245fa refrigerant boils there. Thermocouples and pressure sensors are installed at the inlet and outlet of the test section to measure the temperature and pressure of the refrigerant, respectively.

In the heating loop, water is heated in the boiler by a power supply. The heater capacity is 2.9 kW and the boiler volume is 100 L. After the water is heated to the specified temperature, a variable speed volumetric pump is circulating the heating water to flow through the evaporator. During all the experiments, the pump speed was changed to regulate the water flow rate and thus control the heat flux transferred from the water to the refrigerant. The heating rejection to the cooling water system is obtained by circulation of chilled water in parallel flows the condenser and sub-cooler. By adjusting the mass flow rate and temperature of the chiller water, the condensation pressure of the refrigeration is controlled. Moreover, the temperature and the volume flow rate of heating water are measured at the inlet and outlet of evaporator.

2.2. PHE

The plate heat exchanger used as the evaporator (test section) in the test rig is a commercial BPHE with compact structure. The BPHE has 12 plates in total, 6 water channels and 5 refrigerant channels. The total heat transfer area is 0.204 m². The BPHE is chevron type with chevron angle of 65 degree. Fig. 2 shows the schematic of the corrugated plate, and the main dimensions of the current plate are listed in table 1.

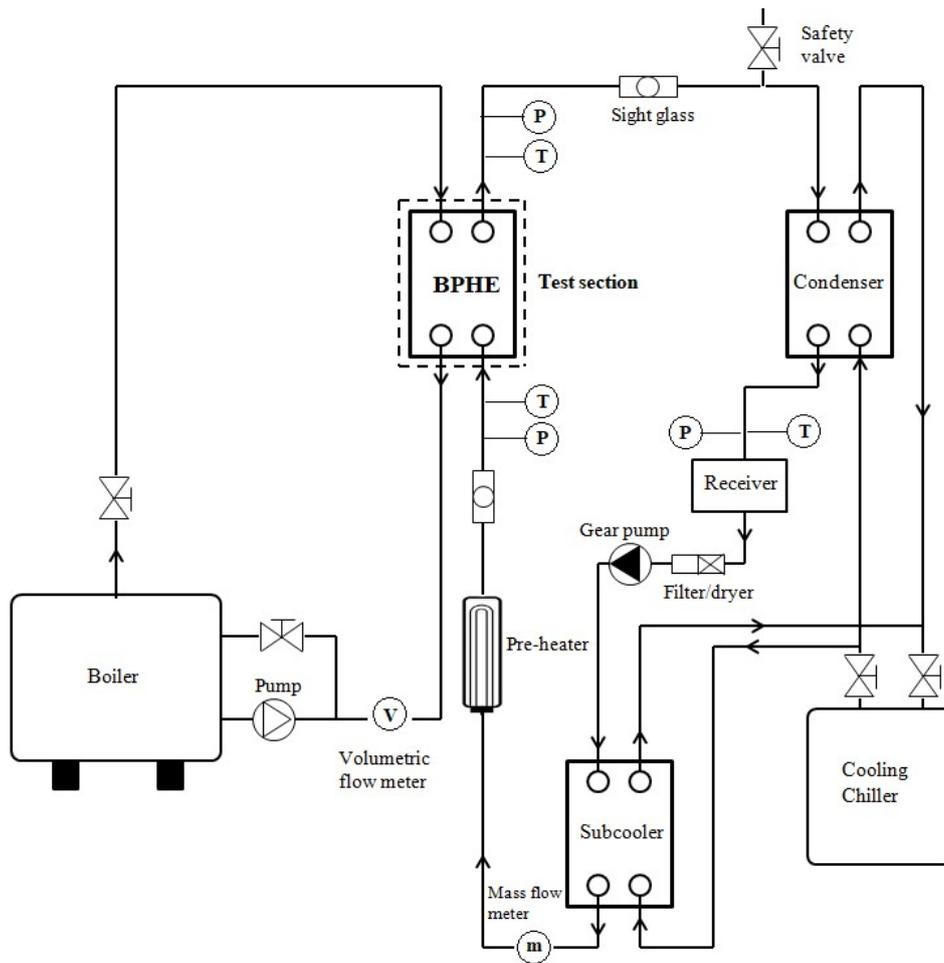


Figure 1. Schematic of the experimental setup.

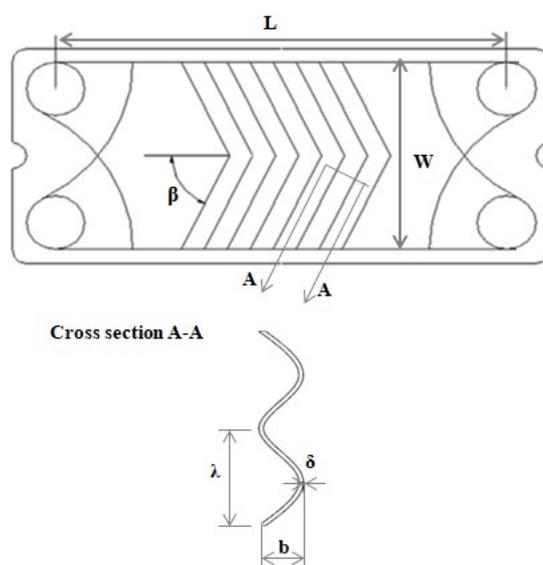


Figure 2. Schematic of the corrugated plate.

Table 1. Geometrical data of the plate.

Parameters	Value
Fluid flow plate length, L	278 (mm)
Plate width, W	73.5(mm)
Plate thickness, δ	0.25 (mm)
Corrugation amplitude, b	1.9 (mm)
Corrugation angle, β	65 ($^{\circ}$)
Corrugation pitch, λ	6 (mm)

2.3. Data reduction

Firstly, the total heat transfer between the counter flows in the PHE is calculated from the hot water side.

$$Q_w = \dot{m}_w c_{p,w} (T_{w,in} - T_{w,out}) \quad (1)$$

where Q_w is the heat transfer rate of the evaporator, \dot{m}_w and $c_{p,w}$ mass flow rate and specific heat of water, respectively, and $T_{w,in}$ and $T_{w,out}$ are the evaporator inlet and outlet temperatures of water side, respectively. Before entering the test section, the liquid refrigerant is heated to the sub-cooled liquid refrigerant (5 $^{\circ}$ C) state in the pre-heater. Then, the heat transfer from the hot water to the refrigerant sides in the test section occurs and thus the sub-cooled liquid refrigerant boils there. The change in the refrigerant vapour quality in the test section, Δx , is then deduced from the heat transfer rate between the water and refrigerant sides in the test section Q_w :

$$\Delta x = x_{out} - x_{in} = \frac{Q_w}{h_{fg} \dot{m}_r} \quad (2)$$

where h_{fg} and \dot{m}_r are the enthalpy of the evaporation and mass flow rate of refrigerant, respectively. The overall heat transfer coefficient can be obtained using the logarithmic temperature difference (LMTD) method:

$$U = \frac{Q_w}{A_{eva} LMTD} \quad (3)$$

$$LMTD = \frac{(\Delta T_1 - \Delta T_2)}{\ln(\Delta T_1 / \Delta T_2)} \quad (4)$$

where $\Delta T_1 = T_{w,in} - T_{sat}$, $\Delta T_2 = T_{w,out} - T_{sat}$

where T_{sat} is saturation temperature in the evaporator. A_{eva} is the nominal heat transfer area of the evaporator which is equal to the nominal projected area $A_p = L \times W$ of the single plate multiplied by the number of the effective plates, as suggested by R K Shah and W W Focke [7].

Finally, the flow boiling heat transfer coefficient of the refrigerant side h_r is evaluated from below:

$$\frac{1}{h_r} = \frac{1}{U} - \frac{1}{h_w} - \frac{t_{wall}}{k_{wall}} \quad (5)$$

where t_w and k_{wall} are the thickness and thermal conductivity of the plate, respectively. Also, heat transfer coefficient of the water side h_w is calculated by eq. (6) [5]:

$$h_w = 0.277 \left(\frac{\lambda_w}{d_h} \right) Re_w^{0.766} Pr_w^{0.33} \quad (6)$$

where λ_w , Re_w and Pr_w are thermal conductivity, Reynolds number and Prandtl number of water respectively, and d_h is the hydraulic diameter of the channel which is twice the corrugation amplitude.

2.4. Uncertainty analysis

S J Kline and F A McClintock [8] reported that the uncertainty can be estimated with a good accuracy using a root-sum-square combination of the effects of each of the individual inputs. When a target parameter F is a function of several variables x_1, x_2, x_3, \dots , whose total errors are $\varepsilon_{x1}, \varepsilon_{x2}, \varepsilon_{x3}, \dots$, the total error of the F (ε) is calculated by eq. (7):

$$\varepsilon_F = \left[\left(\frac{\partial F}{\partial x_1} \varepsilon_{x1} \right)^2 + \left(\frac{\partial F}{\partial x_2} \varepsilon_{x2} \right)^2 + \left(\frac{\partial F}{\partial x_n} \varepsilon_{xn} \right)^2 \right]^{1/2} \quad (7)$$

In this study, the temperature measurement uncertainty was ± 0.13 °C. The error associated with the mass flow rate and volume flow rate were $\pm 0.112\%$ and $\pm 0.35\%$, respectively. The total uncertainty of the heat supplied to the test section was estimated as $\pm 4.5\%$. The maximum uncertainty of the heat transfer coefficient was estimated as $\pm 12\%$.

3. Results and discussion

3.1. Heat transfer

Table 2 provides the main operating conditions in the evaporator: refrigerant saturation temperature T_{sat} and pressure P_{sat} , inlet and outlet refrigerant vapour quality x_{in} and x_{out} , mass flux on the refrigerant side G and heat flux q . Figure 3 presents the heat transfer coefficient variation of R245fa refrigerant as a function of heat flux for two values of mass fluxes $15.5 \text{ kg m}^{-2} \text{ s}^{-1}$ and $17.4 \text{ kg m}^{-2} \text{ s}^{-1}$ at saturation temperature of 61.8 °C. Moreover, the outlet quality is reported for each point. At the given mass flux, the flow boiling heat transfer coefficient increases with the increase of heat flux to its maximum value, at quality of 0.9, and then starts to decrease. For example, at $G = 15.5 \text{ kg m}^{-2} \text{ s}^{-1}$, the heat transfer starts to decrease at heat fluxes higher than 8000 W m^{-2} . The enhancement in the heat transfer coefficient is conjectured to result from the more nucleation density on the plate, higher bubble generation frequency and faster bubble growth for a higher imposed heat flux [6]. The possible reason for decreasing heat transfer coefficient at quality of 0.99 is local dryout phenomenon. However, at low mass flux, i.e., $G = 15.5 \text{ kg m}^{-2} \text{ s}^{-1}$, this phenomenon is more prominent than that high mass flux. From figure 3, it can be further found that the flow boiling heat transfer coefficient enhances with increasing mass flux.

Figure 4 shows the effect of saturation temperature on the flow boiling heat transfer for both mass fluxes. By inspecting figure 3 and figure 4, it is interesting to note that the shape of heat transfer coefficient profiles remains nearly similar by decreasing saturation temperature to 55.5 °C for both mass fluxes $15.5 \text{ kg m}^{-2} \text{ s}^{-1}$ and $17.4 \text{ kg m}^{-2} \text{ s}^{-1}$. However, the enhancement of heat transfer coefficient at saturation temperature of 61.8 °C is higher than that at $T_{sat} = 55.5$ °C. At high saturation temperature, the size of bubbles is conjectured to become smaller and leave the surface of evaporator at higher frequency and thus creating more nucleation sites, resulting in enhancement of heat transfer coefficient [6].

Figure 5 presents the effect of mass flux on the flow boiling heat transfer coefficient at a heat flux of 9000 W m^{-2} for saturation temperature of 55.5 °C and 61.8 °C. As the mass flux increases from $15.5 \text{ kg m}^{-2} \text{ s}^{-1}$ to $17.4 \text{ kg m}^{-2} \text{ s}^{-1}$, the heat transfer coefficient increases by 7% . This is a consequence of the stronger convection in the flow associated with the high mass flux which promotes the earlier departure of the bubbles on the plate and thus enhances the bubble generation frequency [6]. Moreover, the heat

transfer coefficient is found to increase by 3% with increasing saturation temperature from 55.5 to 61.8 °C.

Table 2. Operating conditions.

T_{sat} (°C)	P_{sat} (MPa)	x_{in}	x_{out}	G (kg m ⁻² s ⁻¹)	q (W m ⁻²)
55.5-61.8	0.407-0.487	0	0.69-0.99	15.5-17.4	6400-10120

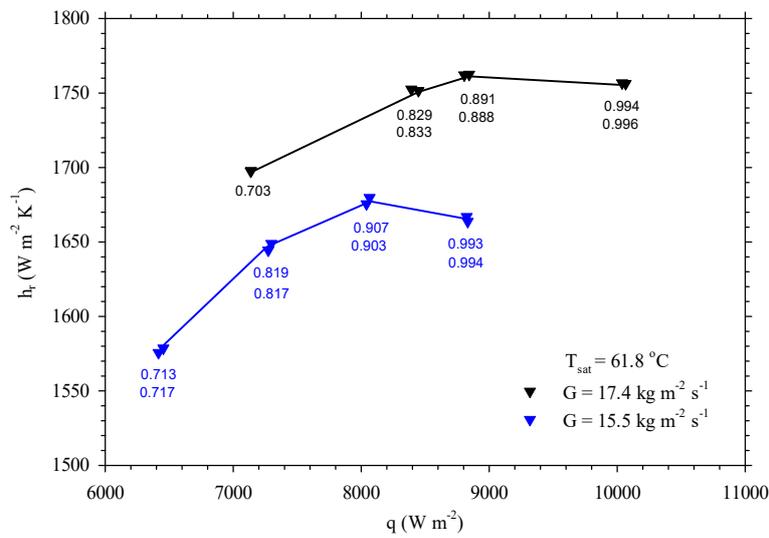


Figure 3. Heat transfer coefficient versus heat flux at $T_{sat} = 61.8$ °C.

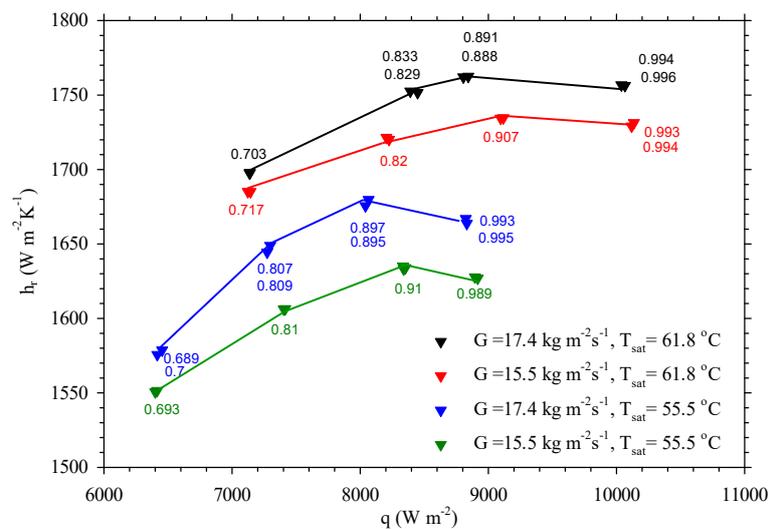


Figure 4. Heat transfer coefficient versus heat flux.

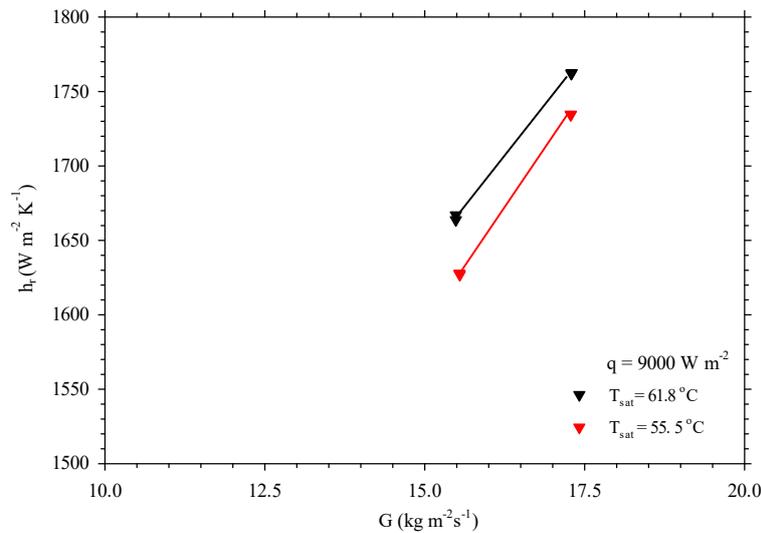


Figure 5. Heat transfer coefficient versus mass flux at $q = 9000 \text{ W m}^{-2}$.

3.2. Comparison with a heat transfer correlation used in PHEs

Figure 6 shows the comparison between the present experimental and calculated heat transfer coefficient from D Han et al. [4] correlation. The heat transfer coefficient in terms of Nusselt number is calculated by eq. (8):

$$Nu = Ge_1 Re_{eq}^{Ge_2} Pr_l^{0.4} \quad (8)$$

$$Ge_1 = 2.81 \left(\frac{\lambda_l}{d_h} \right)^{-0.041} \left(\frac{\pi\beta}{180} \right)^{-2.83}$$

$$Ge_2 = 0.746 \left(\frac{\lambda_l}{d_h} \right)^{-0.082} \left(\frac{\pi\beta}{180} \right)^{0.61}$$

where Re_{eq} is the equivalent Reynolds number, Pr_l , and λ_l are the liquid-phase Prandtl number and thermal conductivity, respectively.

The mean absolute deviation (MAE) has been used to estimate the error of the calculated heat transfer coefficient with respect to the experimental data:

$$MAE = \frac{1}{n} \sum_{i=1}^n \left| \frac{h_{i,exp} - h_{i,cal}}{h_{i,exp}} \right| \times 100\% \quad (9)$$

where n is the number of data points. From the figure, the calculated heat transfer coefficient has a mean absolute error of 5.8% with 69.7% of the present experimental data in a bandwidth of 10%.

4. Conclusion

An experimental investigation was conducted to study the flow boiling heat transfer of R245fa refrigerant in a BPHE. The experimental data were tested with saturation temperature of 55.5 °C and 61.8 °C, mass fluxes of 15.5-17.4 kg m⁻² s⁻¹, and heat fluxes of 6400-10120 W m⁻². The major results can be summarized in the following.

- The flow boiling heat transfer coefficient was found to be dependent on the heat flux. The results showed that the heat transfer coefficient increases with increasing heat flux, and then starts to decrease at quality of 0.99 due to local dry out. Furthermore, at low mass flux the triggering dry out phenomenon was more prominent than that the high mass flux.
- The results showed that the flow boiling heat transfer coefficient at saturation temperature of 61.8 °C is about 3% higher than at $T_{sat} = 55.5 \text{ °C}$.

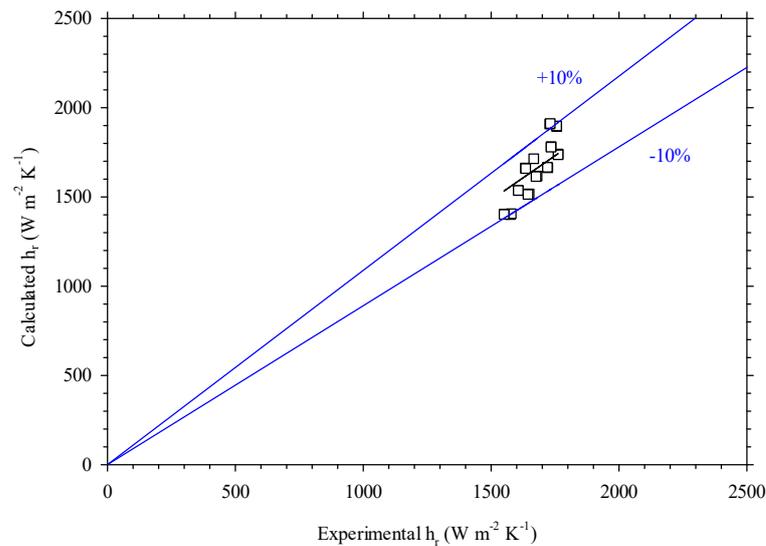


Figure 6. Comparison of eq. (8) with present experimental data.

- The flow boiling heat transfer coefficient showed sensitive to the change in the refrigerant mass flux. The heat transfer coefficient increased by 8% with increasing mass flux from $15.5 \text{ kg m}^{-2} \text{ s}^{-1}$ to $17.4 \text{ kg m}^{-2} \text{ s}^{-1}$.

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