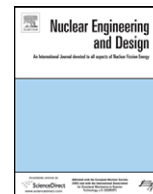




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# Numerical study on effect of gap width of narrow rectangular channel on critical heat flux enhancement

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## ABSTRACT

According to the flow passage characteristic of narrow rectangular channel and liquid film dry-out mechanics of annular flow critical heat flux (CHF), an annular flow CHF analytical model for narrow rectangular channel has been achieved. This model may be used to predict the CHF behavior of boiling two-phase flow in narrow rectangular channel with gap width of not being less than 0.0005 m (the equivalent diameter of this channel is 0.001 m). Through analyzing and calculating, when the inlet dimensionless gap width of narrow rectangular channel is within 30–85, the enhancement of CHF in channel is obvious. At the same time, according to the characteristic of two-phase flow, the new determinant laws of CHF in boiling two-phase flow system have been derived. Through analyzing and calculating, it is substantial that this determinant laws is appropriate. The best dimensionless gap width of heat flux enhancement has been achieved to be 45–75.

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## 1. Introduction

There are many evidence of enhancement of heat transfer as per characteristics of narrow channel (Du, 1997; Fujita et al., 1988; Ishibashi and Nishikawa, 1969; Pan, 2000; Shen, 2002). In the engineering field, there are many applications of narrow channels, for example, research reactor, new style nuclear power reactor and production reactor, etc., whose core are template or thimble and are made up by narrow channels. Narrow channels are also used as cooling system in chemical and aerospace industries. These industries have been developed quickly, and an urgent need of studying of flow and heat transfer characteristics in narrow channels is taken. So, it is an important and active researching aspect in thermal energy science and engineering.

When the gap width of narrow channel is less than 0.003 m, it has been shown that the behavior of surface tension is very significant, and the narrow channel can improve heat transfer obviously (Gan et al., 2004; Ishibashi and Nishikawa, 1969). As for the vapor–liquid two-phase flow in channels, the channels can be categorized according to the equivalent diameters: it is a big channel when the equivalent diameter  $\geq 0.003$  m, a narrow channel when the equivalent diameter is 0.001–0.003 m, a micro-channel when the equivalent diameter is less than 0.001 m. Fedorov and Viskanta's (2000) numerical result has shown that the width and depth of

channel has a very important influence on flow and heat transfer. Peng et al. (1994) have shown that there is a best geometric ratio for heat exchanger in micro-channels. In flow boiling field, researchers have finished many studies for heat transfer coefficient in different narrow channels, most results have shown that the narrow channel configurations have great influences on the heat transfer enhancement. These results have been summarized by researchers as follows (Ishibashi and Nishikawa, 1969; Pan, 2000; Shen, 2002): (1) disturbance of bubbles and frequency of escape have been accelerated by narrow channels, (2) there are micro-liquid film evaporate, convection and conduction of flat bubbles bottom in narrow channels, (3) with the gap width decreasing, the collision and friction is increasing during the procedure of bubble motion. However, Yang et al. (2006) have found that heat transfer is decreased by narrow channels in certain gap width.

The numerical study on effect of gap width of narrow rectangular channel on CHF enhancement can not be found in the open literature nearly. Therefore, in this paper, a method of numerical calculation is used to study this topic.

## 2. CHF model in narrow rectangular channel

The liquid film dry-out mechanics of annular flow CHF (Joel, 1992) has been known very well. The liquid film and liquid are drought out by steam flow carry-over along the wall of channel.

Fig. 1(a) is the schematic of narrow rectangular channel. In Fig. 1(b),  $m_d$ ,  $m_{ew}$ ,  $m_{eb}$  and  $q/h_{lg}$  are the deposition of droplet, the shear carry-over of droplet vapor phase, the droplet carry-over of

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**Nomenclature**

<i>A</i>	dimensionless cross-sectional area of liquid film
<i>B</i>	width of channel (m)
<i>D</i>	diameter (m)
<i>E</i>	rate of droplet for liquid in cross-sectional
<i>f</i>	flow resistance coefficient, $f = (\Delta PD_{hy}) / (2\rho U^2 L)$
<i>G</i>	mass flow flux ( $\text{kg m}^{-2} \text{s}^{-1}$ )
<i>g</i>	acceleration of gravity ( $\text{m s}^{-2}$ )
<i>H</i>	height of channel (m)
<i>h</i>	enthalpy ( $\text{J kg}^{-1}$ )
<i>j</i>	dimensionless coefficient of heat transfer or reduced velocity ( $\text{m s}^{-1}$ )
<i>L</i>	heated length (m)
<i>m</i>	carry-over of droplet ( $\text{kg m}^{-2} \text{s}^{-1}$ )
<i>Nu</i>	Nusselt number, $Nu = hD_{he}/\lambda$
<i>P</i>	system pressure (MPa)
<i>Pr</i>	Prandtl number
$\Delta P$	pressure drop (Pa)
<i>q</i>	heat flux ( $\text{W m}^{-2}$ )
<i>Re</i>	Reynolds number, $Re = (UD_{hy})/\nu$
<i>St</i>	Stanton number, $St = Nu/(RePr)$
<i>U</i>	velocity ( $\text{m s}^{-1}$ )
<i>W</i>	mass flow rate ( $\text{kg s}^{-1}$ )
<i>We</i>	Weber number, $We = G^2 D_{hy} / \sigma \rho$
<i>z</i>	axial length (m)

*Greek letters*

$\alpha$	void fraction
$\gamma$	dimensionless gap width of narrow rectangular channel
$\delta$	average thickness of liquid film (m)
$\mu$	coefficient of dynamic viscosity ( $\text{kg m}^{-1} \text{s}^{-1}$ )
$\nu$	coefficient of kinematic viscosity ( $\text{m}^2 \text{s}^{-1}$ )
$\rho$	density ( $\text{kg m}^{-3}$ )
$\sigma$	surface tension ( $\text{N m}^{-1}$ )
$\tau$	shear stress ( $\text{N m}^{-2}$ )

*Superscripts*

+	dimensionless velocity
---	------------------------

*Subscripts*

an	annular flow
c	critical
d	deposition of droplet
eb	droplet carry-over of steam void broken
ew	shear carry-over of droplet vapor phase
g	vapor phase
he	thermal equivalent
hy	hydraulic equivalent
in	inlet
l	liquid
lf	liquid film
lg	vaporization
m	average
TP	two-phase

steam void broken and the evaporation rate of liquid film, respectively. The thickness of liquid film is becoming less and less by above several factors acted on liquid film in channel. When the mass flow rate of liquid film reaches zero at channel wall, CHF will happen in this point.

The main difference of CHF model for annular flow is the calculated method. There are mainly two methods currently as follows.

(1) Multi-fluid model

The mechanics of physical phenomenon can be shown by multi-fluid model (Joel, 1992; Zhao and Zhu, 1992). However, there are too many field equations, and the calculation is very difficult. At the same time, because the laws of two-phase interface have not been studied deeply enough, and some expressions of structure have not been built very correctly. Therefore, the calculation accuracy of this mathematical model has been affected by the above problems.

(2) Analytical model

Celata and Zummo (1997) analytical model is based on liquid film dry-out mechanics. It was compared with the multi-fluid model, and shown that if the suitable deposition and carry-over of droplets consideration have been implemented in the simple analytical model, this model can predict the CHF of annular flow with a small error in wide range.

The flow patterns and CHF analytical model of narrow rectangular channel cannot be found in the open literature, at the same time, many experimental data show that the effects of mass flow flux, inlet subcooling, system pressure and critical enthalpy on CHF are identical. The related experimental data of effect of channel form on CHF are shown in Fig. 2 (Jackey et al., 1958; Kafengauz and Bocharov, 1959). Therefore, many researchers think that the CHF expression of round channel may be used to predict the CHF of narrow rectangular channel when the length and equivalent diameter of narrow rectangular channel are identical as those of the round channel.

2.1. The distinguish of annular flow region

The Mishima and Ishii (1984) criterion is used to distinguish whether the annular flow happened or not.

$$j_g \geq \frac{(\sigma g \Delta \rho / \rho_g^2)^{0.25}}{[\mu_l (\rho_l \sigma \sqrt{\sigma / g \Delta \rho})^{-0.5}]^{0.2}} \quad (1)$$

where  $j_g$  is the reduced velocity of vapor phase, m/s;  $\sigma$  is the surface tension, N/m;  $g$  is the acceleration of gravity,  $\text{m/s}^2$ ;  $\Delta \rho$  is the difference of density,  $\text{kg/m}^3$ ;  $\rho_g$  is the density of vapor phase,  $\text{kg/m}^3$ ;  $\mu_l$  is the coefficient of liquid dynamic viscosity,  $\text{kg/(ms)}$ ;  $\rho_l$  is the density of liquid phase,  $\text{kg/m}^3$ .

Eq. (1) can be used in round channel or rectangular channel, with the range of some parameters as follows: the pressure is 0.1–20.0 MPa, the mass flow flux is 8–15,000  $\text{kg/(m}^2 \text{s)}$ , the equivalent diameters is 0.001–0.025 m, the heated length-equivalent diameters ratio is 1–400, the subcooling is 10–255 °C, the quality is 0.1–1.

2.2. The calculation of mass flow rate of liquid film in annular flow region

When the onset of annular flow  $z_{an}$  is achieved by iteration, the mass flow rate of liquid film will be integrated by Eq. (2) in  $[z_{an}, L]$ .

$$\frac{dW_{lf}}{dz} = \pi D_{hy} \left( m_d - m_{ew} - m_{eb} - \frac{q}{h_{lg}} \right) \quad (2)$$

where  $W_{lf}$  is the mass flow rate of liquid film,  $\text{kg/s}$ ;  $z$  is the distance along flow direction, m;  $D_{hy}$  is the diameter of hydraulic equivalent, m;  $q$  is the heat flux,  $\text{W/m}^2$ ;  $h_{lg}$  is the latent heat of vaporization,  $\text{J/kg}$ .

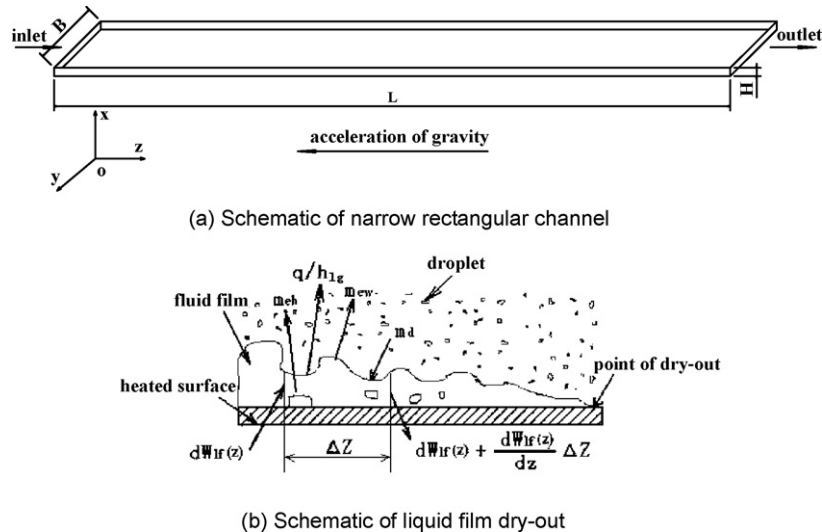


Fig. 1. Schematic of liquid film dry-out mechanics of annular flow in narrow rectangular channel.

The rate of deposition of droplet is calculated by Kataoka et al. (1983) expression.

$$m_d = 0.22Re_1^{0.74}E^{0.74} \left( \frac{\mu_g}{\mu_l} \right)^{0.26} \frac{\mu_l}{D_{hy}} \quad (3)$$

where  $Re_1$  is the Reynolds number of liquid, whose definition is as follows,  $Re = UD_{hy}/\nu$ ;  $E$  is the rate of droplet for liquid in cross-section.

$$E = 1 - \frac{j_{lf}}{j_1} \quad (4)$$

where  $j_{lf}$  is the reduced velocity of liquid film, m/s;  $j_1$  is the reduced velocity of liquid, m/s.

The reduced velocity of liquid film is defined as follows:

$$j_{lf} = \frac{4W_{lf}}{\rho_l \pi D_{hy}^2} \quad (5)$$

Because the unknown quantity  $W_{lf}$  is included in Eq. (2), an iteration method is used to calculate  $E$ .

### 2.3. The calculation of droplet carry-over rate

The droplet carry-over rate is made up by two parts.

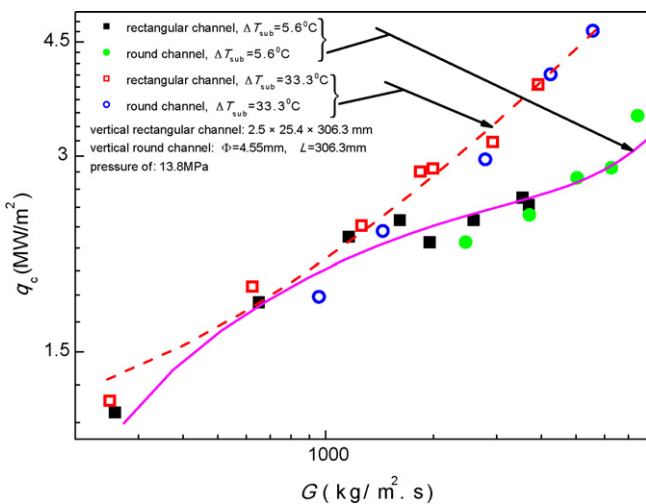


Fig. 2. The  $q_c$  trend of different type of channels with identical equivalent diameter.

(1)When the steam void is generated in liquid film due to nucleate boiling, and the steam void flow through liquid film into steam flow, the carry-over of liquid phase is caused by the broken of steam-liquid interface. Ueda et al. (1981) expression is used to calculate this carry-over behavior.

$$m_{eb} = 477 \left[ \frac{q^2 \delta}{h_{lg}^2 \sigma \rho_g} \right]^{0.75} \frac{q}{h_{lg}} \quad (6)$$

where  $\delta$  is the average thickness of liquid film, m.

In Eq. (6),  $\delta$  is iterated by

$$U_{lf,m}^+ = \frac{\int U^+ dY^+}{\delta} \quad (7)$$

$$U_{lf,m} = \frac{G_{lf}}{A_{lf} \rho_l} \quad (8)$$

$$U_{lf,m}^+ = \frac{U_{lf,m}}{\sqrt{\tau_w / \rho_l}} \quad (9)$$

$$A_{lf} = \frac{D_{hy}^2}{D_{hy}^2 - (D_{hy} - 2\delta)^2} \quad (10)$$

where  $U_{lf,m}^+$ ,  $Y^+$  and  $A_{lf}$  are the mean dimensionless velocity of liquid film, the dimensionless distance from the wall and dimensionless cross-sectional area of liquid film, respectively;  $U_{lf,m}$  is the average flow velocity of liquid film, m/s;  $G_{lf}$  is the mass flow flux of liquid film,  $kg/(m^2 s)$ ;  $\tau_w$  is the shear stress of channel wall,  $N/m^2$ .

(2)The liquid film is wavy normally, the shear carry-over of liquid phase is caused by high speed steam flow in the crest of liquid film wave. It is calculated by Kataoka et al.'s (1983) expression.

$$m_{ew} = 0.22Re_1^{0.74}E_\infty^{0.74} \left( \frac{\mu_g}{\mu_l} \right)^{0.26} \frac{\mu_l}{D_{hy}} \quad (11)$$

where  $E_\infty$  is the rate of equilibrium droplet.

The ratio of balance droplet is calculated by following expression.

$$E_\infty = \tanh(7.25 \times 10^{-7} We^{1.25} Re^{0.25}) \quad (12)$$

where  $We$  is the Weber number,  $We = G^2 D_{hy} / \sigma \rho$ .

Considering the difference of round channel and rectangular channel in geometry, the hydraulic equivalent diameter ( $D_{hy}$ ) and thermal equivalent diameter ( $D_{he}$ ) are used in this model. In this paper, the double wide sides are heated uniformly, except for the

double narrow sides are unheated. Therefore, above two definitions of qualitative dimensions are as follows:

$$D_{hy} = \frac{2BH}{B+H} \quad (13)$$

$$D_{he} = \frac{4BH}{2B} = 2H \quad (14)$$

where  $B$  is the width of narrow rectangular channel, m;  $H$  is the gap width of narrow rectangular channel, m.

Eq. (1) is used to distinguish whether the annular flow happened or not, if the assumed  $q_c$  is lower than the real  $q_c$ , annular flow will not happen; if annular flow is happening, the onset point  $z_{an}$  of annular flow can be achieved.  $m_d$ ,  $m_{ew}$ ,  $m_{eb}$  and  $W_{1f}$  are calculated in every advanced step length from  $z_{an}$  to the outlet of channel. At this time, if the assumed heat flux is lower than the real  $q_c$ , the mass flow rate of liquid film is larger than zero at the outlet; if the assumed heat flux is higher than the real  $q_c$ , the mass flow rate of liquid film is changed to zero before reaching the outlet. By changing  $q$ , until liquid film is zero, at this time,  $q$  is equal to  $q_c$ .

### 3. The testing of model and predicting calculation

#### 3.1. The result of testing

Based on the above model, a code has been programmed in FORTRAN language. And totally 57 experimental data of CHF in narrow rectangular channel, which come from National Key Laboratory of Bubble Physics and Natural Circulation, are used to test this model. The inlet cross-section width of test section is 0.066 m ( $B$ ), gap width is 0.0025 m ( $H$ ), the length of heated section is 0.750 m ( $L$ ), the double wide sides are heated uniformly. The pressure is 0.106–13.789 MPa, and the mass flow flux is 9.0–13,879.8 kg/(m<sup>2</sup> s). The de-ionized water is used as working medium. The results have been shown in Fig. 3.

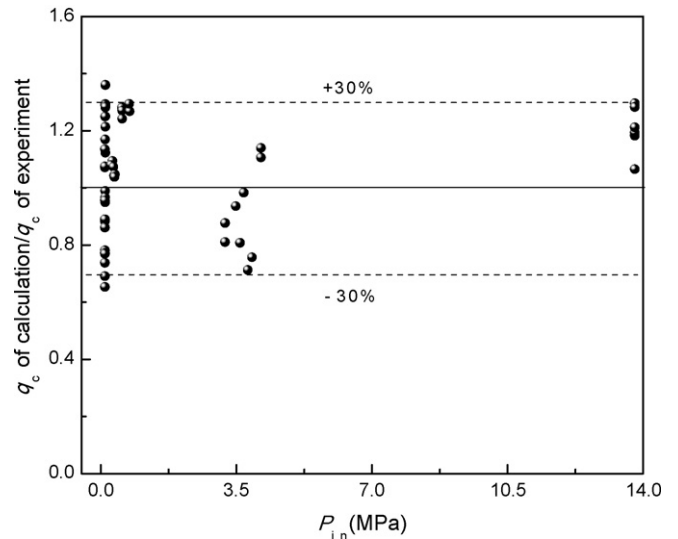
In Fig. 3, the predicting error of 93.0% data is within  $\pm 30\%$ . Generally speaking, the accuracy of this model is good, and it may be used to predict the CHF in narrow rectangular channel.

#### 3.2. The predicting calculation

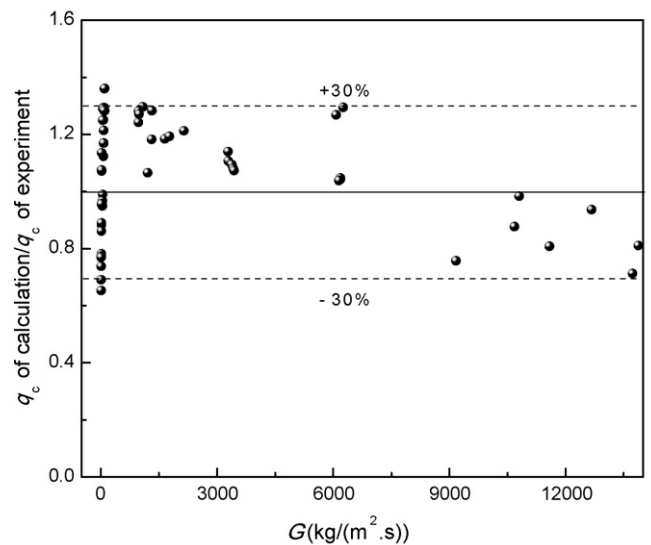
The physical dimensions of rectangular channel have been set again, the width of channel is 0.060 m, the gap width is 0.0005–0.0030 m, and the heated length is 0.750 m. When the width of channel is 0.060 m, the gap width is 0.0005 m, the corresponding hydraulic equivalent diameter is 0.001 m, which is the lowest limit of narrow rectangular channel hydraulic equivalent diameter generally. At the same time, considering about the application value of engineering, 0.0005 m ( $H$ ) is taken as the lowest limit for testing. The dimensionless gap width of inlet cross-section is defined as  $\gamma$ , whose expression can be found in Eq. (15). The change step of gap length is 0.0001 m. Within this paper,  $\gamma$  is between 20 and 120.

$$\gamma = \frac{B}{H} \quad (15)$$

When the inlet mass flow flux is set as 50, 100, 150 and 200 kg/(m<sup>2</sup> s), the inlet subcooling is 30 °C, and the system pressure is 2.0 MPa, the predicted results have been shown in Fig. 4. When the width of channel and the inlet mass flow flux are fixed, the inlet mass flow flux is changed linearly with the gap width of channel. Looking from four curves in Fig. 4, at this same inlet dimension, in other words, when  $\gamma$  is identical,  $q_c$  is decreased linearly with inlet mass flow flux decreased. Other results are compared with the results of linear fit, at the same operating conditions, when  $\gamma$  is equal to 30 and 85, there are two knee points in the slope absolute value of  $q_c$ . When  $\gamma$  is less than 30, with  $\gamma$  decreasing, the width



(a) Different system pressure



(b) Different inlet mass flow flux

Fig. 3. Comparison of calculation and experiment of CHF in narrow rectangular channel.

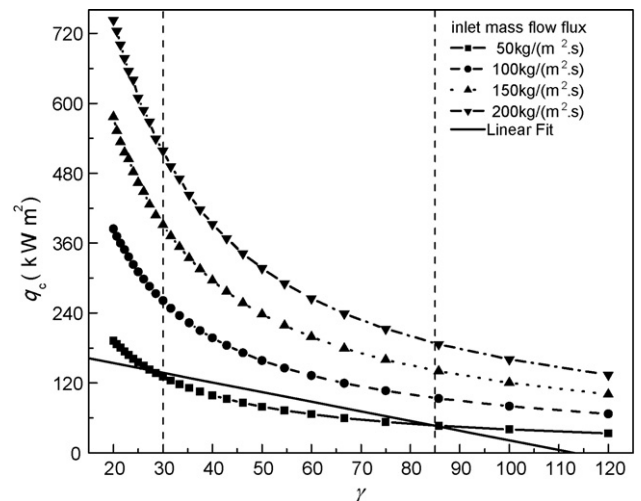


Fig. 4. The effect of predicted  $q_c$  under different  $\gamma$  in different inlet mass flow flux.



of channel and inlet mass flow flux are all fixed, so, the gap width of channel and inlet mass flow rate are increasing. At this time, the gap width is larger than 0.002 m, the characteristic of heat transfer enhancement is similar to big channel, so the slope absolute value of  $q_c$  is not changed nearly. When  $\gamma$  is larger than 85, the phenomenon of heat transfer is similar to when  $\gamma$  is less than 30. However, when  $\gamma$  is in 30–85, looking from the region between two dotted lines in these curves, the change rate of slope absolute value of  $q_c$  is maximum. In this region of  $\gamma$ , the slope absolute value of  $q_c$  is decreased with the increase of  $\gamma$ , in other words, the gap width is decreased, and the mass flow flux is decreased. It is proved that  $q_c$  is decreased with the decrease of mass flow flux. At the same time, with  $\gamma$  increased,  $q_c$  is increased at a definitive degree with gap width decreased. The reason is that, with the gap width decreasing, the collision and friction is increasing during the process of bubble motion, and the disturbance of bubbles and frequency of escape have been accelerated. So, more energy has been moved from the heated plate, and  $q_c$  is increased. Under the effect of above two factors, the slope absolute value of  $q_c$  is decreased largely. Therefore, when  $\gamma$  is in 30–85,  $q_c$  is increased at different inlet mass flow flux.

When, the inlet mass flow flux is set as 150 kg/(m<sup>2</sup> s), the system pressure is 4.0 MPa, but the inlet subcooling is 10, 20, 40 and 60 °C, respectively, the predicted results have been shown in Fig. 5. The results of linear fit are compared with predicted results, when  $\gamma$  is equal to 30 and 85, there are two knee points in the slope absolute value of  $q_c$ . Looking from two dotted lines in this figure, when  $\gamma$  is in 30–85, with the increasing of  $\gamma$ , the slope absolute value of  $q_c$  is decreased. It is shown that  $q_c$  is improved at different inlet subcooling.

When, the inlet mass flow flux is set as 100 kg/(m<sup>2</sup> s), the inlet subcooling is 30 °C, but the system pressure is 1.0, 3.0, 5.5 and 8.0 MPa, respectively, the predicted results have been shown in Fig. 6. The results of linear fit are compared with predicted results, when  $\gamma$  is equal to 30 and 85, there are two knee points in the slope absolute value of  $q_c$ . Looking from two dotted lines in this figure, when  $\gamma$  is in 30–85, with the increasing of  $\gamma$ , the slope absolute value of  $q_c$  is decreased. It can be concluded that  $q_c$  is improved at different system pressure.

Based on the above results, when  $\gamma$  is in 30–85, the enhancement of  $q_c$  is obvious. On the viewpoint of engineering application and its economics, the parameter range stated above may be taken as the reference range of geometry size of inlet cross-section in narrow rectangular channel to enhance  $q_c$ .

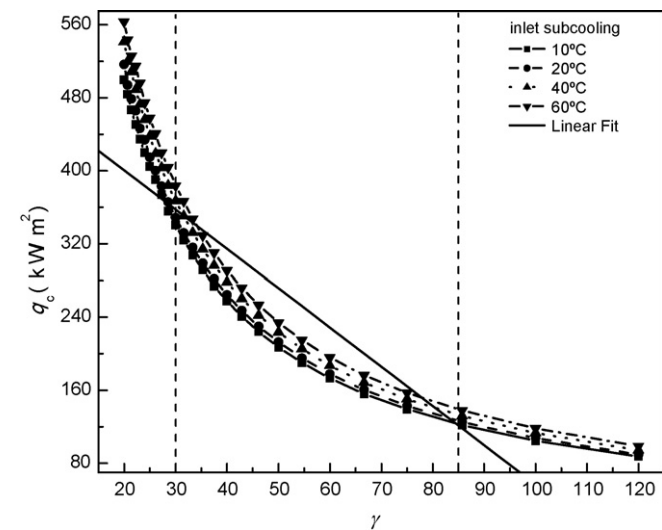


Fig. 5. The effect of predicted  $q_c$  under different  $\gamma$  in different inlet subcooling.

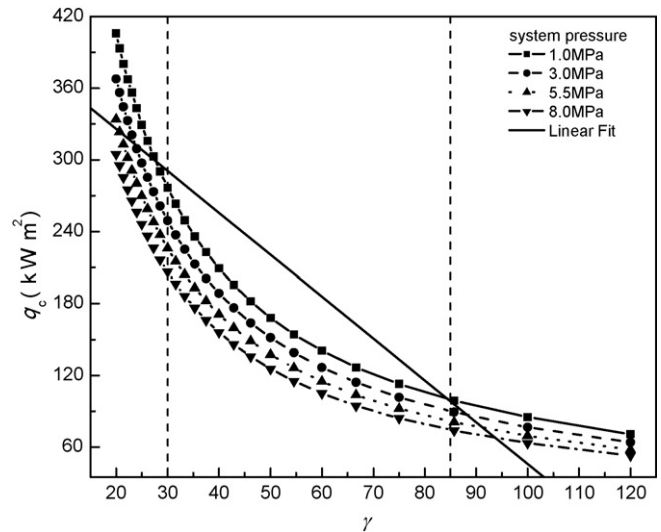


Fig. 6. The effect of predicted  $q_c$  under different  $\gamma$  in different system pressure.

#### 4. The determinant laws of CHF enhancement

When the effect of heat transfer enhancement is evaluated, a more attention has been paid on the coefficient of heat transfer and flow resistance, and a larger increase of heat transfer coefficient with a smaller increase of flow resistance coefficient is expected to be attained. Within the past studies (Akira and Sadanari, 2003; Joardar and Jacobi, 2005; Leu et al., 2004; Torii et al., 2002), the ratio of  $j$  and  $f$  is used to determine the behavior of heat transfer enhancement normally. However, these dimensionless parameters are defined to gas or single-phase liquid, they may be not suitable to vapor–liquid two-phase medium, including CHF in boiling two-phase system especially. Considering the characteristic of vapor–liquid two-phase flow, the determinant laws of heat transfer enhancement is derived again. Coefficient of comprehensive evaluation  $j_c/f_{TP}$  is used to evaluate the effect of CHF enhancement in narrow rectangular channel.

##### 4.1. The derivation of determinant laws of CHF enhancement

The definition of  $j$  is as follows:

$$j = StPr_1^{2/3} \quad (16)$$

$St$  is a kind of modified  $Nu$ , which means that the ratio of real heat flux to the maximum heat flux. According to the definition of  $St$ , when CHF takes place, the real heat flux has reached the maximum heat flux which the fluid can transfer. As a result,  $St$  equals 1. Therefore,

$$j_c = Pr_1^{2/3} \quad (17)$$

The definition of  $f$  is as follows:

$$f = \frac{\Delta P D_{hy}}{2\rho U^2 L} \quad (18)$$

In two-phase flow, the calculation expression of  $f_{TP}$  is:

$$f_{TP} = \frac{\Delta P D_{hy}}{2\rho_{TP} U_{TP}^2 L} \quad (19)$$

where  $\Delta P$  is the pressure drop between test section inlet and outlet, Pa;  $\rho_{TP}$  is the density of two-phase flow, kg/m<sup>3</sup>;  $U_{TP}$  is the average velocity of two-phase flow, m/s;  $L$  is the heated length, m

$$\rho_{TP} = \alpha\rho_g + (1 - \alpha)\rho_l \quad (20)$$

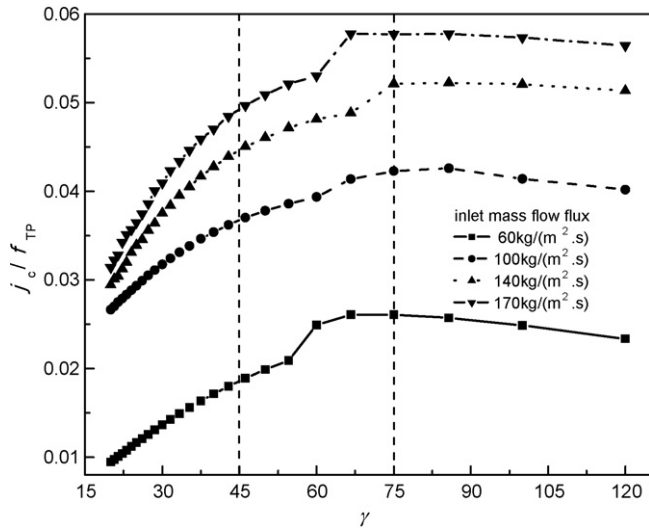


Fig. 7. The trend of  $j_c/f_{TP}$  with  $\gamma$  in different inlet mass flow flux.

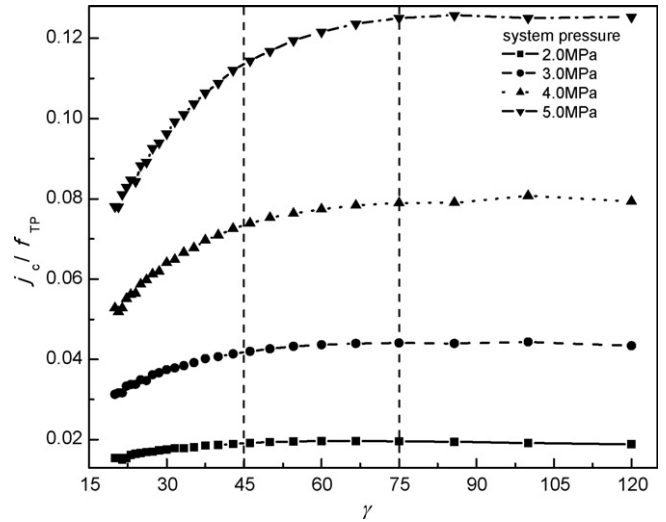


Fig. 9. The trend of  $j_c/f_{TP}$  with  $\gamma$  in different system pressure.

where  $\alpha$  is void fraction.

$$U_{TP} = \frac{G}{\rho_l} \quad (21)$$

Eqs. (17)–(21) are merged, then, the following expression can be achieved

$$\frac{j_c}{f_{TP}} = \frac{2[\alpha\rho_g + (1 - \alpha)\rho_l]G^2 L Pr_1^{2/3}}{\Delta P D_{hy} \rho_l^2} \quad (22)$$

#### 4.2. The analysis of determinant laws

Based on the CHF model from Section 4.1, and the geometric dimensions of narrow rectangular channel in Section 3.2 have been used again.

The inlet subcooling is 40 °C, the system pressure is 3.5 MPa, but the inlet mass flow flux is set as 60, 100, 140 and 170 kg/(m<sup>2</sup> s), respectively. The results have been shown in Fig. 7.

The inlet mass flow flux is set as 150 kg/(m<sup>2</sup> s), the system pressure is 1.5 MPa, but the inlet subcooling is 10, 20, 30 and 70 °C, respectively. The results have been shown in Fig. 8.

The inlet mass flow flux is set as 200 kg/(m<sup>2</sup> s), the inlet subcooling is 30 °C, but the system pressure is 2.0, 3.0, 4.0 and 5.0 MPa, respectively. The results have been shown in Fig. 9.

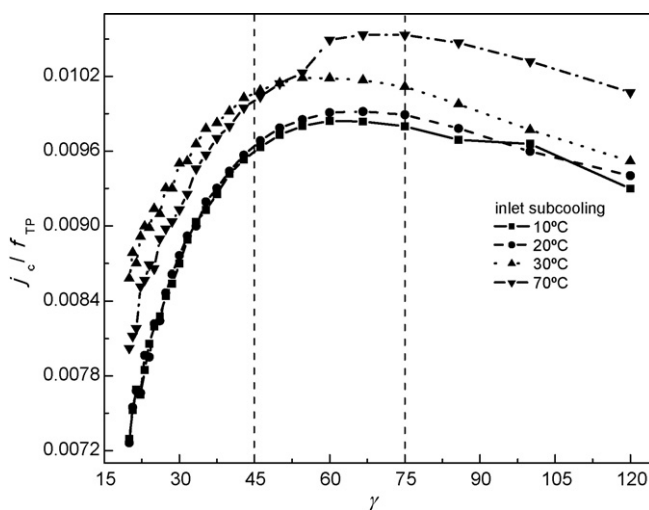


Fig. 8. The trend of  $j_c/f_{TP}$  with  $\gamma$  in different inlet subcooling.

Looking from Figs. 7–9, it can be seen that  $q_c$  is achieved to the maximum value among mostly curves when  $\gamma$  is equal to 60. At the same time, 45–75 can be taken as the reference range of  $\gamma$  based on the trend of curves in Fig. 9. In this reference range, the increasing trend of  $j_c/f_{TP}$  is gradually approaching to a maximum value in different inlet mass flow flux, inlet subcooling and system pressure. In other words, the enhancement of CHF is obvious in this  $\gamma$  region. When the width of channel is fixed in 0.060 m, and  $\gamma$  is in 45–75, the corresponding width of narrow rectangular channel is 0.0008–0.0013 m. This value is within the real engineering parameters of being 0.0005–0.0030 m (Ishibashi and Nishikawa, 1969). Therefore, the determinant laws is appropriate.

#### 5. The effect of width on CHF enhancement

Based on Sections 3.2 and 4.2, if we want to improve CHF value only by changing the gap width, when  $\gamma$  is in 30–85, the CHF value will be improved obviously in narrow rectangular channel. When the gap width is decreased, the CHF value will be improved, and the pressure-loss will be increased simultaneously. At this time, when  $\gamma$  is in 45–75,  $j_c/f_{TP}$  reaches the maximum value in narrow rectangular channel.

In fact, under some conditions, the outlet pressure of circulated pump is high enough, it is not considerable that the pressure-loss is brought by gap width, in other words, the most important concern is whether the CHF value is improved or not. At other conditions, considering the efficiency of heat transfer and economic aspect, the enhancement of CHF and pressure-loss should be considered simultaneously. So, based on above considerations actions, taking  $\gamma$  as 45–75 could be as the reference range of CHF enhancement by changing gap width in narrow rectangular channel.

#### 6. Conclusions

Based on liquid film dry-out mechanics of annular flow, a CHF analytical model has been achieved in narrow rectangular channel. This model may be used to predict the CHF of boiling two-phase flow in narrow rectangular channel with the gap width of not being less than 0.0005 m. According to the characteristic of vapor–liquid two-phase flow, the new determinant laws of CHF in boiling two-phase flow system have been derived. Through analyzing and calculating, it is substantial that the determinant laws are appropriate. The determinant laws may be used to determine the enhancement of CHF in narrow rectangular channel with high

Reynolds number when the flow is forced. Within the range of channel geometry size and operation mode in this paper, the best dimensionless gap width of CHF enhancement has been achieved to be 45–75. This methodology development may be used to design high effective rectangular exchangers or plate-type fuel elements, etc.

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