Research Article

Xiaoyu Hu, Yi Wang, Siyuan Li, Qiang Sun, Guoxiang Li*, Shuzhan Bai*, and Ke Sun*

Investigation on subcooled flow boiling heat transfer characteristics in ICE-like conditions

<https://doi.org/10.1515/phys-2021-0052>

received January 23, 2021; accepted June 06, 2021

Abstract: The increasing demand of cooling in internal combustion engine (ICE) may require the shift of heat removal method from traditional single phase liquid convection to subcooled flow boiling in order to fulfill the desired functional temperature. Thus, the characteristics of subcooled flow boiling heat transfer should be studied exclusively considering the practical conditions in ICEs. Accordingly, in this article, subcooled flow boiling experiments were conducted in a rectangular channel using 50/50 volume mixture of ethylene glycol and water coolant (EG/W) as working fluid. Aluminum and cast iron surfaces were selected as the heated surfaces to simulate the material of cylinder head in gasoline and diesel engines, respectively. Experimental results showed a trend that the aluminum surface had a better performance than the cast iron surface in terms of heat transfer coefficient in the boiling region. The difference between these two surfaces was concluded as results of different surface thermophysical properties. A modified wall heat flux model was proposed based on the power-type addition method. The proposed model modified the nucleation boiling contribution by introducing a new parameter which accounts for the influence of thermophysical properties of heated surface on the boiling process. Thus, one such modified model could be useful for practical engine cooling applications.

Keywords: subcooled flow boiling, ICE cooling, ethylene glycol/water mixture, aluminum and cast iron heated surfaces

1 Introduction

Engine downsizing and strengthening is one of the important technical solutions to deal with increasingly strict fuel consumption and emission regulations [1]. However, an evitable issue associated with these techniques is the enhanced thermal load in internal combustion engine (ICE) cylinder head, which requires to establish optimal cooling strategies. A great amount of heat transfer augmentation techniques have been reported for decades, such as extended surfaces [2,3], boiling [4,5], ferrofluids [6], etc. Among these techniques, subcooled flow boiling heat transfer has been proved to be effective and beneficial to enhance heat transfer, which is largely due to its high heat dissipation rate under relatively small change in wall temperature. This benefit could help enhancing the heat transfer rate in the compact space of ICE cylinder head without essential changes in its architecture. However, the boiling process is extremely complex and contains many different phenomena, such as bubble nucleation, growth, detachment, and collapse, of which the process is still not fully understood yet [7,8]. Therefore, from the point of view of designing advanced engine cooling system, prediction of heat flux through coolant passage wall of ICEs is important and requires the application of empirical or semi-empirical correlations.

One such widely applied model in ICE industry was proposed by Chen [9], in which the overall heat transfer is governed by a combined effect of both forced convection component q_{sp} and nucleate boiling component q_{nb} . Two factors, the enhancement factor F and the suppression factor S that is as a function of flow Reynolds number, were defined taking into account the convective heat transfer enhancement by bubble agitation and the flow suppression effect on the nucleation boiling, respectively.

* **Corresponding author: Guoxiang Li**, School of Energy and Power Engineering, Shandong University, Jinan, Shandong, 250061, China, e-mail: liguox@sdu.edu.cn

* **Corresponding author: Shuzhan Bai**, School of Energy and Power Engineering, Shandong University, Jinan, Shandong, 250061, China, e-mail: baishuzhan@sdu.edu.cn

* **Corresponding author: Ke Sun**, School of Energy and Power Engineering, Shandong University, Jinan, Shandong, 250061, China, e-mail: sunke@sdu.edu.cn

Xiaoyu Hu, Yi Wang, Siyuan Li, Qiang Sun: School of Energy and Power Engineering, Shandong University, Jinan, Shandong, 250061, China

$$q = Fq_{sp} + Sq_{nb}. \quad (1)$$

Several modifications have been made based on the investigation of subcooled flow boiling heat transfer to further improve the accuracy of Chen model in its application to ICE industry. As proposed by Steiner *et al.* [10], the suppression factor in Chen model was calibrated for saturated boiling and is insensitive to local parameters such as wall superheat and wall shear stress, which could cause deviations in application for subcooled flow boiling conditions. Then, a modified model was proposed to improve Chen's superposition method by taking into account the impact of the flow-induced suppression and subcooling. As stated by Torregrosa *et al.* [11], the Chen model fails in boiling regimes for low velocities and subcooling, which is largely due to the significant influence of Prandtl number on suppression factor. More recently, Chen *et al.* [12] made a modification on the enhancement and suppression factors of the Chen model by introducing a non-dimension boiling number into them, and the predictions of this modified correlation coincides well with the subcooled flow boiling experimental data. Although Chen model has been exclusively used in ICE industry [13–15], its application to ICE cooling system design is not totally straightforward. For one thing, Chen model is mainly used in gasoline engines, but it may not be applicable to diesel engines because the Chen model overestimates the subcooled flow boiling heat flux of cast iron heated surfaces under high wall superheat [16]. The other is the difficulty in defining an effective hydraulic diameter and velocity scale for the cooling passage due to its complex architecture [10]. Therefore, an appropriate correlation considering the above issues is still required.

A more general form of the above strategy is the so called power-type addition model or asymptotic model. This kind of model mainly focuses on the component with greater influence regarding the convective and nucleate boiling components, which could connect the overall flow boiling phenomena with the more dominant mechanism in these two regions as well as assure a smooth transition in between.

$$q^n = (q_{sp})^n + (q_{nb})^n, \quad (2)$$

where n is an empirical constant to be determined from the experimental data. In particular case of $n = 1$, it becomes the simple additive model similar to Chen model. The model given in equation (2) has the advantage that the researchers can incorporate the set of correlations which best estimate their problem for

two contributions of forced convection and nucleation components.

$$q_{nb} = \mu_l h_{lg} \left[\frac{\sigma}{g(\rho_l - \rho_g)} \right]^{-1/2} \left(\frac{c_{pl} \Delta T_{sat}}{C_{sf} h_{lg} Pr} \right)^{1/\gamma}. \quad (3)$$

Ramstorfer *et al.* [17] proposed a new type of power-type addition correlation adopting only Rohsenow saturation pool boiling type model [18] in its q_{nb} . The advantages of this type of model are that first, it eliminates the dependence of structure parameters on calculating nucleation boiling term; second, it could reasonably account for the effect of fluid–solid interaction on boiling heat transfer. Similar method was adopted by Hua *et al.* [16]. According to the authors, the prediction results were sensitive to empirical constants C_{sf} and γ , and these constants should be derived from the experimental results of exactly the same surface–fluid combination.

Consequently, there is a great interest in applying such a power-type addition model into ICE coolant jacket's design considering its accuracy and independence on structure parameter. However, proper empirical constants in this type of model for the fluid–solid combination of 50/50 EG/W and aluminum or cast iron is not yet available in literature. Moreover, reprehensive subcooled flow boiling experimental data, especially that of 50/50 EG/W on cast iron heater surface is not sufficient to support in developing a generally improved model beyond the models mentioned above. Therefore, the objective of this work is to carry out subcooled flow boiling experiments under conditions that are similar to practical ICE operating conditions and construct an appropriate power-type addition model for future more advanced ICE engine coolant passage design.

One of the unique characteristics of coolant subcooled boiling in the cylinder head regions in ICEs is that boiling generally occurs only on the cooling channel side facing the flame plate because of the one-sided heating condition. Accordingly, in this study, subcooled flow boiling experiments were conducted in a one-side heated rectangular channel using 50/50 EG/W as working fluid. Aluminum and cast iron were employed as heater surfaces to simulate the material of gasoline and diesel engine cylinder head, respectively. Considering its practical conditions in ICEs, the experimental conditions of inlet temperature, bulk velocity, and system pressure ranged from 70 to 90°C, 0.25–1 m/s, and 1–2 bar, respectively. Finally, a subcooled flow boiling correlation was proposed to characterize the effect of heated surface material on flow boiling heat transfer.

Figure 1: Schematic of boiling system.

2 Experimental setup

2.1 Experimental apparatus

The experiment system was built accordingly to investigate the subcooled flow boiling heat transfer in the cooling passages for cylinder head in ICEs. A schematic of the whole system is shown in Figure 1. The closed flow circulating system consists of storage tank, pump, flowmeter, heating apparatus, heat exchanger, and rectangular test section. The storage tank with a capacity of 150 L was designed to maintain a stable bulk fluid temperature. Four immersion heaters with total maximum power of 4 kW were installed at the bottom of the storage tank. The heaters were controlled by a PID controller so as to achieve and maintain certain bulk fluid temperature. Likewise, a solenoid valve connected to high pressure nitrogen was mounted on the top of the storage tank to adjust the pressure in the loop by the PID controller. Additionally, the circuit was equipped with a variable speed pump to determine flow rate across the test section.

A photo of the real flow boiling test section can be seen in Figure 2. The horizontal rectangular test section, of 300 mm length and with 38 mm × 48 mm cross-sectional dimensions, located at 2,000 mm downstream of a stainless steel rectangular tube, which was to ensure the flow in the test section was fully developed is shown in Figure 3. The test specimen with dimensions of 90 mm × 10 mm was placed on the bottom surface of the test section and was located 110 mm downstream of the test section inlet. The test specimen was heated by seven heating rods inserted at the bottom of the testing surfaces which can supply a maximum power of around 7 kW. As shown in

Figure 4, seven K-type thermocouples with diameter of 1.0 mm were arranged below the test specimen along the vertical direction in two rows, where three thermocouples were in the upper row and the other four were in the lower row. The thermocouple probes were inserted to reach the width-wise centerline below the testing surface. The heating surface temperature during the experiment was obtained based on the thermocouple readings and Fourier's thermal conduction equation. The distance between the upper row and the test section surface is 3 mm and the distance between upper and lower rows is 5 mm. Since PTFE (polytetrafluoroethylene) was coated on the surrounding of test samples to minimize thermal diffusion, most of the heat can only be transferred in the vertical direction (i.e., 1-D) through the thickness of the test specimen.

Figure 2: Photo of the test section.

Figure 3: Details of test section.

Figure 4: Location of thermocouples and dimension of test specimen.

test surface was heated from low heat flux to high heat flux and the data were taken 20 min after the system reached thermally steady state. Heat flux q and surface temperature T_w are calculated as follows:

$$q = k \frac{T_2 - T_1}{H_2}, \quad (4)$$

$$T_w = T_1 - \frac{H_1(T_2 - T_1)}{H_2}, \quad (5)$$

where T_1 and T_2 are average temperatures of the upper and lower thermocouple rows, respectively. H_1 (3 mm) and H_2 (5 mm) are the distances from the heated surface to the upper thermocouples row and the distance between the upper and lower thermocouple rows, as shown in Figure 4. k is the thermal conductivity of the test specimen.

2.2 Experimental procedures

Before the formal testing, the bulk fluid was heated close to its boiling point (around 95°C) for several hours to remove the dissolved gas in the fluid. Besides, subcooled flow boiling heat transfer experiment was continuously running for more than 100 h before generating any experimental data in order to minimize the effect of heater surface morphology on boiling heat transfer [19]. The experimental range of inlet temperature, bulk velocity, and system pressure were set as 70–90°C, 0.25–1 m/s, and 1–2 bar, respectively. In each experimental condition, the

2.3 Experimental uncertainties

The experimental errors of the variables were mainly caused by the accuracy of sensors, manufacturing tolerances, the layout of thermocouples, undesired heat losses, and functions to calculate the indirect variables. Before the experiment, all the thermocouples and corresponding data acquisition system were calibrated according to the thermal conditions between ice and saturation temperature of water. For uncertainties of indirect measurement such as heat flux, error propagation equation (6) was adopted for the calculation. All

Table 1: Experimental uncertainties

Parameters	Uncertainties
Bulk temperature	0.1°C
Bulk velocity	0.022 m/s
Pressure	1,500 Pa
Thermocouple location	0.15 mm
Thermocouple temperature	0.1°C
Heat flux	≤11.2%

the calculated and measured uncertainties are listed in Table 1.

$$\Delta q = \sqrt{\left(\frac{\partial q}{\partial x_1}\right)^2 \delta x_1^2 + \left(\frac{\partial q}{\partial x_2}\right)^2 \delta x_2^2 + \dots + \left(\frac{\partial q}{\partial x_n}\right)^2 \delta x_n^2}, \quad (6)$$

where $x_1, x_2, \dots, x_n$ are direct measured values and $\delta x_1, \delta x_2, \dots, \delta x_n$ are the uncertainties of these variables.

2.4 Fluid thermophysical properties

The thermophysical properties of bulk liquid are required in the data reduction and analysis of this study. These properties are usually a function of the temperature, and the effective thermophysical properties of EG/W mixtures are calculated based on the values of pure water and ethylene glycol, and also the concentration of ethylene glycol in the mixtures. The density, specific heat capacity, thermal conductivity, and dynamic viscosity of pure liquids and mixtures were evaluated by semi-empirical equations as suggested in refs. [20–22].

3 Experimental result and discussion

3.1 Single-phase heat transfer

Before the subcooled flow boiling heat transfer experiment, a series of single-phase heat transfer experiments were carried out to validate the experimental setup and study the heat transfer characteristics of one-side heated rectangular channels. The prediction of convective heat flux q_{sp} was calculated as follows:

$$q_{sp} = h_{sp}(T_w - T_b), \quad (7)$$

where h_{sp} is the single-phase heat transfer coefficient calculated from Dittus–Boelter equation [23]. T_w and T_b are wall temperature and bulk temperature, respectively.

$$h_{sp} = 0.023 \text{Re}_l^{0.8} \text{Pr}_l^{0.4} \left(\frac{k_l}{D_h} \right), \quad (8)$$

where Re_l and Pr_l are the liquid Reynolds number and Prandtl number, respectively, as can be seen in equations (9) and (10). k_l is the thermal conductivity of the bulk liquid. All these values are evaluated at bulk temperature.

$$\text{Re}_l = \frac{\rho_l u_b D_h}{\mu_l}, \quad (9)$$

$$\text{Pr}_l = \frac{\mu_l c_{p,l}}{k_l}, \quad (10)$$

where ρ_l , u_b , μ_l , and $c_{p,l}$ are bulk density, velocity, dynamic viscosity, and specific thermal capacity, respectively. D_h is the hydraulic diameter of the test channel, which can be calculated as follows:

$$D_h = \frac{4A}{P_e}, \quad (11)$$

where A is the cross-sectional area of the channel and P_e is the wetted perimeter of the channel. In this article, the D_h equals to 42.4 mm.

The comparative results between experimental data and predicted values can be seen in Figure 5(a). It is shown that the Dittus–Boelter equation underpredicted the heat flux value, and this trend is in consistence with previous literatures [14,23]. The range of validation of Dittus–Boelter equation can be summarized as smooth pipes, hydraulically and thermally developed flow. Obviously, the experimental conditions in this article do not satisfy the validity criteria. As for this article, the heated surface is located downstream from the duct inlet. This would lead to an enhancement in heat transfer coefficient, since the bulk fluid reaching to the heated surface would be cooler than that of the simultaneously developing flow (fluid dynamic boundary layer and thermal boundary layer developed in the meantime) from which the Dittus–Boelter equation was proposed [23]. As a result, a modified equation was correlated based on the experimental data:

$$h_{sp} = 0.05 \text{Re}_l^{0.76} \text{Pr}_l^{0.3} \left(\frac{k_l}{D_h} \right). \quad (12)$$

The comparison between the experimental heat flux and predictions by modified equation is shown in Figure 5(b). the maximum deviation is within $\pm 10\%$,

Figure 5: Comparison of experimental convective heat flux with predictions: (a) Dittus–Boelter correlation (b) modified Dittus–Boelter correlation.

which means the modified equation could well predict the convective heat transfer coefficient of this apparatus.

3.2 Subcooled flow boiling heat transfer

For the cooling system of ICEs, coolant temperature, velocity, and operation pressure are important control variables. As a result, the effect of these parameters on subcooled flow boiling heat transfer is discussed in more detail.

To investigate the effect of subcooling on subcooled flow boiling heat transfer, boiling curves at constant velocity and system pressure ($u_b = 1.0$ m/s and $P = 1.0$ bar) are presented at various inlet temperatures from 70 to 90°C, as shown in Figure 6. It can be seen that, with the increase in subcooling, the onset of boiling (ONB) is delayed since the wall temperature is decreased with an increase in subcooling at a given heat flux. The boiling curves become steep after the ONB, which implies that the heat transfer coefficient is increased with the boiling process. In this partially developed boiling region (PDB), the heat transfer is determined by forced convection and nucleate boiling simultaneously, and the boiling curves of lower inlet temperature would be shifted to the left. But as the heat flux keeps increasing, the flow boiling comes into fully developed boiling region (FDB) where the agitation of generated bubbles is so vigorous that the effect of macro-forced convection disappears, and the boiling curves merge into an asymptote.

Figure 7 presents the effect of velocity on subcooled flow boiling at constant inlet temperature and system pressure ($T_{in} = 80^\circ\text{C}$ and $P = 1.0$ bar) at various velocities

Figure 6: Effect of bulk subcooling on subcooled flow boiling heat transfer ($u_b = 1.0$ m/s and $P = 1.0$ bar).

from 0.25 to 1.0 m/s. With the increase in bulk velocity, the convective heat transfer coefficient increases, leading to higher boiling curve in PDB region since in this region the overall heat flux is largely dependent on forced convection. But this dependence disappears when the boiling goes into FDB region, where the boiling curves converge into the nucleation boiling curve. During boiling process, the bubble behavior, such as nucleation, growth, and departure, is strongly governed by the thermal boundary layer whose thickness decreases with the increase in bulk velocity [24]. As a result, increased bulk velocity would suppress the boiling process and finally lead to a delay on ONB and convergence to FDB for high bulk velocity cases.

Figure 7: Effect of bulk velocity on subcooled flow boiling heat transfer ($T_{in} = 80^{\circ}\text{C}$ and $P = 1.0$ bar).

The boiling curves at different system pressures are plotted in Figure 8. These data were obtained at constant inlet temperature of 90°C , bulk velocity of 0.25 m/s, and various system pressures of 1.0 , 1.5 , and 2.0 bar, at the saturation temperature of 107.8 , 119.9 , and 129.3°C , respectively. It exhibits that, the effect of system pressure on single-phase heat transfer is considerably small, and the boiling curves at different pressures seem to follow a similar development pattern. However, the ONB is delayed with the increase in system pressure, which is mainly caused by the enhanced saturation temperature of the 50/50 EG/W mixture.

The material of engine cylinder head is mainly aluminum or cast iron, and its influence on boiling heat transfer is shown in Figures 6–8. It can be seen that:

Figure 8: Effect of system pressure on subcooled flow boiling heat transfer ($T_{in} = 90^{\circ}\text{C}$ and $u_b = 0.25$ m/s).

(1) In the single-phase flow heat transfer area, the heated surface material has little effect on the heat transfer. In this area, the heat transfer effect is mainly affected by the flow and fluid physical parameters. (2) The boiling curves of aluminum and cast iron heated surfaces obey the similar law: under the same experimental conditions, as the inlet temperature and fluid velocity increase, the heat flux in the PDB increases. While in the FBD, the boiling heat transfer curves finally converge to nucleate boiling curves. In addition, as the system pressure increases, the ONB point is delayed. (3) In the boiling heat transfer area, under the same experimental conditions, the heat flux of the aluminum heated surface is higher than that of the cast iron heated surface, and this trend is more pronounced at low inlet temperature and fluid velocity conditions. It can be seen that for the design of the coolant passages of the engine cylinder head, in the area where single-phase flow heat transfer mechanism dominates, the effect of the cylinder head material on the heat transfer could be ignored; but in the area with high wall superheat where nucleate boiling heat transfer is the main mechanism, the difference in boiling heat transfer performance caused by different materials must be considered.

The difference of boiling heat flux between these two surfaces is concluded as results of different surface thermophysical properties, such as thermal conductivity. On the heated surface, vapor bubble generating centers are places of intensive heat removal, with the local heat flux being much higher than the average value. Therefore, the time-average temperature near the active vapor bubble generating center in the surface layer of a heated surface is lower than that on the rest of the surface. This will cause the overall temperature of the heating surface to decrease, which can obtain a greater heat transfer coefficient than single-phase heat transfer. Moreover, due to the uneven temperature distribution, heat would be transferred inside the surface layer from the free surface of the uncovered bubble to the vicinity of the nucleation center [25,26]. The thermal conductivity of the heated surface material is the key factor that determines its internal heat conduction. It can be predicted that the higher the thermal conductivity of the heated surface, the stronger the internal heat conduction from the free surface to the nucleation center, and thus better heat transfer efficiency.

From the perspective of bubble dynamics, the evaporation of the micro-liquid layer at the bottom of the bubble is the main mechanism of boiling heat transfer, and its contribution to the entire bubble volume accounts for 30–70% [27]. In the process of bubble growth, due to

the evaporation of the micro-liquid layer, the temperature of the nearby heating surface will decrease. For the surface with high thermal conductivity, the heat conduction from the surrounding free surface to the vicinity of the nucleation point is easier than that of the surface with low thermal conductivity, and the temperature change around the nucleation point caused by evaporation is smaller. In other words, for high thermal conductivity heated surface, the bubble micro-liquid layer can supply more heat to bubble growth. Therefore, the heated surface with high thermal conductivity can obtain better boiling heat transfer performance.

4 Subcooled flow boiling correlation

4.1 Comparison between experimental data and predictions

As discussed previously, the power-type addition model proposed by Ramstorfer *et al.* [17] is attractive for the application in ICE coolant system design, as this model could reasonably account for the effect of fluid–solid combination on subcooled flow boiling heat transfer and no local input related to channel size or velocity is required. As a result, this type of model was chosen as reference model to make a comparison between the predicted and the experimental data of this article.

Ramstorfer *et al.* [17] conducted subcooled flow boiling experiment using 40/60 EG/W bulk fluid and cast iron heated surface combination. The experiment was carried out in a vertical channel under 3.4 bar system pressure, at 43°C subcooling, and at various bulk velocities. An asymptotic type model was proposed by correlating the experimental data, which is presented as follows:

$$q^{10} = (q_{\text{conv}})^{10} + (q_{\text{nb}})^{10}, \quad (13)$$

where q_{conv} is the convective heat flux and q_{nb} is the nucleate heat flux. The natural convective heat transfer should be considered for the cases with very low velocity.

$$q_{\text{conv}}^{2.5} = (q_{\text{sp}})^{2.5} + (q_{\text{nc}})^{2.5}, \quad (14)$$

where q_{sp} is the forced convective heat flux and q_{nc} is the natural convective heat flux.

$$q_{\text{sp}} = 0.0447 \text{Re}_l^{0.75} \text{Pr}_l^{0.4} \left(\frac{k_l}{D_h} \right) (T_w - T_b), \quad (15)$$

$$q_{\text{nc}} = \{0.825 + 0.387[Gr\text{Pr}_l(\text{Pr}_l)]^{1/16}\} \frac{k_l}{L} (T_w - T_b), \quad (16)$$

where the Grashof number Gr and $f_1(\text{Pr}_l)$ are given as follows:

$$Gr = \frac{gL^3}{(\mu_l/\rho_l)^2} \beta_1 (T_w - T_b), \quad (17)$$

$$f_1(\text{Pr}_l) = \left[1 + \left(\frac{0.492}{\text{Pr}_l} \right)^{9/16} \right]^{-16/9}, \quad (18)$$

where g is the gravitational acceleration, L is the length of the heated surface, and β_1 is the coefficient of thermal expansion.

The nucleate heat flux is calculated using the Rohsenow pool boiling equation [18].

$$q_{\text{nb}} = \mu_l h_{\text{lg}} \left[\frac{\sigma}{g(\rho_l - \rho_g)} \right]^{-1/2} \left(\frac{c_{\text{pl}} \Delta T_{\text{sat}}}{C_{\text{sf}} h_{\text{lg}} \text{Pr}} \right)^{1/\gamma}, \quad (19)$$

where h_{lg} is latent heat and ρ_g is the density of vapor. $C_{\text{sf}} = 0.0172$ and $\gamma = 0.9$ were suggested for the combination of 40–60 EG/W and cast iron.

Hua *et al.* [16] carried out subcooled flow boiling experiment in a horizontal rectangular channel heated with cast iron test sample using water as fluid. The experimental conditions of bulk temperature, bulk velocity, and system pressure were 70–90°C, 0.2–2 m/s, and 1–3 bar, respectively. The model is written as:

$$q^{8.5} = (q_{\text{sp}})^{8.5} + (q_{\text{nb}})^{8.5}, \quad (20)$$

$$h_{\text{sp}} = 2.906 \text{Re}_l^{0.36} \text{Pr}_l^{0.48} \left(\frac{\mu_b}{\mu_w} \right)^{-0.22} \left(\frac{k_l}{D_h} \right). \quad (21)$$

The term q_{nb} is calculated based on equation (19) with $C_{\text{sf}} = 0.046$ and $\gamma = 0.624$.

Moreover, Hua *et al.* [16] proposed a new correlation for water and aluminum heated surface combination based on the experimental data in ref. [10], which was conducted in a horizontal rectangular channel with aluminum heated surface and water as fluid. The correlations are given as follows:

$$q^{5.7} = (q_{\text{sp}})^{5.7} + (q_{\text{nb}})^{5.7}, \quad (22)$$

where the q_{sp} term is obtained from standard Dittus–Boelter equation. The q_{nb} term is calculated from equation (19) with $C_{\text{sf}} = 0.0248$ and $\gamma = 0.613$.

Comparisons have been made between the experimental data of this article and the predictions of the above proposed equations. The forced convection term in these equations was obtained from equations (7) and (12). Equations (13) and (20) were adopted to predict the

Figure 9: Comparison of heat flux between predicted and experimental data at $T_{in} = 90^\circ\text{C}$ and $P = 1$ bar. (a) $u_b = 0.25$ m/s and (b) $u_b = 0.5$ m/s.

heat flux value of cast iron heated surface, and equation (22) for aluminum surface.

Some representative comparison results between the predicted and experimental data are shown in Figures 9–12. It can be seen that, for particular measurement conditions considered here, the performance of the reference models is unreasonable, with a deviation larger than $\pm 30\%$ in high wall superheat region, especially for low bulk subcooling and velocity conditions. The main reason is that the empirical parameters C_{sf} and γ in these equations are derived from experimental data of different solid–liquid combinations from this research. Equation (13) is based on the subcooled flow boiling heat transfer data of the cast iron and the 40/60 EG/W. For the same heated surface, the boiling heat transfer performance of 40/60 and 50/50 EG/W

Figure 11: Comparison between experimental data of cast iron heated surface with predictions of equation (20).

Figure 10: Comparison between experimental data of cast iron heated surface with predictions of equation (13).

Figure 12: Comparison between experimental data of aluminum heated surface with predictions of equation (22).

is not much different [28]. However, the heated test sample in the literature is a cylindrical heating rod with a diameter of 12 mm wrapped with a layer of cast iron, and its effusivity value (ρck , the product of density, specific heat capacity, and thermal conductivity) is higher than that of the cast iron heated block used in this research. Experimental studies have shown that the heated surface with a large effusivity value has a greater boiling heat transfer performance [7]. Therefore, the predicted value of heat flux in equation (13) is greater than the experimental value in this article. As for equations (20) and (22), these equations are derived for prediction of water subcooled heat transfer. And the influence of fluid physical parameters on boiling process is somewhat different for pure water and EG/W mixtures. Thus, the unmatched empirical parameters lead to large deviation in these reference models.

4.2 Modified correlation

To develop a comprehensive subcooled flow boiling model, the nucleation boiling heat transfer model is modified initially. In order to reflect the influence of the thermophysical properties of the heated surface on the subcooled flow boiling heat transfer, the Rohsenow pool boiling heat transfer model is modified. The main concept of Rohsenow's method is that the heat transfers from the wall directly to the liquid with an increased heat transfer rate due to the agitation of liquid by the departing vapor bubbles. Based on this logical explanation of the boiling heat transfer mechanism, the usual type of convective heat transfer equation was used:

$$Nu_b = C Re_b^m Pr^n, \quad (23)$$

Where Nu_b is the bubble Nusselt number, Re_b is the bubble Reynolds number, and C is an empirical constant. Equation (23) can be written in the final form as:

$$\frac{c_{pl}\Delta T_{sat}}{h_{lg}} = C_{sf} \left[\frac{q}{\mu h_{lg}} \sqrt{\frac{\sigma}{g(\rho_l - \rho_g)}} \right]^m Pr^n. \quad (24)$$

It should be pointed out that the nucleate boiling curve of subcooled flow boiling is substantially different from that of pool boiling curve, according to the comparison of experimental results of subcooled flow boiling and pool boiling with the same heated surface [29]. In subcooled flow boiling, due to the influence of flow, the gradient of the thermal boundary layer near the wall is larger. Thus, the bubble characteristics in the subcooled flow boiling process will be different from the pool boiling, which will eventually lead to different disturbances of

the liquid near the wall caused by generated bubbles. Therefore, it is necessary to re-establish the relationship between the dimensionless factors in equation (23).

Since only one fluid 50/50 EG/W is considered in this research, its Prandtl number is influenced only by the system pressure. Thus, the relationship between the dimensionless wall temperature and the heat flux of two heated surfaces was initially accessed under various system pressure conditions, as shown in Figure 13. Note that all the data used here were derived from the FDB region where the nucleate boiling heat transfer is the main heat transfer mechanism. It can be seen that $m = 0.75$ can well fit the data of the two heated surfaces.

In order to reflect the influence of the thermophysical properties of the heated surface on the boiling heat transfer, the following parameter is adopted accordingly:

$$\omega = \sqrt{\frac{c_{ps}k_s\rho_s}{c_{pl}k_l\rho_l}}, \quad (25)$$

where c_p , k , and ρ are specific heat capacity, thermal conductivity, and density, respectively, and subscripts l and s refer to liquid and heated surfaces, respectively.

The C_{sf} in equation (24) is set to account for the effect of fluid–solid interaction on boiling heat transfer. According to published references [7,25], ω has an exponential relationship with the boiling heat flux. Thereby, the authors now define $C_{sf} = C\omega^a$. Thus, the equation (24) can be transformed into:

$$CPr^n = \frac{1}{\omega^a} \left(\frac{c_{pl}\Delta T_{sat}}{h_{lg}} \right) / \left[\frac{q_w}{h_{lg}\mu_l} \sqrt{\frac{\sigma}{g(\rho_l - \rho_g)}} \right]^{0.75}, \quad (26)$$

Figure 13: The relationship between dimensionless wall temperature and heat flux.

where $C = 3.62$ and $a = 1.86$ yield the best fit to the experimental data. The final nucleate boiling equation is given as follows:

$$q_{nb} = \mu_l h_{lg} \left[\frac{\sigma}{g(\rho_l - \rho_g)} \right]^{-1/2} \left(\frac{c_{pl} \Delta T_{sat}}{3.62 \omega^{-1.86} h_{lg} \text{Pr}^{0.75}} \right)^{1/0.75}. \quad (27)$$

The comparison between the predicted results of equation (27) and the experimental data of nucleate boiling is shown in Figure 14. It can be seen that the maximum deviation of the predicted results is within $\pm 20\%$, indicating that the proposed nucleate boiling heat transfer model can well reflect the influence of thermophysical parameters of heated surface on boiling heat transfer.

For obtaining the final subcooled flow boiling heat transfer model, the power-type addition method in the literature [17] is adopted:

$$q^b = (q_{sp})^b + (q_{nb})^b, \quad (28)$$

Figure 14: Comparison between experimental data and predictions of the modified nucleation model.

Figure 15: Comparison between experimental data and predictions of modified subcooled flow boiling model.

where b is the parameter that needs to be fitted. q_{sp} is the heat flux of the single-phase flow heat transfer, calculated by equations (7) and (12); q_{nb} is the heat flux of nucleate boiling, calculated by equation (27).

The final fitting shows that $b = 6.5$ can well predict the experimental data in this article. The comparison between the predicted value of the equation (28) and the experimental value is shown in Figure 15. It is shown that prediction results agree with the experimental data very well with the maximum deviation less than 20%.

5 Conclusion

Subcooled flow boiling heat transfer model is important for ICE cylinder head design. The present study reports the subcooled flow boiling experiment of 50/50 EG/W in a horizontal rectangular channel with a flat plane heated surface of aluminum and cast iron. The experiment was conducted under the condition similar to that of practical ICE applications, aiming to simulate the flow boiling characteristics in the real cooling passages of a cylinder head in ICEs. The main concluding remarks are as follows:

- (1) The effects of operation conditions, bulk temperature, bulk velocity, and system pressure, on subcooled flow boiling heat transfer are similar for aluminum and cast iron heated surfaces. And under same system pressure, the flow boiling curves of different bulk temperatures and velocities tend to converge to an asymptote, which implies in this region that the effect of these operation parameters disappears.
- (2) Besides above similarities, the aluminum surface had a better performance than the cast iron surface in terms of heat transfer coefficient in the boiling region. The difference between these two surfaces is concluded as results of different surface thermophysical properties.
- (3) A modified wall heat flux model is proposed based on the power-type addition method. The proposed model modifies the nucleation boiling contribution by introducing a new parameter which accounts for the influence of thermophysical properties of the heated surface on the boiling process. The comparison with experimental data shows good agreement indicating that the proposed model could help to improve the design of ICE cylinder head.

Funding information: This work was supported by the National Key Research and Development Project of

China (Grant No. 2017YFB0103504) and National Natural Science Foundation of China (Grant No.51576116).

Conflict of interest: The authors state no conflict of interest.

s heated surface
sat saturation
sub subcooling
sp single phase
w wall

Nomenclature

Latin symbols

A	Cross-sectional area of the duct, m^2
C_p	specific heat, J/kg K
C_{sf}	fluid–solid combination constant
D_h	hydraulic diameter, mm
g	gravitational acceleration, m/s^2
G	mass flow rate, $\text{kg/m}^2 \text{s}$
h_{fg}	latent heat, J/kg
k	thermal conductivity, W/m K
L	length of heated surface, m
P	system pressure, bar
q	heat flux, W/m^2
T	temperature, $^\circ\text{C}$
u	velocity, m/s
Bo	Boiling number = q/Gh_{fg}
Pr	Prandtl number
Re	Reynolds number

Greek symbols

ΔT_{sat}	wall superheat = $T_w - T_{\text{sat}}$, $^\circ\text{C}$
β	Liquid coefficient of thermal expansion, $1/\text{K}$
σ	surface tension, N/m
ρ	density, kg/m^3
μ	dynamic viscosity, kg/m s
γ	exponent in Rohsenow's correlation
ω	influence factor of surface material

Subscripts

b	bulk flow
fc	forced convection
g	vapor
l	liquid
nb	nucleate boiling
nc	natural convection

References

- [1] Taylor AMKP. Science review of internal combustion engines. *Energy Policy*. 2008;36:4657–67. doi: 10.1016/j.enpol.2008.09.001.
- [2] Wijayanta AT, Yaningsih I, Juwana WE, Aziz M, Miyazaki T. Effect of wing-pitch ratio of double-sided delta-wing tape insert on the improvement of convective heat transfer. *Int J Therm Sci*. 2020;151:106261. doi: 10.1016/j.ijthermalsci.2020.106261.
- [3] Hu X, Sun Q, Li G, Bai S. Numerical investigation of thermo-hydraulic performance of an opposed piston opposed cylinder engine water jacket with helical fins. *Appl Therm Eng*. 2019;159:159. doi: 10.1016/j.applthermaleng.2019.113824.
- [4] Enoki K, Ono M, Okawa T, Kristiawan B, Wijayanta AT. Water flow boiling heat transfer in vertical minichannel. *Exp Therm Fluid Sci*. 2020;117:117. doi: 10.1016/j.expthermflusci.2020.110147.
- [5] Enoki K, Ono M, Okawa T, Akisawa A, Mori H, Kristiawan B, et al. Two-phase flow regimes of refrigerant R134a in an oscillating horizontal rectangular minichannel conduit. *Int J Refrig*. 2020;118:261–8. doi: 10.1016/j.ijrefrig.2020.06.002.
- [6] Zheng D, Yang J, Wang J, Kabelac S, Sundén B. Analyses of thermal performance and pressure drop in a plate heat exchanger filled with ferrofluids under a magnetic field. *Fuel*. 2021;293:120432. doi: 10.1016/j.fuel.2021.120432.
- [7] Paz MC, Conde M, Suárez E, Concheiro M. On the effect of surface roughness and material on the subcooled flow boiling of water: experimental study and global correlation. *Exp Therm Fluid Sci*. 2015;64:114–24. doi: 10.1016/j.expthermflusci.2015.02.016.
- [8] Paz C, Conde M, Porteiro J, Concheiro M. Effect of heating surface morphology on active site density in subcooled flow nucleated boiling. *Exp Therm Fluid Sci*. 2017;82:147–59. doi: 10.1016/j.expthermflusci.2016.11.011.
- [9] Chen JC. Correlation for boiling heat transfer to saturated fluids in convective flow. *Ind Eng Chem Process Des Dev*. 1966;5:322–9. doi: 10.1021/i260019a023.
- [10] Steiner H, Kobor A, Gebhard L. A wall heat transfer model for subcooled boiling flow. *Int J Heat Mass Transf*. 2005;48:4161–73. doi: 10.1016/j.ijheatmasstransfer.2005.03.032.
- [11] Torregrosa AJ, Broatch A, Olmeda P, Cornejo O. Experiments on subcooled flow boiling in I.C. engine-like conditions at low flow velocities. *Exp Therm Fluid Sci*. 2014;52:347–54. doi: 10.1016/j.expthermflusci.2013.10.004.
- [12] Chen L, Zhou P, Huang R, Han X, Hua S, Li Z, et al. Experimental investigation on the suppression factor in subcooled boiling

- flow. *Appl Therm Eng.* 2018;135:549–58. doi: 10.1016/j.applthermaleng.2018.02.026.
- [13] Robinson K, Hawley JG, Campbell NAF. Experimental and modelling aspects of flow boiling heat transfer for application to internal combustion engines. *Proc Inst Mech Eng Part D J Automob Eng.* 2003;217:1209–15. doi: 10.1243/095440703769683289.
- [14] Nourbakhsh A, Bayareh M, Mohammadi A, Jahantighi S. Effect analysis on boiling heat transfer performance of an internal combustion engine at the shutdown time. *Int J of Therm Sci.* 2018;129:365–74. doi: 10.1016/j.ijthermalsci.2018.03.022.
- [15] Gholinia M, Pourfallah M, Chamani HR. Numerical investigation of heat transfers in the water jacket of heavy duty diesel engine by considering boiling phenomenon. *Case Studies in Therm Eng.* 2018;12:497–509. doi: 10.1016/j.csite.2018.07.003.
- [16] Hua S, Huang R, Li Z, Zhou P. Experimental study on the heat transfer characteristics of subcooled flow boiling with cast iron heating surface. *Appl Therm Eng.* 2015;77:180–91. doi: 10.1016/j.applthermaleng.2014.11.082.
- [17] Ramstorfer F, Steiner H, Brenn G, Kormann C, Rammer F. Subcooled boiling flow heat transfer from plain and enhanced surfaces in automotive applications. *J Heat Transf.* 2008;130:81–99. doi: 10.1115/1.2780178.
- [18] Rohsenow WM. A method of correlating heat transfer data for surface boiling of liquids. *Trans of ASME.* 1952;84:969–75.
- [19] Steiner H, Breitschädel B, Brenn G, Petutschnig H, Samhaber C. Nucleate boiling flow experimental investigations and wall heat flux modelling for automotive engine applications. *Adv Comput Methods Heat Transf.* 2008;61:169–78. doi: 10.2495/HT080161.
- [20] Sun T, Teja AS. Density, viscosity, and thermal conductivity of aqueous ethylene, diethylene, and triethylene glycol mixtures between 290 and 450 K. *J Chem Eng Data.* 2003;48:198–202. doi: 10.1021/je025610o.
- [21] Bohne D, Fischer S, Obermeier E. Thermal, conductivity, density, viscosity, and prandtl-numbers of ethylene glycol-water mixtures. *Ber Bunsen Phys Chem.* 1984;88:739–42.
- [22] Popiel CO, Wojtkowiak J. Simple formulas for thermophysical properties of liquid water for heat transfer calculations (from 0 to 150°C). *Heat Transf Eng.* 1998;19:87–101. doi: 10.1080/01457639808939929.
- [23] Robinson K, Hawley JG, Hammond GP, Owen NJ. Convective coolant heat transfer in internal combustion engines. *Proc Inst Mech Eng Part D J Automob Eng.* 2003;217:133–46. doi: 10.1177/095440700321700207.
- [24] Lucic A, Emans M, Mayinger F, Zenger C. Interferometric and numerical study of the temperature field in the boundary layer and heat transfer in subcooled flow boiling. *Int J Heat Fluid Flow.* 2004;25:180–95. doi: 10.1016/j.ijheatfluidflow.2003.11.004.
- [25] Pioro IL, Rohsenow W, Doerffer SS. Nucleate pool-boiling heat transfer. I: review of parametric effects of boiling surface. *Int J Heat Mass Transf.* 2004;47:5033–44. doi: 10.1016/j.ijheatmasstransfer.2004.06.019.
- [26] Zou L, Jones BG. Heating surface material's effect on subcooled flow boiling heat transfer of R134a. *Int J Heat Mass Transf.* 2013;58:168–74. doi: 10.1016/j.ijheatmasstransfer.2012.11.036.
- [27] Chen Z, Wu F, Utaka Y. Numerical simulation of thermal property effect of heat transfer plate on bubble growth with microlayer evaporation during nucleate pool boiling. *Int J Heat Mass Transf.* 2018;118:989–96. doi: 10.1016/j.ijheatmasstransfer.2017.11.083.
- [28] Yu W, France DM, Zhao W, Singh D, Smith RK. Subcooled flow boiling heat transfer to water and ethylene glycol/water mixtures in a bottom-heated tube. *Exp Heat Transf.* 2016;29:593–614. doi: 10.1080/08916152.2015.1046018.
- [29] Sparrow EM. The determination of forced-convection surface-boiling heat transfer. *J Heat Transf.* 1964;86:365–72. doi: 10.1115/1.3688697.