New prediction methods for CO$_2$ evaporation inside tubes: Part II — An updated general flow boiling heat transfer model based on flow patterns

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Abstract

Corresponding to the updated flow pattern map presented in part I of this study, a general flow pattern based flow boiling heat transfer model was developed for CO$_2$ on the basis of the Cheng-Ribatski-Wojtan-Thome flow boiling heat transfer model [1]. Compared to that model, the updated flow boiling model is applicable to a wider range of conditions: tube diameters from 0.6 to 10 mm, mass velocities from 50 to 1500 kg/m$^2$s, heat fluxes from 1.8 to 46 kW/m$^2$ and saturation temperatures from -28 to 25 $^\circ$C (reduced pressures from 0.21 to 0.87). In addition, a heat transfer correlation for mist flow was newly proposed and heat transfer in the bubbly flow regime was added to the model. The updated model was compared to an extensive experimental database of 1108 data points, including 790 new data points in addition to the 318 data points used in the previous Cheng-Ribatski-Wojtan-Thome model [1]. Good agreement between the predicted and experimental data was found, with 71.4% of the entire database and 83.2% of the database without the dryout and mist flow data were predicted within ±30%.

Keywords: Model; Flow patterns; Flow boiling; Heat transfer; Macro-channel; Micro-channel; CO$_2$
1. Introduction

As pointed out in part I of this study, the flow boiling heat transfer characteristics of CO\textsubscript{2} are quite different from those of the conventional refrigerants. Due to its low critical temperature ($T_{\text{crit}} = 31.1^\circ$C) and high critical pressure ($p_{\text{crit}} = 73.8$ bar), CO\textsubscript{2} is utilized at much higher operating pressures compared to other conventional refrigerants. The higher operating pressures result in high vapor densities, very low surface tensions, high vapor viscosities and low liquid viscosities. High pressures and low surface tensions have major effects on nucleate boiling heat transfer characteristics and previous experimental studies have suggested a clear dominance of nucleate boiling heat transfer even at very high mass flux [1, 2]. Therefore, CO\textsubscript{2} has much higher heat transfer coefficients than those of conventional refrigerants at the same saturation temperature and the available heat transfer correlations generally underpredict the experimental data of CO\textsubscript{2}. In addition, previous experimental studies have demonstrated that dryout trends occur earlier at moderate vapor qualities in CO\textsubscript{2}, particularly at high mass flux and high temperature conditions [1, 2].

Thome and Ribatski [2] conducted an overall review of flow boiling heat transfer and two-phase flow of CO\textsubscript{2} in the literature and they found that not one of the available prediction methods was able to predict the experimental data of CO\textsubscript{2} well. Therefore, they suggested that a new flow boiling heat transfer prediction method should be developed and the flow boiling heat transfer model should include CO\textsubscript{2} effects on the annular to mist flow transition in order to more accurately predict heat transfer coefficients at moderate/high vapor qualities. In response, Cheng et al. [1] proposed a new flow boiling heat transfer model and flow pattern map for carbon dioxide evaporating inside horizontal tubes. The new flow boiling heat transfer model and flow pattern map were developed according to the model by Wojtan et al. [3, 4], which is an updated version of the Kattan-Thome-Favrat flow pattern map and flow boiling heat transfer model [5-7]. The Cheng-Ribatski-Wojtan-Thome flow boiling heat transfer model [1] is a flow pattern based flow boiling heat transfer model which relates the flow patterns to the corresponding heat transfer mechanisms, thus, different from the numerous empirical models, such as the correlations of Chen [8], Shah [9], Gungor and Winterton [10], Kandlikar [11], Liu and Winterton [12], etc., which do not include flow pattern information. The Cheng-Ribatski-Wojtan-Thome flow boiling heat transfer model is applicable to a wide range of conditions: tube diameters (equivalent diameters for non-circular channels) from 0.8 to 10 mm, mass fluxes from 170 to 570 kg/m$^2$s, heat fluxes from 5 to 32 kW/m$^2$, saturation temperatures from –28 to 25$^\circ$C (the corresponding reduced pressures are from 0.21 to 0.87). The model reasonably predicts the database and it covers channel diameters found in most of CO\textsubscript{2} flow boiling applications. However, their model is limited by its parameter ranges from being applicable to some important applications, for example the mass velocity ranges from 50 to 1500 kg/m$^2$s in CO\textsubscript{2} automobile air conditioning systems. In addition, the heat fluxes in some applications go beyond the maximum value in the
Cheng-Ribatski-Wojtan-Thome flow boiling heat transfer model. Furthermore, the model does not extrapolate well to these conditions. Therefore, it is necessary to update the model for CO₂ to cover a wider range of conditions.

In the present study, a new flow boiling heat transfer model was developed by modifying the Cheng-Ribatski-Wojtan-Thome model. By incorporating the updated new flow pattern map developed in part I of this study, the new model is physically related to the flow regimes of CO₂ evaporation, and thus correspondingly the new heat transfer model has been extended to a wider range of conditions. The proposed model predicted reasonably well an extensive experimental database derived from the literature.

2. CO₂ flow boiling heat transfer database and comparisons

2.1 Selection of CO₂ flow boiling heat transfer data

Thirteen independent experimental studies from different laboratories have been carefully selected to form the present database for flow boiling heat transfer of CO₂. These comprise 13 experimental studies containing six studies (Knudsen and Jensen [13], Yun et al. [14], Yoon et al. [15], Koyama et al. [16], Pettersen [17] and Yun et al. [18]) used in the Cheng-Ribatski-Wojtan-Thome flow boiling heat transfer model [1] and seven new studies (Gao and Honda [19, 20], Tanaka et al. [21], Hihara [22], Shinmura et al. [23], Zhao et al. [24, 25], Yun et al. [26, 27] and Jeong et al. [28]). The details of the test conditions of the database are summarized in Table 1. The channels tested include single circular channels and multi-channels with circular and rectangle cross-sections for a wide range of test conditions, using either electrical or fluid heating. The data were taken from tables where available or by digitizing the heat transfer data from graphs in these publications. All together, 1124 (1108 data points were used in this study) heat transfer data points including the heat transfer data (total 334 data points and 318 data points were used) in our previous study [1] and new heat transfer data (790 data points) were obtained. Special care was taken in order to extract experimental data in the dryout region and mist flow since these are important regimes in the design of CO₂ evaporators.

In the present study, the physical properties of CO₂ have been obtained using the REFPROP of NIST [29]. For non-circular channels, equivalent diameters rather than hydraulic diameters were used. Using an equivalent diameter gives the same mass flux as in the non-circular channel and thus correctly reflects the mean liquid and vapor velocities, something using a hydraulic diameter does not. The equivalent diameter is calculated with Eq. (1) in part I. The heat flux of the original perimeter is applied to the perimeter of the equivalent channels using the perimeter ratio.

2.2 Analysis of the flow boiling heat transfer database
An extensive literature survey on flow boiling heat transfer related to macro-scale channels when $D > 3$ mm and micro-scale channels when $D \leq 3$ mm was conducted. Such a distinction between macro- and micro-scale by the threshold diameter of 3 mm is adopted due to the lack of a well-established theory, as pointed out by Cheng et al. [1] and Thome and Ribatski [2]. In order to develop a general flow boiling heat transfer prediction model, extensive comparisons of the data available in the literature were made. However, some of the data available have not been selected due to various reasons. By carefully analysing the experimental data, we have found that only 13 papers (4 papers related to macro-scale and 9 papers related to micro scale), including the 6 papers used in our previous study [1], are useful. Therefore, a database including the experimental data from these papers for flow boiling heat transfer have been set up as the first step in this study. Experimental data in some papers were discharged or ignored because: i) the same data were in more than one paper by the same authors; ii) some necessary information of the experimental conditions, viz. saturation temperature, vapor quality or tube length was missing; iii) some data showed extremely strange parametric trends; iv) some data were physically unreasonable; v) the uncertainties of some data were very large; and vi) some data were only presented in correlated form and could not be extracted. Discussion of some data has already been presented in our previous study [1]. Thus, below only some of the new data sets are discussed here.

Fig. 1 shows the experimental heat transfer data of Park and Hrnjak [30, 31] at a mass velocity of 400 kg/m$^2$s, a saturation temperature of -15 °C and a tube diameter of 6.1 mm at heat fluxes of 5, 10 and 15 kW/m$^2$. According to this study, the heat transfer coefficient increases with increasing heat flux, a trend which is reasonable. In many other experimental studies, dryout in flow boiling of CO$_2$ occurs at significantly lower vapor qualities (earlier) than for conventional refrigerants such as R-22, R-134a and so on, while the Park-Hrnjak data do not show any dryout (characterized by a sharp fall off in heat transfer), even at a vapor quality around 0.9. In addition, their heat transfer data are significantly different in value from comparable data for 6 mm and 10.06 mm tubes in two other earlier studies [13, 14], both aspects which were not addressed in their paper. Therefore, the Park-Hrnjak data have been excluded from the present database. Fig. 2 shows the experimental heat transfer data of Hihara [32] at a mass velocity of 360 kg/m$^2$s, a saturation temperature of 15 °C and a heat flux of 18 kW/m$^2$ with two different tube diameters, 4 and 6 mm. It shows that the flow boiling heat transfer coefficients of the 4 mm tube are twice those of the 6 mm tube. In addition, the trends of the heat transfer coefficients are totally different. As both diameters are in the range of macro-scale, it seems unreasonable that the diameter can have such a dramatic effect on the heat transfer values and trends. Therefore, the experimental data of Hihara have been excluded. Fig. 3 shows the experimental heat transfer data of Gasche [33] at a mass velocity of 96 kg/m$^2$s, a saturation temperature of 23.3 °C, a tube diameter of 0.8 mm and a heat flux of 1.81 kW/m$^2$. For the same test conditions, the heat transfer coefficients change by more than 2 times in magnitude. Hence,
his experimental data have been excluded. As already pointed out in part I, the two-phase pressure drop data of Wu et al. [34] seem to be incorrect, and so do their flow boiling heat transfer data since the local saturation temperature is deduced from the pressure drop. Therefore, their experimental heat transfer data have been excluded as well.

Although some anomalous data have already been excluded as pointed out earlier, the heat transfer data in the database show still some different behaviors at similar test conditions. Some of the analysis can be found in our previous study [1] and therefore, none is presented in here. In all, the experimental data from the different studies still show somewhat different heat transfer trends and thus will affect the accuracy of the new general flow boiling heat transfer model developed for CO\textsubscript{2} in the present study since no conclusive reasons for the contradicting trends could be found and it is not possible to say which study is right.

3. Updated flow boiling heat transfer model for CO\textsubscript{2}

To develop a general flow boiling heat transfer prediction method, it is important that the method is not only numerically accurate but that it also captures correctly the trends in the data. Most importantly, the flow boiling heat transfer mechanisms should be related to the corresponding flow patterns and be physically explained according to flow pattern transitions. Accordingly, an updated general flow boiling heat transfer model was developed using the updated flow pattern map for CO\textsubscript{2} developed in part I of this study. Corresponding to the updated flow pattern map, the updated flow boiling model has been extended to a wider range of conditions: tube diameters from 0.6 to 10 mm, mass velocities from 50 to 1500 kg/m\textsuperscript{2}s, heat fluxes from 1.8 to 46 kW/m\textsuperscript{2} and saturation temperatures from -28 to +25 °C (reduced pressures from 0.21 to 0.87). Several new modifications were implemented in the updated heat transfer model. A new dryout inception vapor quality correlation (Eq. (20) in part I) was updated and a new dryout completion vapor quality correlation (Eq. (25) in part I) was newly developed in the present study. Accordingly, the flow boiling heat transfer correlation in the dryout region was updated. In addition, a new mist flow heat transfer correlation for CO\textsubscript{2} was developed based on the CO\textsubscript{2} data. With these modifications, a new general flow boiling heat transfer model for CO\textsubscript{2} was developed to meet a wider range of conditions.

The Kattan-Thome-Favrat [5-7] general equation for the local flow boiling heat transfer coefficients $h_\text{tp}$ in a horizontal tube is used:

$$h_\text{tp} = \frac{\theta_\text{dry} h_\text{tp} + (2\pi - \theta_\text{dry}) h_\text{sat}}{2\pi}$$

(1)

where $\theta_\text{dry}$ is the dry angle defined in part I. The dry angle $\theta_\text{dry}$ defines the flow structures and the ratio of the tube perimeter in contact with liquid and vapor. In stratified flow, $\theta_\text{dry}$ equals the stratified angle $\theta_\text{strat}$ which is calculated
with Eq. (13) in part I. In annular and intermittent flows, \( \theta_{dry} = 0 \). For stratified-wavy flow, \( \theta_{dry} \) varies from zero up to its maximum value \( \theta_{strat} \). Stratified-wavy flow has been subdivided into three subzones (slug, slug/stratified-wavy and stratified-wavy) to determine \( \theta_{dry} \).

For slug zone (Slug), the high frequency slugs maintain a continuous thin liquid layer on the upper tube perimeter. Thus, similar to the intermittent and annular flow regimes, one has:

\[
\theta_{dry} = 0 \quad (2)
\]

For stratified-wavy zone (SW), the following equation is proposed:

\[
\theta_{dry} = \theta_{strat} \left( \frac{G_{strat} - G}{G_{strat} - G_{wavy}} \right)^{0.61} \quad (3)
\]

For slug-stratified wavy zone (Slug+SW) and bubbly flow, the following interpolation between the other two regimes is proposed for \( x < x_{IA} \):

\[
\theta_{dry} = \theta_{strat} \frac{x}{x_{IA}} \left( \frac{G_{strat} - G}{G_{strat} - G_{wavy}} \right)^{0.61} \quad (4)
\]

The vapor phase heat transfer coefficient on the dry perimeter \( h_v \) is calculated with the Dittus-Boelter [35] correlation assuming tubular flow in the tube:

\[
h_v = 0.023 Re_v^{0.8} Pr_v^{0.4} \frac{k_v}{D} \quad (5)
\]

where the vapor phase Reynolds number \( Re_v \) is defined as follows:

\[
Re_v = \frac{Gx_D}{\mu_v \varepsilon} \quad (6)
\]

The heat transfer coefficient on the wet perimeter \( h_{wet} \) is calculated with an asymptotic model that combines the nucleate boiling and convective boiling heat transfer contributions to flow boiling heat transfer by the third power:

\[
h_{wet} = (h_{nb} + h_{cb})^{1/3} \quad (7)
\]

The nucleate boiling heat transfer coefficient \( h_{nb} \) is calculated with the Cheng-Ribatski-Wojtan-Thome nucleate boiling correlation for \( \text{CO}_2 \) [1]:

\[
h_{nb} = 131 \rho_c^{-0.0063} (-\log_{10} \rho_c)^{-0.55} M^{-0.5} q^{0.58} \quad (8)
\]

The Cheng-Ribatski-Wojtan-Thome nucleate boiling heat transfer suppression factor for \( \text{CO}_2 \) is applied to reduce the nucleate boiling heat transfer contribution due to the thinning of the liquid film [1]:

If \( x < x_{IA} \), \( S = 1 \)

If \( x \geq x_{IA} \), \( S = 1 - 1.14 \left( \frac{D}{0.00753} \right) \left( 1 - \frac{\delta}{\delta_{IA}} \right)^{2.2} \quad (10)

Furthermore, if \( D > 7.53 \text{ mm} \), then set \( D = 7.53 \text{ mm} \).
The convective boiling heat transfer coefficient \( h_{cb} \) is calculated with the following correlation assuming an annular liquid film flow:

\[
h_{cb} = 0.0133 \, \text{Re}_d^{0.09} \, \text{Pr}_l^{0.14} \, \frac{k_f}{\delta}
\]

where the liquid film Reynolds number \( \text{Re}_d \) is:

\[
\text{Re}_d = \frac{4G(1-x)\delta}{\mu_l (1-\varepsilon)}
\]

The void fraction \( \varepsilon \) is calculated with Eq. (8) in paper part I and the liquid film thickness \( \delta \) is calculated with the expression proposed by El Hajal et al. [36]:

\[
\delta = \frac{D}{2} - \sqrt{\left(\frac{D}{2}\right)^2 - \frac{2A_p}{2\pi - \theta_{dry}}}
\]

where \( A_p \), based on the equivalent diameter, is cross-sectional area occupied by liquid-phase shown in Fig. 1 in part I. When the liquid occupies more than one-half of the cross-section of the tube at low vapor quality, this expression would yield a value of \( \delta > D/2 \), which is not geometrically realistic. Hence, whenever Eq. (13) gives \( \delta > D/2 \), \( \delta \) is set equal to \( D/2 \) (occurs when \( \varepsilon < 0.5 \)).

Combining the Cheng-Ribatski-Wojtan-Thome nucleate boiling heat transfer correlation for CO\(_2\) (Eq. 8) and the Cheng-Ribatski-Wojtan-Thome nucleate boiling heat transfer suppression factor correlation (Eqs. 9 and 10), the flow boiling heat transfer coefficient on the wet perimeter is calculated as follows:

\[
h_{nfb} = \left[ \left( Sh_{nb} \right)^3 + h_{cb}^3 \right]^{1/3}
\]

The heat transfer coefficient in mist flow is calculated by the new correlation developed in this study, which is a modification of the correlation by Groeneveld [37] for water and old refrigerants:

\[
h_m = 2 \times 10^{-8} \, \text{Re}_{Hr}^{1.97} \, \text{Pr}_l^{1.06} \, \text{Y}^{-1.83} \, \frac{k_f}{D}
\]

where the homogeneous Reynolds number \( \text{Re}_{Hr} \) and the correction factor \( \text{Y} \) are calculated as follows:

\[
\text{Re}_{Hr} = \frac{GD}{\mu_v} \left[ x + \frac{\rho_l}{\rho_v} (1-x) \right]
\]

\[
\text{Y} = 1 - 0.1 \left[ \frac{\rho_l}{\rho_v} - 1 \right] (1-x)^{-0.4}
\]

The heat transfer coefficient in the dryout region is calculated by a linear interpolation as

\[
h_{dryout} = h_p (x_a) - \frac{x-x_a}{x_{dry}-x_a} \left[ h_p (x_{dry}) - h_m (x_{dry}) \right]
\]
where \( h_{df}(x_{di}) \) is the two-phase heat transfer coefficient calculated with Eq. (1) at the dryout inception quality \( x_{di} \) and \( h_{mf}(x_{de}) \) is the mist flow heat transfer coefficient calculated with Eq. (15) at the dryout completion quality \( x_{de} \). Dryout inception quality \( x_{di} \) and dryout completion quality \( x_{de} \) are respectively calculated with Eqs. (20) and (25) in part I. If \( x_{de} \) is not defined at the mass velocity being considered, it is assumed that \( x_{de} = 0.999 \).

Corresponding to the bubbly flow regime in the updated flow pattern map, a heat transfer model for bubbly flow was added. In the absence of any data, the heat transfer coefficients in bubbly flow regime are calculated by the same method as that in the intermittent flow. First, Eq. (4) is used to calculate the dry angle \( \theta_{dry} \). Then, Eq. (1) is used to calculate the local flow boiling heat transfer coefficients.

4 Comparisons of the updated heat transfer model to the database

The updated general flow boiling heat transfer model was compared to the database in Table 1. As the updated model predicts the heat transfer data used in our previous study [1] similar to the Cheng-Ribitski-Wojtan-Thome flow boiling heat transfer model, those comparisons are not shown in this study but are included in the statistical analysis in Table 2. For low mass velocities, both the Cheng-Ribatski-Wojtan-Thome and the new general flow boiling heat transfer models may be used. However, for high mass velocities, only the new general flow boiling heat transfer model is applicable. Figs. 4 to 10 show the comparisons of the predictions by the updated flow boiling heat transfer model to the new flow boiling heat transfer database and the corresponding flow pattern maps. Fig. 4 shows the comparison of the predicted flow boiling heat transfer coefficients to the experimental heat transfer data of Gao and Honda [19, 20]. Fig. 5 shows the comparison of the predicted flow boiling heat transfer coefficients to the experimental heat transfer data of Tanaka et al. [21]. Fig. 6 shows the comparison to the data of Hihara [22]. Fig. 7 shows the comparison to the data of Shinmura et al. [23]. Fig. 8 shows the comparison to the data of Zhao et al. [24, 25]. Fig. 9 shows the comparison to the data of Yun et al. [26, 27]. Finally, Fig. 10 shows the comparison to the data of Jeong et al. [28]. According to these figures, the new general flow boiling heat transfer model not only captures the heat transfer trends well but also predicts the experimental heat transfer data well.

Fig. 11 (a) shows the comparison of the updated CO\(_2\) flow boiling heat transfer model to the new flow boiling heat transfer data in the database and Fig. 11 (b) shows the comparison of the predicted flow boiling heat transfer coefficients to the new flow boiling heat transfer data without the harder to predict (and harder to accurately measure) heat transfer data in the dryout and mist flow regimes. The new general flow boiling heat transfer model predicts these data reasonably well. Also note that many of the experimental data sets have a level of scatter ranging from 10 to 40%
themselves, which reflects the larger experimental errors when measuring large heat transfer coefficients and values near sharp transitions such as dryout.

Three criteria were used to analyze the accuracy of the flow boiling heat transfer model: the standard deviation, the mean error and the percent of data predicted within ±30%. In the error analysis, the entire data point set, the new data points and the data points in our previous study [1] were respectively analyzed. The statistical analysis of the predicted results is presented in Table 2, where 71.4% of the entire database including both the new data and the previous data (total of 1108 data points) and 83.2% of these data without the dryout and mist flow data are predicted within ±30%. Furthermore, 70.1% of the new experimental data (790 data points) and 84.7% of the new data without the dryout and mist flow data (484 data points) are predicted within ±30%. Comparing the updated CO₂ flow boiling heat transfer model to the previous version, it can be seen in Table 2 that the database considered in the previous study [1] is reasonably predicted by both versions.

For such a wide range of experimental data from different laboratory studies, the predictions are quite reasonable and encouraging. However, for the heat transfer in some regions such as stratified-wavy flow, dryout and mist flow, due to the lack of experimental data and the lower accuracy of such data points, the predicted results are not always satisfactory. It is important to highlight here the importance of accurate predictions for dryout and mist flow regions since they are typical working conditions in micro-scale channels of automobile air-conditionings and are confronted over large ranges of vapor qualities, contrary to stratified flow which is rarely observed. To provide more accurate heat transfer predictions for flow boiling of CO₂, more experimental data covering wide ranges of interest in practical applications are needed and it is suggested that more carefully designed experiments of flow boiling of CO₂ be conducted in the future.

5. Conclusions

A general flow pattern based flow boiling heat transfer model was developed for CO₂ on the basis of the Cheng-Ribatski-Wojtan-Thome flow boiling heat transfer model [1] by incorporating the updated flow pattern map presented in part I. Corresponding to the flow pattern map, the general flow boiling heat transfer model is applicable to a wider range of conditions: tube diameters from 0.6 to 10 mm, mass velocities from 50 to 1500 kg/m²s, heat fluxes from 1.8 to 46 kW/m² and saturation temperatures from -28 to 25 °C (reduced pressures from 0.21 to 0.87). In addition, a heat transfer correlation in mist flow was newly proposed and a heat transfer correlation for bubbly flow was added. The general flow boiling heat transfer model was compared to an extensive experimental database (a total of 1108 data points including the 318 data points used in the Cheng-Ribatski-Wojtan-Thome flow boiling heat transfer model [1])
and good agreement between the predicted and experimental data has been found, where 71.4% of the entire experimental database and 83.2% of these data without the dryout and mist flow data are predicted within ±30%.

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Nomenclature

$A_L$ cross-sectional area occupied by liquid-phase, m$^2$
$c_p$ specific heat at constant pressure, J/kgK
$D$ internal tube diameter, m
$D_{eq}$ equivalent diameter, m
$D_h$ hydraulic diameter, m
$G$ total vapor and liquid two-phase mass flux, kg/m$^2$s
$h$ heat transfer coefficient, W/m$^2$K
$k$ thermal conductivity, W/mK
$M$ molecular weight, kg/kmol
$N$ number of data points
$Pr$ liquid phase Prandtl number [$c_p\mu/k$]
$p$ pressure, Pa
$p_r$ reduced pressure [$p/p_{cr}$]
$q$ heat flux, W/m$^2$
$Re_H$ homogeneous Reynolds number [$GD/\mu_c[x + \rho_r/\rho_i(1-x)]$]
$Re_V$ vapor phase Reynolds number [$GD/(\mu_r\epsilon)$]
$Re_s$ liquid film Reynolds number [$4G(1-x)/\delta(\mu_L(1-\epsilon))$]
$S$ nucleate boiling suppression factor
$T$ temperature, °C
\( x \) vapor quality
\( Y \) correction factor

**Greek symbols**

\( \delta \) liquid film thickness, m
\( \varepsilon \) cross-sectional vapor void fraction
\( \bar{\varepsilon} \) average deviation, %
\( |\varepsilon| \) mean deviation, %
\( \mu \) dynamic viscosity, Ns/m\(^2\)
\( \theta \) angle of tube perimeter, rad
\( \rho \) density, kg/m\(^3\)
\( \sigma \) standard deviation, %

**Subscripts**

\( cb \) convection boiling
\( crit \) critical
\( de \) dryout completion
\( di \) dryout inception
\( dry \) dry
\( dryout \) dryout region
\( IA \) intermittent flow to annular flow
\( L \) liquid
\( mist \) mist flow
\( nb \) nucleate boiling
\( sat \) saturation
\( tp \) two-phase flow
\( V \) vapor
\( wet \) on the wet perimeter
<table>
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<th>Description</th>
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[29] REFPROP. NIST Refrigerant Properties Database 23, Gaithersburg, MD, 1998, Version 6.01


List of table captions

Table 1. Database of CO$_2$ flow boiling heat transfer

Table 2. Statistical analysis of the predictions of flow boiling heat transfer coefficients
Fig. 1. Experimental flow boiling heat transfer data of Park and Hrnjak [30, 31] at the experimental conditions: \( G = 400 \text{ kg/m}^2\text{s}, T_{\text{sat}} = -15^\circ\text{C} \) and \( D_{\text{eq}} = 6.1 \text{ mm} \).

Fig. 2. Experimental flow boiling heat transfer data of Hihara [32] at the experimental conditions: \( G = 360 \text{ kg/m}^2\text{s}, T_{\text{sat}} = 15^\circ\text{C} \) and \( q = 18 \text{ kW/m}^2 \).

Fig. 3. Experimental flow boiling heat transfer data of Gasche [33] at the experimental conditions: \( G = 96 \text{ kg/m}^2\text{s}, T_{\text{sat}} = 23.3^\circ\text{C} \), \( D_{\text{eq}} = 0.833 \text{ mm} \) and \( q = 1.81 \text{ kW/m}^2 \).

Fig. 4. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Gao and Honda [19, 20]; (b) The corresponding flow pattern map; \( D_{\text{eq}} = 3 \text{ mm}, G = 390 \text{ kg/m}^2\text{s}, T_{\text{sat}} = 10^\circ\text{C}, q = 20 \text{ kW/m}^2 \).

Fig. 5. (a) Comparison of the predicted flow boiling heat transfer coefficients to the experimental heat transfer data of Tanaka et al. [21]; (b) The corresponding flow pattern map. \( D_{\text{eq}} = 1 \text{ mm}, G = 360 \text{ kg/m}^2\text{s}, T_{\text{sat}} = 15^\circ\text{C}, q = 9 \text{ kW/m}^2 \).

Fig. 6. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Hihara [22]; (b) The corresponding flow pattern map. \( D_{\text{eq}} = 1 \text{ mm}, G = 720 \text{ kg/m}^2\text{s}, T_{\text{sat}} = 15^\circ\text{C}, q = 18 \text{ kW/m}^2 \).

Fig. 7. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Shinmura et al. [23]; (b) The corresponding flow pattern map. \( D_{\text{eq}} = 0.6 \text{ mm}, G = 400 \text{ kg/m}^2\text{s}, T_{\text{sat}} = 5.83^\circ\text{C}, q = 20 \text{ kW/m}^2 \).

Fig. 8. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Zhao et al. [24, 25]; (b) The corresponding flow pattern map. \( D_{\text{eq}} = 1.15 \text{ mm}, G = 300 \text{ kg/m}^2\text{s}, T_{\text{sat}} = 10^\circ\text{C}, q = 11 \text{ kW/m}^2 \).

Fig. 9. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Yun et al. [26, 27]; (b) The corresponding flow pattern map. \( D_{\text{eq}} = 2 \text{ mm}, G = 1500 \text{ kg/m}^2\text{s}, T_{\text{sat}} = 5^\circ\text{C}, q = 30 \text{ kW/m}^2 \).

Fig. 10. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Jeong et al. [28]; (b) The corresponding flow pattern map. \( D_{\text{eq}} = 2.3 \text{ mm}, G = 450 \text{ kg/m}^2\text{s}, T_{\text{sat}} = 10^\circ\text{C}, q = 8 \text{ kW/m}^2 \).

Fig. 11. Comparison of the predicted flow boiling heat transfer coefficients to the new data in the database (1- Gao and Honda [19, 20], 2- Tanaka et al. [21], 3-Hihara [22], 4-Shinmura et al. [23], 5-Zhao et al. [24, 25], 6-Yun et al. [26, 27], 7-Jeong et al. [28]). (a) Comparison of the predicted results to the entire new database; (b) Comparison of the predicted results to the new data without the dryout and mist flow data.
Table 1 Database of flow boiling heat transfer for CO₂

<table>
<thead>
<tr>
<th>Data source</th>
<th>Channel configuration and material</th>
<th>Equivalent diameter $D_{eq}$ (mm)</th>
<th>Saturation temperature $T_{sat}$ (°C)</th>
<th>Reduced pressure $p_r$</th>
<th>Mass flux $G$ (kg/m²s)</th>
<th>Heat flux $q$ (kW/m²)</th>
<th>Data points</th>
<th>Heating method</th>
</tr>
</thead>
<tbody>
<tr>
<td>*Knudsen and Jensen [13]</td>
<td>Single circular tube, stainless steel</td>
<td>10.06</td>
<td>-28</td>
<td>0.21</td>
<td>80</td>
<td>8, 13</td>
<td>16</td>
<td>Heated by condensing R-22 vapor</td>
</tr>
<tr>
<td>*Yun et al. [14]</td>
<td>Single circular tube, stainless steel</td>
<td>6</td>
<td>5, 10</td>
<td>0.54, 0.61</td>
<td>170, 240, 340</td>
<td>10, 15, 20</td>
<td>53</td>
<td>Electrical heating</td>
</tr>
<tr>
<td>*Yoon et al. [15]</td>
<td>Single circular tube, stainless steel</td>
<td>7.35</td>
<td>0, 5, 10, 15, 20</td>
<td>0.47, 0.54, 0.61, 0.69 0.78</td>
<td>318</td>
<td>12.5, 16.4, 18.6</td>
<td>127</td>
<td>Electrical heating</td>
</tr>
<tr>
<td>*Koyama et al. [16]</td>
<td>Single circular tube, stainless steel</td>
<td>1.8</td>
<td>0.26, 9.98, 10.88</td>
<td>0.47, 0.61, 0.62</td>
<td>250, 260</td>
<td>32.06</td>
<td>36</td>
<td>Electrical heating</td>
</tr>
<tr>
<td>*Pettersen [17]</td>
<td>Multi-channel with 25 circular channels, aluminium</td>
<td>0.8</td>
<td>0, 10, 20, 25</td>
<td>0.47, 0.61, 0.78, 0.87</td>
<td>190, 280, 380, 570</td>
<td>5, 10, 15, 20</td>
<td>46</td>
<td>Heated by water</td>
</tr>
<tr>
<td>*Yun et al. [18]</td>
<td>Multi-channels with rectangle channels</td>
<td>1.52 (1.14)*</td>
<td>5</td>
<td>0.54</td>
<td>200, 300, 400</td>
<td>10, 15, 20</td>
<td>56</td>
<td>Electrical heating</td>
</tr>
<tr>
<td>Gao and Honda [19, 20]</td>
<td>Single circular tube, stainless steel</td>
<td>3</td>
<td>-7, 10</td>
<td>0.39, 0.61</td>
<td>236, 390, 393, 590, 786, 1179</td>
<td>10, 20, 21</td>
<td>150</td>
<td>Electrical heating</td>
</tr>
<tr>
<td>Tanaka et al. [21]</td>
<td>Single circular tube, Stainless steel</td>
<td>1</td>
<td>15</td>
<td>0.69</td>
<td>360</td>
<td>9, 18, 36</td>
<td>119</td>
<td>Electrical heating</td>
</tr>
<tr>
<td>Hihara [22]</td>
<td>Single circular tube, stainless steel</td>
<td>1</td>
<td>15</td>
<td>0.69</td>
<td>720, 1440</td>
<td>9, 18, 36</td>
<td>150</td>
<td>Electrical heating</td>
</tr>
<tr>
<td>Shinmura et al. [23]</td>
<td>Multi-circular channels, aluminium</td>
<td>0.6</td>
<td>5.83</td>
<td>0.55</td>
<td>400</td>
<td>10, 20</td>
<td>48</td>
<td>Heated by hot water</td>
</tr>
<tr>
<td>Zhao et al. [24, 25]</td>
<td>Multi-triangle channels, aluminium</td>
<td>1.15 (0.86)*</td>
<td>10</td>
<td>0.61</td>
<td>300</td>
<td>11</td>
<td>11</td>
<td>Electrical heating</td>
</tr>
<tr>
<td>Yun et al. [26, 27]</td>
<td>Single circular channel</td>
<td>0.98, 2</td>
<td>5, 10</td>
<td>0.54, 0.61</td>
<td>1000, 1500</td>
<td>7.2, 7.3, 15.9, 16.2, 20, 26, 26.5, 30, 36, 46</td>
<td>224</td>
<td>Electrical heating</td>
</tr>
<tr>
<td>Jeong et al. [28]</td>
<td>Multi-rectangle channels, aluminium</td>
<td>2.3 (2)*</td>
<td>0, 5, 10</td>
<td>0.47, 0.54, 0.61</td>
<td>450, 600, 750</td>
<td>4, 8, 12</td>
<td>88</td>
<td>Electrical heating</td>
</tr>
</tbody>
</table>

* The data used in the previous study [1] and the values in the brackets are hydraulic diameters, for circular channels, the hydraulic diameter equals to equivalent diameter.
Table 2 Statistical analysis of the predicted flow boiling heat transfer coefficients

| Data used for comparison | Data points | Percentage of predicted points within ±30 % | Mean error $|\bar{e}|$ | Standard deviation $\sigma$ |
|--------------------------|-------------|---------------------------------------------|------------------|------------------|
| All data points including both new and previous data [14-28] | 1108        | 71.4 %                                      | 34 %             | 68.8 %           |
| All data points including both new and previous data points without the dryout and mist flow data [14-18] | 757         | 83.2 %                                      | 20.9 %           | 35.6 %           |
| All new data points [19-28] | 790         | 70.4 %                                      | 36.3 %           | 72.9 %           |
| All new data points without the dryout and mist flow data [19-28] | 484         | 84.7 %                                      | 18.7 %           | 26.5 %           |
| All data points in our previous study [14-18] | 318         | 74.2 %                                      | 28.4 %           | 50.2 %           |
| All data points without the dryout and mist flow data in our previous study [14-18] | 273         | 81 %                                        | 24.5 %           | 44.1 %           |
| All data points [14-18] in our previous study compared with Cheng-Ribatski-Wojtan-Thome model [1] | 318         | 75.5 %                                      | 27.1%            | 47.2 %           |
| All data points without the dryout and mist flow data [14-18] in our previous study compared with Cheng-Ribatski-Wojtan-Thome model [1] | 287         | 79.1%                                       | 23.5%            | 47.5%            |

$$\sigma = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (e_i - \bar{e})^2} ; \quad |\bar{e}| = \frac{1}{N} \sum_{i=1}^{N} |e_i| ; \quad e_i = \frac{\text{Predicted} - \text{Measured}}{\text{Measured}}$$
Vapor quality

Heat transfer coefficient \([\text{W/m}^2\text{K}]\)

\(q = 5 \text{ kW/m}^2\)
\(q = 10 \text{ kW/m}^2\)
\(q = 15 \text{ kW/m}^2\)

Fig.1. Experimental flow boiling heat transfer data of Park and Hrnjak [30, 31] at the experimental conditions: \(G = 400\) kg/m²s, \(T_{\text{sat}} = -15^\circ\text{C}\) and \(D_{\text{eq}} = 6.1\) mm.
Fig. 2. Experimental flow boiling heat transfer data of Hihara [32] at the experimental conditions: $G = 360 \text{ kg/m}^2\text{s}$, $T_{\text{sat}} = 15^\circ\text{C}$ and $q = 18 \text{ kW/m}^2$. 
Fig. 3. Experimental flow boiling heat transfer data of Gasche [33] at the experimental conditions: $G = 96 \text{ kg/m}^2\text{s}$, $T_{sat} = 23.3 \, ^\circ\text{C}$, $D_{eq} = 0.83 \, \text{mm}$ and $q = 1.81 \text{ kW/m}^2$. 
Fig. 4. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Gao and Honda [19, 20]; (b) The corresponding flow pattern map; ($D_{eq} = 3$ mm, $G = 390$ kg/m²s, $T_{sat} = 10$ °C, $q = 20$ kW/m²).

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Fig. 5. (a) Comparison of the predicted flow boiling heat transfer coefficients to the experimental heat transfer data of Tanaka et al. [21]; (b) The corresponding flow pattern map. ($D_{eq} = 1$ mm, $G = 360$ kg/m$^2$s, $T_{sat} = 15$ °C, $q = 9$ kW/m$^2$).
Fig. 6. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Hihara [22]; (b) The corresponding flow pattern map. ($D_{eq} = 1$ mm, $G = 720$ kg/m$^2$s, $T_{sat} = 15 \, ^\circ$C, $q = 18$ kW/m$^2$).
Fig. 7. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Shinmura et al. [23]; (b) The corresponding flow pattern map. ($D_{eq} = 0.6$ mm, $G = 400$ kg/m²s, $T_{sat} = 5.83$ °C, $q = 20$ kW/m²).
Fig. 8. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Zhao et al. [24, 25]; (b) The corresponding flow pattern map. ($D_{eq} = 1.15$ mm, $G = 300$ kg/m$^2$s, $T_{sat} = 10$ °C, $q = 11$ kW/m$^2$).
Fig. 9. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Yun et al. [26, 27]; (b) The corresponding flow pattern map. ($D_{eq} = 2$ mm, $G = 1500$ kg/m$^2$s, $T_{sat} = 5$ °C, $q = 30$ kW/m$^2$).
Fig. 10. (a) Comparison of the predicted heat transfer coefficients to the experimental data of Jeong et al. [28]; (b) The corresponding flow pattern map. \( D_{eq} = 2.3 \text{ mm}, \ G = 450 \text{ kg/m}^2\text{s}, \ T_{sat} = 10^\circ \text{C}, \ q = 8 \text{ kW/m}^2 \).
Fig. 11. Comparison of the predicted flow boiling heat transfer coefficients to the new data in the database (1- Gao and Honda [19, 20], 2- Tanaka et al. [21], 3-Hihara [22], 4-Shinmura et al. [23], 5-Zhao et al. [24, 25], 6-Yun et al. [26, 27], 7-Jeong et al. [28]). (a) Comparison of the predicted results to the entire new database; (b) Comparison of the predicted results to the new data without the dryout and mist flow data.