

行政院國家科學委員會補助專題研究計畫 ☒ 成果報告
☐ 期中進度報告

小管中新冷媒流動沸騰熱傳特性和主要機制研究(三)

計畫類別：☒ 個別型計畫 ☐ 整合型計畫

計畫編號：NSC 94-2212-E-009-002-

執行期間：民國 94 年 08 月 01 日 起至民國 95 年 07 月 31 日

計畫主持人：林清發 教授

共同主持人：

計畫參與人員：

成果報告類型(依經費核定清單規定繳交)：☐ 精簡報告 ☒ 完整報告

本成果報告包括以下應繳交之附件：

- ☐ 赴國外出差或研習心得報告一份
- ☐ 赴大陸地區出差或研習心得報告一份
- ☐ 出席國際學術會議心得報告及發表之論文各一份
- ☐ 國際合作研究計畫國外研究報告書一份

處理方式：除產學合作研究計畫、提升產業技術及人才培育研究計畫、
列管計畫及下列情形者外，得立即公開查詢

☐ 涉及專利或其他智慧財產權，☐ 一年☐ 二年後可公開查詢

執行單位：國立交通大學機械工程研究所

中 華 民 國 九 十 五 年 十 月 日

摘要

本研究以實驗方式探討管道尺寸對飽和流動沸騰熱傳及相關氣泡特徵之影響。以環保冷媒 R-134a, R-407C 和 R-410A 為工作流體流入一水平狹窄雙套管之測試段，流道之間隙由 0.2 至 2.0mm。測試段則由玻璃外管及加熱內銅管組成來量測熱傳係數及流場觀測。一電橋式加熱棒置於銅管內部提供熱通量加熱狹窄流道內流動之冷媒。此外，針對 R-407C 冷媒在 28 根(內徑 2mm、外徑 3mm)的水平小圓管內蒸發熱傳及壓降特性進行實驗研究，實驗的目的是探討改變冷煤質通量、測試段的熱通量、測試段的進口蒸氣乾度及系統的飽和壓力(相對應之飽和溫度)對 R-407C 冷煤在本熱交換器的影響。

首先，提出冷媒 R-134a 在飽和流動沸騰之測試結果。從所量測之沸騰曲線發現，沸騰初始之溫度突降現象在飽和流動沸騰中並不明顯。飽和流動沸騰熱傳係數隨尺寸減少而增加。此外，熱通量增加亦會使得熱傳係數明顯增加。然而，冷媒之質通量及飽和溫度對熱傳係數影響甚小。從流場觀測之結果顯示氣泡脫離尺寸隨著質通量增加而稍微減小。並且在高熱通量下，許多氣泡從加熱表面之成核孔產生並趨向合併成大氣泡。在較小間隙如 0.2 及 0.5mm，流譜將由氣泡流轉變為彈狀流。在管道中之彈狀流的特徵則是由液體彈狀消長伴隨大、小氣泡。

接著在小管中經由實驗的結果可知，蒸發熱傳係數與摩擦壓降係數除了在較低的冷煤質通量狀態下，皆會隨著冷煤質通量的加大與進口乾度的增加而上升。在熱通量的影響方面，除了在較低的熱通量情況下，蒸發熱傳係數也會隨著熱通量的加大而上升。但相較於熱通量對蒸發熱傳的影響，改變熱通量對摩擦壓降係數的影響則是很微弱的。而在飽和溫度的影響上，蒸發熱傳係數會隨著系統的飽和溫度上升而增加；但相反的情況則發生在降低系統的飽和溫度會造成摩擦壓降係數上升。因此，成核與對流效應對本實驗系統的熱傳機制皆有重要的影響。

然後，提出冷媒 R-407C 和 R-410A 在飽和流動沸騰之測試結果並與 R-134a 比較。從所量測之沸騰曲線發現，所有冷煤的沸騰初始之溫度突降現象在飽和流動沸騰中並不明顯。R-410A 的流動沸騰熱傳係數比其他冷煤來的高。此外，熱通量增加亦會使得熱傳係數明顯增加。然而，冷媒之質通量及飽和溫度對熱傳係數影響甚小。從流場觀測之結果顯示氣泡脫離尺寸隨著質通量以及飽和溫度增加而稍微減小。R-410A 的有效成核密度明顯大於其他冷媒。

最後，我們把這個實驗中流動沸騰氣泡脫離直徑、產生頻率及成核密度的資料作分析，求出氣泡特徵之 3 個經驗式，並利用此經驗公式推得氣泡流區之熱傳係數。至於彈狀流區之熱傳係數則則是修正 Cornwell and Kew 經驗式所得。此外，亦在小管中蒸發熱傳係數與摩擦壓降係數的實驗資料作分析，找出經驗公式。

關鍵字：狹窄雙套管，飽和流動沸騰，冷媒 R-134a，冷媒 R-407C，熱傳及氣泡特徵，沸騰曲線，蒸發熱傳係數，摩擦壓降係數

ABSTRACT

Experiments have been conducted here to investigate how the channel size affects the saturated flow boiling heat transfer and associated bubble characteristics of the ozone friendly refrigerant R-134a, R-407C and R-410A in the horizontal annular duct with the gap between the inner and outer pipes of the duct being relatively small. The gap of the duct is varied from 0.2 to 2.0 mm in this study. The test section for the horizontal annular duct consists of an outer pipe made of Pyrex glass and an inner heated copper pipe, intending to measure the boiling heat transfer coefficient and to facilitate the visualization of boiling processes. A cartridge heater is installed inside the inner pipe to provide the required heat flux to the refrigerant flow in the narrow annular duct. Besides, An experiment is carried out here to unravel the characteristics of the evaporation heat transfer and pressure drop for refrigerant R-407C flowing in horizontal small circular tubes, each having an inside diameter of 2.0 mm. The effects of the imposed wall heat flux, mass flux, and vapor quality and saturated temperature of R-407C on the measured evaporation heat transfer coefficient and frictional pressure drop are examined in detail.

First, the experimental heat transfer data for the saturated flow boiling of R-134a in the horizontal narrow annular duct are examined. From the measured boiling curves, the temperature undershoot at ONB are insignificant for the saturated flow boiling of R-134a in the duct. The saturated flow boiling heat transfer coefficient increases with a decrease in the gap size of the duct. Besides, raising the imposed heat flux can also cause a significant increase in the boiling heat transfer coefficients. However, the effects of the refrigerant mass flux and saturated temperature on the boiling heat transfer coefficient are small. The results from the flow visualization show that the mean diameter of the bubbles departing from the heating surface decreases slightly at increasing mass flux. Besides, at a high imposed heat flux many bubbles generated from the cavities in the heating surface tend to merge together to form a big bubbles and in the smaller ducts with the gap at 0.2 and 0.5 mm the bubbly flow changes to a slug flow. The slug flow in the ducts is characterized by a number of liquid slugs interdispersed with big bubbles along with small bubbles in the liquid slugs.

The experimental data in small circular tubes clearly show that both the evaporation heat transfer coefficient and frictional pressure drop increase with the refrigerant mass flux

and vapor quality except at low mass fluxes. A rise in the imposed heat flux results in a significant increase in the evaporation heat transfer coefficient except at low heat fluxes. Nevertheless the influence of the imposed heat flux on the frictional pressure drop is rather slight. At increasing saturated temperature of the refrigerant the evaporation heat transfer coefficient increases significantly. But an opposite trend is noted for the frictional pressure drop. As a result, both the convection and nucleation can be important. This is indeed the case according to the experimental data presented above.

Then, the experimental heat transfer data for the saturated flow boiling of R-407C and R-410A in the horizontal narrow annular duct are examined and compared with R-134a. From the measured boiling curves, the temperature undershoot at ONB are insignificant for the saturated flow boiling of all refrigerants in the duct. The saturated flow boiling heat transfer coefficient of R-410A is higher than other refrigerants. Besides, raising the imposed heat flux can also cause a significant increase in the boiling heat transfer coefficients. However, the effects of the refrigerant mass flux and saturated temperature on the boiling heat transfer coefficient are small. The results from the flow visualization show that the mean diameter of the bubbles departing from the heating surface decreases slightly at increasing mass flux and saturated temperatures. Besides, the active nucleation site density for R-410A is significantly higher than other refrigerants.

Correlations for the saturated flow boiling heat transfer coefficients of R-134a in the narrow annular duct are proposed. Besides, data for some quantitative bubble characteristics such as the mean bubble departure diameter, departure frequency and the mean active nucleation site density have been correlated. Besides, the empirical correlations for the evaporation heat transfer coefficient and friction factor in small circular tubes are proposed.

Keywords: narrow annular duct, saturated flow boiling, R-134a, R-407C, heat transfer and bubble characteristics, boiling curve, evaporation heat transfer coefficient, friction factor

CONTENTS

ABSTRACT	I
CONTENTS	V
NOMENCLATURE	VII
CHAPTER 1 INTRODUCTION	1
1.1 Alternatives to HCFC-22	1
1.2 Literature Review – Two Phase Heat Transfer and Pressure Drop	3
1.3 Review of Flow patterns and bubble characteristics	8
1.4 The Objective of This Study	11
CHAPTER 2 EXPERIMENTAL APPARATUS AND PROCEDURES	13
2.1 Refrigerant flow loop	13
2.2 Test Section	14
2.3 Water loop for preheater	17
2.4 Water-glycol loop	17
2.5 DC Power Supply	17
2.6 Photographic System	18
2.7 Data acquisition	18
2.8 Experimental Procedures	19
CHAPTER 3 DATA REDUCTION	28
3.1 Flow Boiling Heat Transfer Coefficient in Annular Duct	28
3.2 Two Phase Heat Transfer in Small Circular Tubes	29
3.3 Friction Factor in Small Circular Tubes	30
3.4 Uncertainty Analysis	32
CHAPTER 4 SATURATED FLOW BOILING OF R-134a IN a HORIZONTAL NARROW ANNULAR DUCT	35

4.1 Single-phase Heat Transfer	35
4.2 Saturated Flow Boiling Curves	36
4.3 Saturated Flow Boiling Heat Transfer Coefficient	37
4.4 Bubbles Characteristics in Saturated Flow Boiling	38
4.5 Correlation Equations	44
4.6 Concluding Remarks	47
CHAPTER 5 EVAPORATION HEAT TRANSFER IN SMALL CIRCULAR TUBES	89
5.1 Single Phase Heat Transfer	89
5.2 Two-Phase Evaporation Heat Transfer	90
5.3 Correlation Equations	94
5.4 Concluding Remarks	95
CHAPTER 6 TWO-PHASE PRESSURE DROP IN SMALL CIRCULAR TUBES	110
6.1 Two-Phase Frictional Pressure Drop	110
6.2 Correlation Equations	112
6.3 Concluding Remarks	112
CHAPTER 7 SATURATED FLOW BOILING OF R-407C and R-410A IN a HORIZONTAL NARROW ANNULAR DUCT	126
7.1 Single-phase Heat Transfer	126
7.2 Saturated Flow Boiling Curves	127
7.3 Saturated Flow Boiling Heat Transfer Coefficient	128
7.4 Bubbles Characteristics in Saturated Flow Boiling	129
7.5 Concluding Remarks	131
CHAPTER 8 CONCLUDING REMARKS	152
8.1 Concluding Remarks	152
REFERENCES	155

NOMENCLATURE

A	heat transfer area of the annular duct, m ²
Bo	Boiling number, $Bo = \frac{q}{G \cdot i_{fg}}$, dimensionless
c _p	specific heat, J/kg°C
D	diameter of smooth duct, m
D _h	hydraulic diameter, m, $D_h = 4A_c/P$
d _p	bubble departure diameter, m
f _b	bubble generation frequency
f _f	friction factor
Fr	Froude number, $Fr = \frac{G^2}{\rho_l^2 \cdot g \cdot D_h}$, dimensionless
g	acceleration due to gravity, m/s ²
G	mass flux, kg/m ² s
h	heat transfer coefficient, W/m ² °C
i _{fg}	enthalpy of vaporization, J/kg
I	measured current from DC power supply, A
Ja	Jakob number based on ΔT_{sat} , $Ja = \frac{\rho_l \cdot C_p \cdot \Delta T_{sat}}{\rho_g \cdot i_{fg}}$, dimensionless
Ja'	Jakob number based on ΔT_{sub} , $Ja = \frac{\rho_l \cdot C_p \cdot \Delta T_{sub}}{\rho_g \cdot i_{fg}}$, dimensionless
k	thermal conductivity, W/m°C
N _{ac}	Active nucleation sit density, n/m ²
N _{conf}	Confinement number, $N_{conf} = \frac{(\sigma/(g\Delta\rho))^{0.5}}{D_h}$, dimensionless

Nu	Nusselt number, $Nu = \frac{hD_h}{k}$, dimensionless
P	system pressure, kpa
Pr	Prandtl number, $Pr = \frac{\mu \cdot C_p}{k}$, dimensionless
q	average imposed heat flux, W/m^2
Q	heat transfer rate, W
Re	Reynolds number, $Re = \frac{G \cdot D_h}{\mu}$, dimensionless
T	temperature, °C
T _{sat}	saturated temperature of the refrigerant R-134a, °C
V	measured voltage from DC power supply, V
v _s	slug bubble velocity, m/s
X _{tt}	Martnelli parameter
z	coordinate (downstream coordinate for annular duct flow), mm

Greek Symbols

ΔT	temperature difference, °C
δ	gap size, mm
μ	viscosity, $N \cdot s/m^2$
ρ	density, kg/m^3
$\Delta \rho$	density difference, $\Delta \rho = \rho_l - \rho_g$, kg/m^3
σ	surface tension, N/m

Subscripts

ave	average
cs	cross-section of annular duct
exp	experimental value

fg	difference between liquid phase and vapor phase
g	vapor
i, o	inside and outside diameter of the smooth pipe
l	all-liquid nonboiling heat transfer
n	net power input to the refrigerant R-134a
r	refrigerant side
s	heater surface
sat	saturated flow boiling
sub	subcooled flow boiling
w	wall of the smooth pipe surface

CHAPTER 1

INTRODUCTION

In the past decade following the signing of the Montreal Protocol, extensive research has been undertaken to search for the alternatives that can replace chlorofluorocarbons (CFCs) refrigerants. It is well known that the use of the CFCs refrigerants, which contain chlorine and carbon, would lead to the ozone depletion and a consequent increase in ultraviolet radiation, and in turn would cause global warming. Due to these disadvantages, some hydrochlorofluorocarbons (HCFCs) and hydrofluorocarbons (HFCs) refrigerants have been developed and they are found to be the most useful alternatives to substitute for CFCs. The HCFCs refrigerants contain less chlorine than the CFCs and therefore were developed to serve as interim for CFCs. The HFCs refrigerants have zero ozone depletion potential (ODP) and were developed to offer long-term replacement to CFCs. In order to properly use these new refrigerants, we need to understand their thermodynamic, transport and heat transfer properties. In particular, a detailed understanding of the characteristics of the evaporation and condensation heat transfer for the HFCs refrigerants is very important in the design of evaporators and condensers for many refrigeration and air conditioning systems.

1-1 Alternatives to HCFC-22

Due to the serious destruction of the ozone layer in the outer atmosphere by the CFCs refrigerants, various new refrigerants such as R-134a, R-407C, and R-410A were recently developed and introduced into the refrigeration and air conditioning systems. The HCFC-22, one of the HCFCs, offers excellent performance and is

widely used for air conditioners. Unfortunately, HCFC-22 still has some ODP and therefore is planned to be phased out shortly. According to the Montreal Protocol, the production of the HCFCs will be limited to 65% of the reference level chosen to be that of 1989 by January 1, 2004, to 35% by January 1, 2010, to 10% by January 1, 2015, and to 0.5% of the reference level by January 1, 2020. Complete ban on the production of the HCFCs is set on January 1, 2030. The European Union has accelerated the reduction schedule for the HCFCs with an eventual phase-out in 2015. For comfort air conditioning, refrigerants R-134a, R-404A, R-407C and R-410A are commonly used to substitute for HCFC-22. Meyer [1] has evaluated five refrigerants as replacement for R-22 by comparing the cooling capacity, coefficient of performance, compressor discharge temperature, input power, refrigerant mass flow, pressure ratio, and volumetric refrigeration capacity. Besides, alternatives to HCFC-22 for air conditioners were assessed by Devotta et al. [2].

Because there are no single-component HFCs that have thermodynamic properties close to those of HCFC-22, Binary or ternary refrigerant mixtures have been introduced. Many long-term alternative refrigerants are mixtures of two or more HFCs and fall into two major groups, zeotropes and azeotropes. The refrigerant R-134a is a pure refrigerant to substitute for R-12 and R-22. The refrigerant R-410A, which is a binary mixture consisting of 50 wt% of R-32 and 50 wt% of R-125, is a higher pressure, near-azeotropic refrigerant. In particular, the refrigerant R-407C is a ternary zeotropic mixture consisting of 23 wt% of R-32, 25 wt% of R-125, and 52 wt% of R-134a. It is the most widely used refrigerant with zero ozone depletion potential and is current recognized as the main replacement to HCFC-22. It is realized that R-404A, R-407C and R-410A are the mixture refrigerants with respective temperature glides of 0.1°C , 7.2°C and 0.8°C during evaporation and condensation. In

certain applications, the higher temperature glides in the zeotropes offer the potential for higher efficiencies [3, 4].

1-2 Literature Review – Two Phase Heat Transfer and Pressure Drop

In the following the relevant literature on the boiling heat transfer of refrigerants in various tubes, such as smooth, micro-fin and other enhanced tubes, is briefly reviewed. Wang et al. [5] studied the two phase heat transfer of R-22 and R-407C within a smooth tube with a nominal diameter of 9.52 mm. They chose the evaporation pressure of 600 kPa at which the transport properties of R-407C are similar to those of R-22. The heat transfer coefficients for R-407C were found to be 50 to 70 % lower than those for R-22. But the pressure drop of R-407C was similar to that of R-22 at a low mass flux ($G=100 \text{ kg/m}^2\text{s}$) and was 45 % lower compared to R-22 at a higher mass flux ($G=300 \text{ kg/m}^2\text{s}$). The degradation in the R-407C evaporation heat transfer coefficient and friction gradient was attributed to the difference in the two-phase flow patterns. In a continuing study for a 9.52 mm micro-fin tube [6], they found that the heat transfer coefficients for R407C were 50 to 80 % lower and the pressure drop of R-407C is 30 to 50 % lower, as compared to R-22. In a further study [7] they considered a smaller smooth tube of 6.5 mm in diameter and investigated the flow pattern difference between R-22 and R-407C. Their results suggested that the heat transfer degradation for R-407C was the consequence of the delay of the flow pattern transition. Shin et al. [8] studied forced convective boiling heat transfer of pure refrigerants and refrigerant mixtures in a horizontal tube. They concluded that due to the gravity the liquid film at the top of the tube is thinner than that at the bottom and the heat transfer coefficient at the top is higher than that at the bottom. They also noted that the heat transfer coefficients were

strongly dependent on the heat flux at a low vapor quality and became independent at a high quality.

Recently, there is growing interest in the boiling and condensation heat transfer in small tubes and capillary tubes. The small tubes can be used to increase the compactness of heat exchangers. In the traditional large diameter tube, the boiling mechanism is governed by two fundamental mechanisms - forced convection of flow and bubble nucleation. But in small diameter tubes, the bubble nucleation becomes more important. The evaporative heat transfer, pressure drop and critical heat flux in a small vertical tube ($D_i = 3.1$ mm) with R-113 flowing in it were investigated experimentally by Lazarek and Black [9]. They found that the wall heat transfer process was controlled by the nucleate boiling. A similar study from Wambsganss et al. [10] for boiling heat transfer in a horizontal small diameter tube ($D_i = 2.92$ mm) showed that the associated high boiling number (Bo) and slug flow pattern in the small tube lead to the domination of the two-phase heat transfer by the nucleation. Besides, the local evaporation heat transfer coefficient is a strong function of the heat flux, but only weakly dependent on the mass flux and quality. Later, they [11] studied boiling of R-12 and R-113 in a small circular tube ($D_i = 2.46$ mm) and a rectangular channel with a hydraulic diameter of 2.40 mm. The results also support the above conclusion that the bubble nucleation is the dominant boiling heat transfer mechanism in the small channels. They expected that at sufficiently low values of heat flux and very low wall superheat, the forced convection could become dominant. Besides, very small difference was noted for the heat transfer coefficients in a small circular tube and in a small rectangular channel with the same hydraulic diameter.

Experimental measurement was carried out by Yan and Lin [12] to study the evaporation heat transfer and pressure drop of R-134a in a bank forming by small

pipes ($D_i = 2.0$ mm). Their results revealed that the evaporation heat transfer in the small pipes was more effective than that in large tubes. Flow boiling heat transfer of R-11 and R-123 in a smooth copper tube ($D_i = 1.95$ mm) was investigated by Bao et al. [13]. The heat transfer coefficients were found to be independent of the mass flux and vapor quality, but are strong functions of heat flux and system pressure in the saturation region, suggesting that the heat transfer mechanism is dominated by the bubble nucleation. The two phase heat transfer of R-141b in a small tube ($D_i = 1.0$ mm) performed recently by Lin et al. [14] showed that at a low vapor quality, the heat transfer was dominated by the nucleation. But in the high vapor quality region, the heat transfer was dominated by the force convection. Yu et al. [15] studied the two-phase pressure drop, boiling heat transfer, and critical heat flux of water in a small-diameter horizontal tube ($D_i = 2.98$ mm). They showed that the transition boiling was a function of heat flux only. In addition, the boiling heat transfer was found to be dependent of the heat flux but essentially independent of mass flux. Single phase forced convection and subcooled and saturated boiling heat transfer and pressure drop of FC-84 in narrow rectangular channels with a hydraulic diameter of 0.75 mm were recently investigated by Warriar et al. [16]. Two new heat transfer correlations were proposed, one for the subcooled flow boiling and the other for the saturated flow boiling. Fujita et al. [17] measured flow boiling heat transfer of R-123 in a uniformly heated horizontal small tube ($D_i = 1.12$ mm). They explained that the domination of the boiling heat transfer in small tubes by the bubble nucleation was owing to the fact that the boiling number became rather high in small tubes when compared to large diameter tubes. The flow pattern observed in the small tube indicated that the stratified flow did not exist. And the influences of the surface tension became more substantial in smaller tubes. Besides, the friction factor was nearly independent of the Reynolds number of the flow.

Based on the boiling processes in channels of various size, Kandlikar [18] classified the channels into three categories : the convectional channels for $D_h > 3\text{mm}$, the minichannels for $200\ \mu\text{m} < D_h < 3\text{mm}$, and the microchannels for $D_h < 200\ \mu\text{mm}$. For the conventional channels the heat transfer mechanism is dominated by both the convective boiling and nucleate boiling. In the convective boiling, the heat transfer coefficient is mainly affected by the mass flux G and vapor quality x . However in the nucleate boiling the heat transfer is only dominated by the imposed heat flux q . Heat transfer in the microchannels is only affected by the nucleate boiling and hence the imposed heat flux. While for the minichannels some investigators suggest that the flow boiling heat transfer mechanism is only affected by the heat flux but others do not agree. The results are quite different for different working fluids and test conditions. Most of the studies for the minichannels use the low pressure refrigerants or air-water. The higher pressure refrigerants like R-134a, R-407C and R-410A have not received enough attention. To delineate the importance of the channel size on the boiling processes, a dimensionless group named as the “Confinement Number” is normally used [19, 20]. It is defined as

$$N_{conf} = \sqrt{\sigma / g(\rho_l - \rho_g)} / D_i \quad (1.1)$$

The number signifies the relative magnitudes of the bubble departure diameter to the channel diameter. For $N_{conf} > 0.5$ the effects of the channel confinement on the boiling flow and heat transfer are considered to be important.

An interesting issue on the boiling in small tubes is the possibility that when the dimension of a channel is small enough, the vapor bubble growth can be restricted. They even asserted that the vapor bubble formation was impossible when the channel was below certain specific dimension. Brutin et al. [21] examined two phase flow and heat transfer with n-pentane in capillaries with a hydraulic diameter of 0.889 mm. In

their study the flow boiling phenomenology consists of two parts, flow excursion and confinement instability. They noted that when the instabilities developed in the channel, the surface temperature increased and the average heat transfer coefficient decreased.

Two phase flow and adiabatic single phase flow pressure drops of R-12 flowing in both rectangular channel with a hydraulic diameter of 2.64 mm and micro-fin small tubes ($D_i = 1.56$ mm) were measured by Yang et al. [22]. The two phase pressure gradient was found to increase with the mass velocity and vapor quality of the refrigerant. The surface tension force played no significant role in affecting the frictional pressure drop in the micro-fin tube. Additionally, the pressure drop in micro-fin tube is higher than that in the plain tube in most situations. In a continuing study [23] for water, R-134a and air in small tubes ($D_i = 0.173$ to 4.01 mm), they showed that the single phase pressure drop with R-134a in small tubes could be empirically correlated by the conventional Blasius and Poiseuille equations. In capillary tubes ($D_i = 1.2$ to 1.6 mm) with pure refrigerant and their mixtures, Chang et al. [24] suggested that the pressure drop in the tubes resulted mainly from the friction on the tube wall and the flow acceleration associated with the variation in the specific volume of the fluid. The homogeneous flow model was shown to be appropriate for calculating the pressure drop of two phase flow in capillary tubes at high liquid flow rate. Tong et al. [25] measured the pressure drop for highly subcooled flow boiling of water in small tubes ($D_i = 1.05$ to 2.44 mm). Their data manifested that both single and two phase pressure drops increased with increasing mass flux and length-to-diameter ratio but decreased with increasing internal diameter of the tube. Inlet and exit temperatures of water exhibited significant influences on the two phase pressure drop but had invisible effects on the single phase pressure drop.

Tran et al. [26] measured flow boiling pressure drops of refrigerants (R-134a, R-12 and R-113) in two round tubes ($D_i = 2.46$ and 2.92 mm) and one rectangular channel with a hydraulic diameter of 2.40 mm. The measured data for R-134a were compared with five state-of-art correlations for predicting two phase frictional pressure drop in larger diameter tubes. These correlations failed to predict the pressure drop in small tubes for all test conditions.

1-3 Review of Flow patterns and bubble characteristics

To elucidate flow boiling heat transfer mechanism in small channels, we require to delineate the prevailing flow regimes. Based on visualization of the flow and measurement of the heat transfer, three flow regimes have been suggested, namely, isolated bubble, confined bubble and annular-slug flows [28-30]. However, Bubble behavior such as bubble generation frequency, growth, sliding and departure size plays an important role in flow boiling heat transfer. The bubble characteristics in the boiling flow have been examined by a number of research groups. For instance, Sheng and Palm[30] visualized the flow pattern and bubble shape for water in a single glass tube with small tube diameters ($D_h = 1.0, 1.6, 2.0, 4.0$ mm). They indicated the bubble departure diameter was dependent much on the mass flow rate but was slightly affected by the imposed heat flux. The behavior of near-wall bubbles in subcooled flow boiling for water and R-134a in vertical rectangular channel ($D_h = 6.1$ mm) was investigated photographically by Bang et al[31]. They described the coalescence of the bubbles and showed that the bubbles are smaller at a higher mass flux. Bibeau and Salcudean [32] visualized the bubble cycling of water flow in a vertical annular pipe ($D_h = 10.0$ mm) with a high speed photography. They noted that the maximum bubble diameter varied from 0.8 to 0.3 mm and at ejection the bubble was smaller than the maximum diameter, since the bubble condensed on the wall while sliding. The

subcooled flow boiling and the associated bubble characteristics of R-134a in a horizontal annular channel ($D_h = 10.3$ mm) were examined recently by Yin et al. [27]. Results from their flow visualization indicated that the bubble generation was suppressed by raising the mass flux and subcooling, and only the liquid subcooling showed a significant effect on the bubble size. Finally, an empirical correlation for the bubble departure diameter was provided.

Thorncroft et al. [33] experimentally investigated upflow and downflow boiling of FC-87 in vertical rectangular channel ($D_h = 12.7$ mm). They indicated that both bubble growth and bubble departure rates increased with the Jacob number, but decreased with the mass flux. Besides, the bubble waiting time decreased with an increase in the heat flux. This increase in the heat transfer was directly attributed to the sliding vapor bubbles, which remained attached to the wall during the upflow and lift off from the wall during the downflow. The sliding of the bubbles on the boiling surface was found to enhance heat transfer in forced convection boiling of FC-87 in a vertical upflow and downflow by Thorncroft and Klausner[34]. An experimental investigation of low pressure subcooled flow boiling inside a vertical concentric annulus($D_h = 13.0$ mm) from Zeitoun and Shoukri [35] confirmed that the bubble departure was not the reason for the net vapor generation (NVG). However, the mean size and lift duration of the bubbles increased at decreasing liquid subcooling. Moreover, a new correlation for the mean bubble diameter in terms of the Reynolds number, boiling number, local Jacob number, and fluid properties was proposed. Klausner et al. [36] developed a criterion for the bubble departure from the heated surface in the forced convection boiling. The study was carried out for a saturated two-phase mixture of refrigerant R-113 flowing through a 25×25 mm²($D_h = 25.0$ mm²) visual boiling section. They found that the mean bubble departure diameter

decreased with increasing mass flux and with decreasing heat flux. Their analytical prediction further showed the strong influences of the liquid velocity and wall superheat on the bubble departure diameter. They also noted that before lifting off from the heated wall, the bubbles would slide a finite distance along the surface. They concluded that not only the surface tension force but also the asymmetrical bubble growth acting in the direction opposite to the fluid motion were important in holding the bubbles at the nucleation sites before departure. Chien and Webb [37] studied the bubble dynamics for pool boiling on an enhanced tubular surface, which consists of an integral-fin tube having a copper foil wrapped over the fin tips. They showed that the mean bubble diameter was smaller for a higher heat flux and the bubble growth period was shorter for smaller bubbles and higher heat flux.

Yang and Kim [38] attempted to quantitatively predict the active nucleation sites from knowing the size and cone angle β of the cavities actually present on the surface. Using an electron microscope and a differential interference contrast microscope, they obtained the cavity probability density function involving cavity size (ranging from 0.65 to 6.2 μm) and β (cone half angle). The size distribution was found to fit a Poisson distribution while a normal distribution was used for β . Gaertner [39] observed that active nucleation sites were randomly located and could be expressed in terms of Poisson's distribution function. Sultan and Judd [40] reached the same conclusion from their observations. Zeng and Klausner [41] obtained experimental data for the active nucleation site density N_{ac} during flow boiling of R-113 on a horizontal 25 mm $\times$ 325 mm test section with a nichrome heating strip. Their experiments were performed for varying vapor quality at inlet, system pressure and wall heat flux. They examined the effects of vapor velocity, liquid velocity, liquid film thickness, system pressure, and wall heat flux on N_{ac} . They concluded that even if N_{ac}

is dependent on the critical radius of the nucleus r_c , their data are not sufficient for correlating N_{ac} . From their data, they concluded that the vapor velocity, heat flux, and system pressure have strong effects on N_{ac} . Kocamustafaogullari and Ishii [42] developed a relation for active nucleation site density in pool boiling from the data available in the literature. They also applied the correlation to the few available forced convection nucleate boiling data. Their correlation expressed the active nucleation site density in dimensionless form as a function of dimensionless minimum cavity size and density ratio. The correlation was valid for the system pressure ranging from 1.0 to 198.0 bar. Basu et al [43] proposed an empirical correlation including the effect of the contact angle on the active nucleation site density during forced convective boiling of water on a vertical surface based on their experimental data. They performed subcooled boiling experiment at the atmospheric pressure. In the experiments, they utilized mirror-finished copper surfaces prepared by a well-defined procedure. They varied the wettability of the surface by controlling the degree of oxidation of the surface.

1-4 The Objective of This Study

In recent years, environmental concerns over the use of CFCs as working fluids in refrigeration and air-conditioning systems have led to the development of alternative fluids. Among these alternatives, R-134a, R-407C and R-410A are used as substitutes for the HCFC-22. Moreover, considerable effort has recently been devoted to improve the design of more compact and efficient evaporators for the process and refrigeration industries. While a great number of papers have been written on the flow boiling heat transfer, only several of them consider the associated bubble characteristics for the widely used new refrigerants (like R-134a, R-407C, R-410A) particularly in small channels. In this study, the saturated flow boiling of new

refrigerants in an annular duct with a small gap between the inner and outer ducts are investigated by measuring the boiling heat transfer coefficient and by visualizing the bubble behavior. The effects of the imposed heat flux, gap size, mass flux and saturation temperature of new refrigerants on the boiling heat transfer characteristics will be examined in detail. The results will be presented in terms of the boiling curves and the heat transfer coefficients. Particularly, flow visualization is conducted here to examine some bubble characteristics associated with the flow boiling such as the mean bubble departure diameter and generation frequency from the heating surface to improve our understanding of the flow boiling processes in the narrow channel. Besides, the pressure drop and heat transfer coefficient for the evaporation of R-407C in horizontal small tubes will be measured. The effects of the vapor quality, refrigerant mass flux, imposed heat flux and system pressure on the evaporation heat transfer coefficient and friction pressure drop will be investigated in detail.

CHAPTER 2

EXPERIMENTAL APPARATUS AND PROCEDURES

The experimental apparatus used in the present study to investigate the characteristics of saturated flow boiling and evaporation heat transfer of new refrigerant (R-134a or R-407C or R-410A) in a horizontal narrow annular duct and in horizontal small tubes are schematically shown in Figs.2.1 and 2.2 respectively. The apparatus consists of three main loops, namely, a refrigerant loop, a water-glycol loop and a hot-water loop for preheater, along with a data acquisition system and a 50V-25A DC power supply. Refrigerants are circulated in the refrigerant loop. We need to control the temperature and flow rate in the water-glycol loop to obtain enough cooling capacity for condensing the refrigerant vapor and for maintaining the refrigerant liquid at a preset temperature. The liquid refrigerant in the narrow duct is heated by passing a DC current through the inner pipe in the test section from the DC power supply. To unravel the bubble characteristics in the boiling of refrigerant on the heating surface, a high-speed movie camera connected to a photographic microscope is set up beside the test section to observe the boiling flow.

2.1 Refrigerant flow loop

The main components in the refrigerant loop include an oil-free variable-speed refrigerant pump, an accumulator, a mass flow meter, a test section, a condenser, a sub-cooler, a receiver, a filter/dryer and four sight glasses. An AC motor is used to control the refrigerant mass flow rate through the change of the inverter frequency. The flow rate can also be adjusted by regulating the by-pass valve installed in the by-pass flow path. The refrigerant at the outlet of the refrigerant pump is kept subcooled to avoid any vapor flow

through the mass flow meter. The flow meter has an accuracy of $\pm 1\%$. The preheater is used to heat the refrigerants to a specified refrigerant state before entering the test section. The vapor generated in the test section is reliquified in an oversized condenser/subcooler in the cold water-glycol loop. Leaving the subcooler, the liquid refrigerants flow back to the receiver at the bottom of the system. An accumulator is connected to a high-pressure nitrogen tank to dampen the fluctuations of the flow rate and pressure. The filter/dryer is used to filter the impurities and noncondensable gas possibly existing in the loop. Varying the temperature and flow rate of the water-glycol mixture flowing through the condenser and subcooler allows us to control the pressure of the refrigerant loop. Two absolute pressure transducers are respectively installed at the inlet and exit of the test section with a resolution up to $\pm 2\text{kPa}$. All the refrigerant and water temperatures are measured by copper-constantan thermocouples (T-type) with a calibrated accuracy of $\pm 0.2^\circ\text{C}$. The test section is thermally insulated with a polyethylene insulation layer so that heat loss from it can be reduced significantly.

2.2 Test Section

The flow boiling for refrigerant flow in an annular duct with a small clearance between the inner and outer pipes is explored. As schematically shown in Fig. 2.3, the test section of the experimental apparatus is a horizontal annular duct with the outer pipe made of Pyrex glass to permit the visualization of boiling processes in the refrigerant flow. The outer Pyrex glass pipe is 160 mm long and 4 mm thick with inside diameter of 20 mm. Both ends of the pipe are connected with a copper tube of the same size by means of flanges and are sealed by O-rings. The inner smooth pipe has 16.0, 18.0, 19.0 or 19.6 mm nominal outside diameter (the pipe wall thickness is 1.5, 2.5, 3.0 or 3.3 mm) and is 0.41 m long, so that the hydraulic diameter of the annular duct D_h is 4.0, 2.0, 1.0 or 0.4 mm

(corresponding to the gap size of 2.0, 1.0, 0.5 or 0.2 mm for the duct). In order to insure the gap between the ducts being uniform, we first measure the average outside diameter of the inner pipe and the mean inside diameter of the Pyrex glass, with the accuracy of the measurement in the range of ± 0.01 mm. Then we photo the top and side view pictures and measure the average distance between the inside surface of the Pyrex glass pipe to the outside surface of the inner tube, with the accuracy also in the range of ± 0.01 mm. From the above procedures the duct gap is ascertained and its uncertainty is estimated to be 0.02 mm. An electric cartridge heater of 160 mm in length and 13.0 mm in diameter with a maximum power output of 800W is inserted into the inner pipe. Furthermore, the pipe has an inactive heating zone of 10 mm long at each end and is insulated with Teflon blocks and thermally nonconducting epoxy to minimize heat loss from it. Thermal contact between the heater and the inner pipe is improved by coating a thin layer of heat-sink compound on the heater surface before the installation of the heater. Then, 8 T-type calibrated thermocouples are electrically insulated by covering their beads with the electrically nonconducting thermal bond before they are fixed on the inside surface of the inner pipe so that the voltage signals from the thermocouples are not interfered by the DC current passing through the cartridge heater. The thermocouples are positioned at three axial stations along the smooth pipe. At each axial station, two to four thermocouples are placed at top, bottom, or two sides of the pipe circumference with 180° or 90° apart. The outside surface temperature T_w of the inner pipe is then derived from the measured inside surface temperature by taking into account the radial thermal conduction through the pipe wall. Figure 2.4 shows the detailed thermocouple locations at each axial station and the arrangement of the cartridge heater.

The test section for the new refrigerant in small tubes is described next. Due to the tubes tested being relatively small, the refrigerant flow rates in them are very low and

direct measurement of heat transfer and pressure drop in the tubes is difficult and is subject to large error. Thus 28 small tubes all made of copper, each having the same diameter and length, are put together to form the test section, as shown in Fig. 2.5. Each small tube has an inside diameter of 2.0 mm, outside diameter of 3.0 mm and length of 150 mm. Specifically, these 28 tubes are placed together side by side forming a plane tube bundle. In order to allow the refrigerant to flow smoothly into the small tubes, a section including divergent, convergent and straight portions is connected to the inlets of the tubes. Besides, another section including straight and convergent parts is attached to the exits of the tubes. Both the inlet and exit sections are formed by the stainless steel plates. At the middle axial location of the small tubes 14 thermocouples are soldered onto the outer surface of the tubes. Specifically, these thermocouples are soldered onto 14 selected tubes at the circumferential position of 45° from the top of the tube or from the bottom of the tube, as shown in Fig. 2.6. Two copper plates of 5-mm thick are respectively soldered on the upper and lower sides of the tube bundle shown in Fig. 2.7. Obviously small crevices exist between the tube outside surface and copper plates. Due to the good thermal contact of the copper plates and the tubes, there is no need to fill the crevices with conducting grease. Instead, the crevices provide the space for the thermocouple wires leading to the data logger. The copper plates contacting with the small tubes are heated directly by an electric-resistance heater of 2.6-mm wide, 0.5-mm thick and 2.5-m long. The heater is connected to a 500W DC power supply. Mica sheet is placed in the narrow space between the heater and copper plates to prevent leak of electric current to the copper plates. The power input to the heater is measured by a power meter with an accuracy of $\pm 0.5\%$. In order to reduce the heat loss from the heaters, the whole test section is wrapped with a 10-cm thick polyethylene layer.

2.3 Water loop for preheater

In order to maintain the preset refrigerant temperature at the test section inlet, a water loop is used to preheat the refrigerants before it arrives at the inlet. The water loop for the preheater system includes a double-tube, having a heat transfer area of 0.12m^2 , a 125-liter hot water container with three 2.0-kW heaters in it, and a 0.5-hp water pump which can deliver the hot water at specified temperature and flow rate to the preheater. In the preheater the hot water passes through the outer pipe while the liquid refrigerant flows in the inner pipe. The water flow rate is controlled by an AC motor through the change of the inverter frequency and by the by-pass valve. The connecting pipe between the preheater and test section is thermally insulated with a 5-cm thick polyethylene layer to reduce the heat loss from the pipe.

2.4 Water-glycol loop

The water-glycol loop is designed for condensing the refrigerant vapor and for subcooling the liquid refrigerant. The water-glycol loop is cooled by a water cooled R-22 refrigeration system. The cooling capacity is 3.5-kW for the water-glycol mixture at -20°C . The cold water-glycol mixture at a specified flow rate is driven by a 0.5-hp pump to the condenser as well as to the subcooler. A by-pass loop is provided to adjust the flow rate. By adjusting the mixture temperature and flow rate, the bulk temperature of the refrigerant in the subcooler can be controlled at a preset level.

2.5 DC Power Supply

As described above, the inner pipe in the test section is heated by a 800-W cartridge heater. A 50V-25A DC power supply delivers the required electric current to the cartridge heater. A Yokogawa DC meter is used to measure the DC current through the cartridge

heater with an accuracy of $\pm 1\%$. Then the voltage drop across the heater is measured by a Yokogawa multimeter. Thus the power input to the heater can be calculated.

2.6 Photographic System

The photographic apparatus established in the present study to record the bubble characteristics in the subcooled and saturated flow boiling in the annular duct consists of a Kodak Motion Corder Analyzer. The high-speed motion analyzer can take photographs up to 10,000 frames/s. Here, a recording rate of 2,000 frames/s is adopted to obtain the images of the bubble ebullition processes in the flow boiling. The data for some bubble characteristics are collected in the regions around the middle axial location ($z = 80$ mm). Note that the symbol z denotes the axial coordinate measuring from the inlet of the test section. After the experimental system reaches a statistically steady state, we start to record the boiling activity. The high-speed motion analyzer stores the images which are later downloaded to a personal computer. Then, the bubble departure diameter, bubble generation frequency and active nucleation site density are calculated by viewing more than 500 frames at each location. In order to achieve the highest possible resolution and to eliminate errors in calibration, the camera lens is fixed at a constant focal length, resulting in a fixed viewing area. Typically, a total of over 150 bubble diameter measurements are used to construct the present data. The bubble generation frequency is measured by counting the total number of bubbles that emerge from the heating surface during 1 second.

2.7 Data acquisition

The data acquisition system includes a recorder (Yokogawa HR-2300), a 24V-3A power supply, and a controller. The water flowmeter and differential pressure transducer need the power supply as a driver to output an electric current of 4 to 20 mA. The data

signals are collected and converted by a data acquisition system (Hybrid recorder). The converted signals are then transmitted to a host computer through a GPIB interface for further calculation.

The system automatically monitors all the T-Type thermocouples, pressure transducers, differential pressure transducer and mass flowmeters. The thermodynamic and transport properties of the refrigerants are obtained by the manual from AlliedSignal Co. Ltd. and by the ASHRAE handbook.

2.8 Experimental procedures

Before a test is started, the temperature of the refrigerant under investigation in the test section is compared with its saturation temperature corresponding to the measured saturation pressure and the allowable difference is kept in the range of 0.2-0.3K. Otherwise, the system is re-evacuated and then re-charged to remove the air existing in the refrigerant loop. In the test the liquid refrigerant or liquid-vapor refrigerant mixture at the inlet of the test section is first maintained at a specified temperature and/or quality by adjusting the water-glycol temperature and flow rate. In addition, we adjust the thermostat temperature in the water loop to stabilize the refrigerant temperature and/or quality at the test section inlet. Then, we regulate the refrigerant pressure at the test section inlet by adjusting the opening of the gate valve locating right after the exit of the test section. Meanwhile, by changing the current of the DC motor connecting to the refrigerant pump, the refrigerant flow rate can be varied. The imposed heat flux from the heater to the refrigerant is adjusted by varying the electric current delivered from the DC power supply. By measuring the current delivered to and voltage drop across the heater and photographing the bubble activity, we can calculate the heat transfer rate to the refrigerant and obtain the bubble characteristics. All tests are run at statistically steady-state conditions. The whole system is

considered to be at a statistically steady state when the time variations of the system pressure and imposed heat flux are respectively within $\pm 1\%$ and $\pm 4\%$, and the time variations of the heated wall temperature are less than $\pm 0.2^{\circ}\text{C}$ for a period of 100 minutes. Then all the data channels are scanned every 5 seconds for a period of 50 seconds.

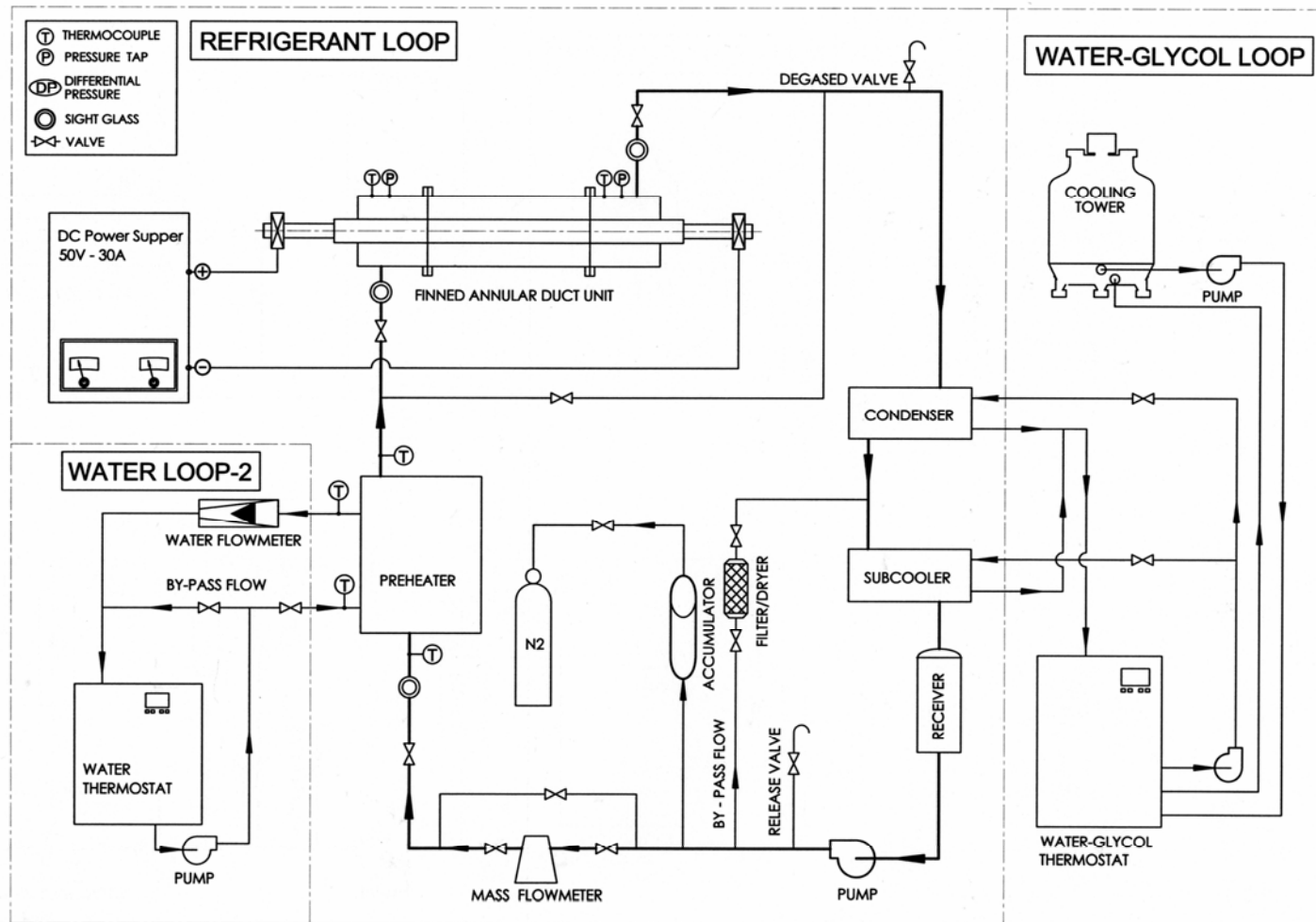


Fig. 2.1 Schematic of experimental system for the annular duct

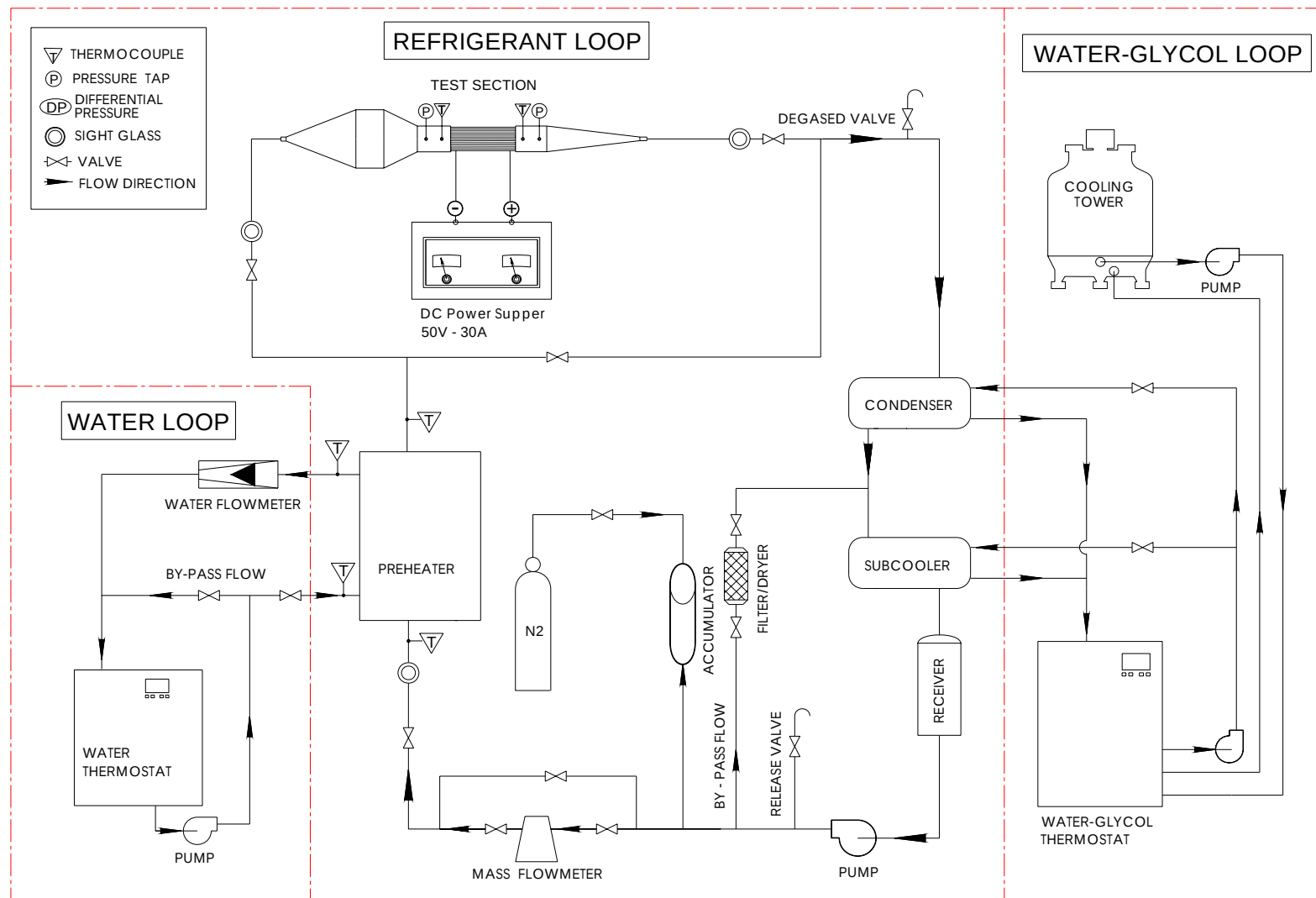


Fig 2.2 Schematic layout of the experimental system for new refrigerant in small tubes

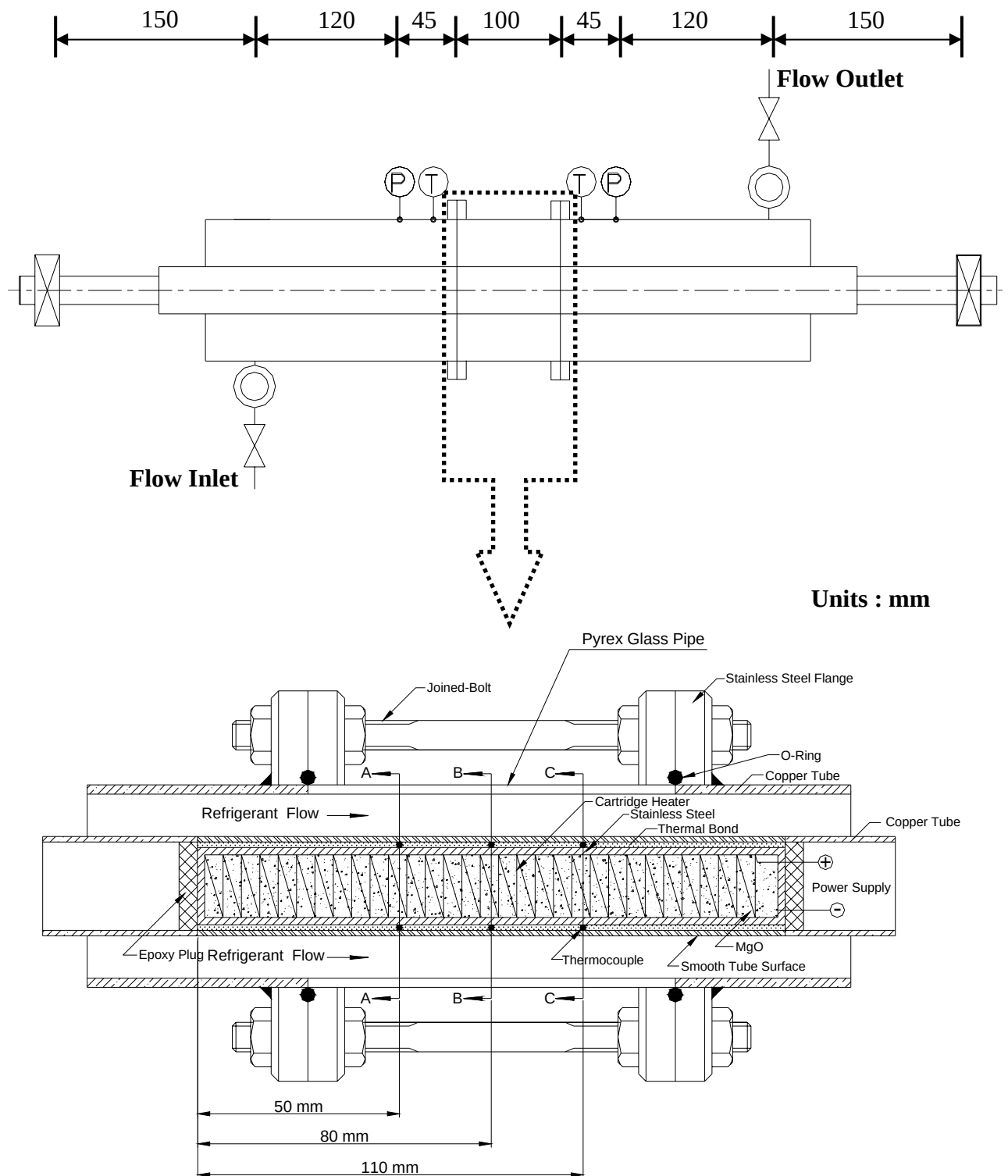


Fig. 2.3 The detailed arrangement of the test section for the annular duct

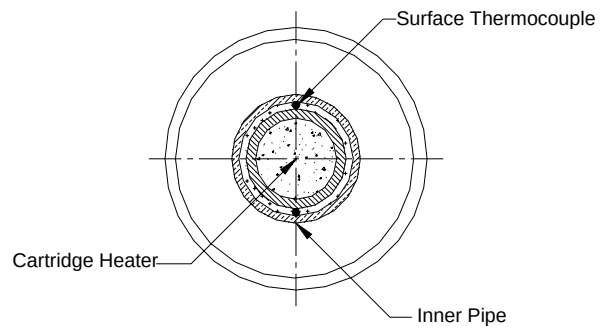
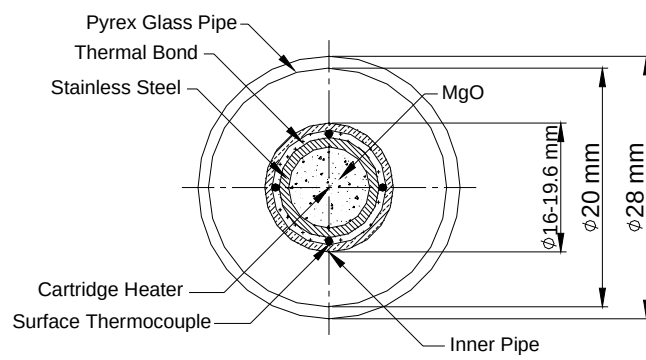
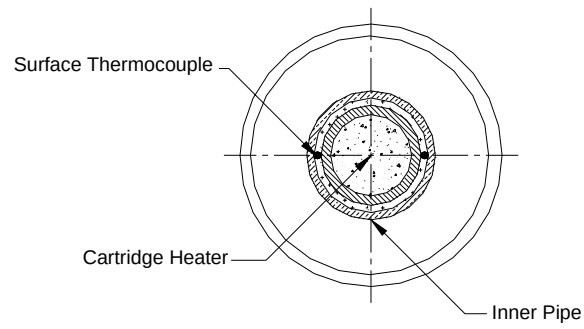


Fig. 2.4 The cross-sectional view of the annular duct showing the heater and locations of the thermocouples.

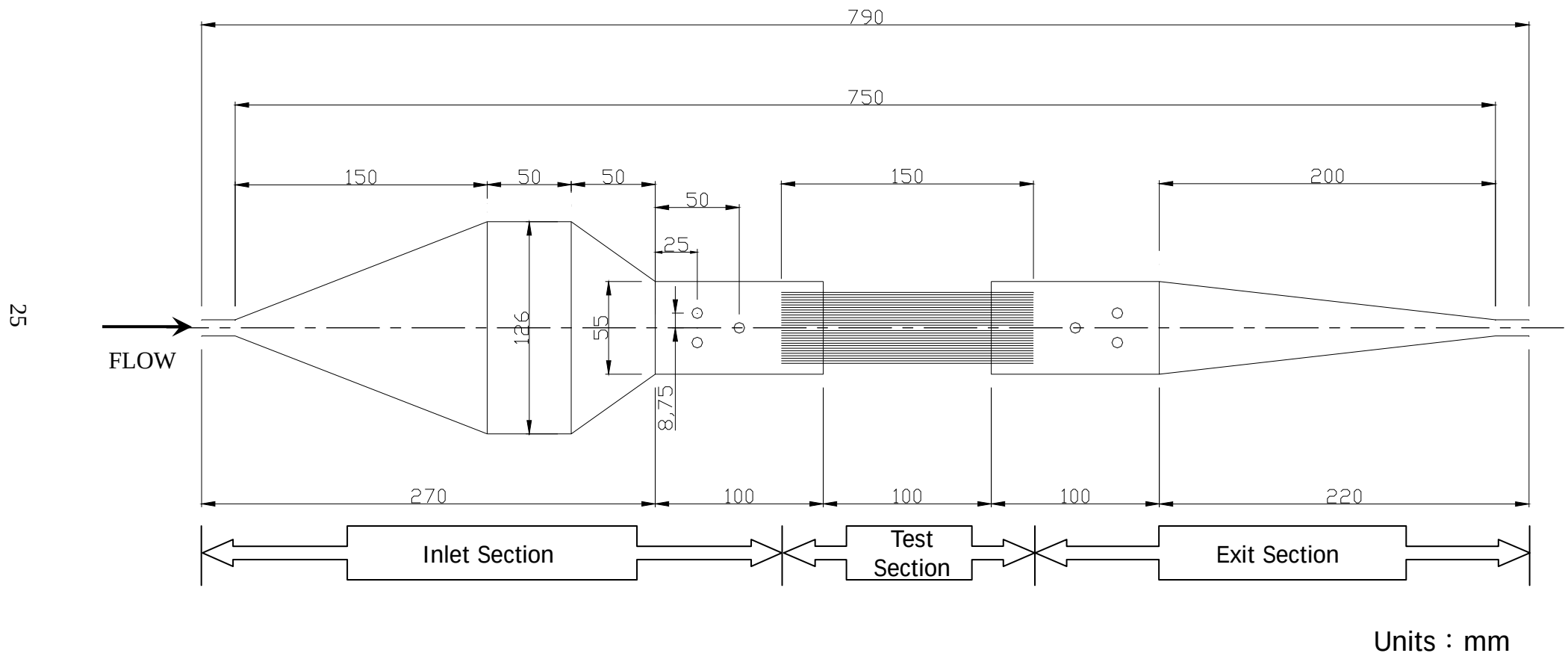
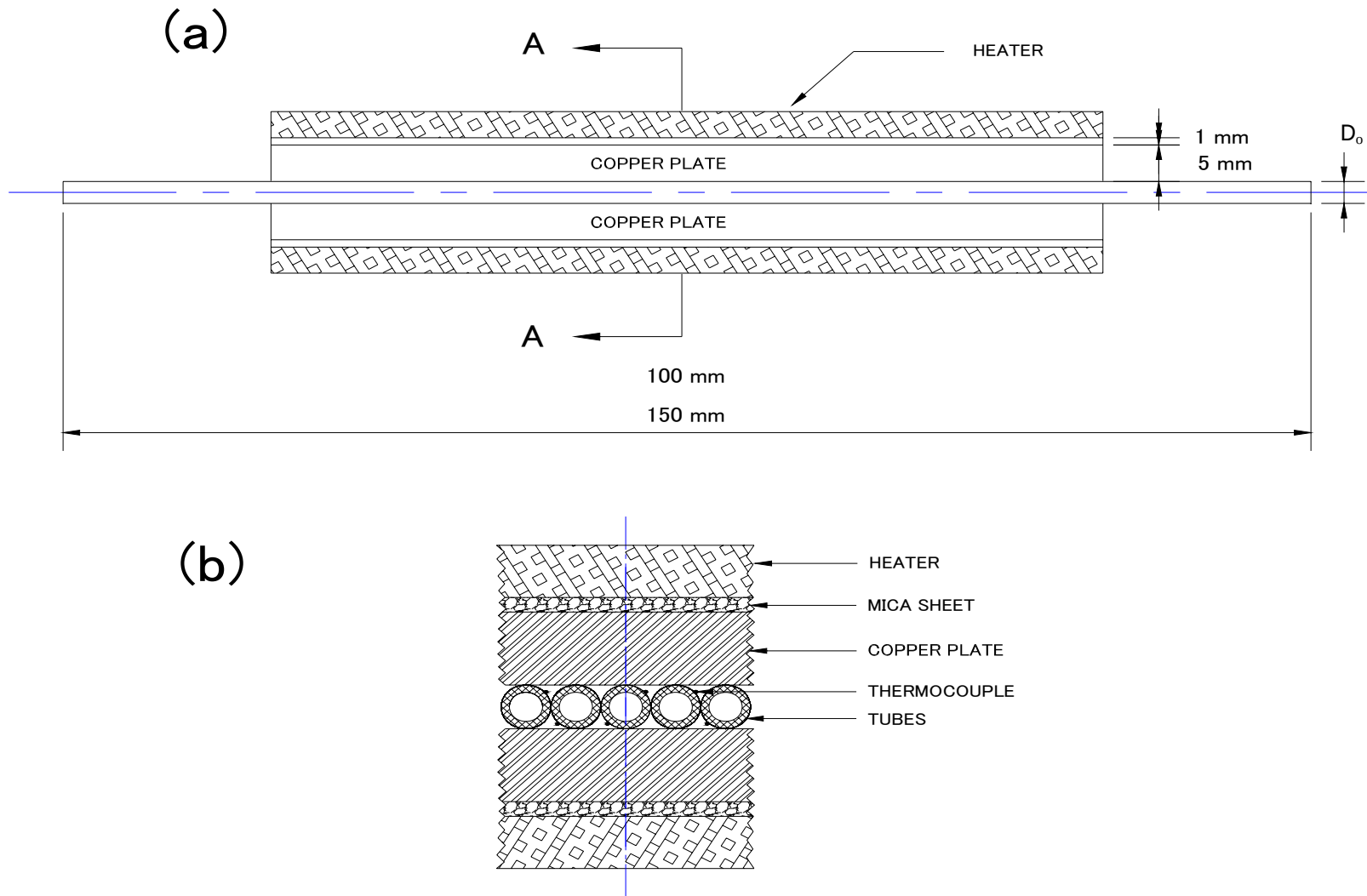


Fig 2.5 Schematic diagram of test section for the small tubes along with the inlet and exit sections.



A-A CROSS SECTIONAL VIEW

Fig 2.6(a) Test section of the small tubes and (b) locations of the thermocouples.

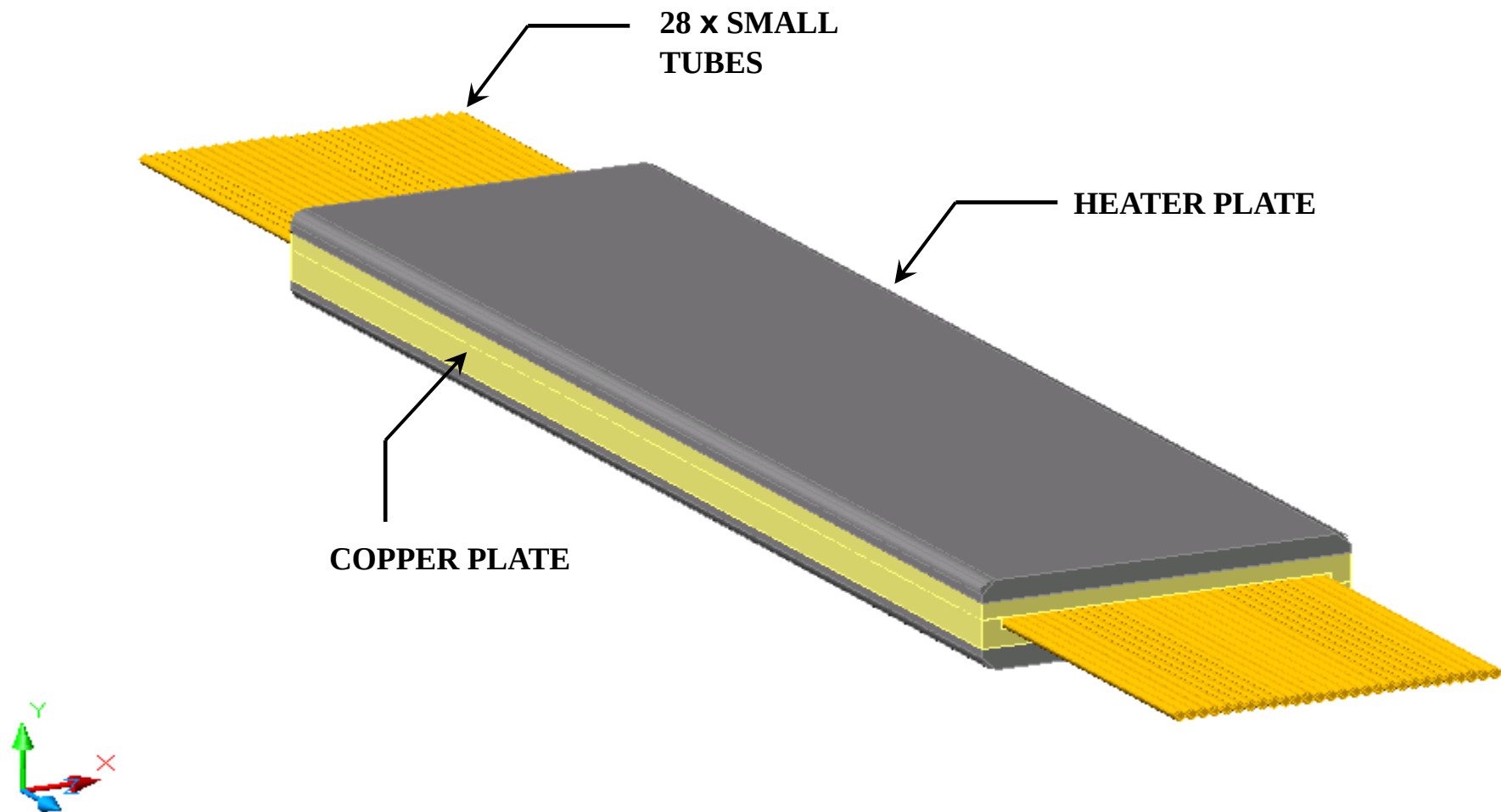


Fig 2.7 The detail of test section.

CHAPTER 3

DATA REDUCTION

In the present study of heat transfer and bubble characteristics in the flow boiling of new refrigerants in an annular duct, the gap between the inner and outer circular pipes is varied from 0.2 to 2.0 mm. Besides, the evaporation heat transfer coefficient and pressure drop of R-407C flow in the small tubes are measured. The data reduction analyses are needed to calculate the flow boiling and evaporation heat transfer coefficient and pressure drop from the raw data measured in the test sections.

3.1 Flow Boiling Heat Transfer Coefficient in Annular Duct

The imposed heat flux to the refrigerant flow in the annular duct is calculated on the basis of the total power input and the total outside heat transfer area of the inner pipe of the annular duct. The total power input is computed from the product of the measured voltage drop across the cartridge heater and the electric current passing through it. The imposed heat flux at the outside surface of the inner pipe is then evaluated from the relation

$$q = VI / A_s \quad (3.1)$$

where V and I respectively represent the measured voltage drop and current. Before the two-phase experiments, the total heat loss from the test section is evaluated by comparing the total power input from the power supply Q_s with the total heat transfer rate to the single phase refrigerant flow expressed as $G \cdot A_{cs} \cdot c_{pr} \cdot \Delta T$, where A_{cs} is the cross-sectional area of the annular duct and ΔT is the difference in the refrigerant temperature at the exit and inlet of the test section. The relative heat loss from the test section is defined as

$$\varepsilon = (Q_s - G \cdot A_{cs} \cdot c_{pr} \cdot \Delta T) / Q_s \quad (3.2)$$

The results from this heat loss test indicate that the heat loss from the test section is

generally less than 3% of the total power input. Then, the average single-phase convection heat transfer coefficient over the entire heated surface is defined as

$$h_{r,l} = \frac{Q_n}{A_s \cdot (\overline{T_w} - T_{r,ave})} \quad (3.3)$$

and

$$T_{r,ave} = \frac{T_{r,i} + T_{r,o}}{2} \quad (3.4)$$

where Q_n is the net power input to the liquid refrigerant in the annular duct, which is estimated from the product $G \cdot A_{cs} \cdot c_{pr} \cdot \Delta T$, and $T_{r,ave}$ is the average of the measured inlet and outlet temperatures of the refrigerant flow through the test section, which is taken as the average bulk refrigerant liquid temperature. Note that $\overline{T_w}$ denotes the average outside surface temperature of the inner pipe measured at the selected thermocouple locations. The outside surface temperature at each thermocouple location is deduced from the measured inside surface temperature of the inner pipe by subtracting the radial temperature drop due to the radial heat conduction in the pipe wall. Thus we have

$$T_w = T_{w,i} - Q_n \frac{\ln(D_o / D_i)}{2\pi k_w l} \quad (3.5)$$

In the two-phase test, the local saturated flow boiling heat transfer coefficients at a given axial location are respectively defined as

$$h_{r,sat} = \frac{Q_n / A_s}{(T_w - T_{sat})} \quad (3.6)$$

Here T_r is the local mean liquid refrigerant temperature and is estimated by assuming that it varies linearly in the axial direction.

3-2 Two-Phase Heat Transfer in Small Circular Tubes

The vapor quality of the R-407C entering the test section inlet is evaluated from the energy balance for the preheater. Based on the temperature drop on the water side the heat transfer in the preheater is calculated from the relation,

$$Q_{w,p} = W_{w,p} C_{p,w} (T_{w,p,i} - T_{w,p,o}) \quad (3.7)$$

While the heat transfer to the refrigerant in the preheater is the summation of the sensible heat transfer (for the temperature rise of the refrigerant to the saturated value) and latent heat transfer (for the evaporation of the refrigerant),

$$Q_{w,p} = Q_{sens} + Q_{lat} \quad (3.8)$$

Where

$$Q_{sens} = W_r C_{p,r} (T_{r,sat} - T_{r,p,i}) \quad (3.9)$$

$$Q_{lat} = W_r \cdot i_{fg} \cdot x_{p,o} \quad (3.10)$$

The above equations can be combined to evaluate the refrigerant quality at the exit of the preheater that is considered to be the same as the vapor quality of the refrigerant entering the test section. Specifically,

$$x_{in} = x_{p,o} = \frac{1}{i_{fg}} \left[\frac{Q_{w,p}}{W_r} - c_{p,r} (T_{r,sat} - T_{r,p,i}) \right] \quad (3.11)$$

The total change of the refrigerant vapor quality in the test section is then deduced from the net heat transfer rate from the heater to the refrigerant in the test section, Q_n . Thus

$$\Delta x = \frac{Q_n}{W_r i_{fg}} \quad (3.12)$$

Finally, the average heat transfer coefficient for the evaporation of R-407C in the test section is determined from the definition

$$h_r \equiv \frac{Q_n}{A_s (T_{wall} - T_{r,sat})} \quad (3.13)$$

3-3 Friction Factor in Small Circular Tubes

Note that in the evaporation of R-407C in the tubes the flow accelerates, causing the pressure to drop, as it moves downstream. Besides, the refrigerant pressure also drops due to the contraction at the test section inlet and rises due to the expansion at the exits of the small tubes. Thus, in the refrigerant flow the two phase frictional pressure drop ΔP_f associated with the R-407C evaporation in the small tubes is calculated by subtracting the pressure drop due to flow acceleration ΔP_a and pressure drop at the test section inlet ΔP_i and by adding the pressure rise at the test section exit ΔP_o from the measured total pressure drop ΔP_{exp} . The frictional pressure drop is hence given as

$$\Delta P_f = \Delta P_{exp} - \Delta P_a - \Delta P_i + \Delta P_o \quad (3.14)$$

Note that the acceleration pressure drop is estimated by the homogeneous model for two phase flow [44] as

$$\Delta P_a = G^2 v_{fg} \Delta x \quad (3.15)$$

Moreover, the pressure drops associated with the sudden contraction and the pressure rises associated with the sudden expansion ΔP_o and ΔP_i for two phase flow moving through the inlet and exit ports estimated by Collier [36] based on a separated flow model are chosen here. They can be expressed as

$$\Delta P_i = \left(\frac{G}{C_c}\right)^2 (1 - C_c) \left\{ \frac{(1 + C_c) [x_{in}^3 v_g^2 / \alpha^2 + (1 - x_{in})^3 v_l^2 / (1 - \alpha)^2]}{2 [x_{in} v_g + (1 - x_{in}) v_l]} - C_c \left[\frac{x_{in}^2 v_g}{\alpha} + \frac{(1 - x_{in})^2 v_g}{(1 - \alpha)} \right] \right\} \quad (3.16)$$

and

$$\Delta P_o = G^2 \sigma (1 - \sigma) v_l \left[\frac{(1 - x_{in})^2}{(1 - \alpha)} + \left(\frac{v_g}{v_l}\right) \frac{x_{in}^2}{\alpha} \right] \quad (3.17)$$

where C_c in equation (3-16) is the coefficient of contraction and it is a function of the contraction ratio σ . The void fraction α in the above equations is calculated from the

correlation given by Zivi [45] as

$$\alpha = \frac{1}{1 + \left(\frac{1-x_{in}}{x_{in}} \right) \left(\frac{\rho_g}{\rho_l} \right)^{2/3}} \quad (3.18)$$

Finally, for the evaporation of R-407C in the small tubes the two phase friction factor is expressed as

$$f_{tp} = \frac{\Delta P_f D_i}{2G^2 v_m L} \quad (3.19)$$

where L is the length of the small tubes and v_m is the mean specific volume of the vapor-liquid mixture in the small tubes when they are homogeneously mixed and can be expressed as

$$v_m = [x_{in} v_g + (1-x_{in}) v_l] = (v_l + x_{in} v_{fg}) \quad (3.20)$$

3.2 Uncertainty Analysis

Uncertainties of the heat transfer coefficients are estimated according to the procedures proposed by Kline and McClintock for the propagation of errors in physical measurement [46]. The results from this uncertainty analysis are summarized in Table 3.1 and 3.2.

Table 3.1 Summary of the uncertainty analysis for boiling in annular duct

Parameter	Uncertainty
Annular pipe geometry	
Length, width and thickness (%)	$\pm 0.5\%$
Area (%)	$\pm 1.0\%$
Parameter measurement	
Temperature, T ($^{\circ}\text{C}$)	± 0.2
Temperature difference, ΔT ($^{\circ}\text{C}$)	± 0.3
System pressure, P (kPa)	± 2
Mass flux of refrigerant, G (%)	± 2
Single-phase heat transfer on inner pipe	
Imposed heat flux, q (%)	± 4.5
Heat transfer coefficient, $h_{r,l}$ (%)	± 12.5
Saturated flow boiling heat transfer on inner pipe	
Imposed heat flux, q (%)	± 4.5
Heat transfer coefficient, $h_{r,sat}$ (%)	± 14.5

Table 3.2 Summary of the uncertainty analysis for evaporation in small circular tubes

Parameter	Uncertainty
tube geometry	
Length, width and thickness (%)	$\pm 0.5\%$
Area (%)	$\pm 1.0\%$
Parameter measurement	
Temperature, T ($^{\circ}\text{C}$)	± 0.2
Temperature difference, ΔT ($^{\circ}\text{C}$)	± 0.4
Pressure drop, ΔP (Pa)	± 200
Mass flux of refrigerant, G (%)	± 2
System pressure, P (kPa)	± 2
Single-phase heat transfer in small tubes	
Imposed heat flux, q (%)	± 4.5
Heat transfer coefficient, $h_{r,l}$ (%)	± 10.5
Evaporation heat transfer in small tubes	
Imposed heat flux, q (%)	± 4.5
Inlet vapor quality, x_{in} (%)	± 9.5
Heat transfer coefficient, h_r (%)	± 14.5
Friction pressure drop, ΔP_f (%)	± 16.5

CHAPTER 4

SATURATED FLOW BOILING OF R-134a IN a HORIZONTAL NARROW ANNULAR DUCT

In this chapter the measured heat transfer data and observed bubble characteristics for the saturated flow boiling of R-134a in the narrow annular duct affected by the refrigerant mass flux, system pressure and duct size are presented. In the tests we have saturated liquid R-134a at the inlet of the duct ($x_{in} = 0$). The experiments are performed for the refrigerant mass flux varying from 400 to 700 kg/m²s ($G = 400$ -500 kg/m²s for $\delta = 2.0$ mm, $G = 500$ -600 kg/m²s for $\delta = 0.5$ & 1.0 mm and $G = 600$ -700 kg/m²s for $\delta = 0.2$ mm), imposed heat flux q from 1 to 55 kW/m² and the system pressure set at 414 kPa and 488 kPa (corresponding to the R-134a saturation temperature $T_{sat} = 10^\circ\text{C}$ and 15°C) for the annular gap of the duct $\delta = 0.2$ mm to 2.0 mm. The measured boiling heat transfer data are expressed in terms of the boiling curves and boiling heat transfer coefficient. Moreover, the overview of the boiling flow in the duct and the close view flow photos taken at a small region around the middle axial station $z = 80$ mm are presented to illustrate the bubble characteristics in the boiling flow. Finally, empirical correlations will be proposed to correlate the present data for the saturated flow boiling heat transfer coefficient, mean bubble departure diameter, mean bubble departure frequency and average active nucleation site density.

4.1 Single-phase Heat Transfer

Before beginning the two-phase flow boiling experiments, the single-phase convective heat transfer experiments are conducted for liquid R-134a. The measured single-phase convection heat transfer coefficients are compared with the correlations proposed by Dittus-Boelter [47] and Gnielinski [48] for $Re_1 > 2,300$ and by Choi et al. [51] for $Re_1 < 2,300$. In the single-phase heat transfer tests the refrigerant mass flux is varied from 200 to 1,200 kg/m²s, gap size δ from 0.2 to 2.0 mm (corresponding to the Reynolds number of the refrigerant flow from 714 to 12,506) for the refrigerant inlet temperature fixed at 15°C . Selected results from these tests are plotted in Figure 4.1.

The Dittus-Boelter correlation is

$$Nu_{1\phi} = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \quad \text{for } Re > 10^4 \quad (4.1)$$

and the Gnielinski correlation is

$$Nu_{1\phi} = \frac{(f_f/2)(Re-1000)Pr}{1.07 + 12.7\sqrt{f_f/2}(Pr^{2/3}-1)} \quad \text{for } 2,300 < Re < 10^5 \quad (4.2)$$

$$\text{where } f_f = (1.58 \ln Re - 3.28)^{-2} \quad (4.3)$$

and the Choi correlation is

$$Nu_{1\phi} = 0.000972 \cdot Re^{1.17} \cdot Pr^{1/3} \quad \text{for } Re < 2,000 \quad (4.4)$$

The results from the single-phase heat transfer tests indicate that the energy balance between the heater and the refrigerant flow defined in Eq.(3.2) is within 5% for all run. This insures the heat loss from the test section is rather small and the design of test section is suitable for our measurement. The results in Figure 4.1 manifest that for $\delta = 0.5\text{-}2.0$ mm the present data for h_l can be well correlated with the Gnielinski correlations. But for the smaller gap of 0.2 mm the Choi correlation is closer to the present data.

4.2 Saturated Flow Boiling Curves

The effects of the refrigerant mass flux, gap size of the duct and refrigerant saturated temperature on the saturated flow boiling characteristics at the middle axial location ($z = 80$ mm) of the narrow annular duct are shown in Figures 4.2-4.6 by presenting the boiling curves for various G , δ and T_{sat} .

First, the effects of the refrigerant mass flux on the saturated flow boiling curves are shown in Figures 4.2 and 4.3. The results indicate that at a low imposed heat flux the wall superheat is lower than that for the onset of nucleate boiling (ONB) and no bubble nucleates from the heating surface. Hence heat transfer in the flow results completely from

the single-phase forced convection. As the imposed wall heat flux is raised gradually, the wall superheat increases correspondingly. At a certain wall superheat bubbles start to nucleate from the heating surface and we have onset of nucleate boiling in the flow. Beyond the ONB there is a significant increase in the slope of the boiling curves, implying that a small rise in the wall superheat causes a large increase in the heat transfer rate from the wall to refrigerant. Note that at increasing refrigerant mass flux the boiling curves shift slightly to the left when the gap size reduces (Figure 4.3), which indicates that at a higher refrigerant mass flux the heat transfer in the saturated boiling is slightly better. This small increase in the heat transfer rate with the mass flux is mainly due to the bubbles in the liquid refrigerant move more vigorously and turbulently due to the higher refrigerant mass flux. The results also indicate that the required imposed heat flux to achieve ONB is influenced noticeably by the change in the mass flux. Specifically, the required imposed heat flux to achieve ONB is slightly higher for a higher mass flux except for the smallest δ of 0.2 mm tested here (Figure 4.3(b)). At this δ the ONB heat flux is nearly the same for $G = 600$ & $700 \text{ kg/m}^2\text{s}$. When compared with the results for the subcooled flow boiling to be presented in chapter 5, no apparent temperature undershoot occurs during the onset of boiling when the flow is saturated.

Then, the effects of the duct size on the saturated flow boiling curves are shown in Figure 4.4. It is noted that the boiling curve shifts significantly to the left as the gap in the duct is reduced from 2 mm to 0.2 mm, indicating that the boiling heat transfer in the smaller duct is substantially better. It is also evident from the data that a lower imposed heat flux is needed to initiate boiling on the heated surface for the smaller duct. Then, the data shown in Figures 4.5 and 4.6 suggest that the effects of the refrigerant saturation temperature on the boiling curves including the single-phase forced convection and nucleate boiling regimes are insignificant. Finally, the relation between the imposed heat flux and wall superheat temperature at the incipient boiling is shown in Figure 4.7. The results indicate that at the same q_{ONB} the wall superheat at ONB is smaller for the narrower duct.

4.3 Saturated Flow Boiling Heat Transfer Coefficient

The effects of the refrigerant mass flux, duct size and refrigerant saturated temperature on the saturated flow boiling heat transfer of R-134a at the middle axial

location ($z = 80$ mm) in the narrow annular duct are shown in Figures 4.8-4.12 by presenting the saturated flow boiling heat transfer coefficient against the imposed heat flux for various G , δ and T_{sat} . The results indicate that at given G , δ and T_{sat} the saturated boiling heat transfer coefficient increases substantially with the imposed heat flux. For example, at $T_{\text{sat}} = 15^\circ\text{C}$, $\delta = 0.5$ mm and $G = 500$ kg/m²s, the saturated boiling heat transfer coefficient for $q = 46$ kW/m² is about 91% higher than that for $q = 6$ kW/m² (Figure 4.9(a)). This large increase in the saturated boiling heat transfer coefficient is ascribed to the higher active nucleation site density on the heating surface and higher bubble departure frequency for a higher imposed heat flux. The effects of the refrigerant mass flux on the saturated flow boiling heat transfer coefficient shown in Figures 4.8 and 4.9 manifest that the saturated boiling heat transfer coefficient rises slightly with the refrigerant mass flux only at a high q for $\delta = 0.5$ and 0.2 mm. For instance, the saturated boiling heat transfer coefficient at $q = 46$ kW/m², $T_{\text{sat}} = 15^\circ\text{C}$ and $\delta = 0.5$ mm for $G = 600$ kg/m²s is about 5% higher than that for $G = 500$ kg/m²s (Figure 4.9(a)). For the larger δ of 1.0 and 2.0 mm the effects of G and h_r are negligible, as evident from the data in Figure 4.8.

Then, the effects of the gap size of the duct on the saturated flow boiling heat transfer coefficient are shown in Figure 4.10. The data show that the saturated boiling heat transfer increases noticeably with a decrease in the channel gap. For example, at $q = 46$ kW/m², $T_{\text{sat}} = 15^\circ\text{C}$ and $G = 500$ kg/m²s the saturated boiling heat transfer coefficient for $\delta = 0.5$ mm is about 31% higher than that for $\delta = 2.0$ mm (Figure 4.10(a)). Besides, at $G = 600$ kg/m²s and the same q and T_{sat} the saturated boiling heat transfer coefficient for $\delta = 0.2$ mm is about 32% higher than that for $\delta = 1.0$ mm (Figure 4.10(b)). Since the shear stress of the flow acting on the heated surface in a smaller channel becomes higher, the cavities on the heating surface can be more easily wetted. Moreover, the flow pattern changes from the bubbly flow for $\delta = 2.0$ & 1.0 mm to become a slug flow for $\delta = 0.5$ & 0.2 mm. These effects are thought to be the main reasons for the enhancement of nucleate and convection boiling heat transfer when the channel size is reduced. Finally, the data shown in Figures 4.11 and 4.12 suggest that the saturation temperature of the refrigerant exhibits negligible effects on the boiling heat transfer coefficient.

4.4 Bubble Characteristics in Saturated Flow Boiling

To illustrate the bubble behavior in the duct, selected photos of the boiling flow of R-134a from the side and top view covering the whole narrow duct to illustrate the effects of the gap size of the duct are shown in Figures 4.13 to 4.16. The results in Figure 4.13 for the large duct with $\delta = 2.0$ mm clearly indicate that in the relatively upstream region near the duct inlet a great number of bubbles already exist in the flow. Bubbles of varying size can be seen. Specifically, in the upper part of the duct larger bubbles dominate obviously due to the buoyancy effects. These photos suggest that for $\delta = 2.0$ mm the onset of nucleation boiling occurs immediately after the refrigerant enters the heated section of the duct. Because of the presence of the flanges, this bubble cannot be seen from the photos (Figure 2.2). Note that for a high G of $600 \text{ kg/m}^2\text{s}$ and a smaller δ of 1.0 mm some delay in the bubble nucleation is seen in Figure 4.14. Besides, the bubble nucleation is earlier in the lower part of the duct. This is due to the difference in the buoyancy effect in different parts of the duct. More specifically, in the lower portion of the duct the flow is heated from above and hence is thermally stable. This in turn results in a lower convection heat transfer coefficient and obviously the heated surface temperature is higher for a fixed wall heat flux. This higher T_w causes the earlier inception of the bubbles from the surface in the lower portion of the duct. Because the ONB somewhat unstable, the ONB location is not symmetry with respect to the vertical central plane through the duct axis. In fact, the ONB locations move back and forth with time in an irregular manner. But for a further reduction of the duct gap to 0.5mm an opposite trend appears with bubbles nucleated on the heated surface earlier in the upper part of the duct (Figure 4.15). This results from the fact that at the smaller Re for the smaller δ of 0.5 and 0.2 mm laminar forced convective liquid flow dominates in the duct since the buoyancy-to-inertia ration is also small ($Gr/Re^2 = 4.14 \times 10^{-2}$ & 6.63×10^{-3}). In this laminar flow ($Re = 2670$ & 1070) subject to the inner duct heating Ciampi et al. [52] found that a helicoidal flow motion appears in the duct which maintains a supply of warm liquid at the top of the cylinder and cold liquid to the bottom[52], which causes higher T_w in the upper portion of the duct. Thus the bubble nucleation takes place earlier in the upper part of the heating surface. Comparing the results in Figures 4.13-4.15 clearly shows that in the smaller duct the coalescence of bubbles is more pronounced. For $\delta = 1.0$ mm in the exit half of the duct many big bubbles exist. When the duct gap is reduced to 0.5 mm Figure 4.15 shows that big bubbles even dominate in the upper portion of the exit half of the duct. Thus in this region we have a slug flow. For a further reduction

of δ to 0.2 mm the bubble coalescence is so intense that the slug flow dominates the upper portion of the entire duct (Figure 4.16). The above results manifest that in the smaller ducts of 0.5& 0.2 mm the bubbly flow and slug flow coexist.

The photos of the boiling flow taken for the cases at different duct sizes and imposed heat fluxes in the small region around the middle axial location marked on Figures 4.13(a)-4.16(a) are shown in Figure 4.17. First of all, it is noted from the photo taken from the duct for $\delta = 1.0$ mm shown in Figure 4.17(a) for the case at $T_{\text{sat}} = 15^\circ\text{C}$ and $G = 600$ kg/m²s at the imposed heat flux $q = 15$ kW/m² that a number of discrete bubbles nucleate from the cavities and slide along the heating surface. As the imposed heat flux is increased to $q = 25$ kW/m², the active bubble nucleation density increases and a lot more bubbles appear and they move faster (Figure 4.17(b)). Many coalescence bubbles are seen as the heat flux is raised to $q = 35$ kW/m² (Figure 4.17(c)). Then, the photos taken from the smaller duct with $\delta = 0.5$ mm shown in Figure 4.17(d) for the same G , q , and T_{sat} indicate that a large number of bubbles generated from the cavities in the heating surface tend to merge together to form big bubbles. As the bubbles become larger, they become distorted and elongated as they slide on the heating surface. As the imposed heat flux is increased slightly to $q = 25$ kW/m² (Figure 4.17(e)), the active bubble nucleation density increases and bubbles collide and coalesce more frequently. The coalescence bubbles rise faster than the tiny bubbles due to the larger buoyancy force associated with them. As the heat flux is raised to $q = 35$ kW/m² (Figure 4.17(f)), coalescence of the bubbles occurs irregularly at a very high rate. The coalescence bubbles can be very large. In fact, the liquid slugs and discrete bubbles coexist in the duct. At even higher imposed heat flux for $q > 30$ kW/m², the bubble departure frequency is very high so that it is difficult to clearly distinguish the individual bubbles. In general, the bubble departure frequency increases substantially with the imposed heat flux due to the fact that an increase in the imposed heat flux directly provides more energy to the cavities and more cavities on the heating surface can be activated. Besides, the bubble departure diameter increases slightly at increasing imposed heat flux due to the rise in the wall superheat. Then, the corresponding photo taken from the even smaller duct with $\delta = 0.2$ mm are shown in Figures 4.17(g)~(i) for the same G , q and T_{sat} . Note that for the smaller δ more coalescence bubbles are seen and they are even bigger especially at a higher imposed heat flux. This causes less bubbles nucleated at the heated surface.

The bubble characteristics in the narrow duct around the middle axial location for $\delta = 0.5$ mm affected by the refrigerant mass flux and saturated temperature are illustrated by the photos in Figure 4.18. The results in Figures 4.18(a)-(f) indicate that at a higher mass flux the liquid refrigerant flow moves at a higher speed, which in turn tends to sweep the bubbles more quickly away from the heating surface. Collision and coalescence of bubbles are still significant. Besides, the bubble departure frequency is higher and the bubbles are smaller and in violent agitating motion. However, the active nucleation site density is lower. Note that at the low mass flux the bubble coalescence is more important and a number of bigger bubbles form in the duct. Then, the effects of the refrigerant saturation temperature on the bubble characteristics are illustrated by comparing the photos in Figures 4.18(a)~(c) with Figures 4.18(g)~(i). The results indicate that at a lower saturation temperature the bubbles grow bigger and move slower due to the higher surface tension. Small bubbles are easier to merge into big bubbles.

To be more quantitative on the bubble characteristics, we move further to estimate the average bubble departure diameter and frequency and the average active bubble nucleation site density on the heating surface for the cases with the bubbly flow dominated in the duct from the images of the boiling flow stored in the video tapes. The results from this estimation are examined in the following. The effects of the three parameters, namely, the refrigerant mass flux, duct size and refrigerant saturated temperature, on the mean bubble departure diameter for the saturated flow boiling of R-134a at the middle axial location ($z = 80$ mm) in the annular duct are shown in Figures 4.19-4.23 by presenting the average bubble departure diameter against the imposed heat flux for various G , δ and T_{sat} . First, the effects of the refrigerant mass flux on the average bubble departure diameter shown in Figures 4.19 and 4.20 indicate that the average departing bubble is only slightly larger for a lower refrigerant mass flux. For example, at $q = 30 \text{ kW/m}^2$, $T_{\text{sat}} = 15^\circ\text{C}$ and $\delta = 0.5$ mm, the average bubble departure diameter for $G = 500 \text{ kg/m}^2\text{s}$ is only about 11% larger than that for $G = 600 \text{ kg/m}^2\text{s}$ (Figure 4.20(a)). Then, the data given in Figure 4.21 also suggest that the average bubble departing from the heated surface is slightly larger in the smaller duct only at the lower G of $500 \text{ kg/m}^2\text{s}$ for the imposed heat flux $q > 20 \text{ kW/m}^2$. For example, at $q = 30 \text{ kW/m}^2$, $T_{\text{sat}} = 15^\circ\text{C}$ and $G = 500 \text{ kg/m}^2\text{s}$, the average bubble departure diameter for $\delta = 0.5$ mm is only about 18% higher than that for $\delta = 2.0$ mm (Figure 4.21(a)). Otherwise the effects of the duct gap on the bubble departure diameter are

relatively small. Finally, the results in Figures 4.22 and 4.23 indicate that the average bubble departure diameter is smaller for a higher refrigerant saturated temperature. For instance, at $q = 30 \text{ kW/m}^2$, $G = 500 \text{ kg/m}^2\text{s}$ and $\delta = 2.0 \text{ mm}$, and the average departing bubble for $T_{\text{sat}} = 10^\circ\text{C}$ is about 18% larger than that for $T_{\text{sat}} = 15^\circ\text{C}$ (Figure 4.22(a)).

How the bubble departure frequency is affected by the three parameters for the saturated flow boiling of R-134a at the middle axial location ($z = 80 \text{ mm}$) in the annular duct are shown in Figures 4.24-4.28 by presenting the average bubble departure frequency against the heat flux for various G , δ and T_{sat} . Note that the increase of the bubble departure frequency with the imposed heat flux is rather significant for all cases presented here. First, the effects of the refrigerant mass flux on the saturated flow boiling average bubble departure frequency are shown in Figures 4.24 and 4.25. The results indicate that the average bubble departure frequency is somewhat higher for a higher refrigerant mass flux especially at high imposed heat flux. For example, at $q = 26 \text{ kW/m}^2$, $T_{\text{sat}} = 15^\circ\text{C}$ and $\delta = 0.2 \text{ mm}$, the average bubble departure frequency for $G = 700 \text{ kg/m}^2\text{s}$ is about 14% higher than that for $G = 600 \text{ kg/m}^2\text{s}$ (Figure 4.25(b)). Then, the effects of the duct size on the saturated flow boiling average bubble departure frequency shown in Figure 4.26 manifests that the average bubble departure frequency is noticeably higher in the smaller duct. For instance, at $q = 26 \text{ kW/m}^2$, $T_{\text{sat}} = 15^\circ\text{C}$ and $G = 500 \text{ kg/m}^2\text{s}$ the average bubble departure frequency for $\delta = 0.5 \text{ mm}$ is about 23% higher than that for $\delta = 2.0 \text{ mm}$ (Figure 4.26(a)). Besides, at $G = 600 \text{ kg/m}^2\text{s}$ and at the same q , and T_{sat} the average bubble departure frequency for $\delta = 0.2 \text{ mm}$ is about 29% higher than that for $\delta = 1.0 \text{ mm}$ (Figure 4.26 (b)). Finally, the data given Figures 4.27 and 4.28 indicate that the average bubble departure frequency is significantly higher for a higher saturated temperature. As an example, at $q = 26 \text{ kW/m}^2$, $G = 600 \text{ kg/m}^2\text{s}$ and $\delta = 0.2 \text{ mm}$ the average bubble departure frequency for $T_{\text{sat}} = 15^\circ\text{C}$ is about 21% higher than that for $T_{\text{sat}} = 10^\circ\text{C}$ (Figure 4.28(b)).

The number density of the active nucleation sites for ONB affected by the three parameters for the saturated flow boiling of R-134a at the middle axial location ($z = 80 \text{ mm}$) in the annular duct are shown in Figures 4.29-4.33 by presenting the average active nucleation site density against the imposed heat flux for various G , δ and T_{sat} . The data clearly show the substantial increase of the active nucleation site density with the imposed heat flux for all cases examined here. First, the effects of the refrigerant mass flux on the

saturated flow boiling average active nucleation site density are shown in Figures 4.29 and 4.30. The results indicate that the average active nucleation site density is significantly higher for a smaller refrigerant mass flux especially at high imposed heat flux for the smaller ducts with $\delta = 0.5$ & 0.2 mm. For example, at $q = 30 \text{ kW/m}^2$, $T_{\text{sat}} = 15^\circ\text{C}$ and $\delta = 0.5$ mm, the average active nucleation site density for $G = 500 \text{ kg/m}^2\text{s}$ is about 22 % higher than that for $G = 600 \text{ kg/m}^2\text{s}$ (Figure 4.30(a)). Then, the effects of the duct size on the saturated flow boiling average active nucleation site density shown in Figure 4.31 manifest that the average active nucleation site density is significantly higher for a larger duct when the imposed heat flux exceeds 20 kW/m^2 . For instance, at $q = 30 \text{ kW/m}^2$, $T_{\text{sat}} = 15^\circ\text{C}$ and $G = 500 \text{ kg/m}^2\text{s}$ the average active nucleation site density for $\delta = 2.0$ mm is about 45% higher than that for $\delta = 0.5$ mm (Figure 4.31(a)). In addition, at $G = 600 \text{ kg/m}^2\text{s}$ and at the same q and T_{sat} the average active nucleation site density for $\delta = 1.0$ mm is about 50% higher than that for $\delta = 0.2$ mm (Figure 4.31 (b)). These results seem to introduce adverse effects for heat transfer for a reduction in the duct size. But a large portion of bubbles are merged to become large confined bubbles in the smaller duct for $\delta = 0.5$ & 0.2 mm at higher imposed heat flux. Hence in the small duct the boiling heat transfer is better. Finally, the data shown in Figures 4.32 and 4.33 suggest that the average active nucleation site density increases significantly with T_{sat} especially at high heat flux. As an example, at $q = 30 \text{ kW/m}^2$, $G = 500 \text{ kg/m}^2\text{s}$ and $\delta = 0.5$ mm the average active nucleation site density for $T_{\text{sat}} = 15^\circ\text{C}$ is about 22% higher than that for $T_{\text{sat}} = 10^\circ\text{C}$ (Figure 4.33(a)).

In the small ducts with $\delta = 0.5$ & 0.2 mm subject to a high imposed heat flux, the slug flow prevails and the flow is dominated by the big bubbles. The mean axial speeds of these bubbles are measured for the saturated flow boiling of R-134a at the middle axial location ($z = 80$ mm) in the annular duct. The data are shown in Figures 4.34-4.36. First, the effects of the refrigerant mass flux on the average speed of the bubbles shown in Figure 4.34 indicate that the average bubble speed is substantially larger for higher refrigerant mass flux and higher heat flux. For example, at $q = 40 \text{ kW/m}^2$, $T_{\text{sat}} = 15^\circ\text{C}$ and $\delta = 0.5$ mm, the average bubble speed for $G = 600 \text{ kg/m}^2\text{s}$ is about 30% larger than that for $G = 500 \text{ kg/m}^2\text{s}$ (Figure 4.34(a)). For instance, at $T_{\text{sat}} = 15^\circ\text{C}$, $\delta = 0.2$ mm and $G = 600 \text{ kg/m}^2\text{s}$, the average bubble speed for $q = 28 \text{ kW/m}^2$ is about 54% higher than that for $q = 46 \text{ kW/m}^2$ (Figure 4.34(b)). Then, the data given in Figure 4.35 suggests that the average bubble speed is

substantially larger when δ is reduced from 0.5 mm to 0.2 mm. This is more pronounced at high heat flux. For example, at $q = 40 \text{ kW/m}^2$, $T_{\text{sat}} = 15^\circ\text{C}$ and $G = 600 \text{ kg/m}^2\text{s}$, the average bubble speed for $\delta = 0.2 \text{ mm}$ is about 29% higher than that for $\delta = 0.5 \text{ mm}$ (Figure 4.35). Finally, the results in Figure 4.36 indicate that the average bubble speed is significantly higher for a higher saturated temperature especially at a high heat flux. For instance, at $q = 40 \text{ kW/m}^2$, $G = 500 \text{ kg/m}^2\text{s}$ and $\delta = 0.5 \text{ mm}$ the average bubble speed for $T_{\text{sat}} = 10^\circ\text{C}$ is about 25% lower than that for $T_{\text{sat}} = 15^\circ\text{C}$ (Figure 4.36(a)).

4.5 Correlation Equations

According to flow boiling mechanisms, the heat transfer in the flow boiling in the bubbly flow regime can be roughly considered as a combination of single-phase convection heat transfer q_c and pool boiling heat transfer q_b . Thus the total heat flux input to the boiling flow q_t can be expressed as

$$q_t = q_b + q_c \quad (4.5)$$

Here q_b and q_c can be calculated from the relations

$$q_b = \rho_g V_g f N_{ac} i_{fg} \quad (4.6)$$

and

$$q_c = E h_{1\phi} \Delta T_{\text{sat}} \quad (4.7)$$

Note that q_b expressed in Equation(4.6) in fact represents the latent heat carried away from the heating surface during the departure of bubbles from the surface. The single-phase forced convection heat transfer coefficient is estimated from the correlation for the enhanced factor E and the Nusselt number $Nu_{1\phi}$ as

$$h_{1\phi} = Nu_{1\phi} k_l / D_h \quad (4.8)$$

with

$$E = 0.97 N_{\text{conf}}^{0.28} Fr_l^{0.01} (1 - 330 Bo)^{5.51} \quad (4.9)$$

Note that $Nu_{1\phi}$ is estimated from the Gnielinski and Choi correlations [48, 51],

$$Nu_{1\phi} = \frac{(f_f / 2)(Re_l - 1000) Pr_l}{1.07 + 12.7 \sqrt{f_f / 2} (Pr_l^{2/3} - 1)} \quad , \text{ for } Re_l \geq 2,300$$

and

$$Nu_{1\phi} = 0.000972 Re_l^{1.17} Pr_l^{1/3} \quad , \text{ for } Re_l < 2,300 \quad (4.10)$$

Here the friction factor f_f is evaluated from the Gnielinski correlation

$$f_f = (1.58 \ln Re_l - 3.28)^{-2} \quad (4.11)$$

Moreover, the Reynolds number of the liquid flow is defined as

$$Re_l = GD_h(1-x) / \mu_l \quad (4.12)$$

In the above equations ρ_g is the vapor density, V_g is the mean vapor volume of a departing

bubble which is equal to $\frac{4\pi}{3} \left(\frac{d_p}{2} \right)^3$, f is bubble departure frequency, N_{ac} is the active

nucleation site density, i_{fg} is the enthalpy of vaporization. Because the range of experimental Re_l is between 800 to 6000, we use Gnielinski correlation for $Re_l > 2,300$ but Choi correlation for $Re_l < 2,300$ to represent single phase convection heat transfer term($h_{1\phi}$). It is difficult to distinguish the individual bubbles at a higher imposed heat flux.

Hence the above correlations do not apply to the data for $q > 40 \text{ kW/m}^2$ at $\delta = 2.0 \text{ mm}$, $q > 35 \text{ kW/m}^2$ at $\delta = 1.0 \text{ mm}$, $q > 30 \text{ kW/m}^2$ at $\delta = 0.5 \text{ mm}$, and $q > 25 \text{ kW/m}^2$ at $\delta = 0.2 \text{ mm}$.

To enable the usage of the above correlation for the flow boiling heat transfer in the bubbly flow regime, the mean bubble size and departure frequency and the active nucleation density on the heating surface need to be correlated in advance. The average bubble departure diameter in the saturated flow boiling of R-134a in the narrow annular duct estimated from the present flow visualization can be correlated as

$$\frac{d_p}{\sqrt{\sigma / (g \cdot \Delta \rho)}} = 0.605 \left(\frac{\rho_l}{\rho_g} \right)^{0.5} Re_l^{-0.2} \cdot Bo^{0.195} \cdot N_{conf}^{-0.19} \quad (4.13)$$

Figure 4.37 shows that almost all the present experimental data for d_p fall within $\pm 25\%$ of the above correlation and the mean absolute error is 9.5%. Besides, an empirical equation

is proposed for the product of the mean bubble departure diameter and departure frequency as

$$\frac{f \cdot d_p}{\mu_l / (\rho_l D_h)} = 1.61 Re_l^{1.33} \cdot Pr_l^2 \cdot Bo^{0.725} \cdot N_{conf}^{0.59} \quad (4.14)$$

Note that more than 85 % of the experimental data for $f \cdot d_p$ collected in this study can be correlated within $\pm 30\%$ by Equation (4.14) and the mean absolute error is 13.1%(Figure 4.38). Finally, we propose an empirical correlation for the average active nucleation site density in the saturated flow boiling of R-134a as

$$N_{ac} d_p^2 = -0.009 + 1095.25 Bo^{1.23} Re_l^{0.049} N_{conf}^{0.058} \quad (4.15)$$

Figure 4.39 shows that the present experimental data fall within $\pm 30\%$ of the above correlation and the mean absolute error is 29.6%.

When the correlations for d_p , f , and N_{ac} given in Equations (4.13)-(4.15) are combined with Equations (4.5)-(4.12) for q_t , the heat transfer data measured in the present study fall within $\pm 30\%$ of the correlation proposed here with the mean absolute error of 29.17%(Figure 4.40).

For the smaller ducts with $\delta = 0.5$ & 0.2 mm the slug flow (confined bubbles) prevails at a high imposed heat flux, the associated flow boiling heat transfer data from the present measurement are correlated by modifying the correlation of Cornwell and Kew [29] for the confined bubble regime as

$$Nu / Nu_{lo} = 2.5 Bo^{0.02} N_{conf}^{0.15} \quad (4.16)$$

with

$$Nu_{lo} = 0.023 Re_l^{0.8} Pr_l^{0.4} \quad (4.17)$$

Figure 4.41 shows that the present experimental data fall within $\pm 30\%$ of the above correlation and the mean absolute error is 14%.

4.6 Concluding Remarks

The experimental heat transfer data for the saturated flow boiling of R-134a in the narrow annular duct have been presented here. Meanwhile, the bubble behavior in the boiling flow is examined. The effects of the imposed heat flux, refrigerant mass flux, system pressure and duct size on the saturated flow boiling heat transfer coefficient and associated bubble characteristics have been investigated in detail. In addition, empirical equations to correlate the measured boiling heat transfer data, bubble departure diameter, bubble departure frequency and active nucleation site density are proposed. The major results obtained here can be summarized in the following.

- (1). The temperature undershoot at ONB are insignificant for the saturated flow boiling of R-134a in the horizontal narrow annular duct.
- (2). The saturated boiling heat transfer coefficients increase with a decrease in the gap size. Besides, raising the imposed heat flux can cause a significant increase in the boiling heat transfer coefficient. However, the effects of the refrigerant mass flux and saturated temperature on the boiling heat transfer coefficient are small.
- (3). Correlation equations were provided for the boiling heat transfer coefficient, mean bubble departure diameter, mean bubble departure frequency and mean active nucleation site density in the saturated flow boiling.
- (4). The results from the flow visualization show that the mean diameter of the bubbles departing from the heating surface decreases slightly with increasing refrigerant mass flux. Besides, at a high imposed heat flux many bubbles generated from the cavities in the heating surface tend to merge together to form big bubbles. The bubble departure frequency increases with the increasing refrigerant mass flux and saturated temperature and with the decreasing duct size. The active nucleation site density is much lower at a higher refrigerant mass flux and lower saturation temperature and smaller gap. Besides, in the smaller ducts with $\delta = 0.5$ & 0.2 mm the slug flow prevails at a high imposed heat flux. Many big bubbles appear in the duct.

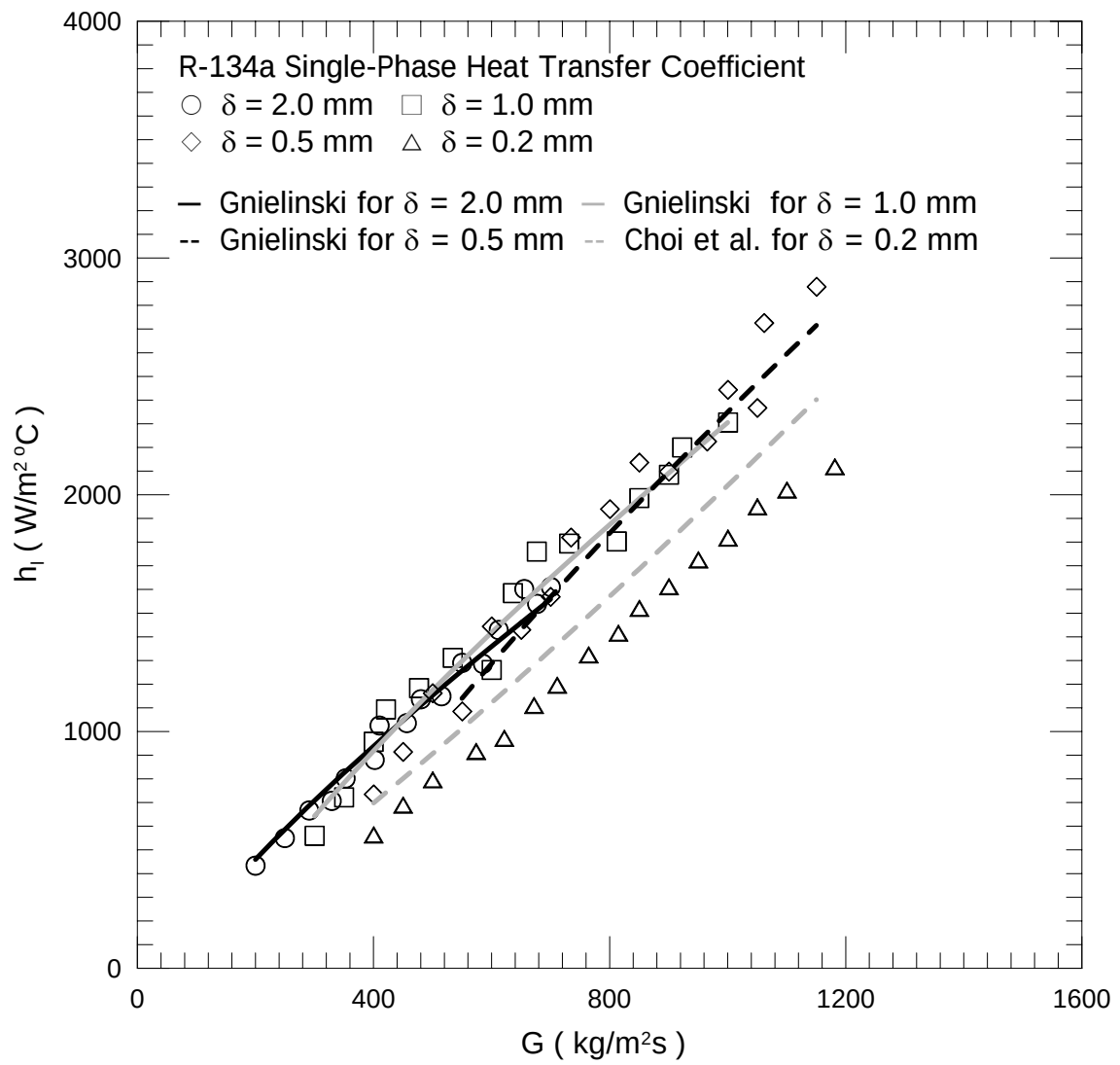


Fig. 4.1 Comparison of the present single-phase liquid convection heat transfer data with the correlation of Gnielinski and Choi et al.

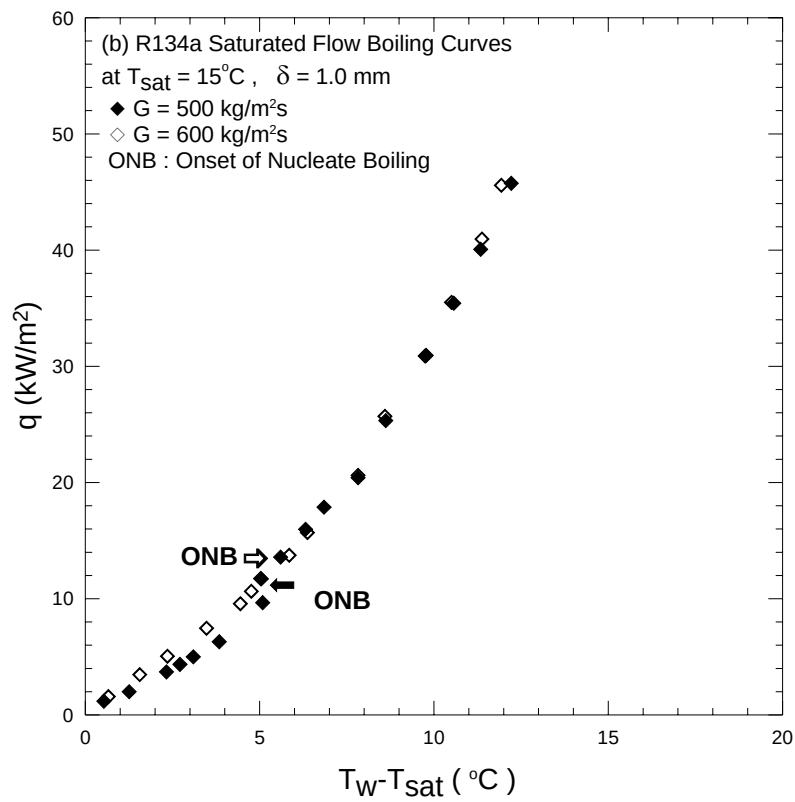
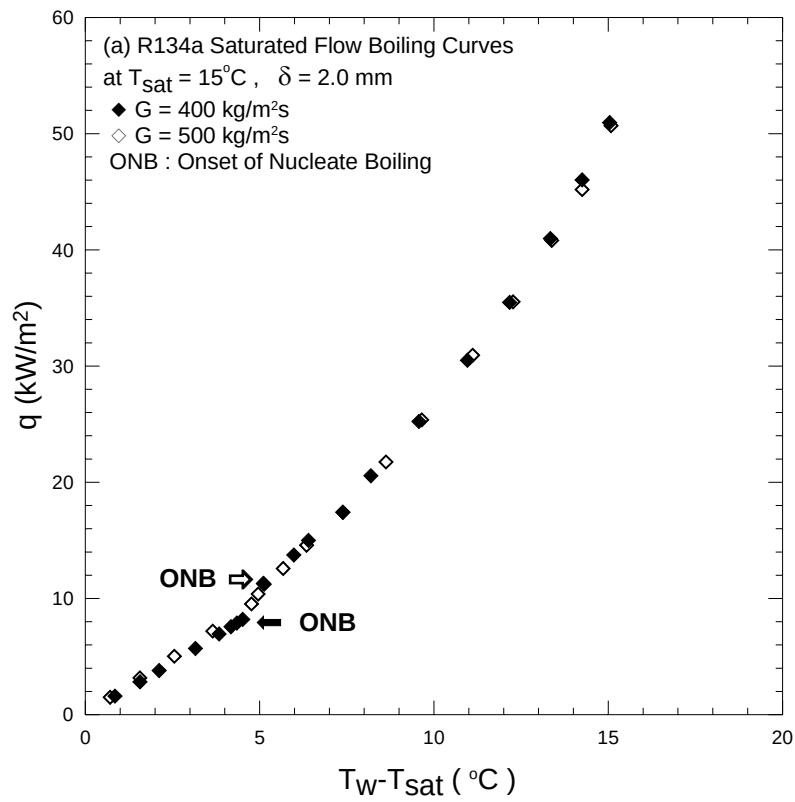


Fig. 4.2 Saturated flow boiling curves for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 2.0\text{ mm}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 1.0\text{ mm}$.

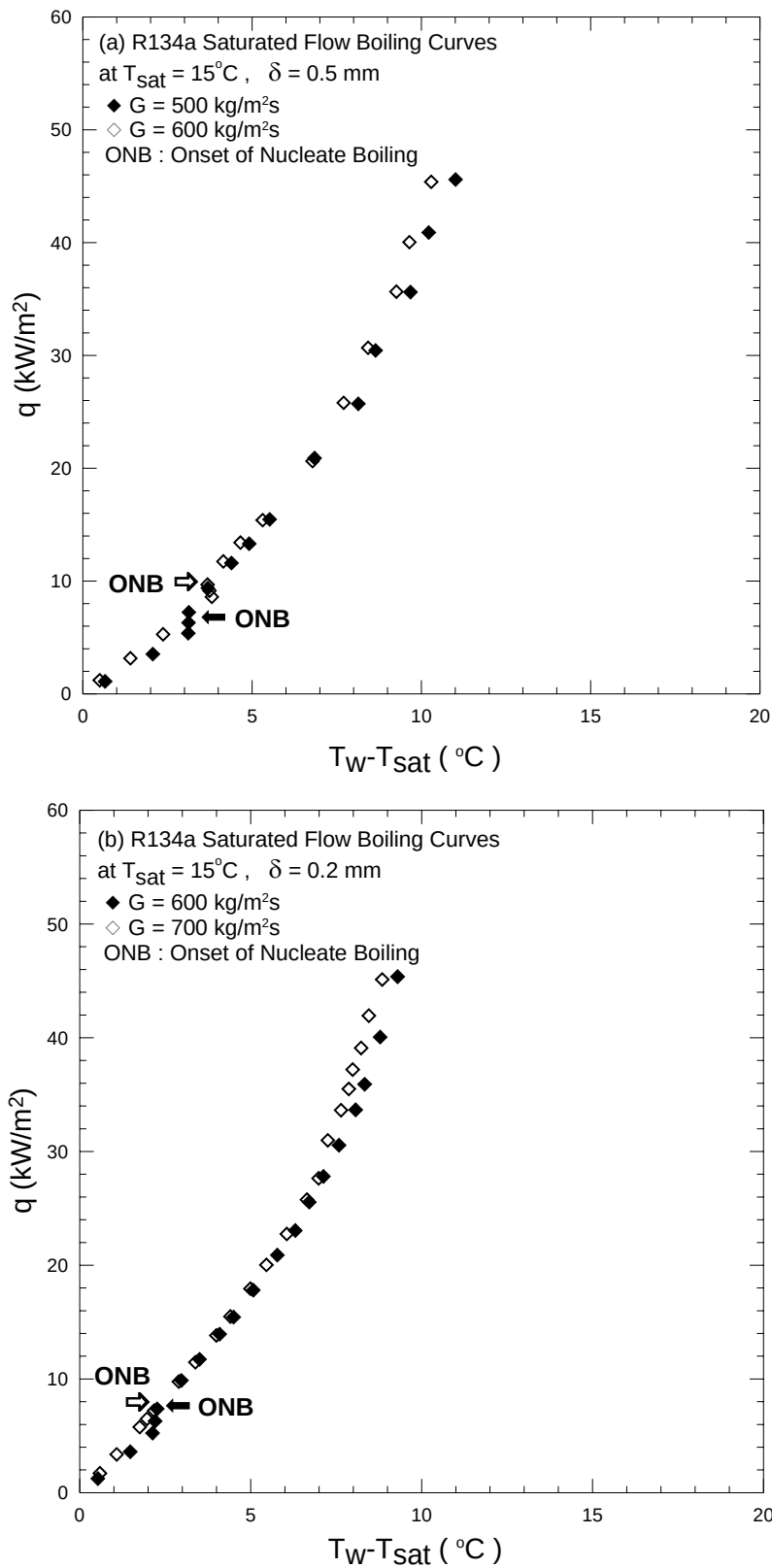


Fig. 4.3 Saturated flow boiling curves for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 0.5 \text{ mm}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 0.2 \text{ mm}$.

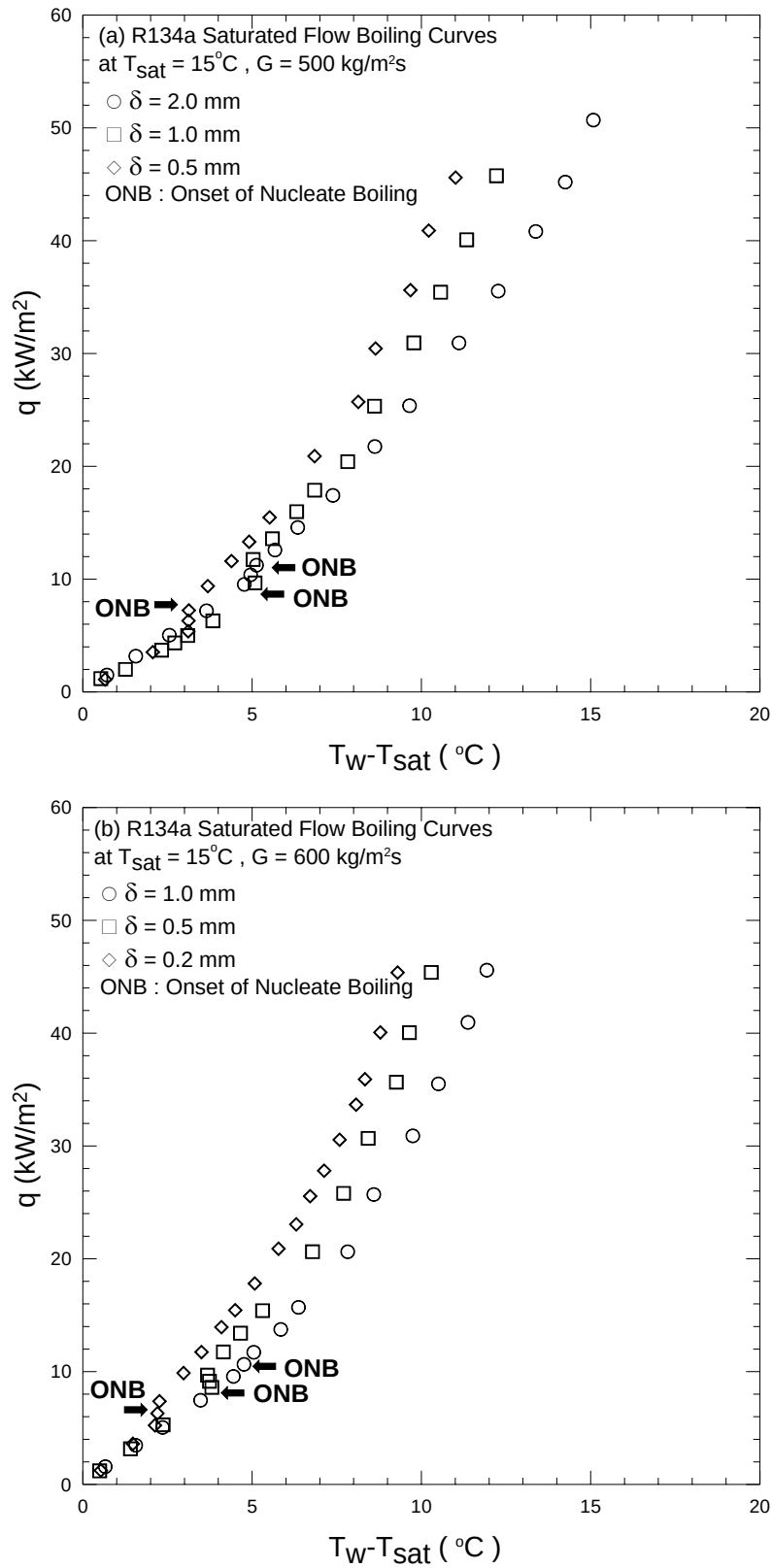


Fig. 4.4 Saturated flow boiling curves for various gap sizes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 500 \text{ kg/m}^2\text{s}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$.

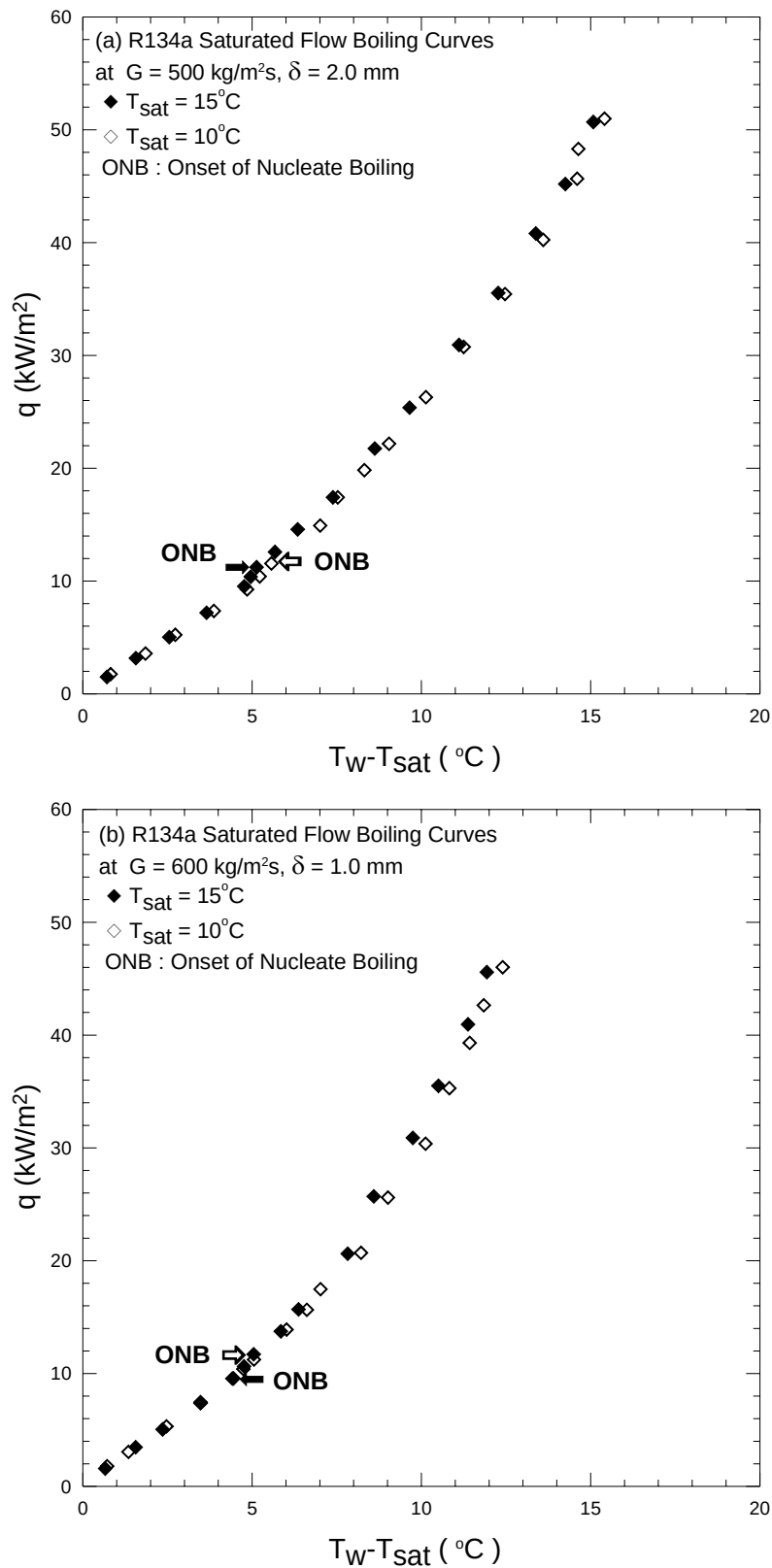


Fig. 4.5 Saturated flow boiling curves for various refrigerant saturated temperatures at
 (a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 2.0 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 1.0 \text{ mm}$.

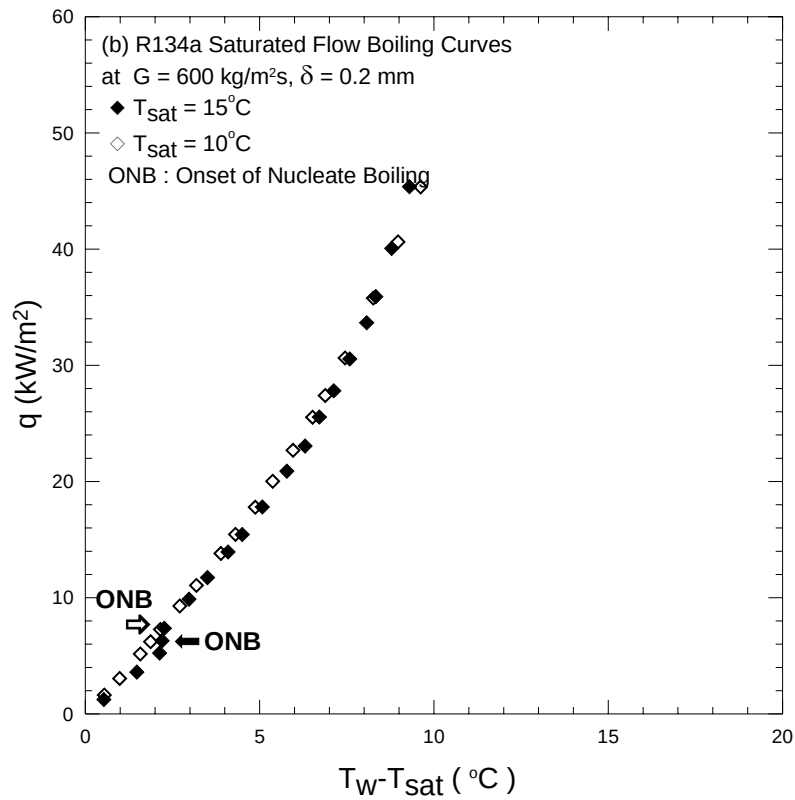
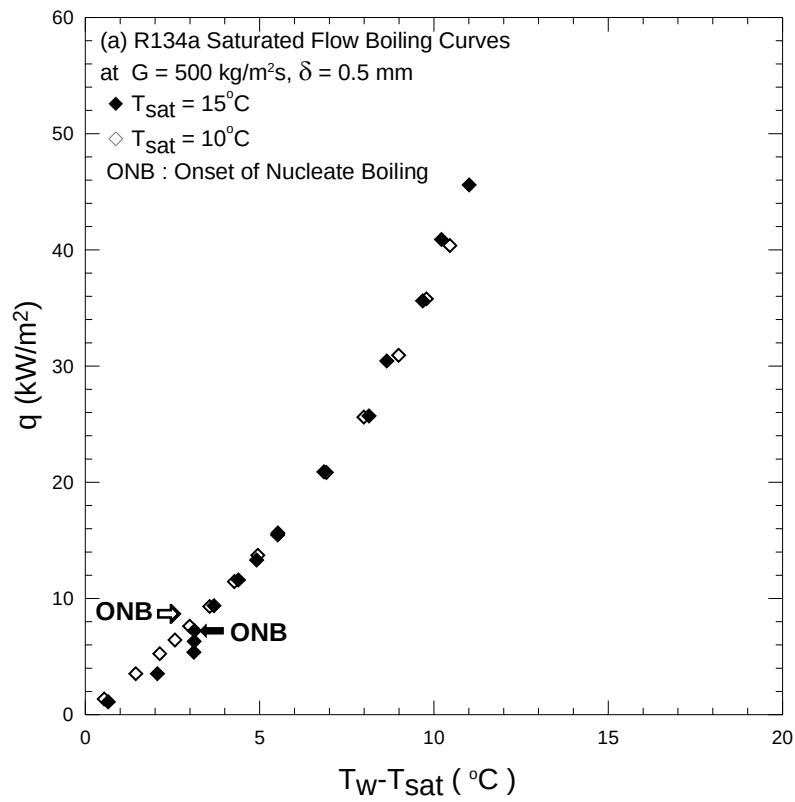


Fig. 4.6 Saturated flow boiling curves for various refrigerant saturated temperatures at
(a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 0.5 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 0.2 \text{ mm}$.

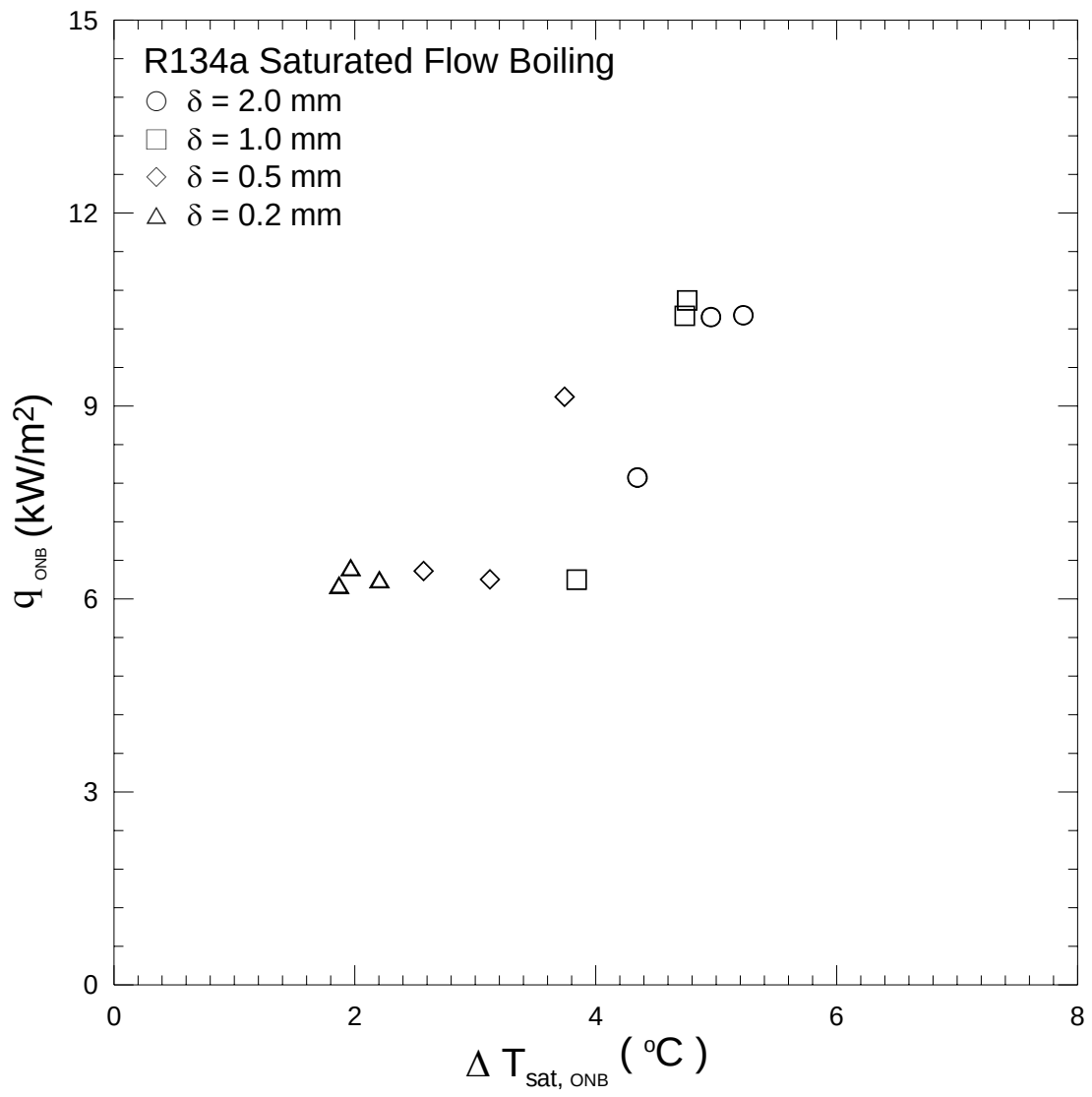


Fig. 4.7 Relation between imposed heat flux and wall superheat at the incipient boiling.

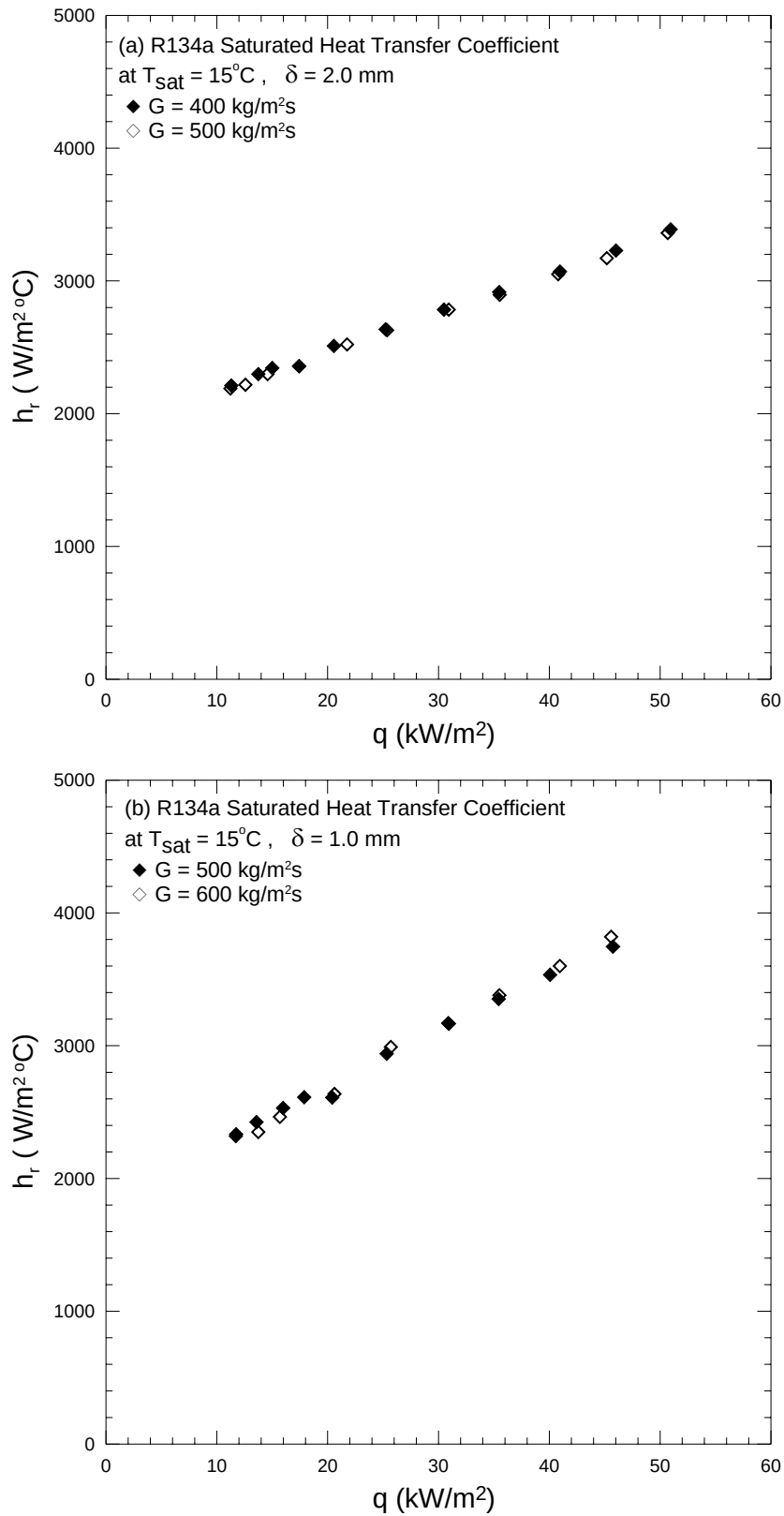


Fig. 4.8 Saturated flow boiling heat transfer coefficient for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 2.0 \text{ mm}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 1.0 \text{ mm}$.

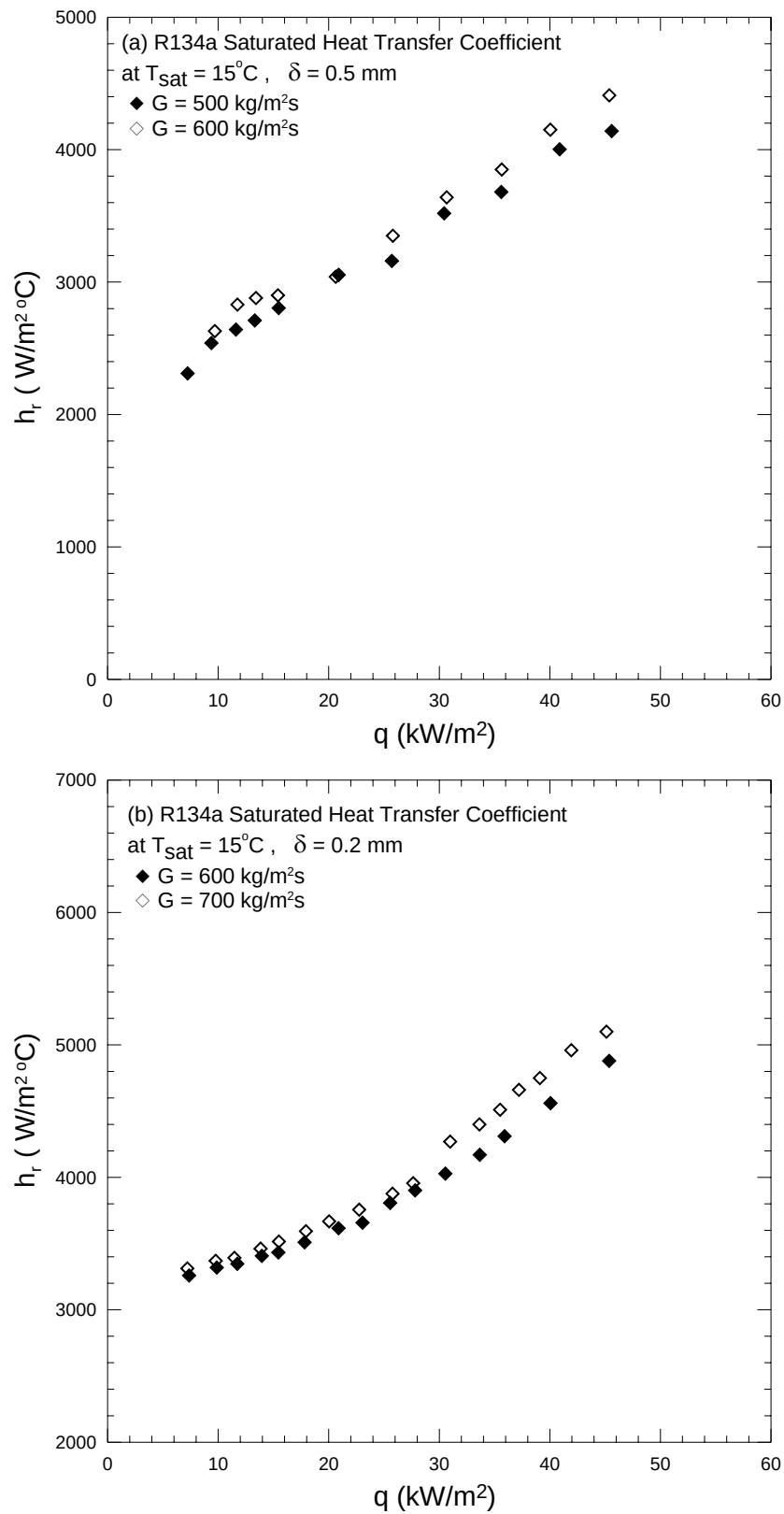


Fig. 4.9 Saturated flow boiling heat transfer coefficient for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 0.5 \text{ mm}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 0.2 \text{ mm}$.

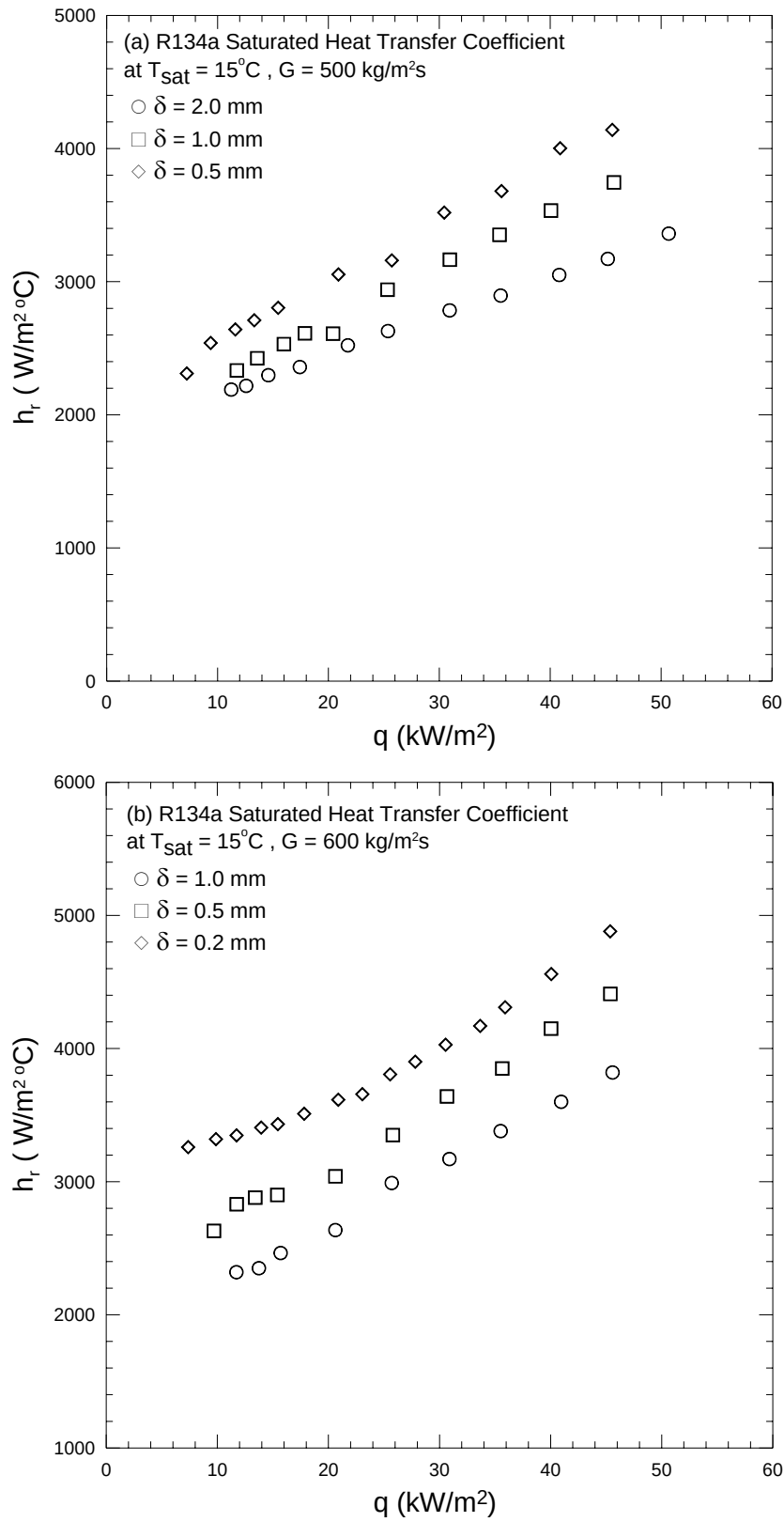


Fig. 4.10 Saturated flow boiling heat transfer coefficient for various gap sizes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 500 \text{ kg/m}^2\text{s}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$.

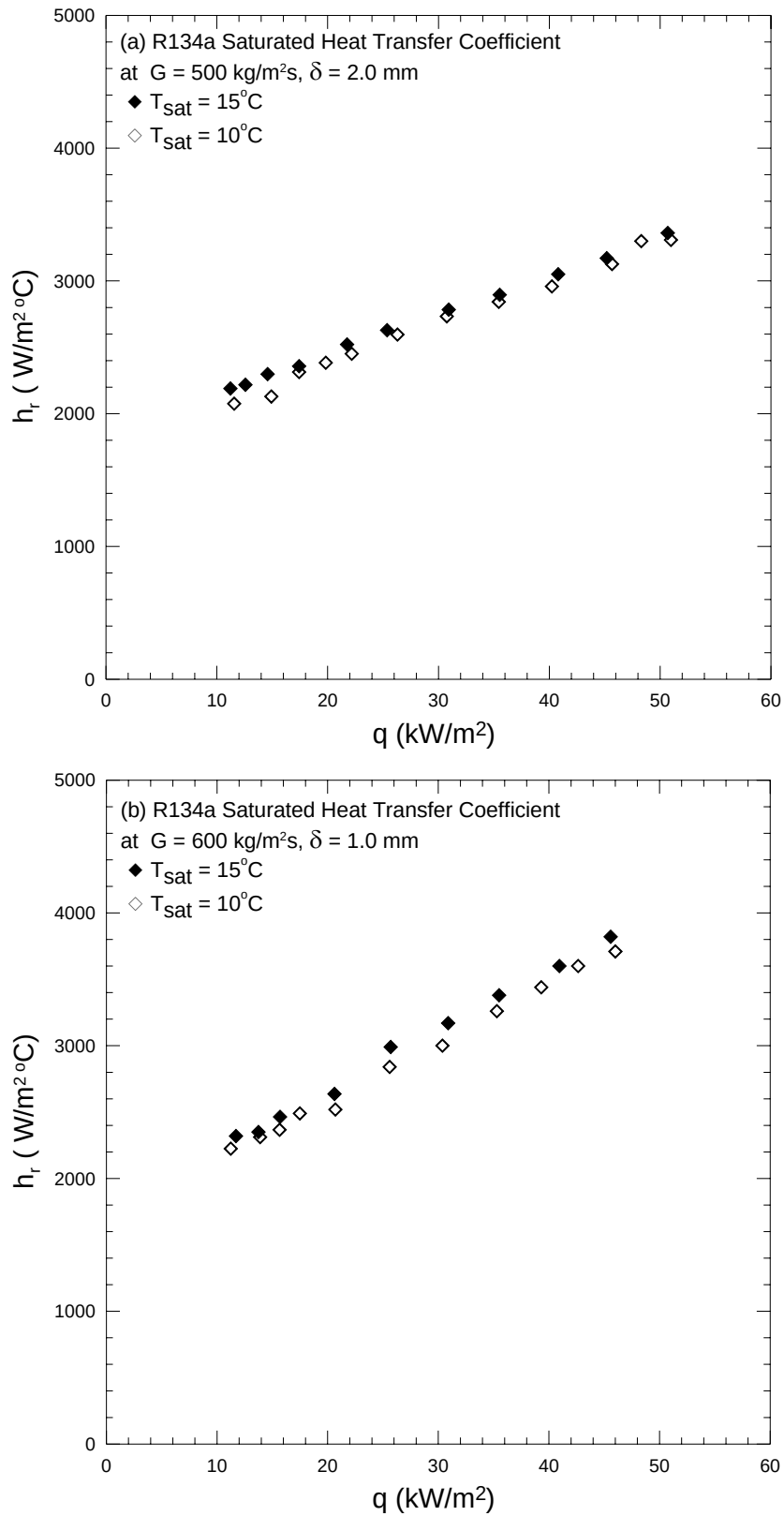


Fig. 4.11 Saturated flow boiling heat transfer coefficient for various refrigerant saturated temperatures at (a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 2.0 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 1.0 \text{ mm}$.

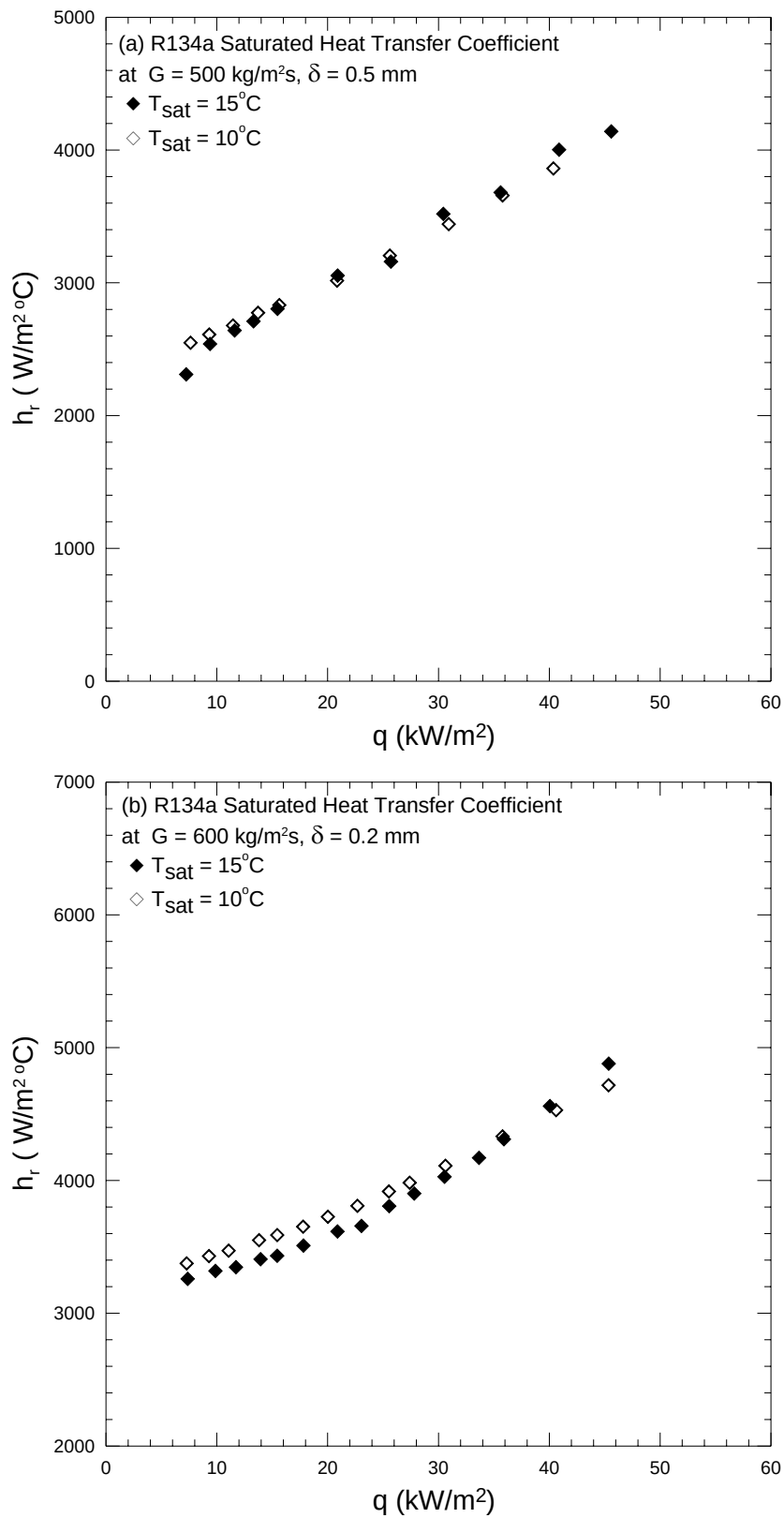


Fig. 4.12 Saturated flow boiling heat transfer coefficient for various refrigerant saturated temperatures at (a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 0.5 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 0.2 \text{ mm}$.

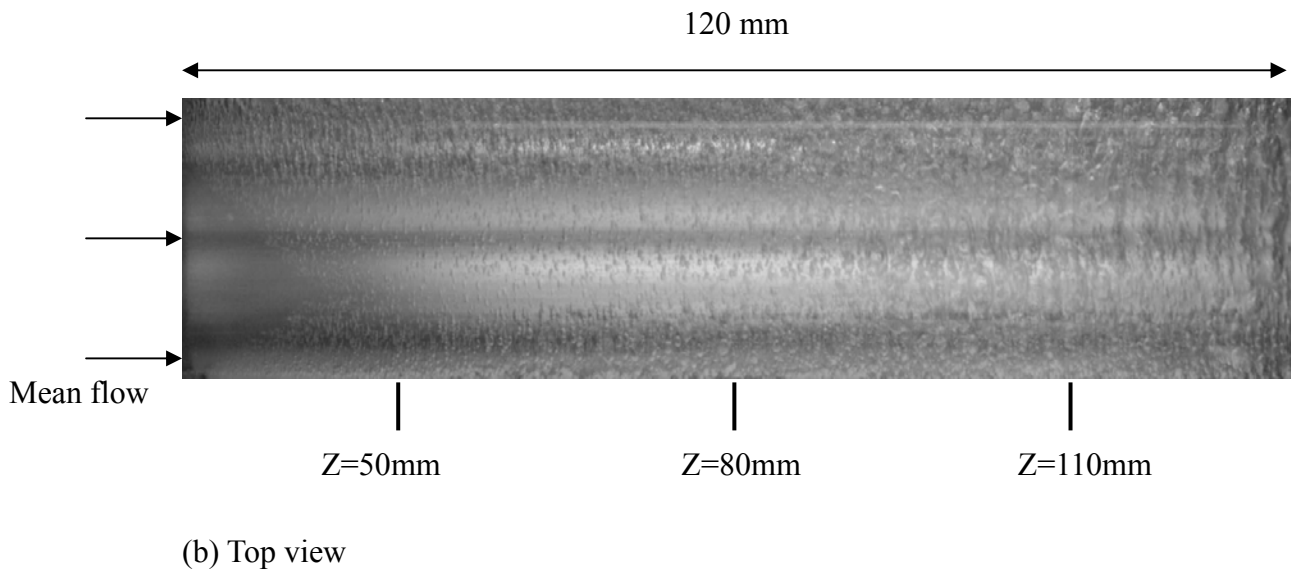
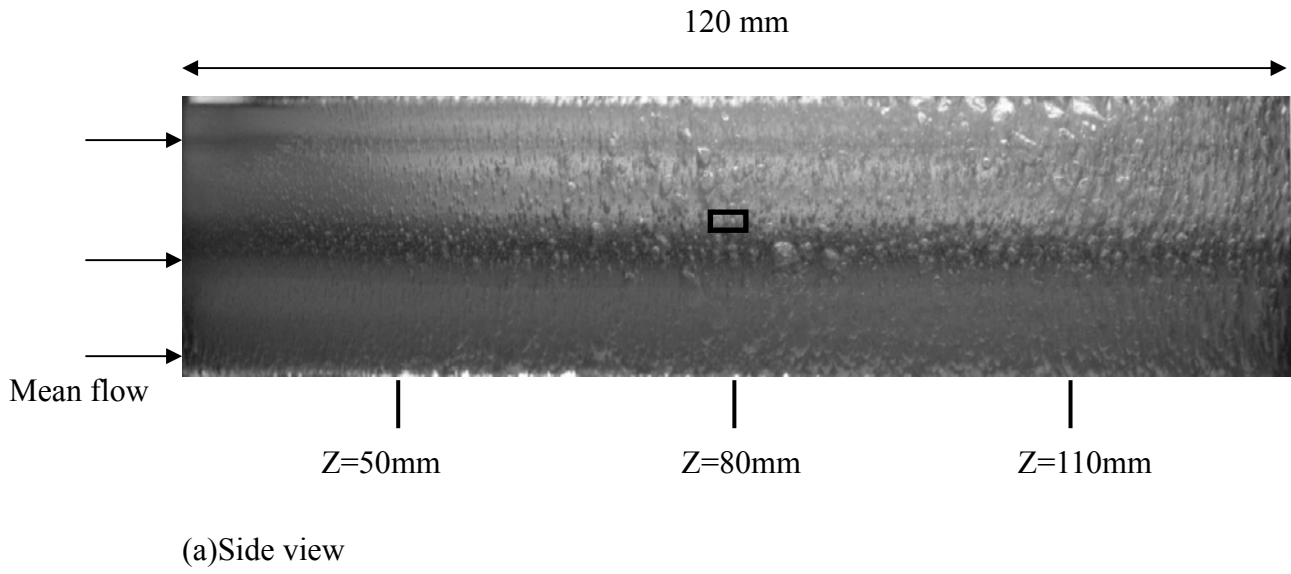
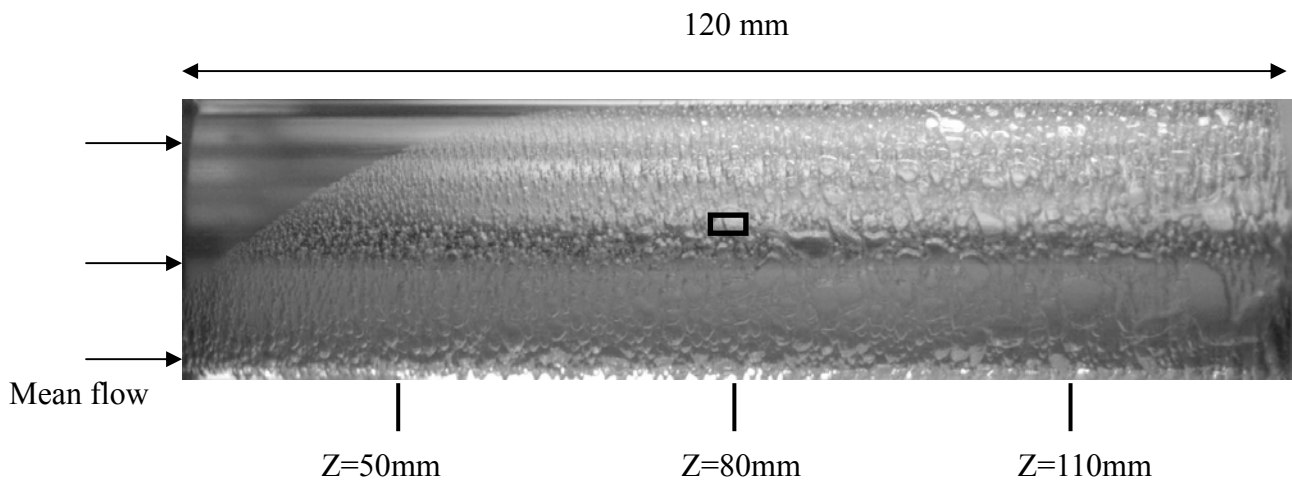
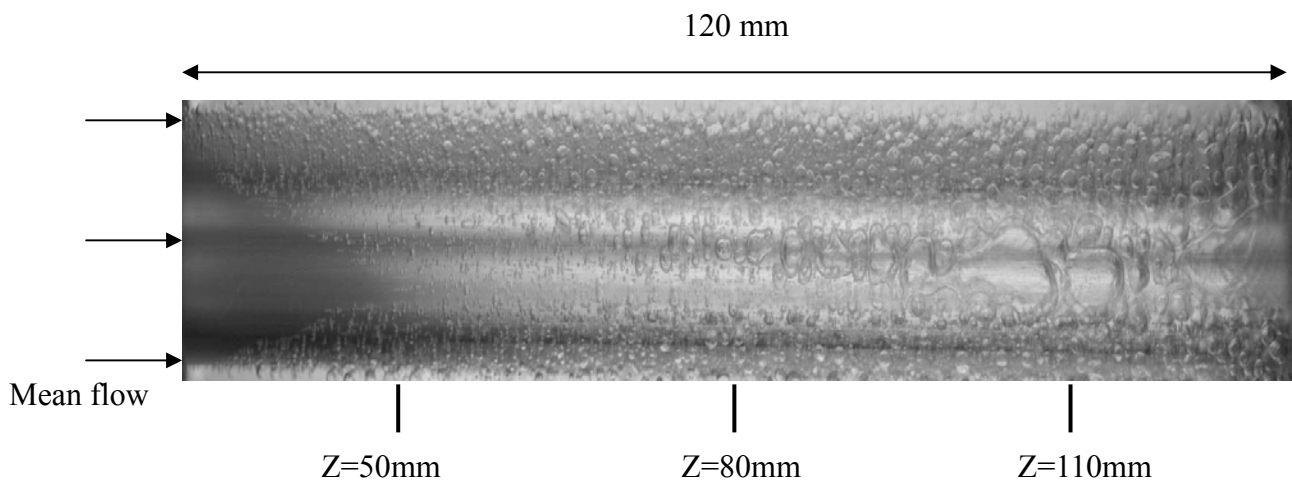


Fig. 4.13 Photos of boiling flow in the saturated flow boiling of R-134a in the entire duct at $G= 500 \text{ kg/m}^2\text{s}$, $T_{\text{sat}}= 15^\circ\text{C}$, $\delta= 2.0 \text{ mm}$ and $q= 30 \text{ kW/m}^2$ from (a) side view and (b) top view.

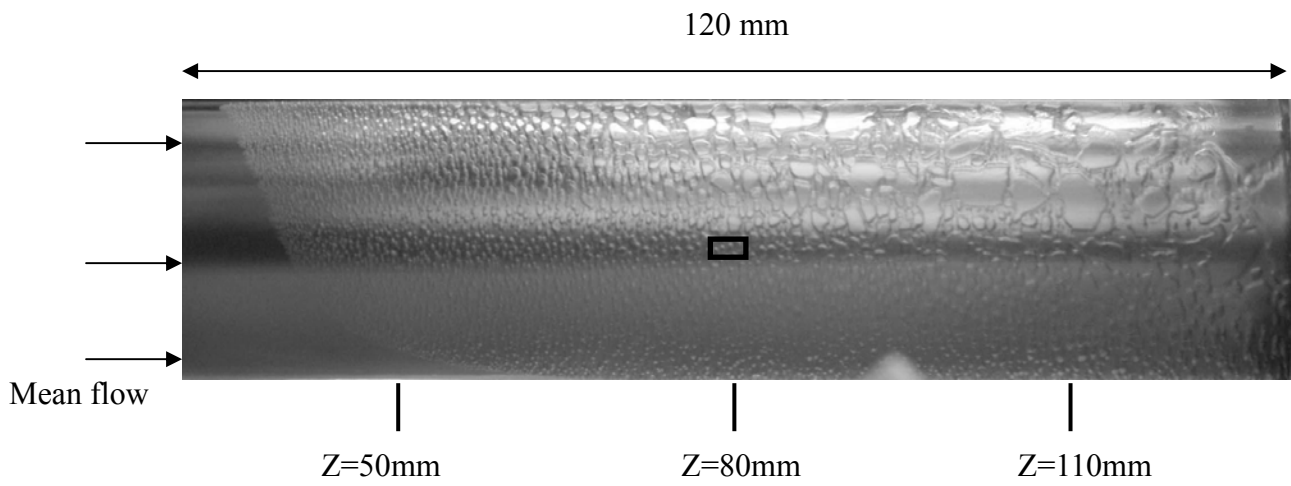


(a) Side view

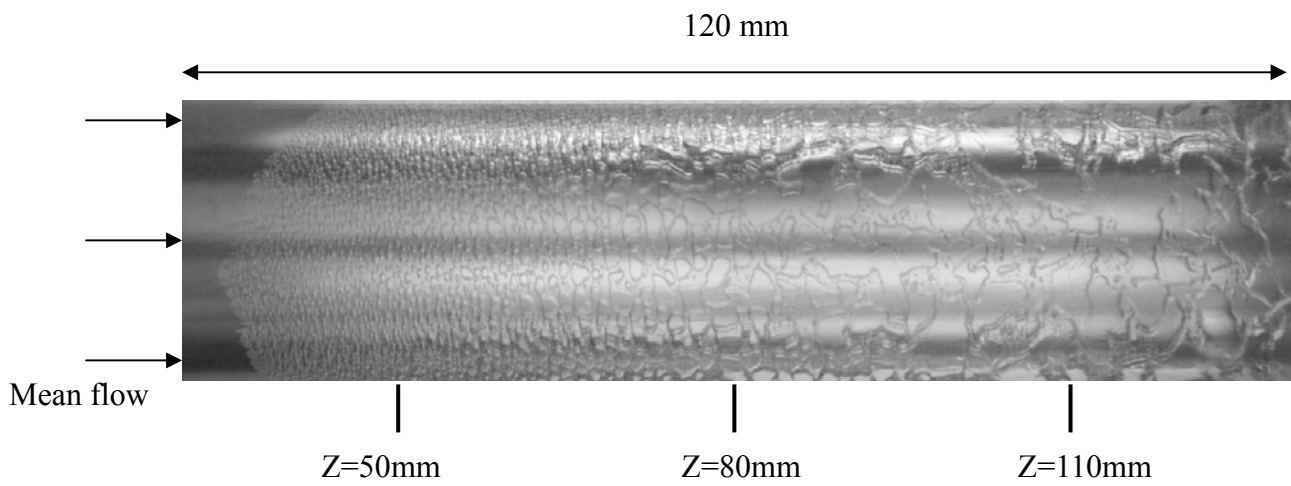


(b) Top view

Fig. 4.14 Photos of boiling flow in the saturated flow boiling of R-134a in the entire duct at $G= 600 \text{ kg/m}^2\text{s}$, $T_{\text{sat}}= 15^\circ\text{C}$, $\delta= 1.0 \text{ mm}$ and $q= 30 \text{ kW/m}^2$ from (a) side view and (b) top view.



(a) Side view



(b) Top view

Fig. 4.15 Photos of boiling flow in the saturated flow boiling of R-134a in the entire duct at $G= 600 \text{ kg/m}^2\text{s}$, $T_{\text{sat}}= 15^\circ\text{C}$, $\delta= 0.5 \text{ mm}$ and $q= 30 \text{ kW/m}^2$ from (a) side view and (b) top view.

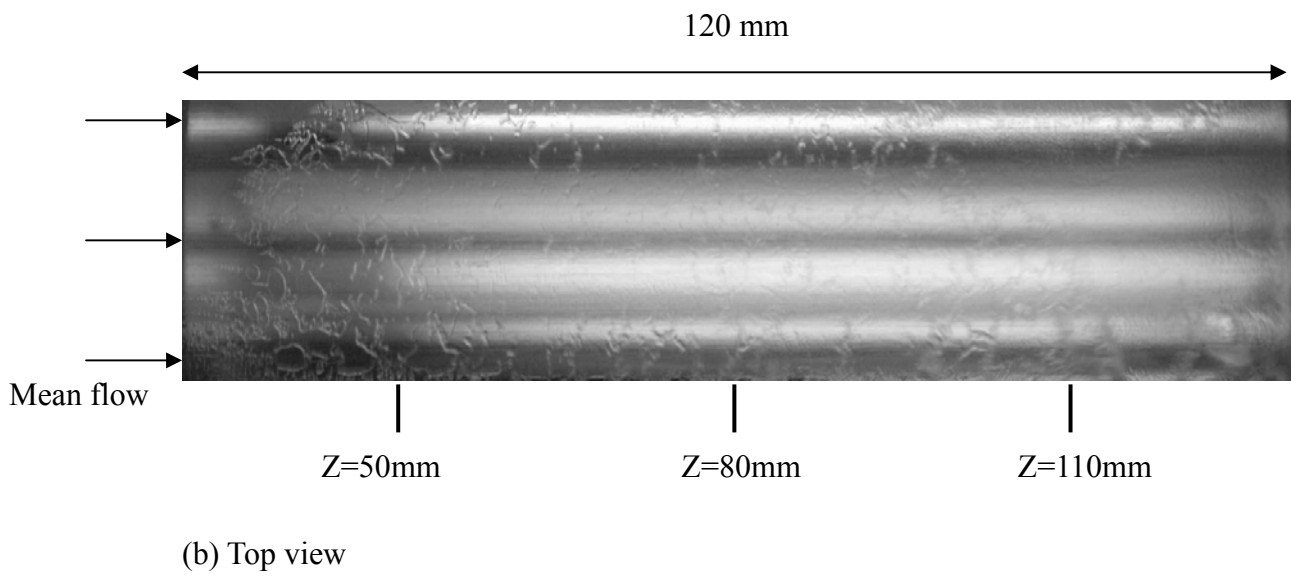
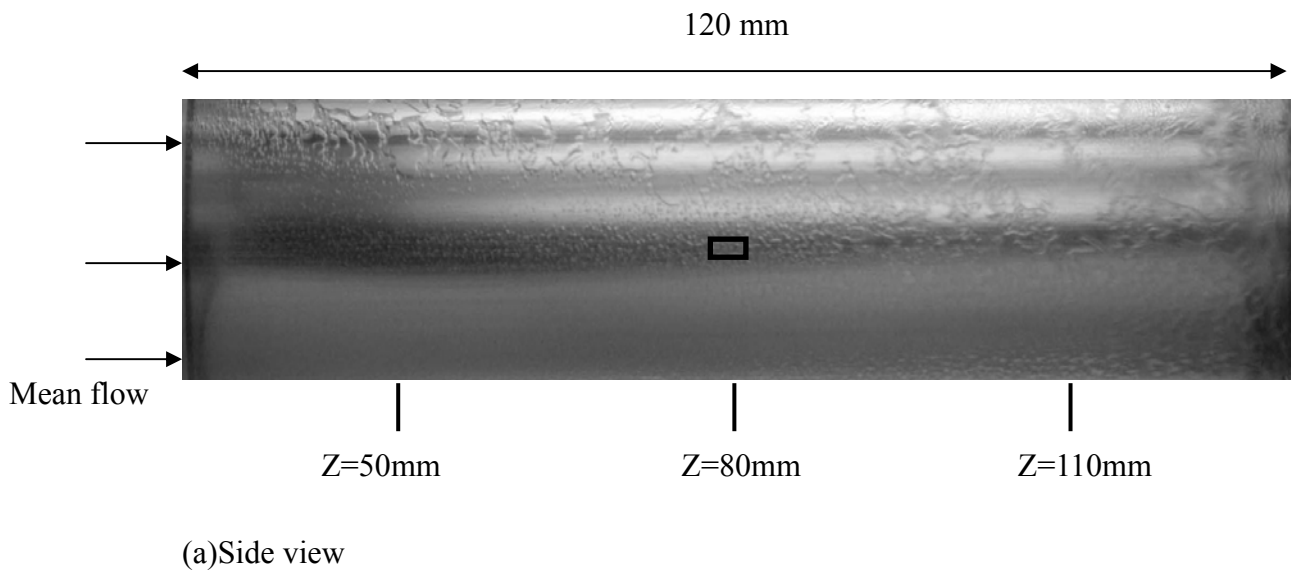


Fig. 4.16 Photos of boiling flow in the saturated flow boiling of R-134a in the entire duct at $G= 600 \text{ kg/m}^2\text{s}$, $T_{\text{sat}}= 15^\circ\text{C}$, $\delta= 0.2 \text{ mm}$ and $q=30 \text{ kW/m}^2$ from (a) side view and (b) top view.

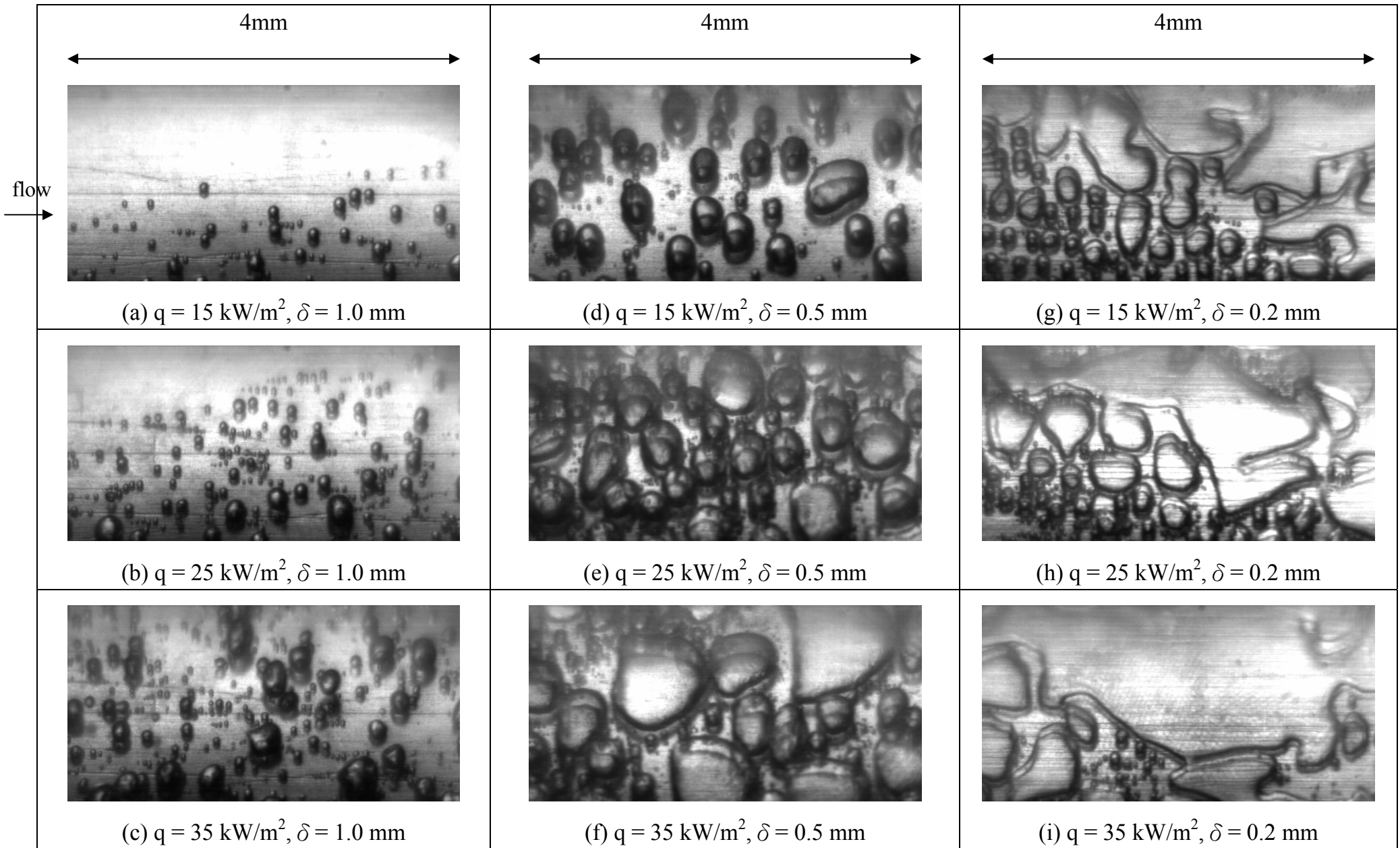


Fig. 4.17 Photos of bubbles in the saturated flow boiling of R-134a in a small region around middle axial location at $T_{\text{sat}} = 15^\circ\text{C}$ and $G = 600\text{-kg/m}^2\text{s}$ for various imposed heat flux and gap sizes.

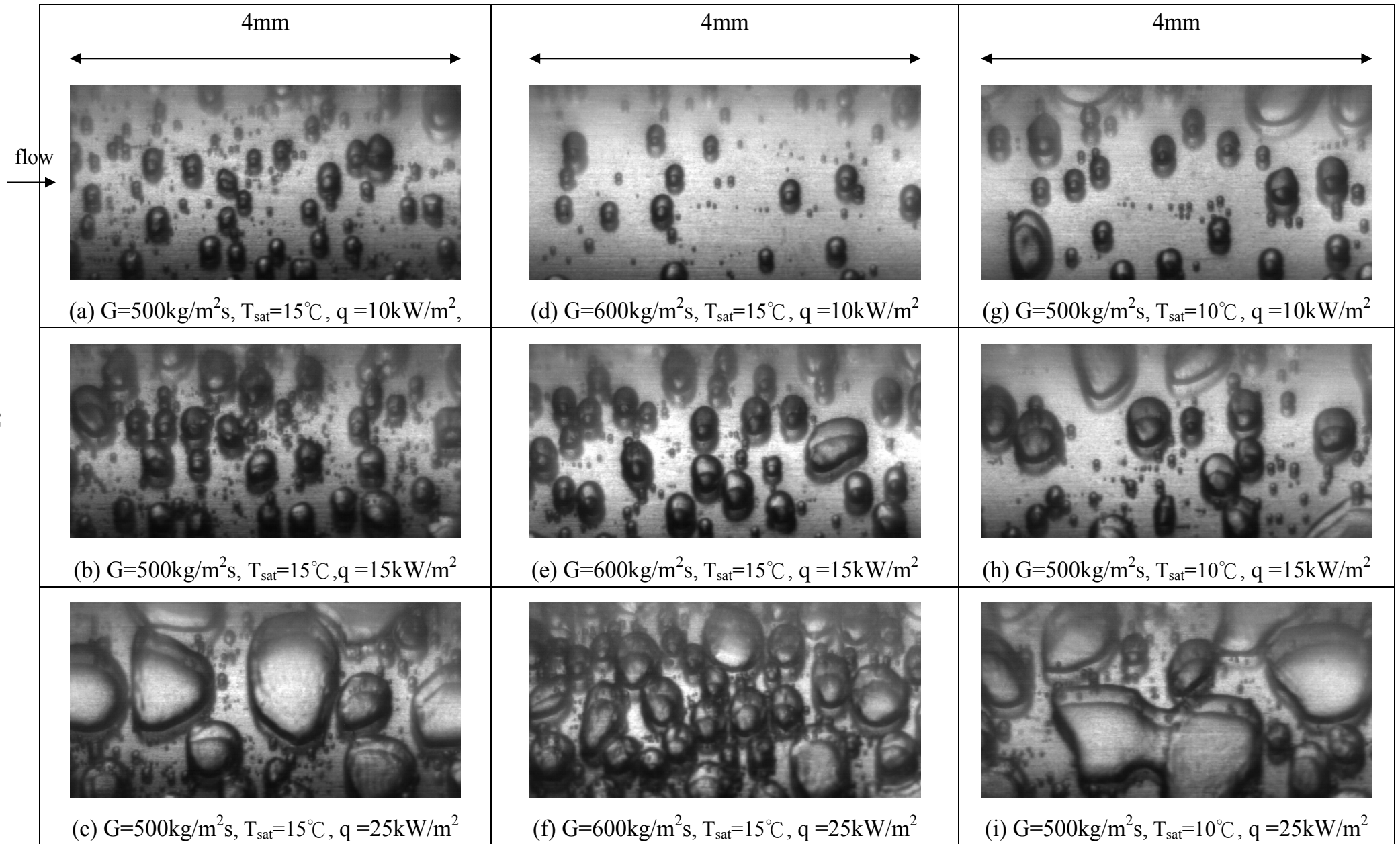


Fig. 4.18 Photos of bubbles in the saturated flow boiling of R-134a in a small region around middle axial location at $\delta=0.5\text{ mm}$ for various imposed heat flux, mass fluxes and refrigerant saturated temperatures.

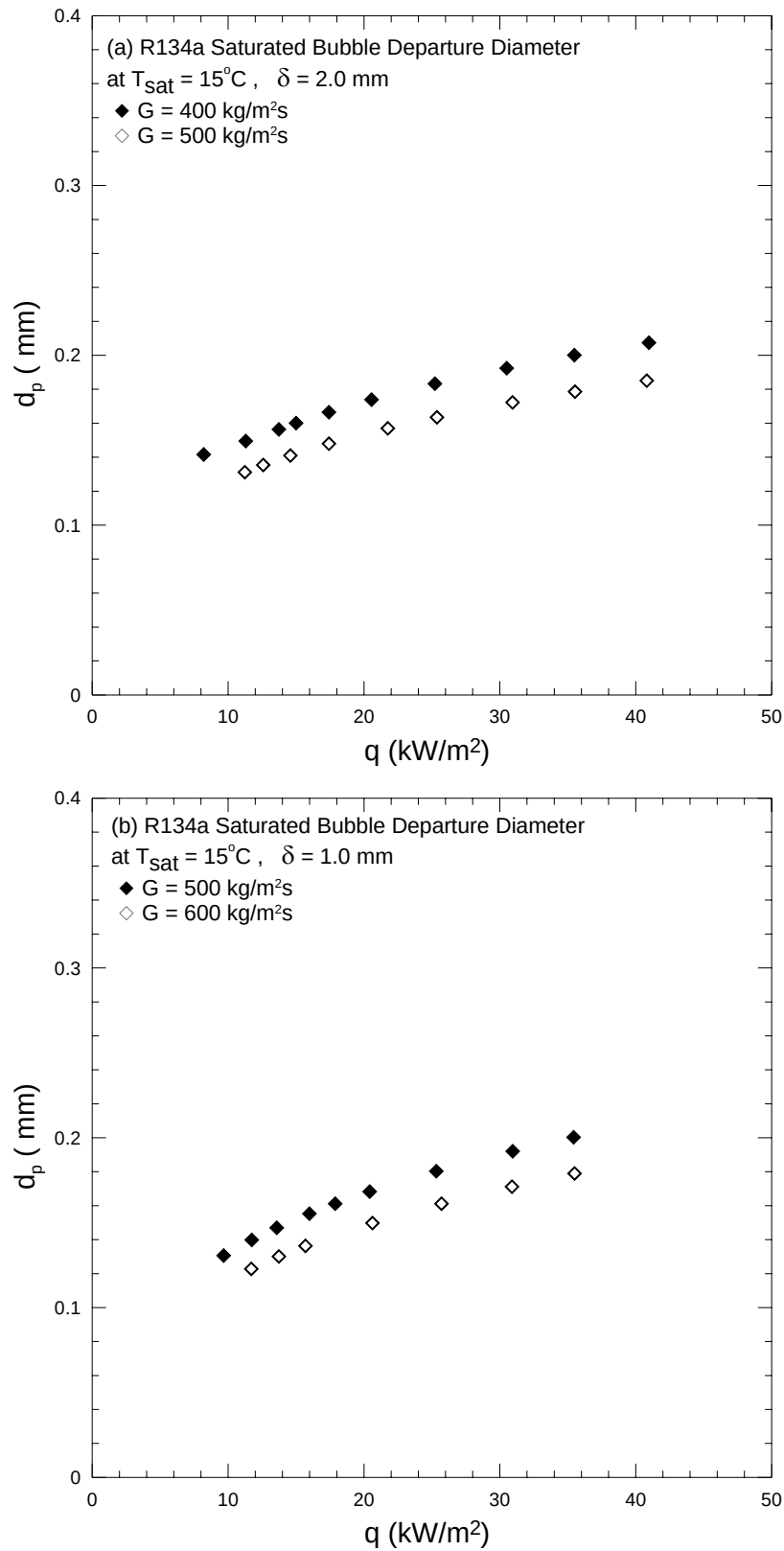


Fig. 4.19 Saturated flow boiling mean bubble departure diameter for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 2.0 \text{ mm}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 1.0 \text{ mm}$.

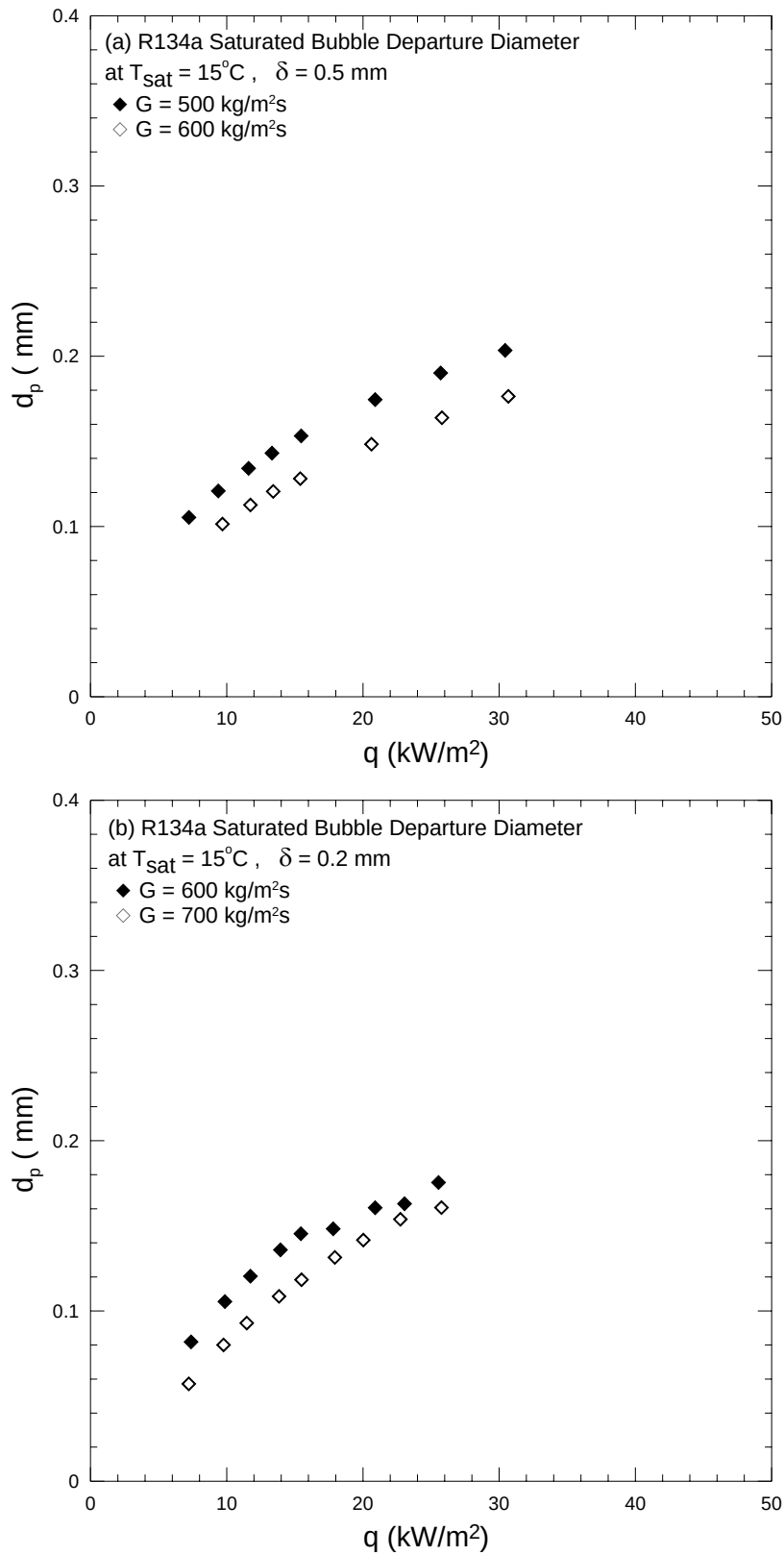


Fig. 4.20 Saturated flow boiling mean bubble departure diameter for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 0.5 \text{ mm}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 0.2 \text{ mm}$.

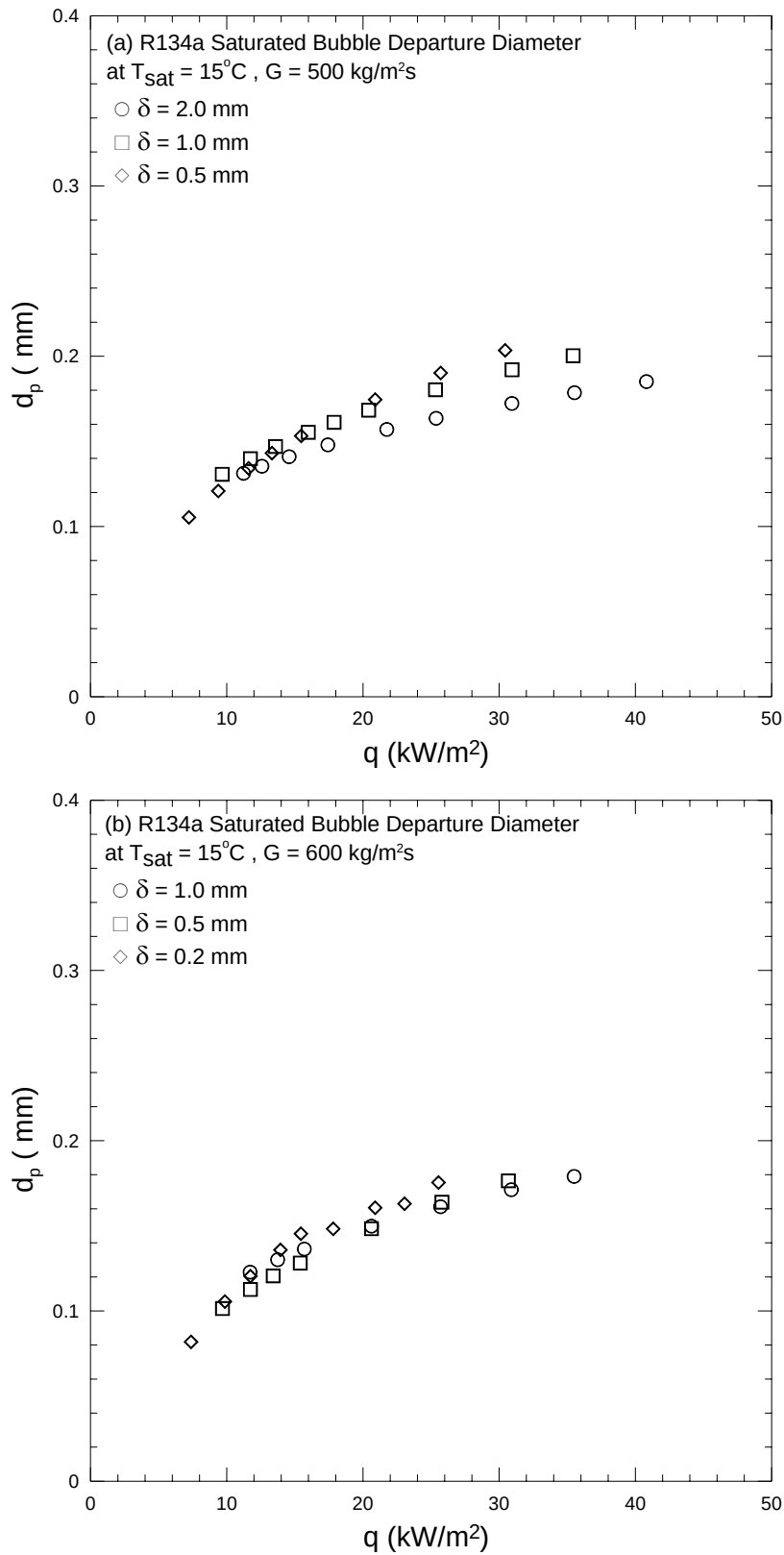


Fig. 4.21 Saturated flow boiling mean bubble departure diameter for various gap sizes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 500 \text{ kg/m}^2\text{s}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$.

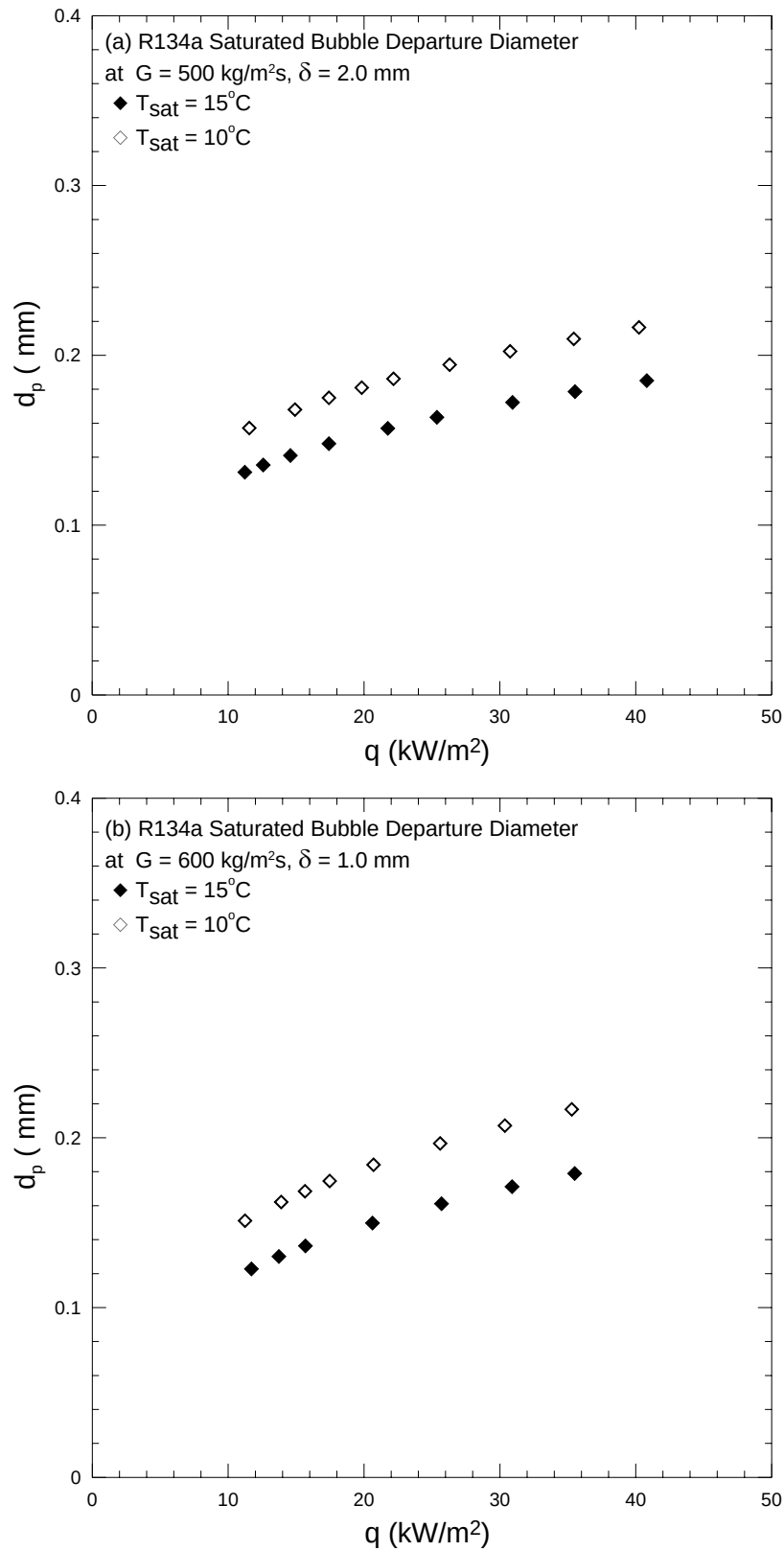


Fig. 4.22 Saturated flow boiling mean bubble departure diameter for various refrigerant saturated temperatures at (a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 2.0 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 1.0 \text{ mm}$.

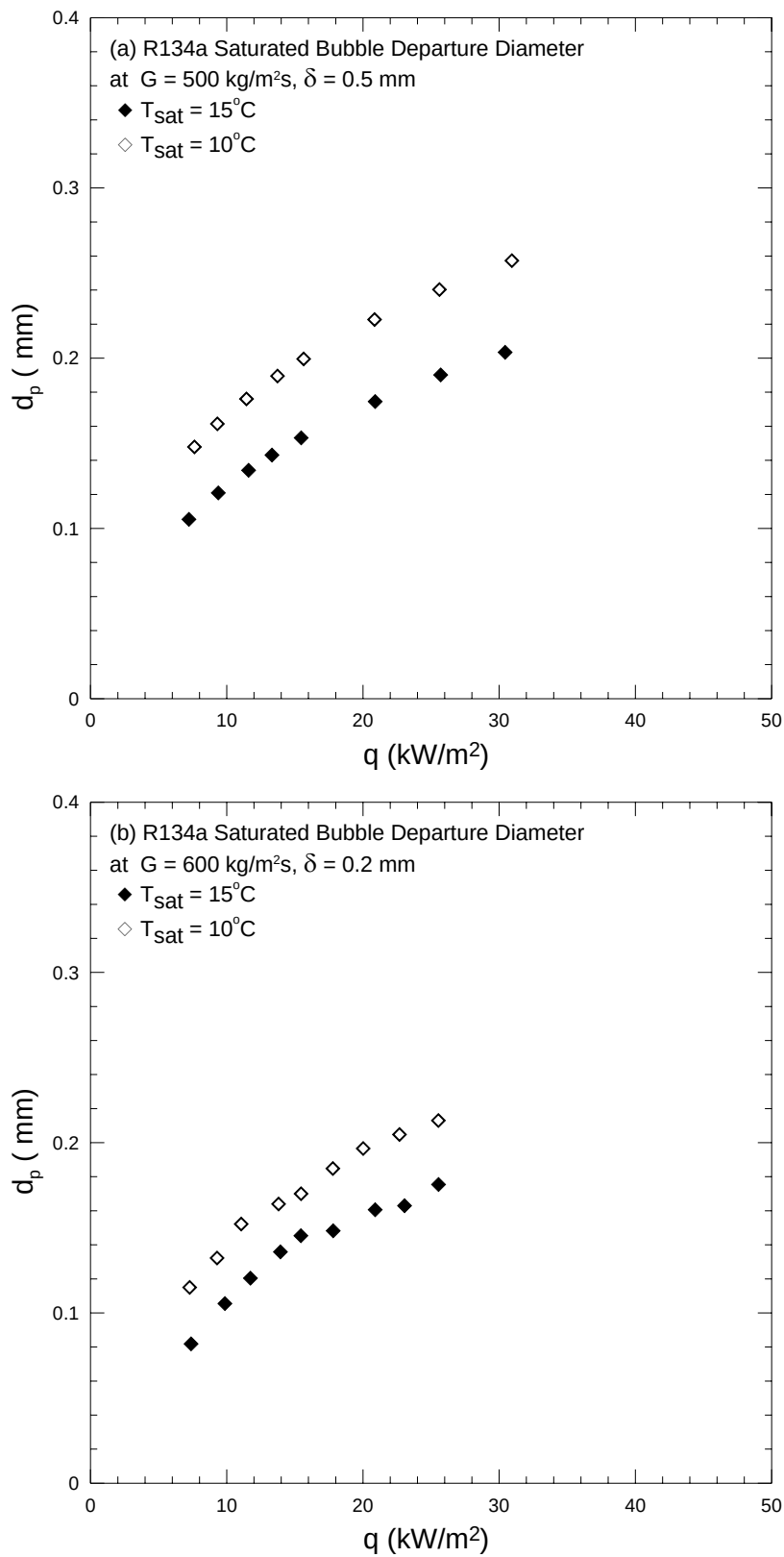


Fig. 4.23 Saturated flow boiling mean bubble departure diameter for various refrigerant saturated temperatures at (a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 0.5 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 0.2 \text{ mm}$.

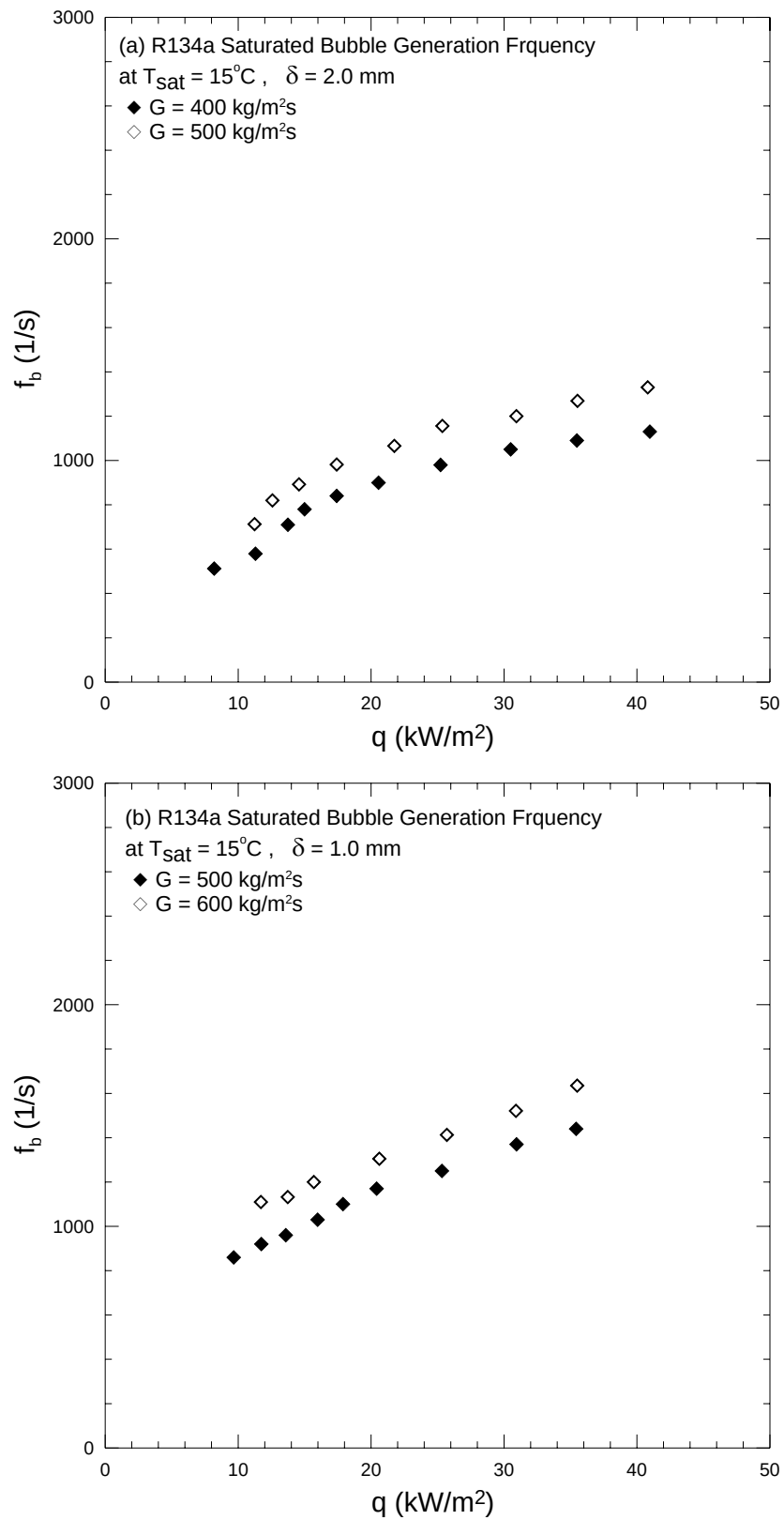


Fig. 4.24 Saturated flow boiling mean bubble departure frequency for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 2.0$ mm and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 1.0$ mm.

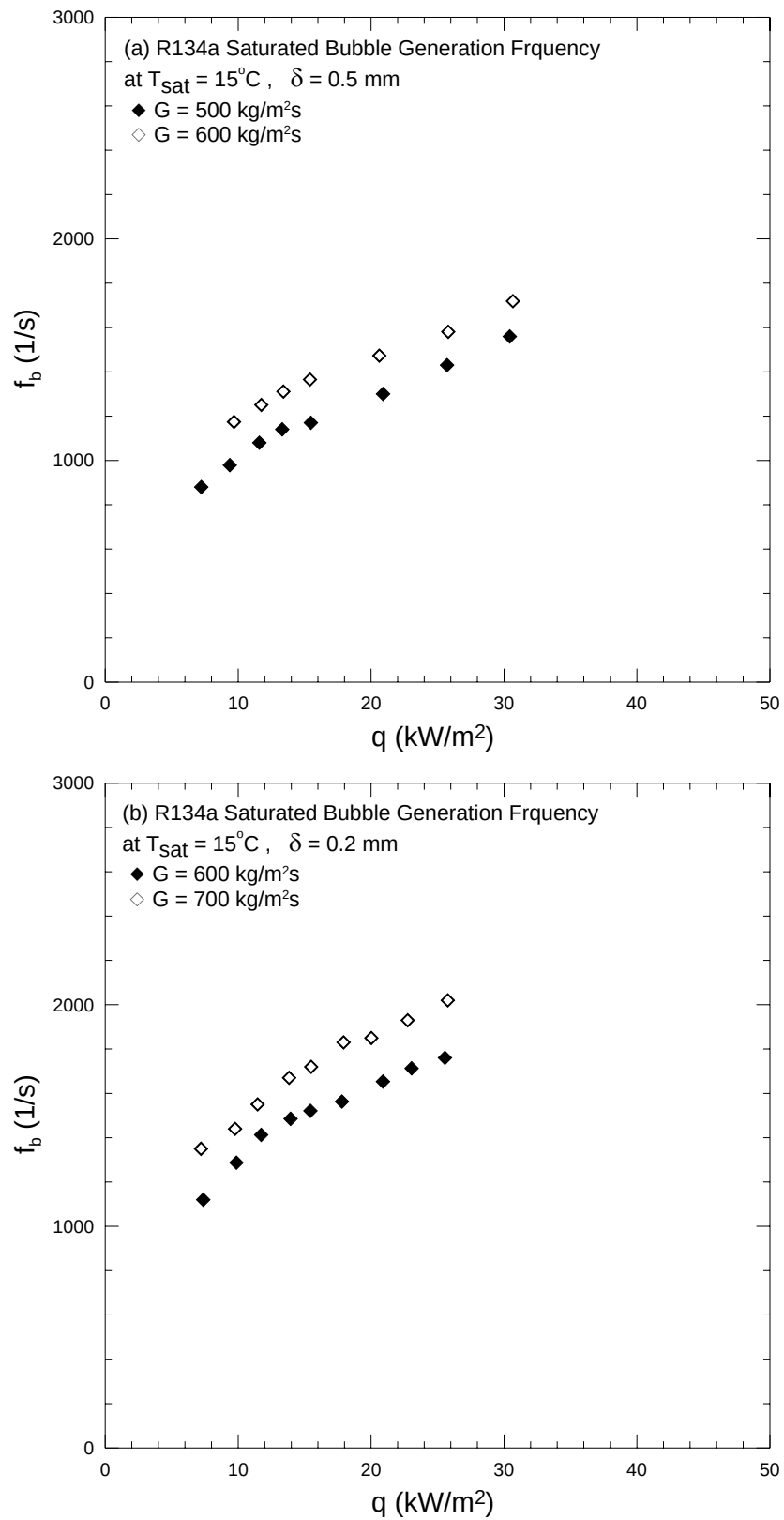


Fig. 4.25 Saturated flow boiling mean bubble departure frequency for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 0.5 \text{ mm}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 0.2 \text{ mm}$.

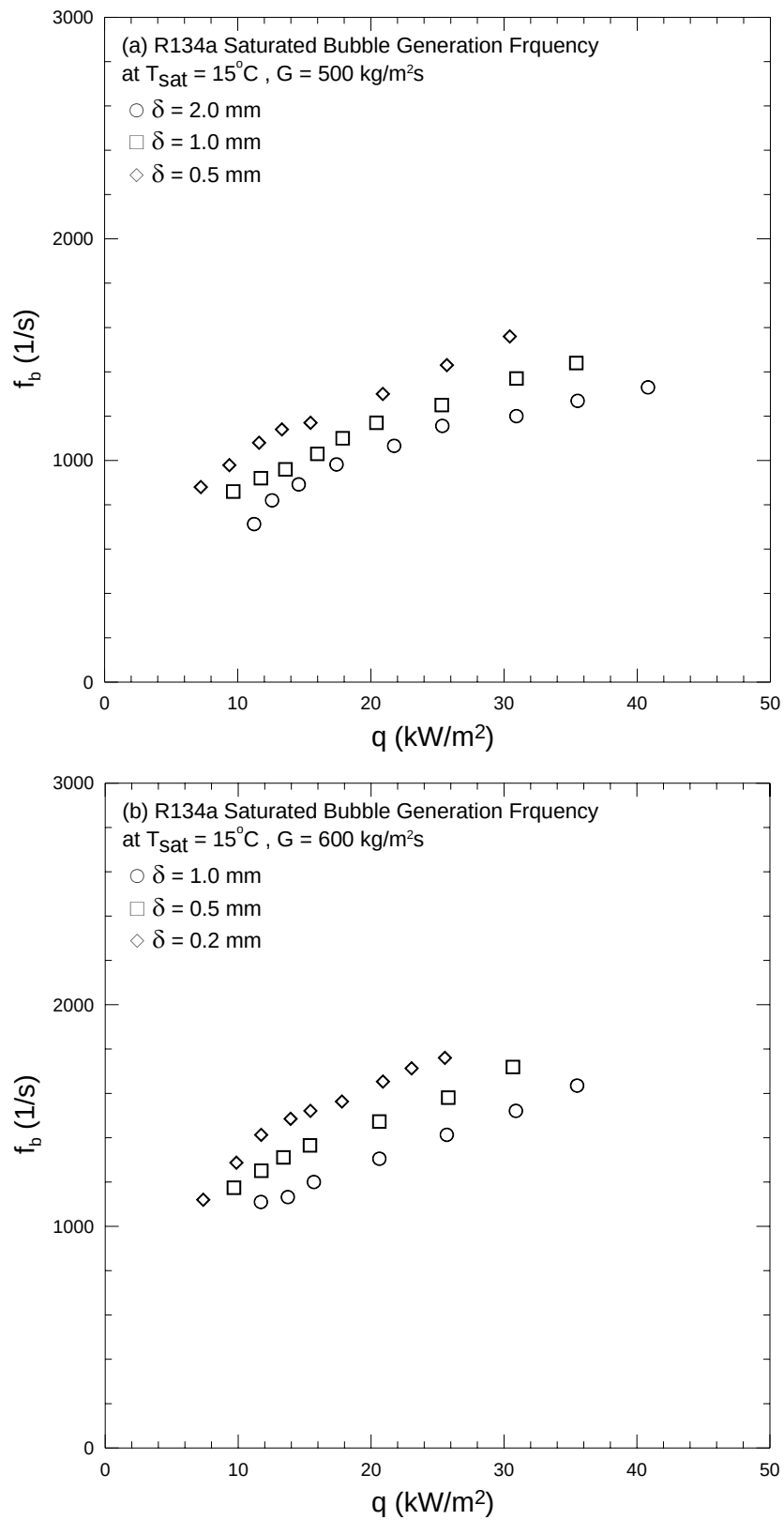


Fig. 4.26 Saturated flow boiling mean bubble departure frequency for various gap sizes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 500 \text{ kg/m}^2\text{s}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$.

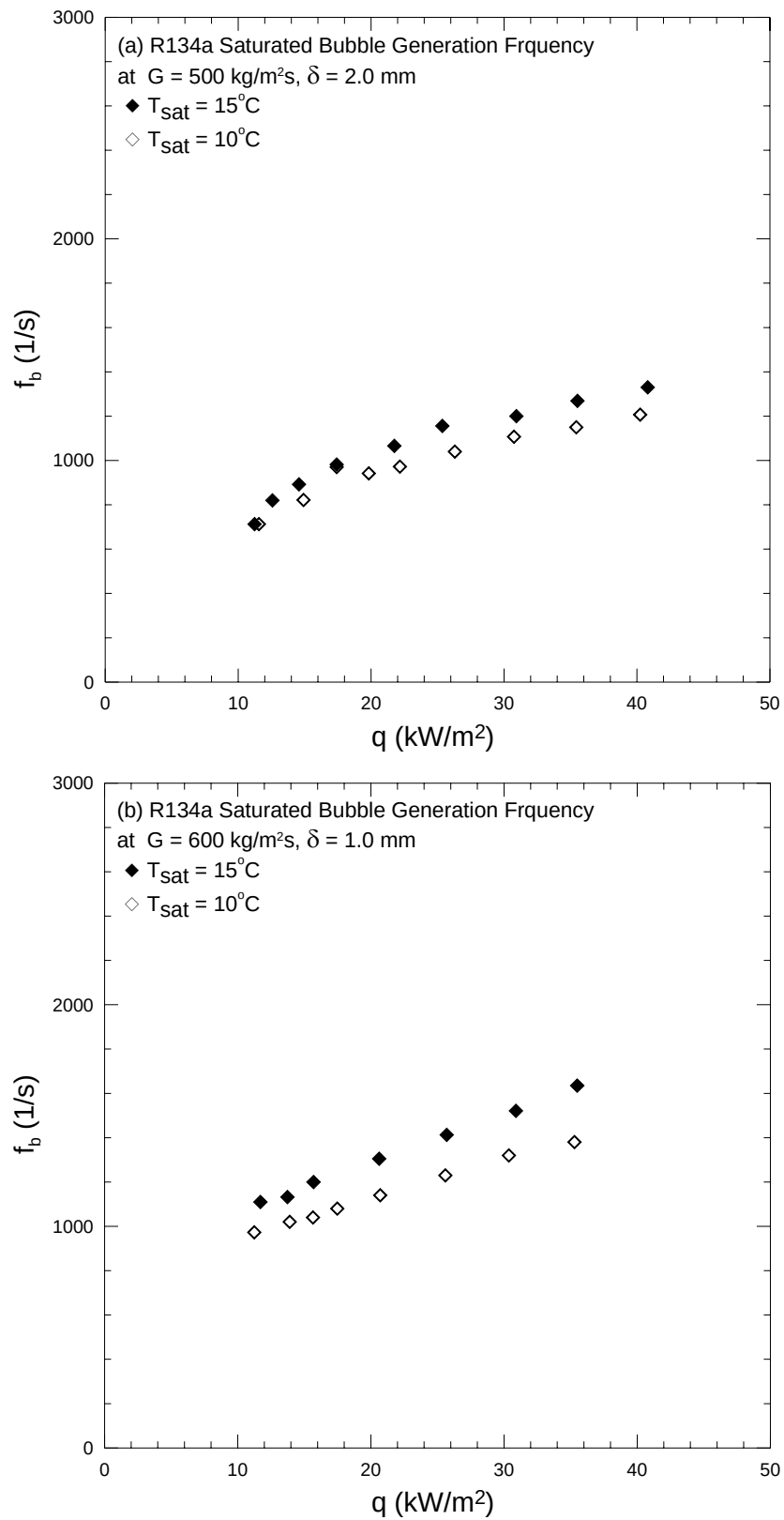


Fig. 4.27 Saturated flow boiling mean bubble departure frequency for various refrigerant saturated temperatures at (a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 2.0 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 1.0 \text{ mm}$.

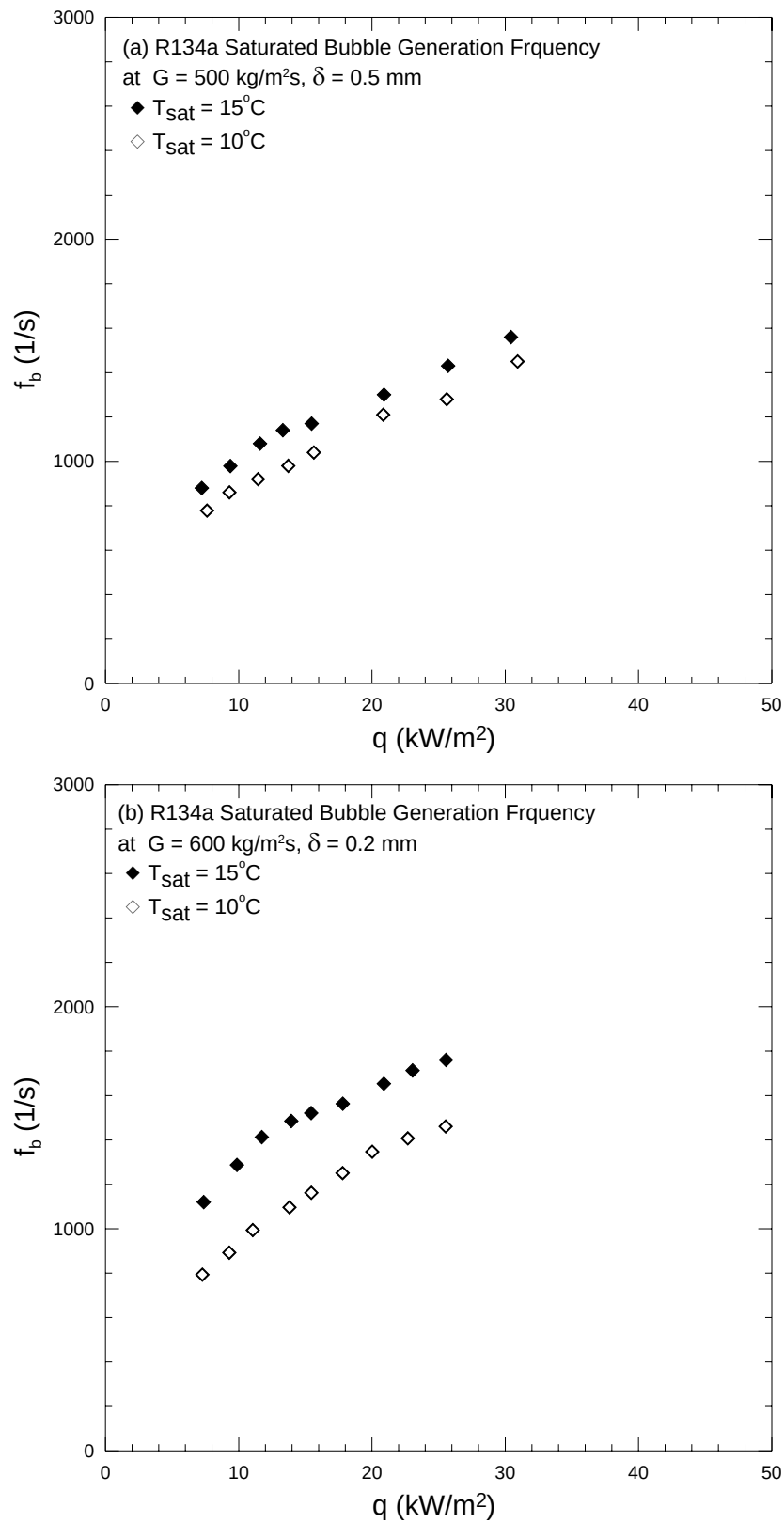


Fig. 4.28 Saturated flow boiling mean bubble departure frequency for various refrigerant saturated temperatures at (a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 0.5 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 0.2 \text{ mm}$.

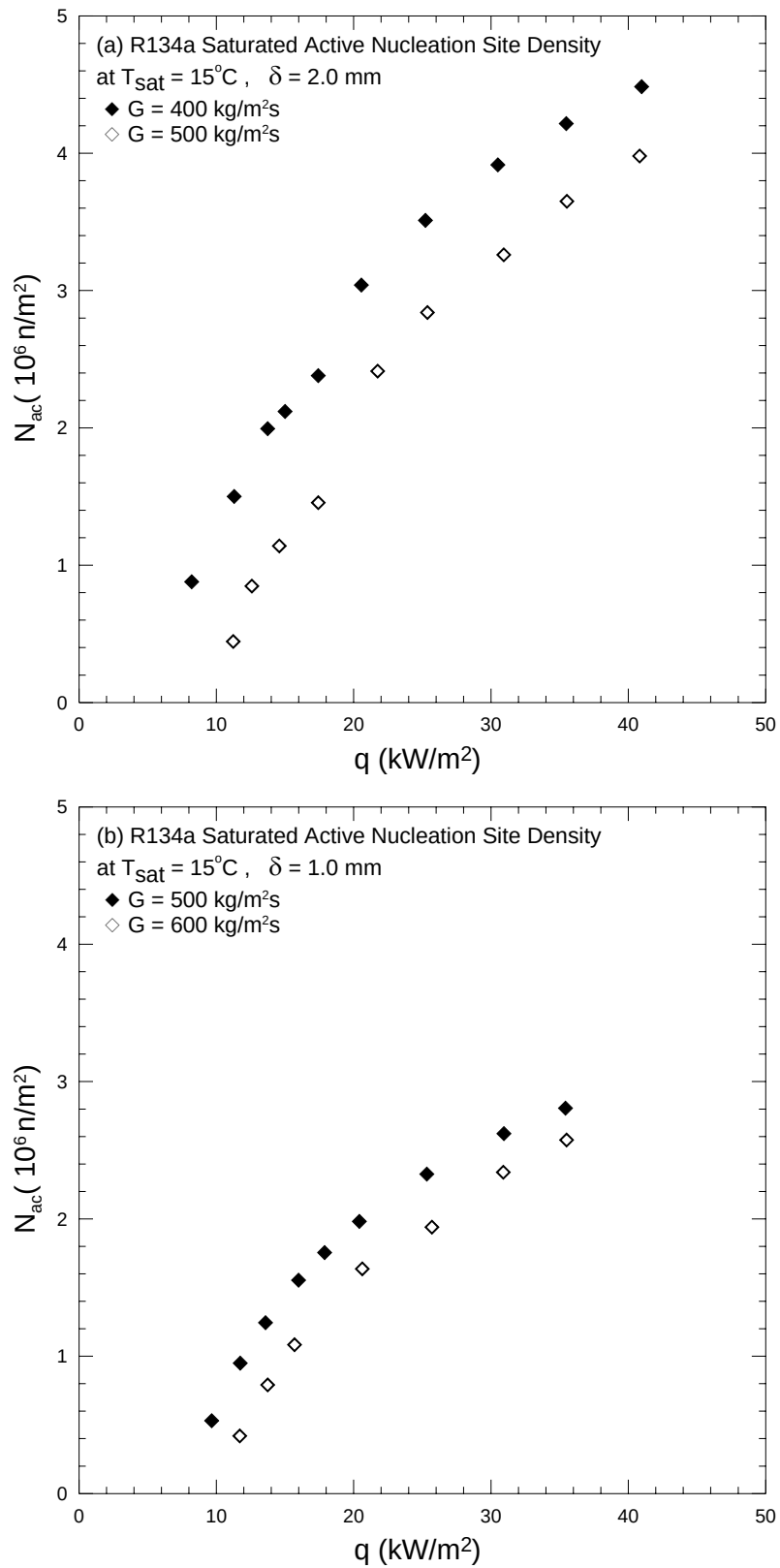


Fig. 4.29 Saturated flow boiling mean active nucleation site density for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 2.0$ mm and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 1.0$ mm.

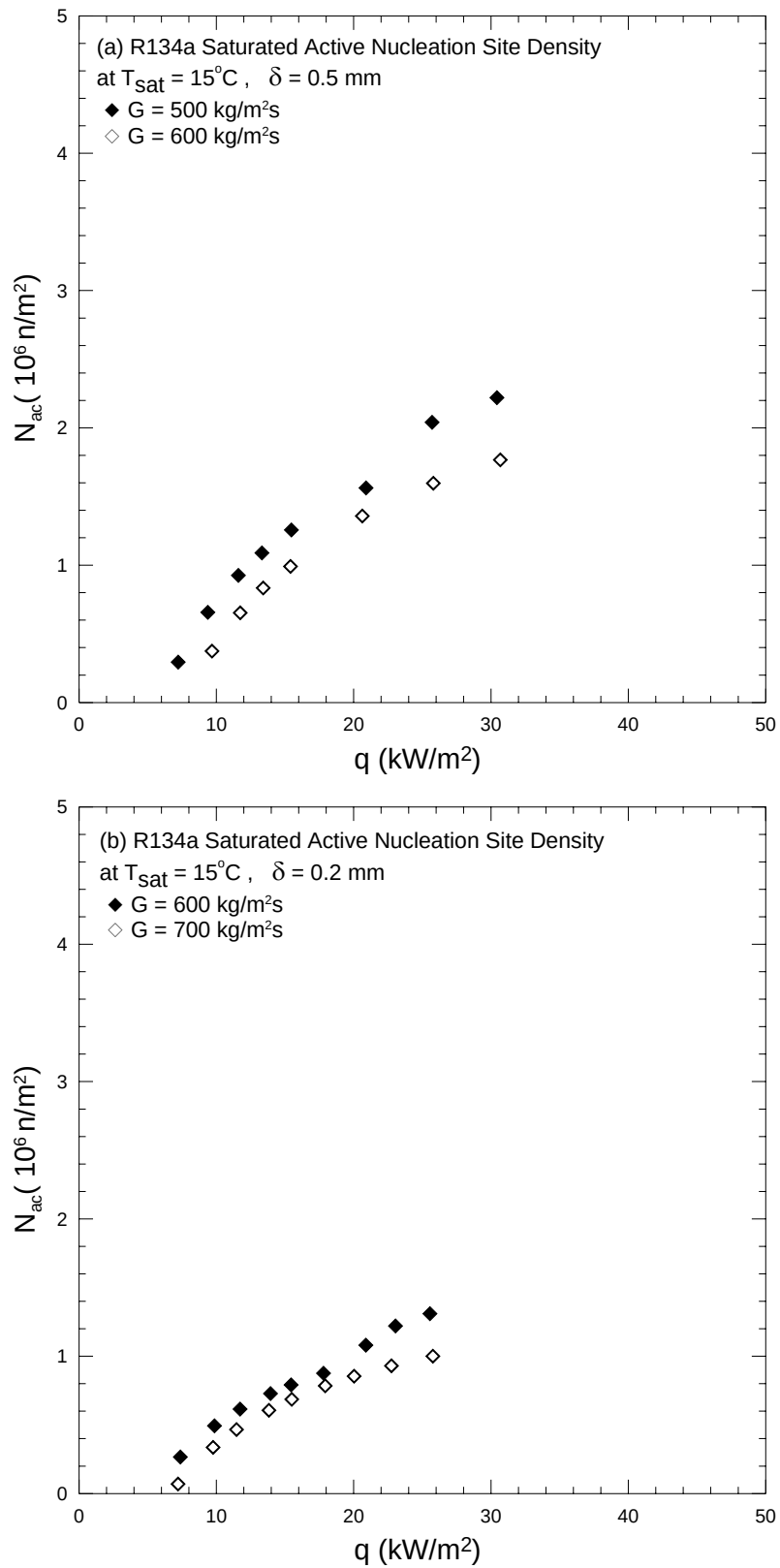


Fig. 4.30 Saturated flow boiling mean active nucleation site density for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 0.5 \text{ mm}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $\delta = 0.2 \text{ mm}$.

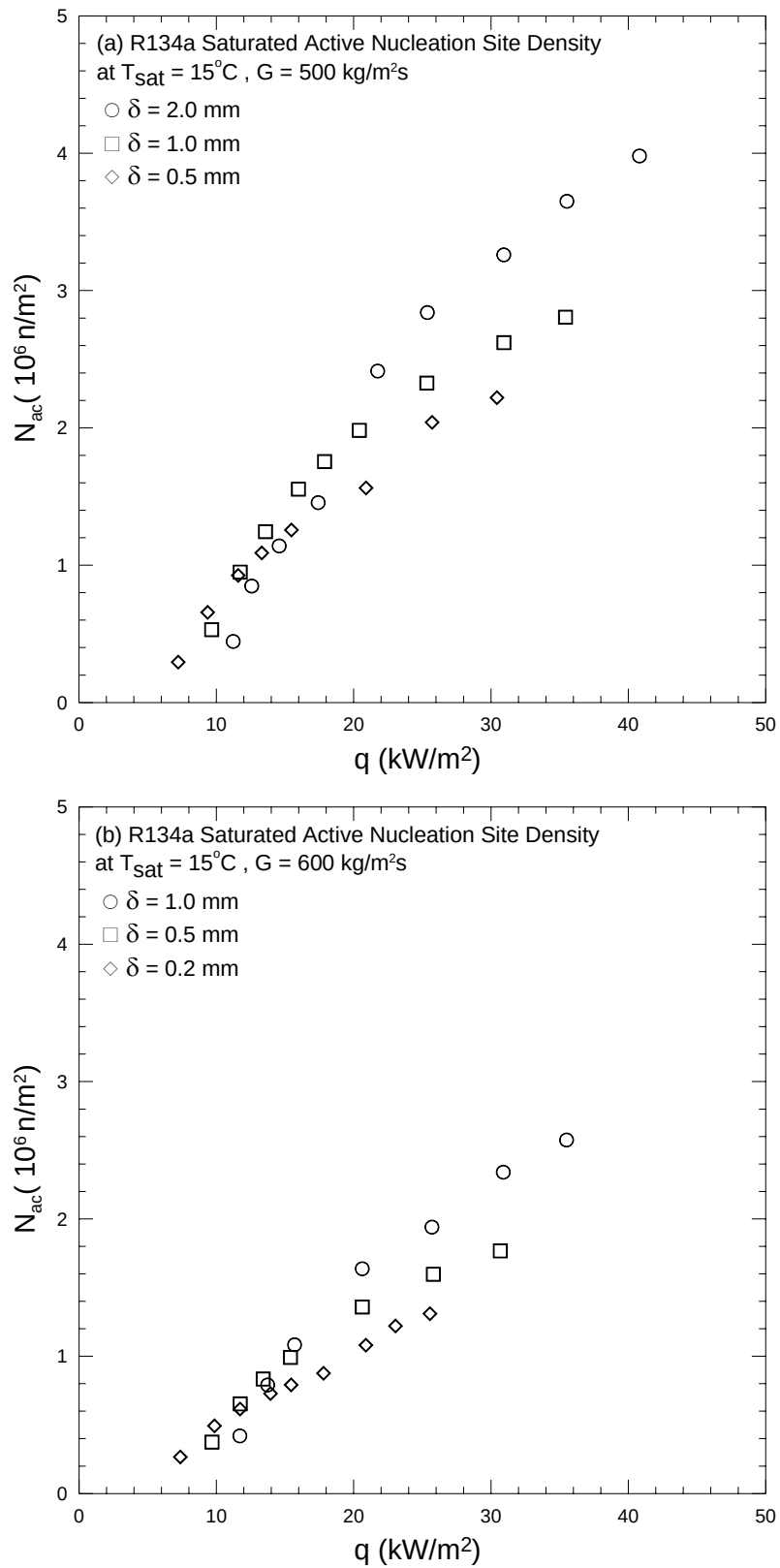


Fig. 4.31 Saturated flow boiling mean active nucleation site density for various gap sizes at (a) $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 500 \text{ kg/m}^2\text{s}$ and (b) $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$.

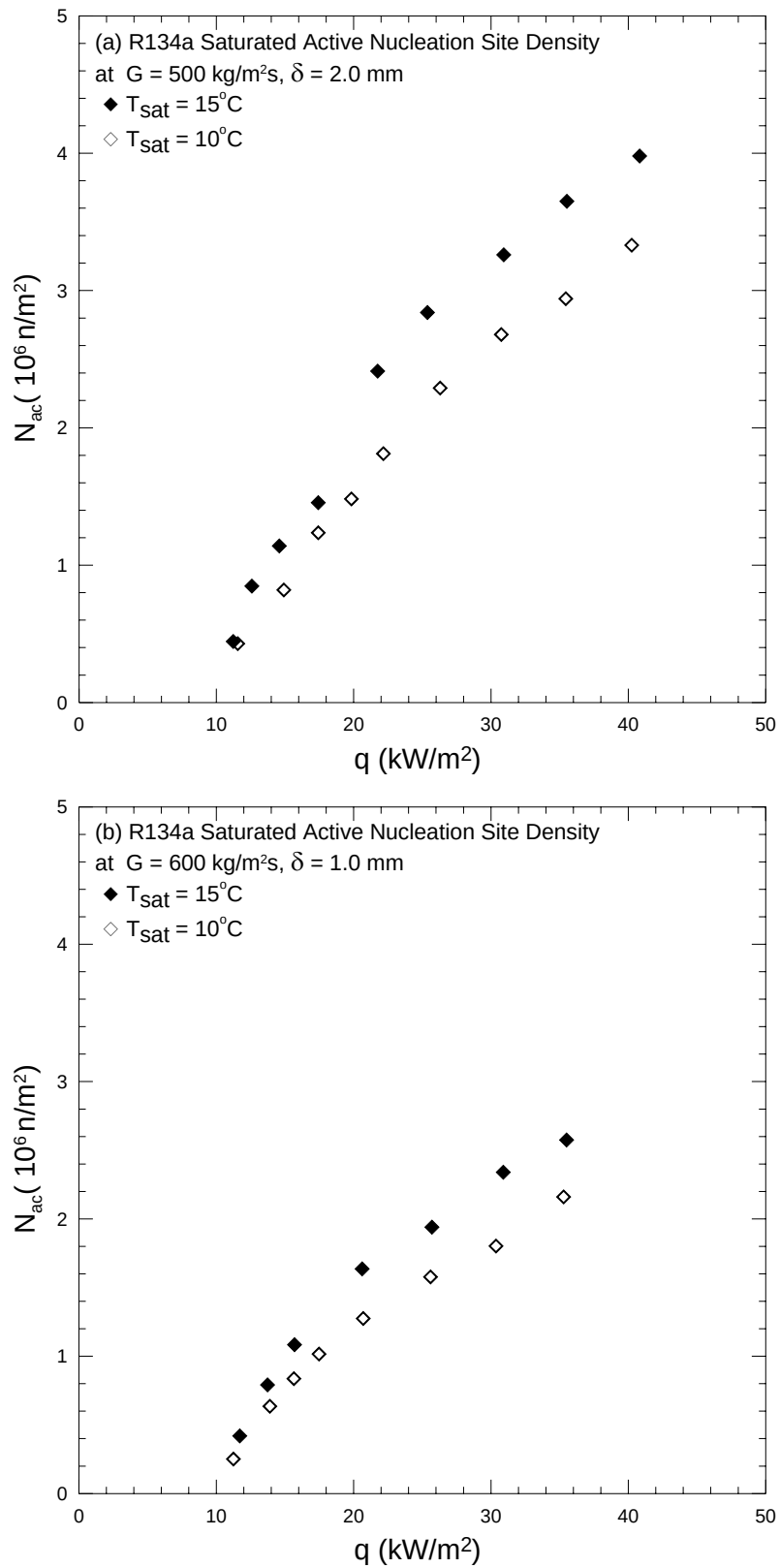


Fig. 4.32 Saturated flow boiling mean active nucleation site density for various refrigerant saturated temperatures at (a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 2.0 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 1.0 \text{ mm}$.

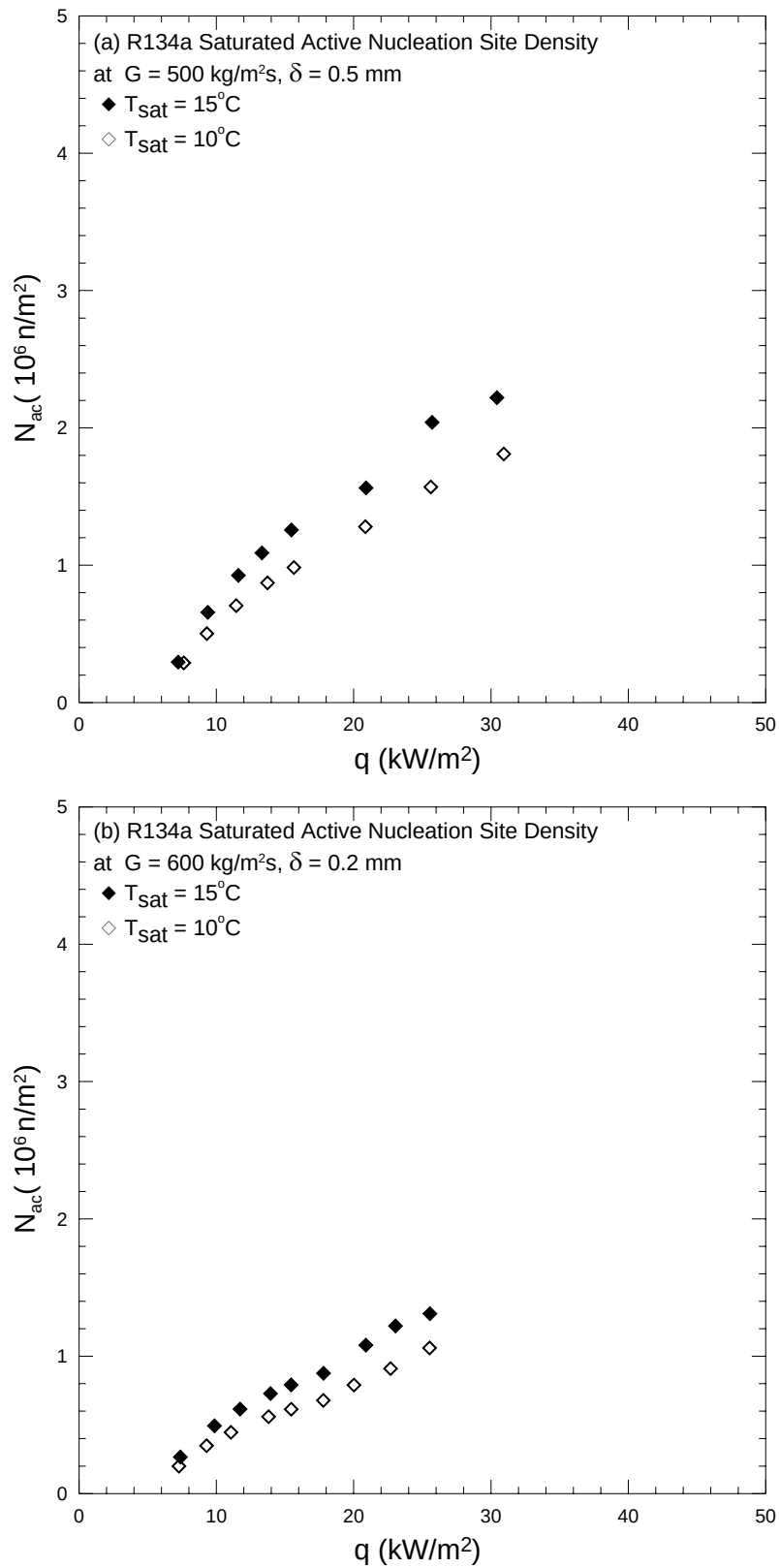


Fig. 4.33 Saturated flow boiling mean active nucleation site density for various refrigerant saturated temperatures at (a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 0.5 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 0.2 \text{ mm}$.

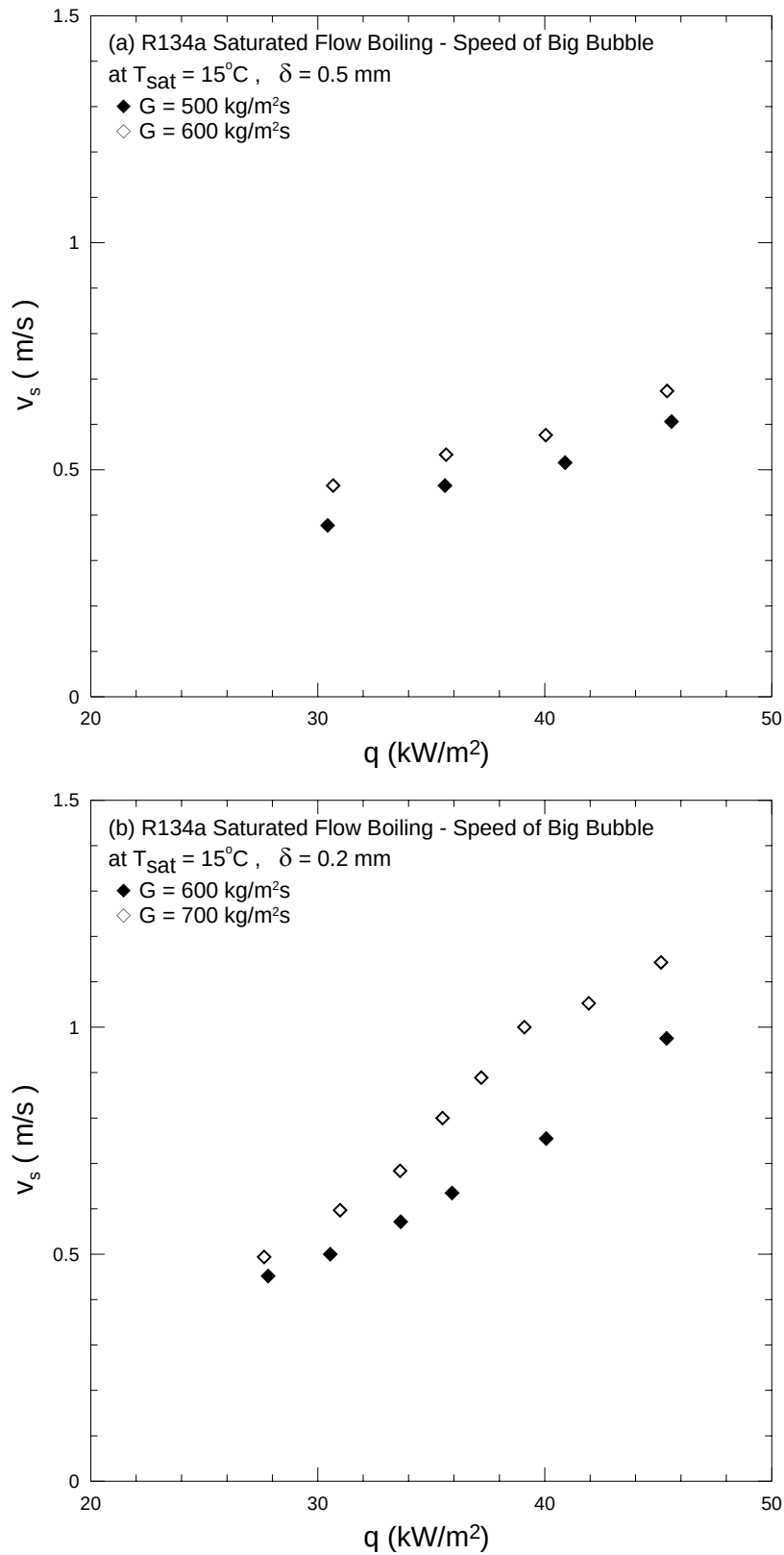


Fig. 4.34 Mean axial speed of the big bubbles in slug flow for various refrigerant mass fluxes at (a) $T_{\text{sat}} = 15^{\circ}\text{C}$ & $\delta = 0.5 \text{ mm}$ and (b) $T_{\text{sat}} = 15^{\circ}\text{C}$ & $\delta = 0.2 \text{ mm}$.

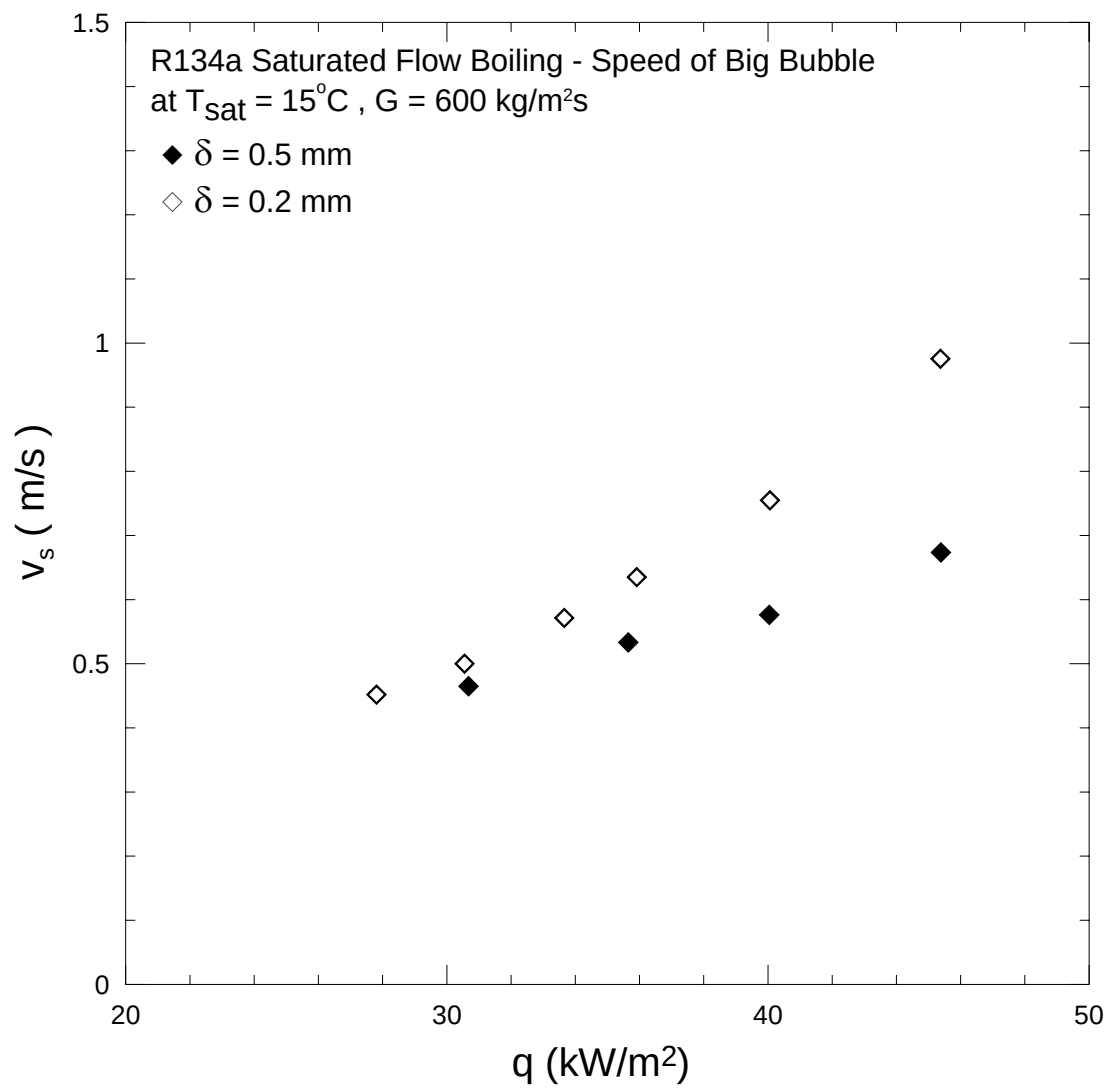


Fig. 4.35 Mean axial speed of the big bubbles in slug flow for various gap sizes at $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$.

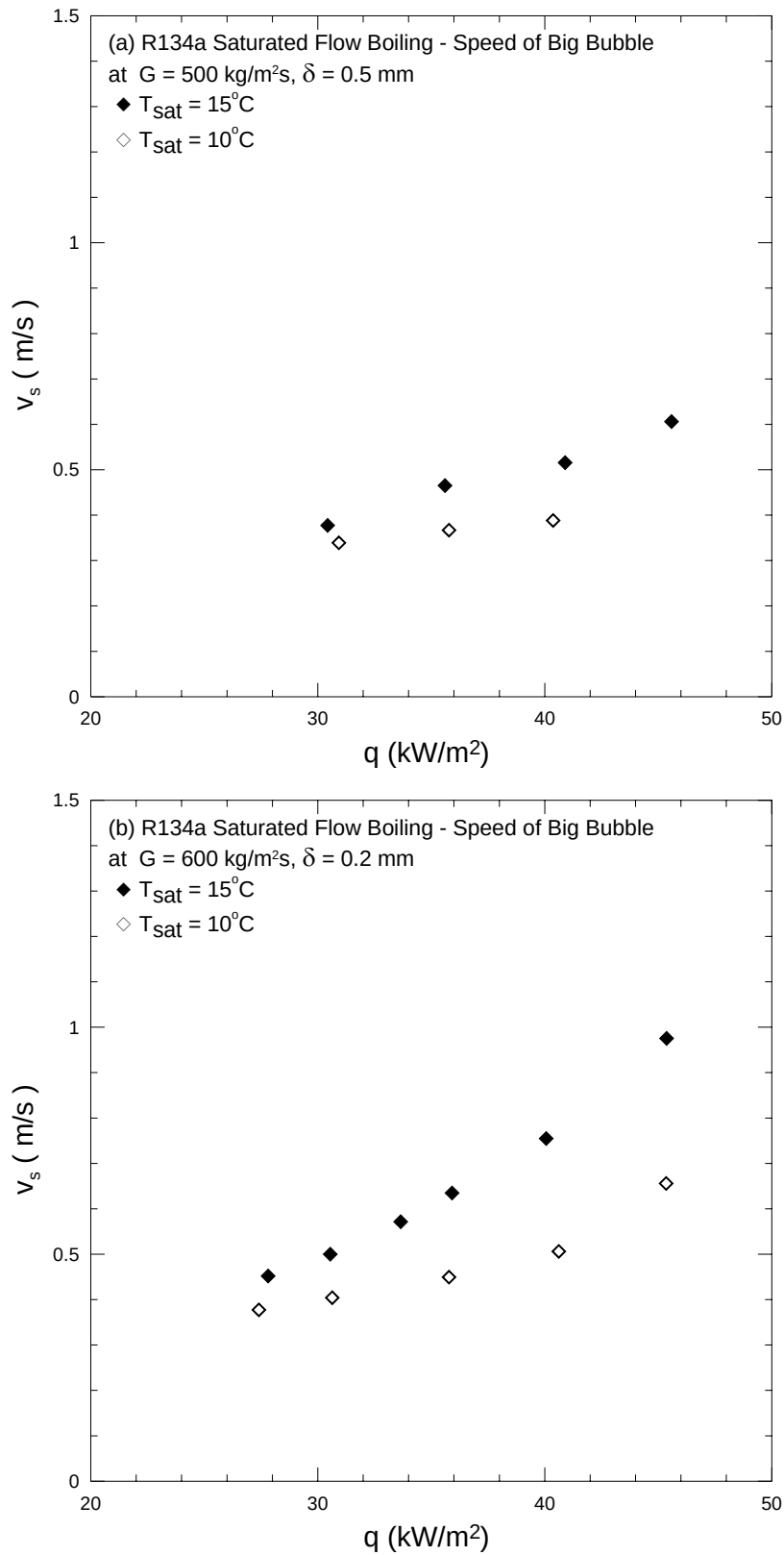


Fig. 4.36 Mean axial speed of the big bubbles in slug flow for various refrigerant saturated temperatures at (a) $G = 500 \text{ kg/m}^2\text{s}$ & $\delta = 0.5 \text{ mm}$ and (b) $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 0.2 \text{ mm}$.

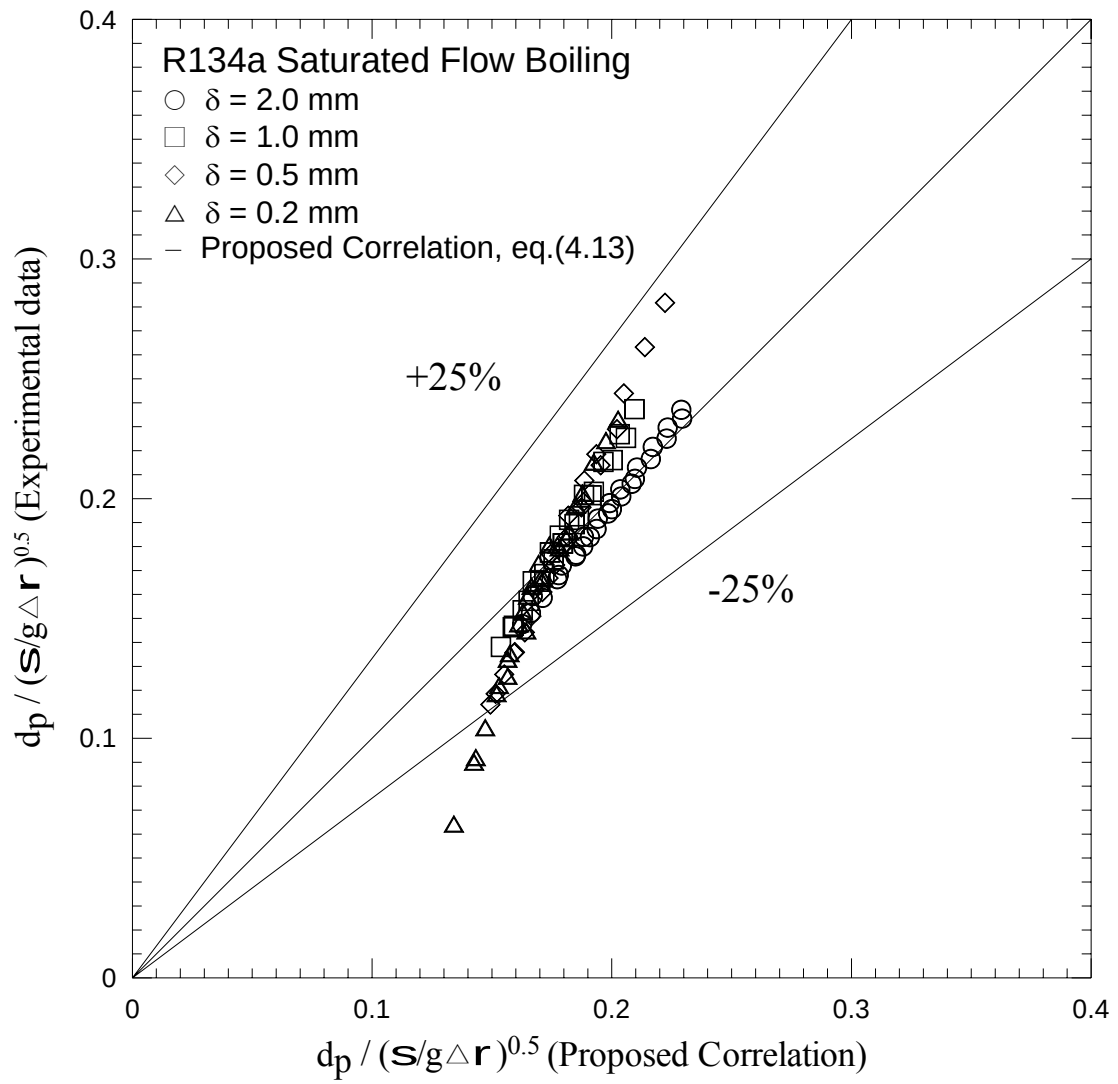


Fig. 4.37 Comparison of the measured data for mean bubble departure diameter in the saturated flow boiling of R-134a with the proposed correlation.

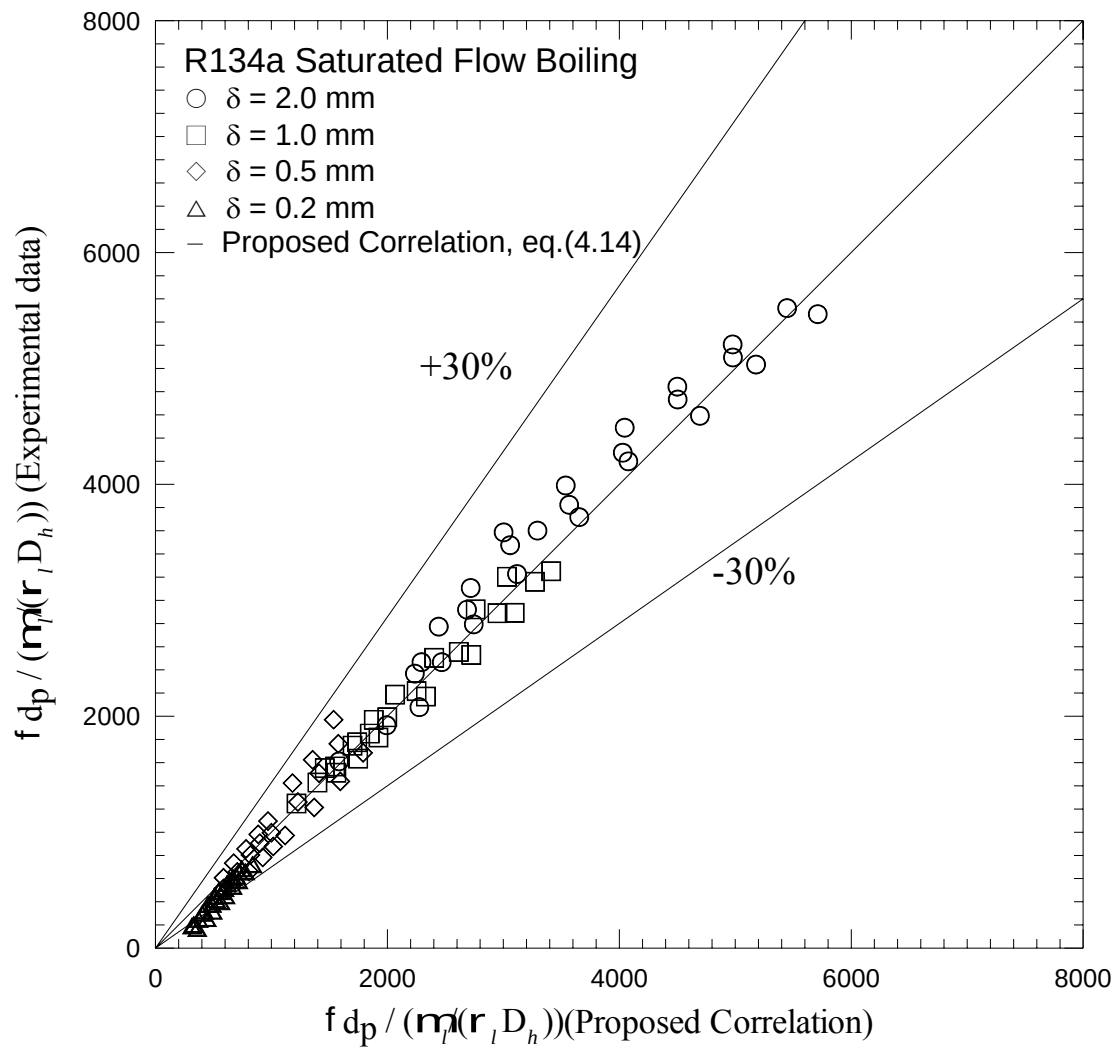


Fig. 4.38 Comparison of the measured data for mean bubble departure frequency in the saturated flow boiling of R-134a with the proposed correlation.

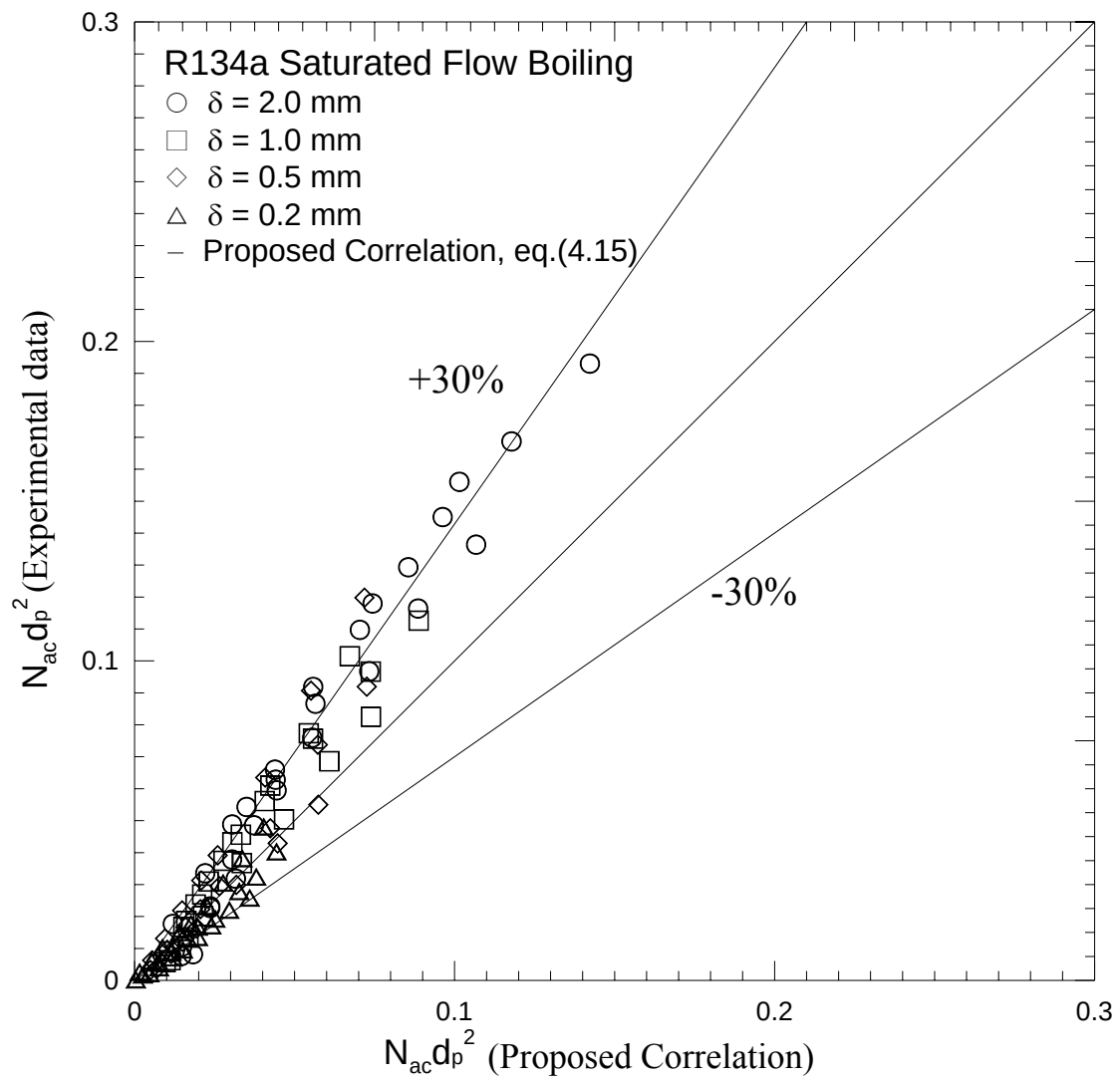


Fig. 4.39 Comparison of the measured data for mean active nucleation site density in the saturated flow boiling of R-134a with the proposed correlation.

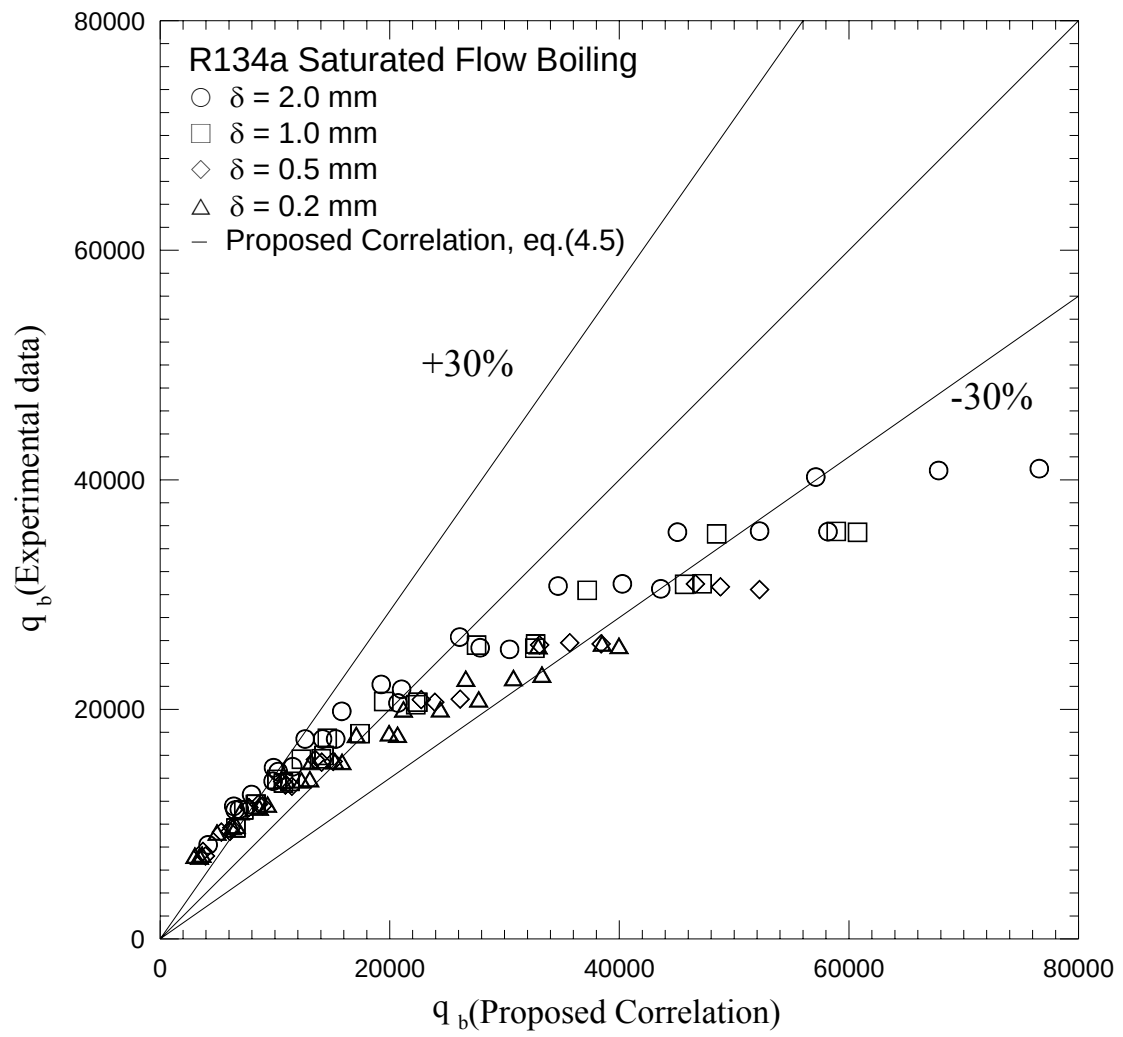


Fig. 4.40 Comparison of the measured data for heat transfer coefficient in the saturated flow boiling of R-134a in the bubbly flow regime with the proposed correlation.

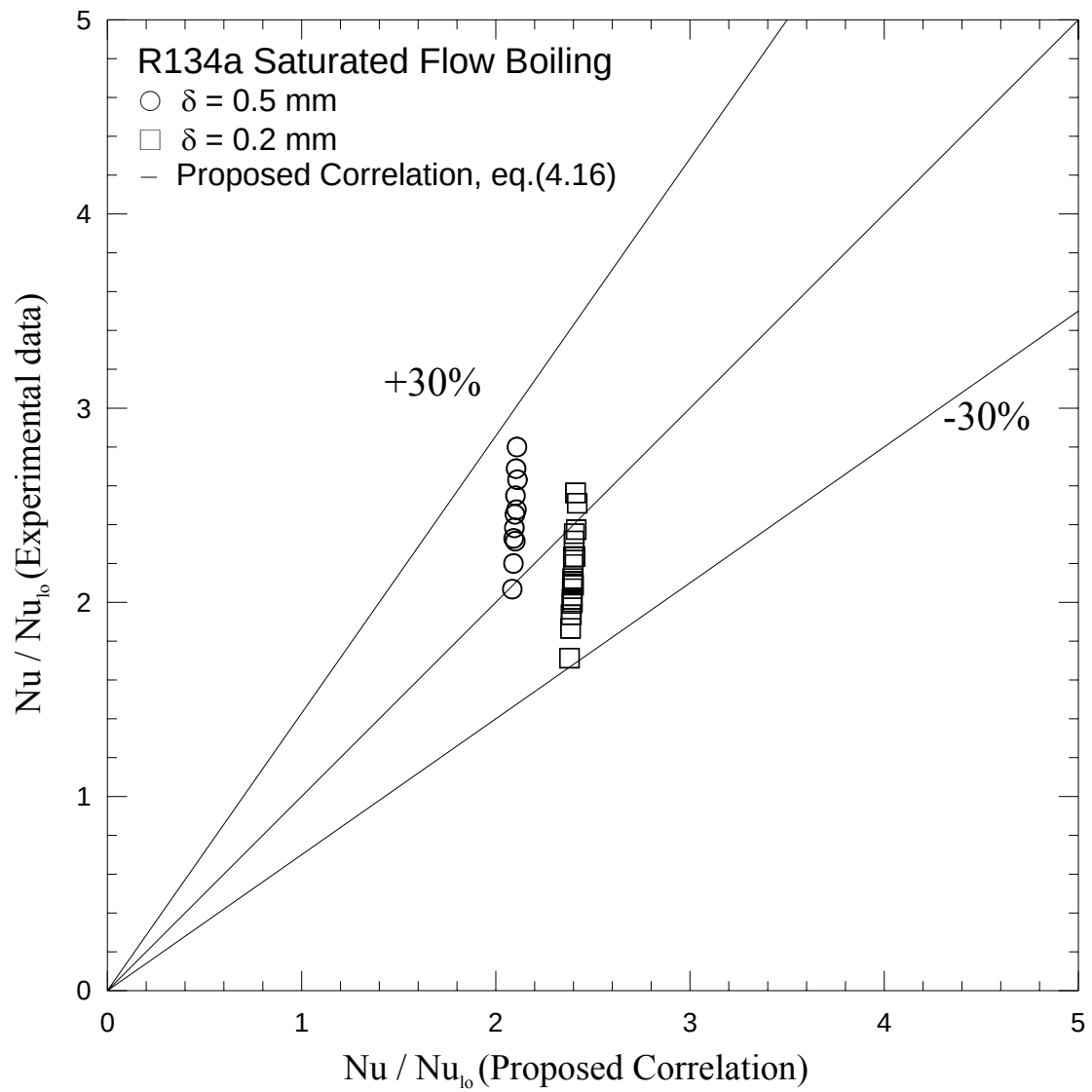


Fig. 4.41 Comparison of the measured data for heat transfer coefficient in the saturated flow boiling of R-134a in the slug flow regime with the proposed correlation.

CHAPTER 5

EVAPORATION HEAT TRANSFER

IN SMALL CIRCULAR TUBES

In this chapter selected results from the data measured here are presented to illustrate the evaporation heat transfer of refrigerant R-407C in the horizontal small circular tubes. The experiments are performed in the present study for the vapor quality at the tube inlet varied from 0.15 to 0.8, mass flux from 200 to 400 kg/m²s, imposed heat flux from 5 to 15 kW/m², and system pressure from 665 to 900 kPa, which corresponds to the saturated temperature of R-407C from 5 to 15°C. Note that this void fraction of R-407C is very high and the evaporation heat transfer of the refrigerant in the small tubes is therefore dominated by the evaporation of liquid film on the heating surface. In what follows the effects of the vapor quality, imposed heat flux, refrigerant mass flux and saturated temperature on the average evaporation heat transfer coefficient are examined in detail. Besides, the present heat transfer data are also compared with those of Yan and Lin [12], Wang and Chiang [7] and certain correlations. Finally, an empirical correlation for the present R-407C evaporation heat transfer coefficient is proposed.

5-1 Single Phase Heat Transfer

Before measuring the R-407C evaporation heat transfer coefficient, single phase liquid R-407C convection heat transfer coefficient was obtained for the refrigerant inlet temperature fixed at 12°C and refrigerant mass flux varied from 183 to 811 kg/m²s (corresponding to the Reynolds number of 2050 to 9100) for the imposed heat flux of 5 kW/m². To check the suitability of the experimental system for the present

measurement, the single phase liquid R-407C heat transfer coefficient is compared with the Dittus-Boelter and Gnielinski correlations [47, 48] in Fig. 5.1. The correlations from Dittus-Boelter and Gnielinski are respectively

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad (5.1)$$

and

$$Nu = \frac{(f/2)(Re-1000)Pr}{1. + 12.7\sqrt{f/2}(Pr^{2/3}-1)} \quad (5.2)$$

where

$$f = (1.58 \ln Re - 3.28)^{-2} \quad (5.3)$$

The results showed that the measured single-phase heat transfer coefficients for R-407C in the small tubes are in reasonable agreement with the two correlations.

5-2 Two-Phase Evaporation Heat Transfer

The effects of the refrigerant vapor quality, mass flux, saturated temperature and imposed heat flux on the evaporation heat transfer coefficient of refrigerant R-407C in the horizontal small circular tubes are shown in Figs. 5.2-5.10. The measured heat transfer coefficients are inspected by checking their variations with the vapor quality at the test section inlet x_{in} . Since the tubes are short, the total vapor quality change Δx in the tubes for various cases in the present test is relatively small, ranging from 0.01 to 0.03. At first, Fig. 5.2 shows the variations of the evaporation heat transfer coefficient with the inlet vapor quality of the refrigerant at $q = 5 \text{ kW/m}^2$ for different mass fluxes and saturated temperatures of R-407C. The results show that at this low imposed heat flux the evaporation heat transfer coefficient only increases very slightly with the inlet vapor quality for given G , T_{sat} and q . But the increases of h_r with the refrigerant mass flux cannot be ignored. For example in Fig. 5.2(a), at $T_{sat} = 5^\circ\text{C}$ the quality-average evaporation heat transfer coefficient at $G = 400 \text{ kg/m}^2\text{s}$ is about 2700

$\text{W/m}^2\text{C}$ and is higher than $2050 \text{ W/m}^2\text{C}$ for $G = 200 \text{ kg/m}^2\text{s}$ by 32%. It means that the effect of the mass flux on the R-407C evaporation heat transfer is significant. Next, the effects of the refrigerant saturated temperature on the R-407C evaporation heat transfer coefficient are shown in Fig. 5.3 again for the low heat flux of 5 kW/m^2 . Note that the increase of h_r with T_{sat} is rather significant especially when T_{sat} is raised from 5°C to 10°C . This is due to the lower latent heat of vaporization for the higher saturated temperature and liquid film vaporization is stronger, which in turn results in a higher vapor velocity and enhances the interfacial heat transfer between the liquid film and vapor flow. Thus, it results in a higher evaporation heat transfer coefficient.

The variations of the evaporation heat transfer coefficient with the inlet vapor quality of the refrigerant at the higher imposed heat flux with $q = 10 \text{ kW/m}^2$ are illustrated in Figs. 5.4 and 5.5. The data in Fig. 5.4 for the higher q clearly indicate that at higher mass fluxes of 300 and $400 \text{ kg/m}^2\text{s}$ the evaporation heat transfer coefficient increases substantially with the inlet vapor quality of the refrigerant in small tubes. For instance in Fig. 5.4(a), the heat transfer coefficient for $G = 400 \text{ kg/m}^2\text{s}$ at $x_{in} = 0.78$ is about $5800 \text{ W/m}^2\text{C}$ and is higher than $3900 \text{ W/m}^2\text{C}$ for $x_{in} = 0.21$ by 48 %. This significant increase of h_r with x_{in} obviously results from the fact that at the high vapor quality the liquid film on the inside surface of the small tubes is thinner. This, in turn, reduces the resistance of heat transfer from the heating surface to the refrigerant. We also note that at the higher q the increase of h_r with the refrigerant mass flux is also large. For example in Fig. 5.4(a) for $x_{in} = 0.78$, the heat transfer coefficient at $G = 400 \text{ kg/m}^2\text{s}$ is about $5800 \text{ W/m}^2\text{C}$ and is much higher than $2500 \text{ W/m}^2\text{C}$ for $G = 200 \text{ kg/m}^2\text{s}$ by 132 %. This results from the simple fact that for the higher refrigerant mass flux the vapor velocity is higher and the convective evaporation is stronger. Then, the effects of the refrigerant saturated temperature on the evaporation heat transfer coefficient are inspected in Fig. 5.5. The results again

suggest that the evaporation heat transfer is much better at the higher T_{sat} .

Figures 5.6 and 5.7 further reveal the variations of the evaporation heat transfer coefficient with the inlet vapor quality of the refrigerant in the small tubes at an even higher imposed heat flux of 15 kW/m^2 for different mass fluxes and refrigerant saturated temperatures. The data clearly indicate that the evaporation heat transfer coefficient also increases significantly with the refrigerant mass flux, saturated temperature and vapor quality. These trends are similar to those at the lower heat fluxes. Obviously, at different imposed heat fluxes the quantitative changes in h_r with G and T_{sat} are somewhat different. For example, at $q=15 \text{ kW/m}^2$ and $T_{\text{sat}} = 10^\circ\text{C}$ the quality-average evaporation heat transfer coefficients at $G = 300$ & $400 \text{ kg/m}^2\text{s}$ are respectively about 47 % and 86 % larger than that at $G = 200 \text{ kg/m}^2\text{s}$ (Fig. 5.6(b)). And at $q=15 \text{ kW/m}^2$ and $G = 400 \text{ kg/m}^2\text{s}$ the quality-average evaporation heat transfer coefficient at $T_{\text{sat}} = 10$ & 15°C are respectively about 24 % and 49 % larger than that at $T_{\text{sat}} = 5^\circ\text{C}$ (Fig. 5.7(c)).

Next, results are presented in Figs. 5.8 to 5.10 to further illustrate the effects of the imposed heat flux on the R-407C evaporation heat transfer coefficient for the small circular tubes. These results clearly manifest that the increase of the evaporation heat transfer coefficient with the vapor quality is almost linear for all G , T_{sat} and q . Moreover, the increase is rather significant at the high G , T_{sat} and q . Note that at the lower heat flux there is lower nucleation density on the heating surface, lower bubble generation frequency, and slower bubble growth. Besides, at the lower mass flux we have lower vapor and liquid velocity. Thus, at low q and G the variations of the evaporation heat transfer coefficient with the vapor quality are less significant.

Finally, the R-407C evaporation heat transfer coefficients measured here are compared with those from Yan and Lin [12] for R-134a in the same small tubes and

with Wang et al. [7] for R-407C in a larger tube of 6.5 mm in diameter. Due to the limited amount of the available data for R-134a in the small tubes which were conducted at similar conditions covered in the present study, the comparison is only possible for a few cases. Figure 5.11 compare the present R-407C evaporation heat transfer coefficients with those from Yan and Lin [12] for R-134a at selected cases for various imposed heat fluxes and refrigerant saturated temperatures at the same refrigerant mass flux. The comparison indicates that at the low q of 5 kW/m^2 refrigerant R-407C has a slightly higher h_r when the inlet quality is not high with $x_{in} < 0.5$ (Fig. 5.11(a)). But at the higher q of 10 and 15 kW/m^2 we have higher h_r for R-407C only at a higher vapor quality for $x_{in} > 0.65$ (Fig. 5.11(b) and (c)). It is of interest to mention that in the previous study for R-134a [12] the decline of h_r with x_{in} at a high vapor quality for the high imposed wall heat flux is attributed to the possibility that at a higher vapor quality for R-134a the tube wall may become partially dry when the wall heat flux is high enough. Note that at the $q = 5 \text{ kW/m}^2$ and at the high vapor quality the R-134a evaporation heat transfer coefficient is higher than that of the R-407C (Fig. 5.11(a)). These reverse trends in different vapor quality ranges for the test heat fluxes are attributed mainly to the different thermal conductivities of the two refrigerants for the liquid and vapor phases. Specifically, the thermal conductivity of the liquid R-407C is higher than that of the R-134a by 17.7 %. However, the thermal conductivity of the vapor R-407C is lower than that of the R-134a by 8 %. The much higher rise of the R-134a with the vapor quality is mainly due to the larger liquid-to-vapor density ratio when compared to that of the R-407C for the refrigerant pressures tested here. The liquid-to-vapor density ratios are respectively 31.6 and 30.6 for R-134a and R-407C for the cases examined in Fig. 5.11. The comparison shown in Fig 5.12 reveals that the R-407C evaporation heat transfer coefficient in the small tubes ($D_i = 2 \text{ mm}$) are much higher than that in a larger tube

($D_i = 6.5$ mm). Specifically, the quality-average heat transfer coefficient given in Fig. 5.12(a) is about $2700 \text{ W/m}^2\text{C}$ for $T_{\text{sat}} = 5^\circ\text{C}$ in the small tubes and exceeds $1000 \text{ W/m}^2\text{C}$ for $T_{\text{sat}} = 2^\circ\text{C}$ in the large tube by 180 %. Similarly, for the lower mass flux the quality-average heat transfer coefficient given in Fig. 5.12(b) is about $2400 \text{ W/m}^2\text{C}$ for $T_{\text{sat}} = 5^\circ\text{C}$ in the small tubes and is also higher than $1200 \text{ W/m}^2\text{C}$ for $T_{\text{sat}} = 2^\circ\text{C}$ in the large tube by 100 %.

5-3 Correlation Equations

For practical application the present data for the R-407C evaporation in the small tubes need to be correlated empirically. We first compare our data with those predicted from the correlations proposed by Bao et al. [13] and Liu and Winterton [49] in Fig. 5.13. The comparison indicate that the correlation from Liu and Winterton over predicted our data and about 50% of our data lie outside $\pm 20\%$ of the correlation of Bao et al. Then, the correlation proposed by Gungor and Winterton [50] is modified here to better fit our data. Their correlation is expressed as

$$h_{tp} = h_l \cdot E + h_{pool} \cdot S \quad (5.4)$$

where the h_l is the single phase all liquid convection heat transfer coefficient evaluated from the Dittus-Boelter correlation, equation(5-1) and h_{pool} is pool boiling heat transfer coefficient. Based on the present data h_{pool} can be correlated as

$$h_{pool} = 44 \cdot P_r^{0.49} \cdot (-0.83 \ln P_r)^{-0.9} \cdot M^{-0.54} \cdot q^{0.699} \quad (5.5)$$

Here P_r is the reduced pressure. Besides, E and S respectively represent the enhancement and suppression factors, which are dependent on the boiling number Bo , the Martinelli parameter X_{tt} and the liquid Reynolds number Re_l . The expressions for E and S are [50]

$$E = 1 + 24000 \cdot Bo^{1.79} + 0.05 \cdot (1/X_{tt})^{0.1} \quad (5.6)$$

$$S = (1 + 1.15 \times 10^{-6} \cdot E^{1.8} \cdot \text{Re}_l^{1.1})^{-1} \quad (5.7)$$

The Boiling number, Martinelli parameter and liquid Reynolds number are defined respectively as

$$Bo = \frac{q}{G \cdot i_{fg}} \quad (5.8)$$

$$X_u = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_g}{\rho_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_g} \right)^{0.1} \quad (5.9)$$

$$\text{Re}_l = \frac{G \cdot (1-x) \cdot D_i}{\mu_l} \quad (5.10)$$

Figure 5.14 shows that more than 65% of the present data for h_r fall within $\pm 25\%$ of Eq. (5-4), and the mean deviation between the present data for h_r and the proposed correlation is about 18%.

5-4 Concluding Remarks

Experiments have been conducted here to investigate the evaporation heat transfer of R-407C in the horizontal small circular tubes. The effects of the refrigerant mass flux, imposed heat flux, saturation temperature and vapor quality of R-407C on the evaporation heat transfer coefficient were examined in detail. The results show that the R-407C evaporation heat transfer coefficient increases linearly with the vapor quality and the increase are significant except at low imposed heat flux, refrigerant mass flux and saturated temperature. Moreover, the increase of the R-407C evaporation heat transfer coefficient with the imposed heat flux, refrigerant mass flux and saturated temperature is also substantial. An empirical correlation modified slightly from that of Gungor and Winterton [50] is proposed here to correlate our data.

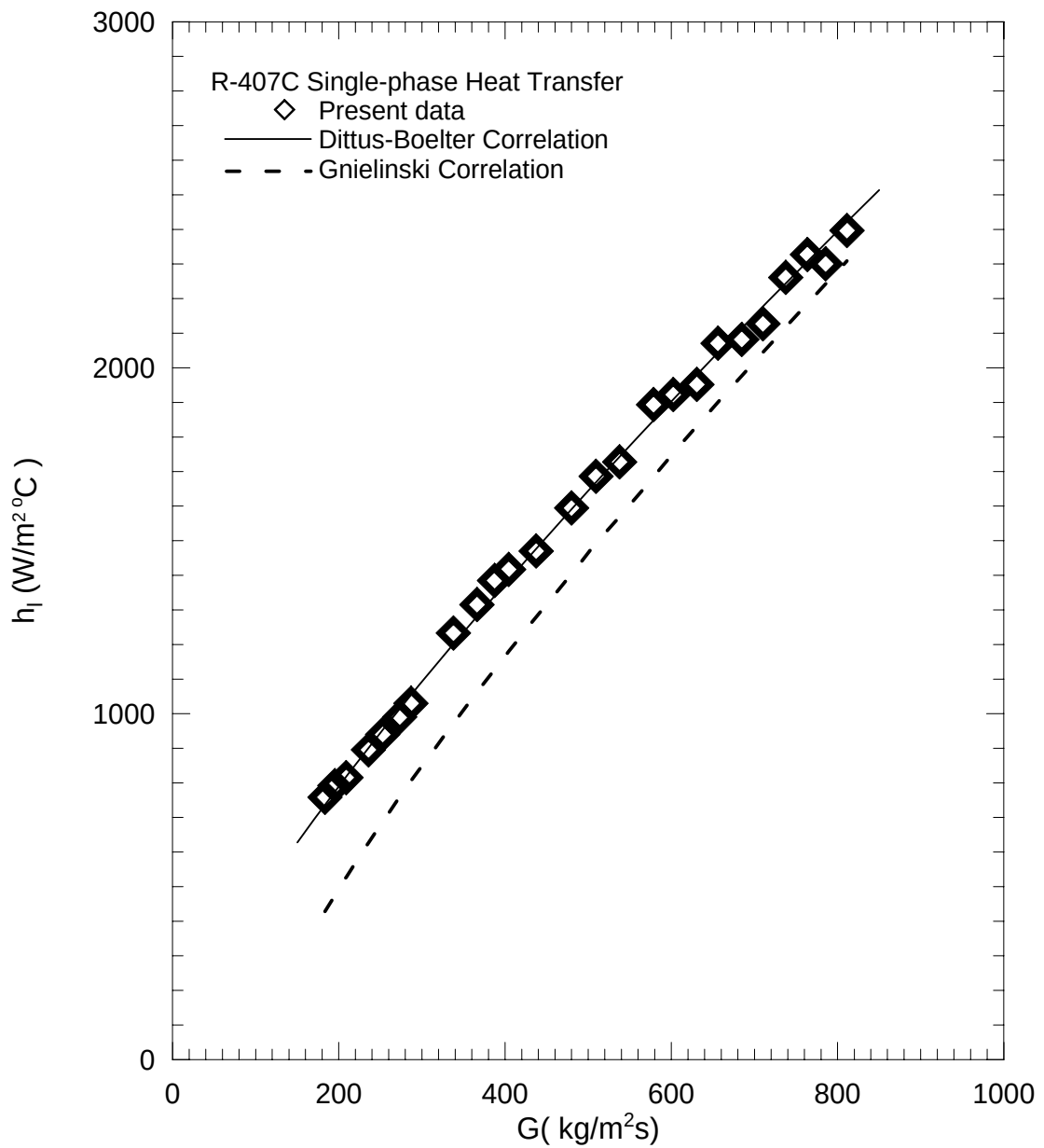


Fig. 5.1 Comparison of the present data for the liquid phase heat transfer coefficient with the Dittus-Boelter and Gnielinski correlations.

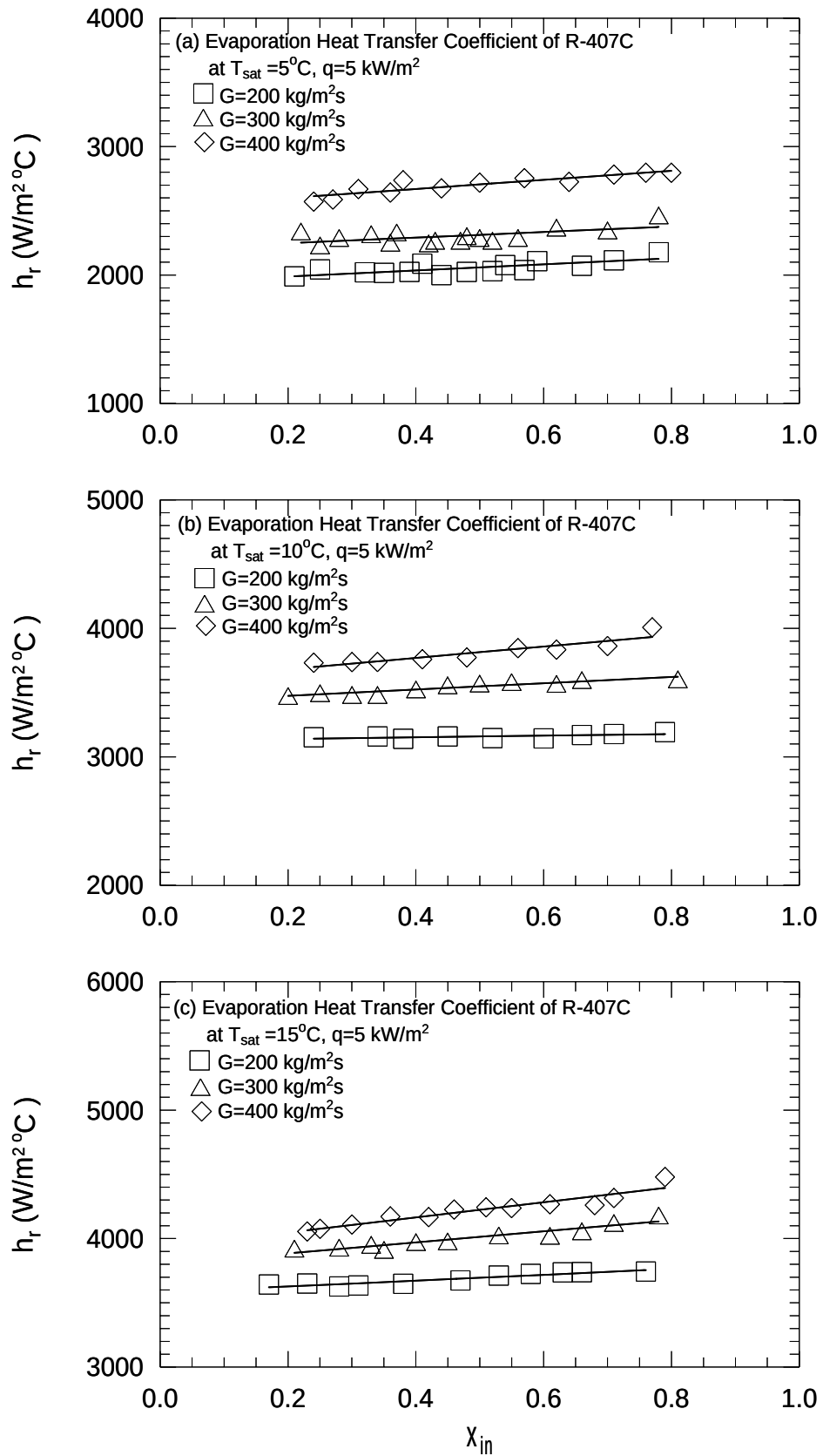


Fig. 5.2 Variations of evaporation heat transfer coefficient with inlet vapor quality at $q = 5 \text{ kW/m}^2$ for various mass fluxes for (a) $T_{\text{sat}} = 5^\circ\text{C}$, (b) $T_{\text{sat}} = 10^\circ\text{C}$ and (c) $T_{\text{sat}} = 15^\circ\text{C}$.

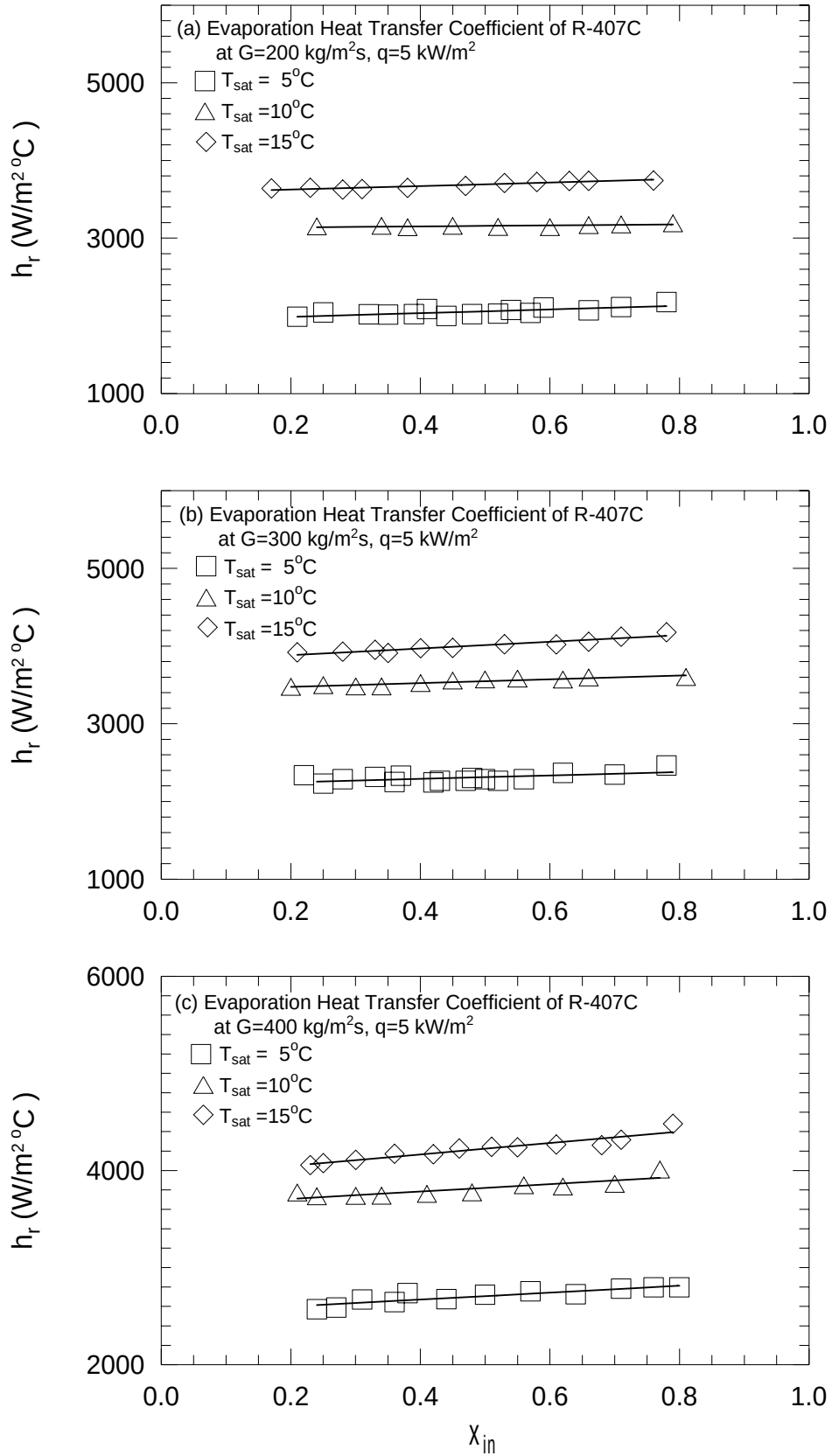


Fig. 5.3 Variations of evaporation heat transfer coefficient with inlet vapor quality at $q=5 \text{ kW/m}^2$ for various saturated temperatures for (a) $G= 200 \text{ kg/m}^2\text{s}$, (b) $G= 300 \text{ kg/m}^2\text{s}$ and (c) $G= 400 \text{ kg/m}^2\text{s}$.

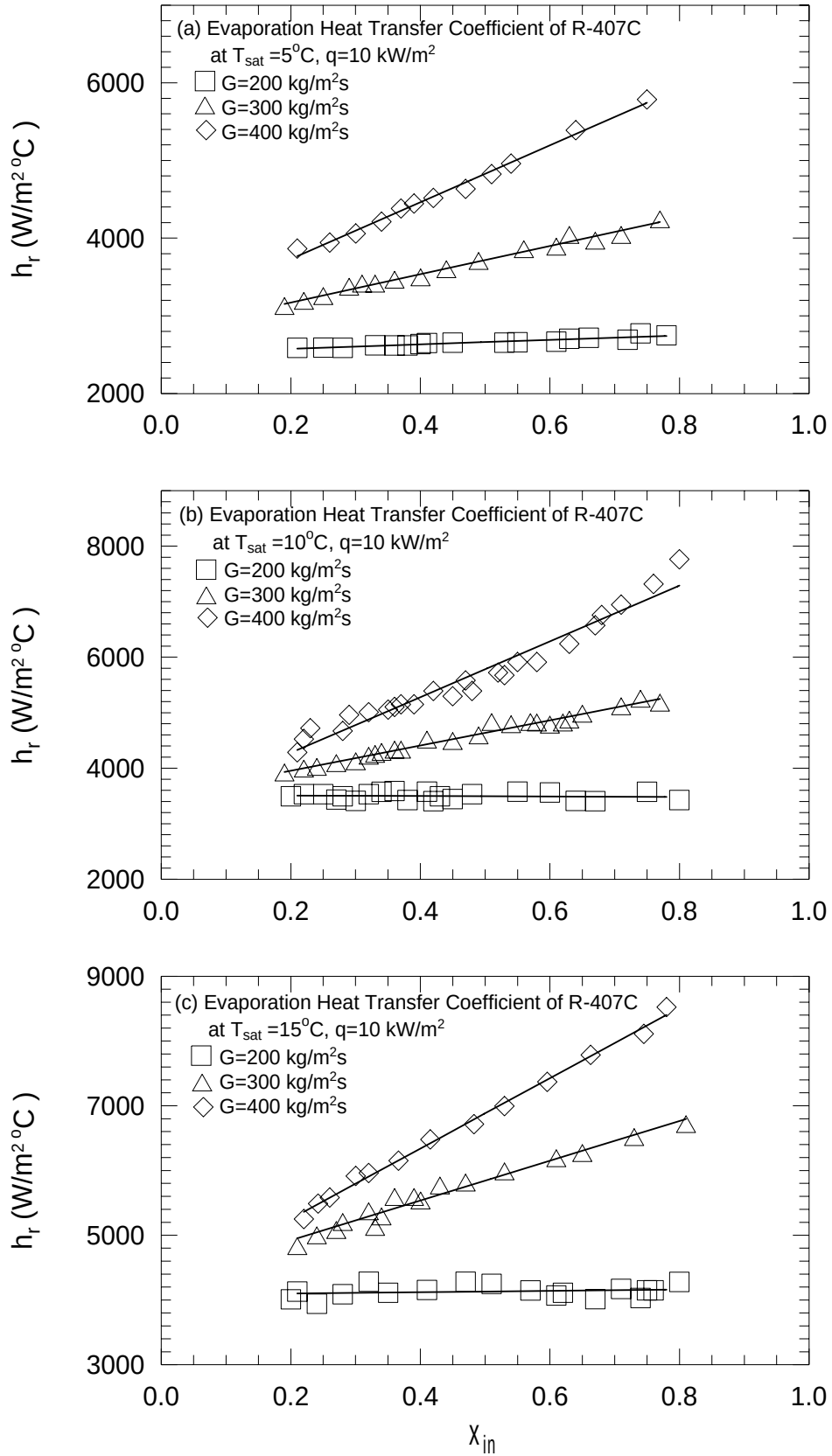


Fig. 5.4 Variations of evaporation heat transfer coefficient with inlet vapor quality at $q = 10 \text{ kW/m}^2$ for various mass fluxes for (a) $T_{sat} = 5^\circ\text{C}$, (b) $T_{sat} = 10^\circ\text{C}$ and (c) $T_{sat} = 15^\circ\text{C}$.

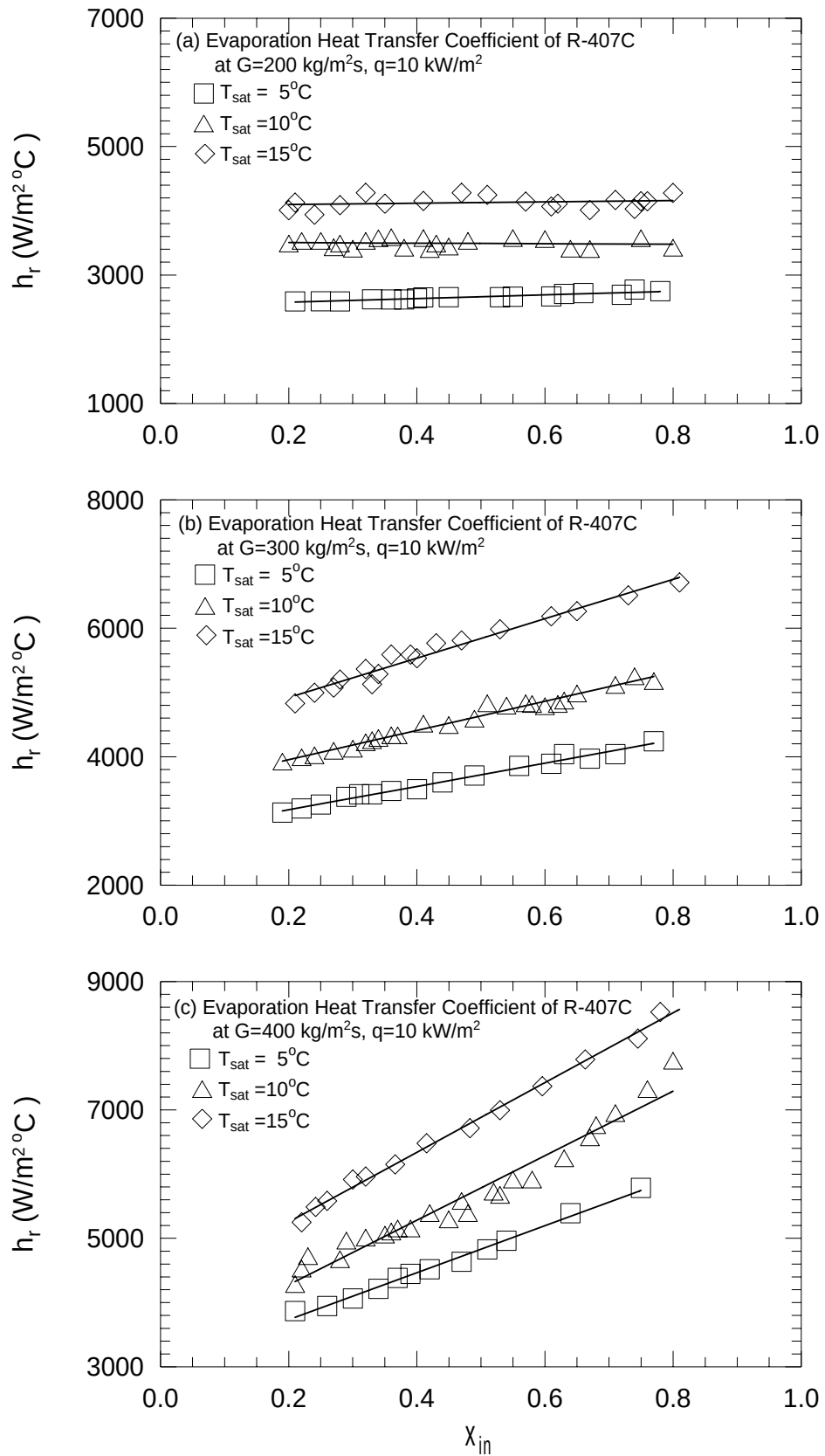


Fig. 5.5 Variations of evaporation heat transfer coefficient with inlet vapor quality at $q=10 \text{ kW/m}^2$ for various saturated temperatures for (a) $G= 200 \text{ kg/m}^2\text{s}$, (b) $G= 300 \text{ kg/m}^2\text{s}$ and (c) $G= 400 \text{ kg/m}^2\text{s}$.

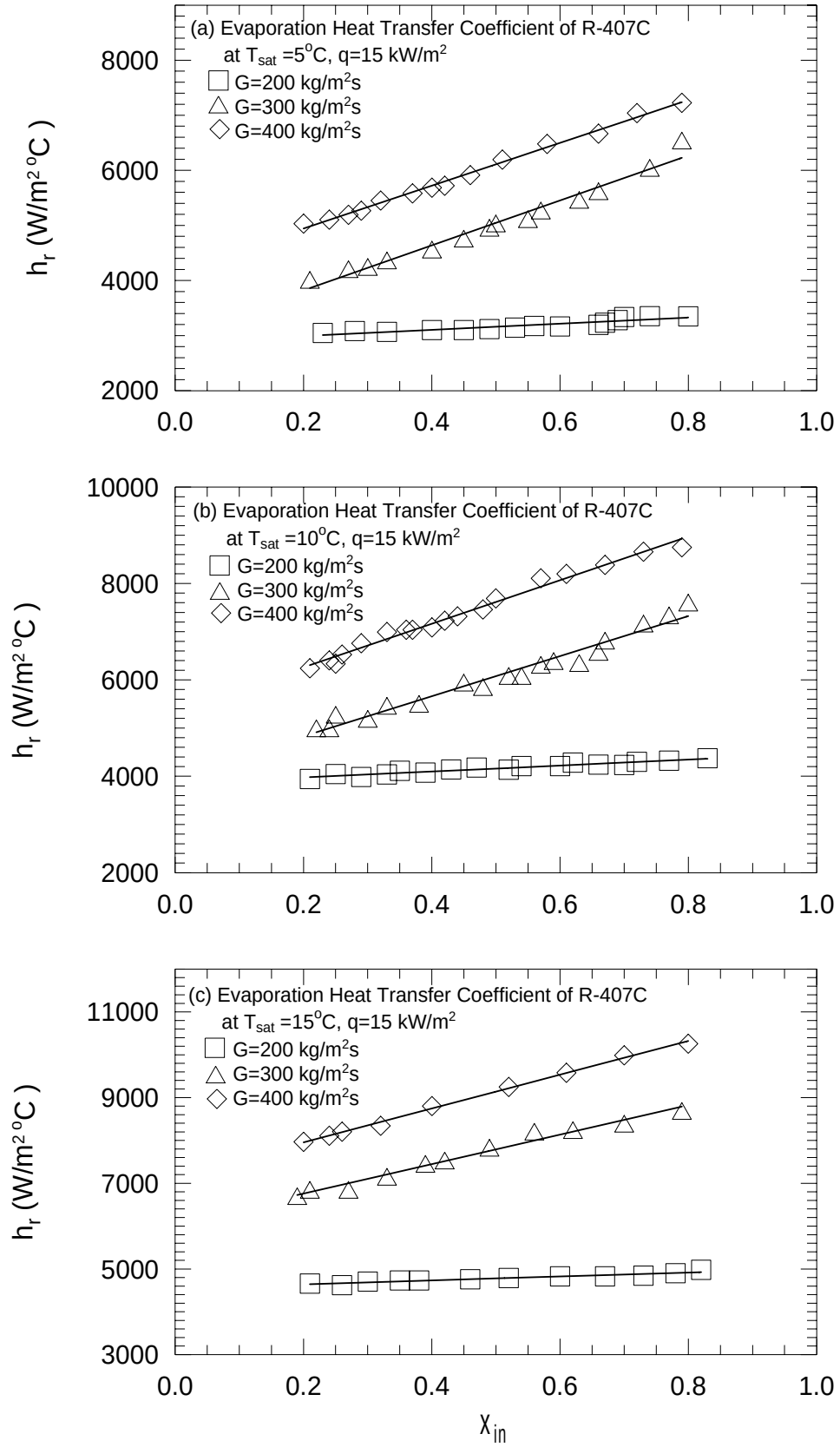


Fig. 5.6 Variations of evaporation heat transfer coefficient with inlet vapor quality at $q = 15 \text{ kW/m}^2$ for various mass fluxes for (a) $T_{sat} = 5^\circ\text{C}$, (b) $T_{sat} = 10^\circ\text{C}$ and (c) $T_{sat} = 15^\circ\text{C}$.

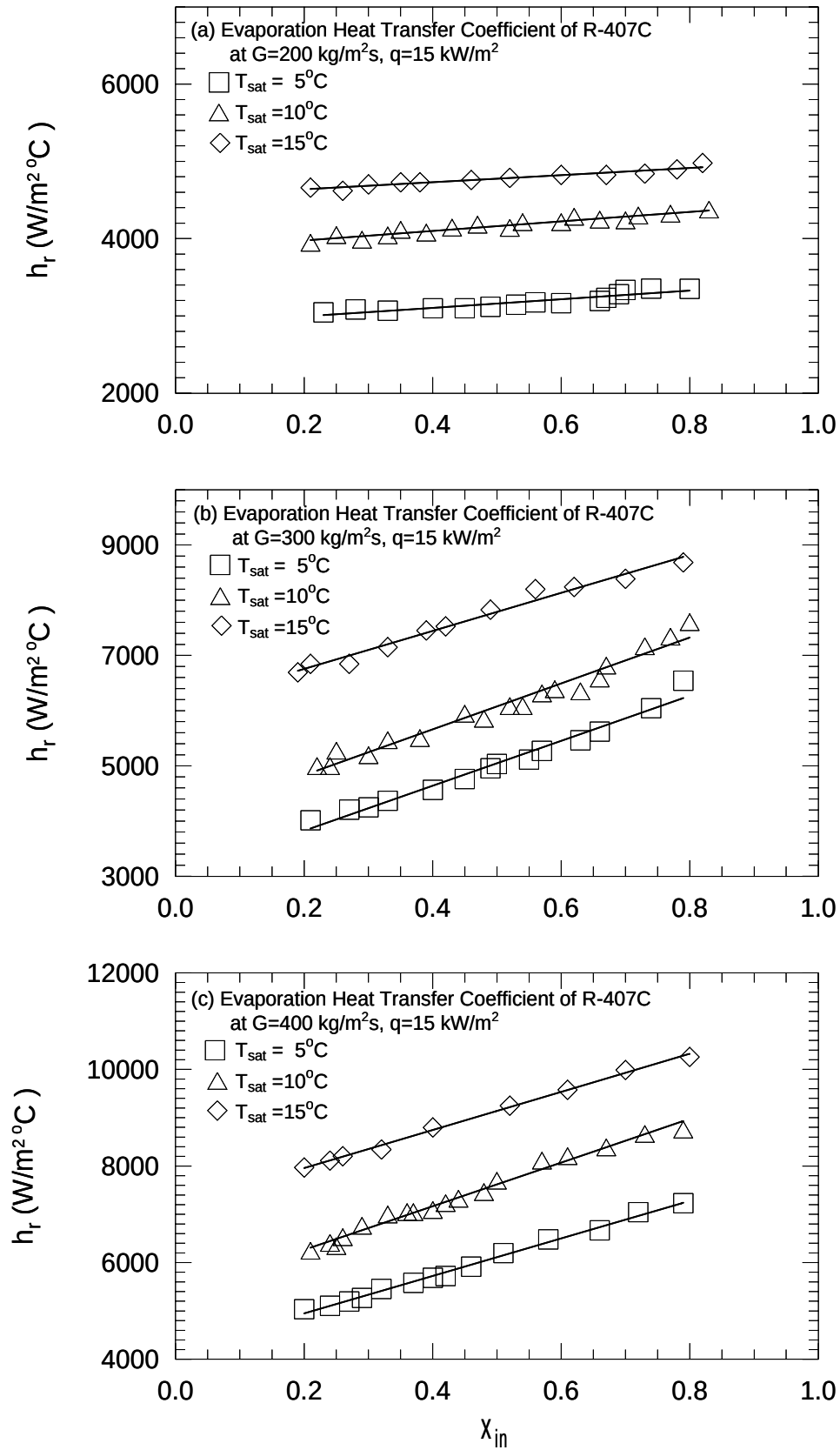


Fig. 5.7 Variations of evaporation heat transfer coefficient with inlet vapor quality at $q=15 \text{ kW/m}^2$ for various saturated temperatures for (a) $G= 200 \text{ kg/m}^2\text{s}$, (b) $G= 300 \text{ kg/m}^2\text{s}$ and (c) $G= 400 \text{ kg/m}^2\text{s}$.

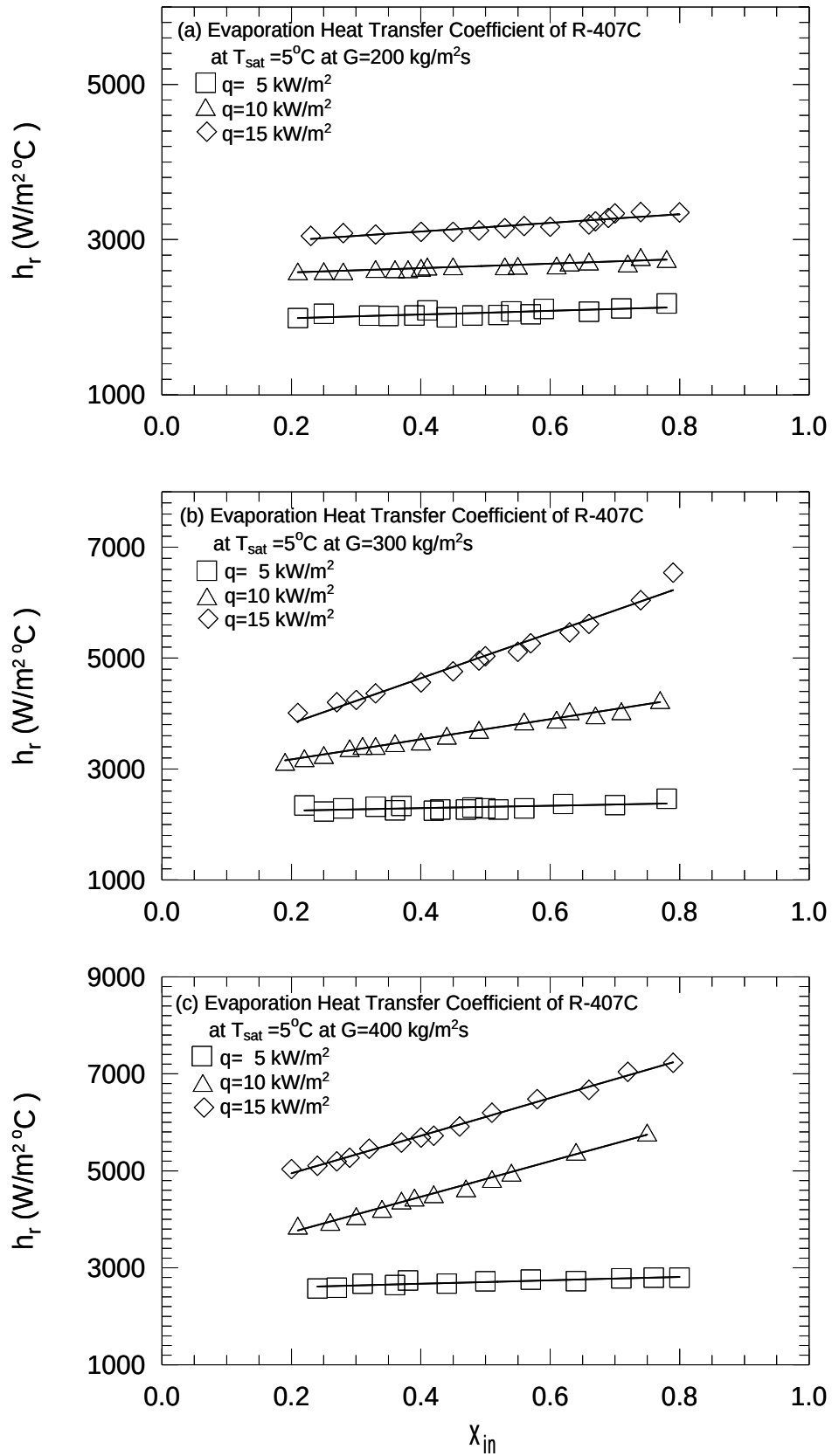


Fig. 5.8 Variations of evaporation heat transfer coefficient with inlet vapor quality at $T_{\text{sat}} = 5^\circ\text{C}$ for various heat fluxes for (a) $G = 200 \text{ kg/m}^2\text{s}$, (b) $G = 300 \text{ kg/m}^2\text{s}$ and (c) $G = 400 \text{ kg/m}^2\text{s}$.

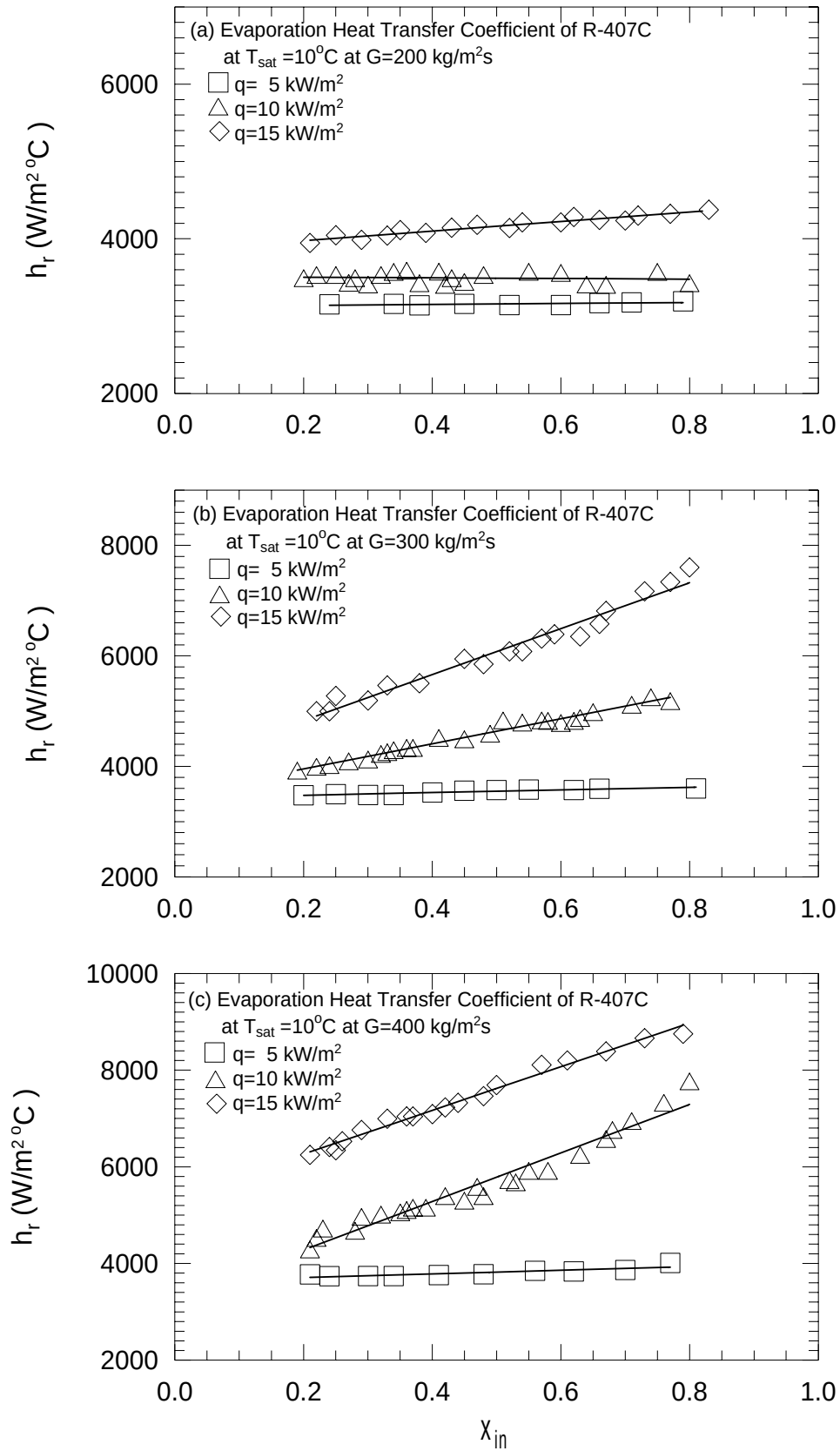


Fig. 5.9 Variations of evaporation heat transfer coefficient with inlet vapor quality at $T_{\text{sat}} = 10^\circ\text{C}$ for various heat fluxes for (a) $G = 200 \text{ kg/m}^2\text{s}$, (b) $G = 300 \text{ kg/m}^2\text{s}$ and (c) $G = 400 \text{ kg/m}^2\text{s}$.

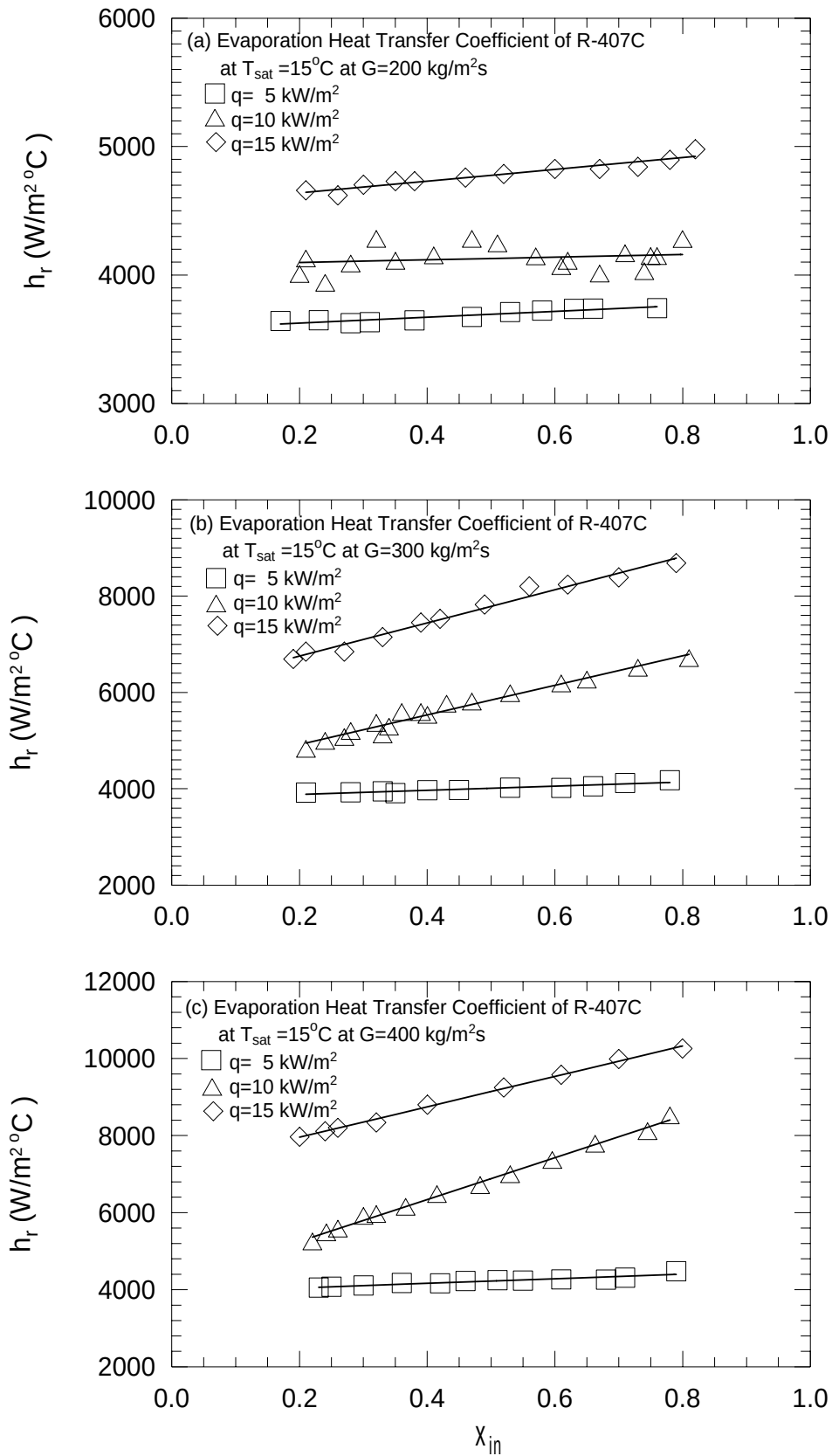


Fig. 5.10 Variations of evaporation heat transfer coefficient with inlet vapor quality at $T_{\text{sat}} = 15^\circ\text{C}$ for various heat fluxes for (a) $G = 200 \text{ kg/m}^2\text{s}$, (b) $G = 300 \text{ kg/m}^2\text{s}$ and (c) $G = 400 \text{ kg/m}^2\text{s}$.

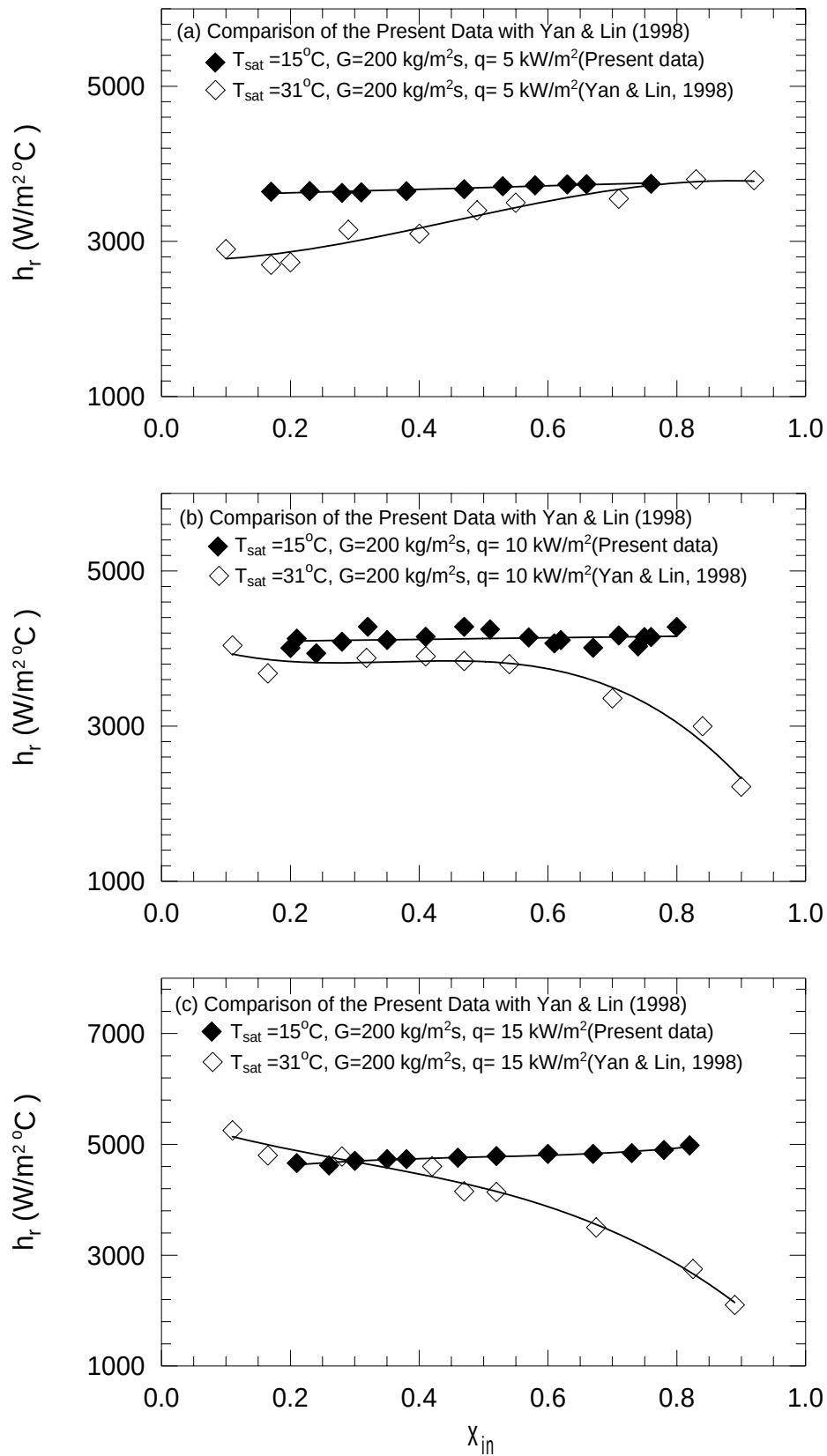


Fig. 5.11 Comparison of the R-407C evaporation heat transfer coefficient for small tubes with those for R-134a in the same small tubes from Yan and Lin (1998) for (a) $q = 5 \text{ kW/m}^2$, (b) $q = 10 \text{ kW/m}^2$ and (c) $q = 15 \text{ kW/m}^2$.

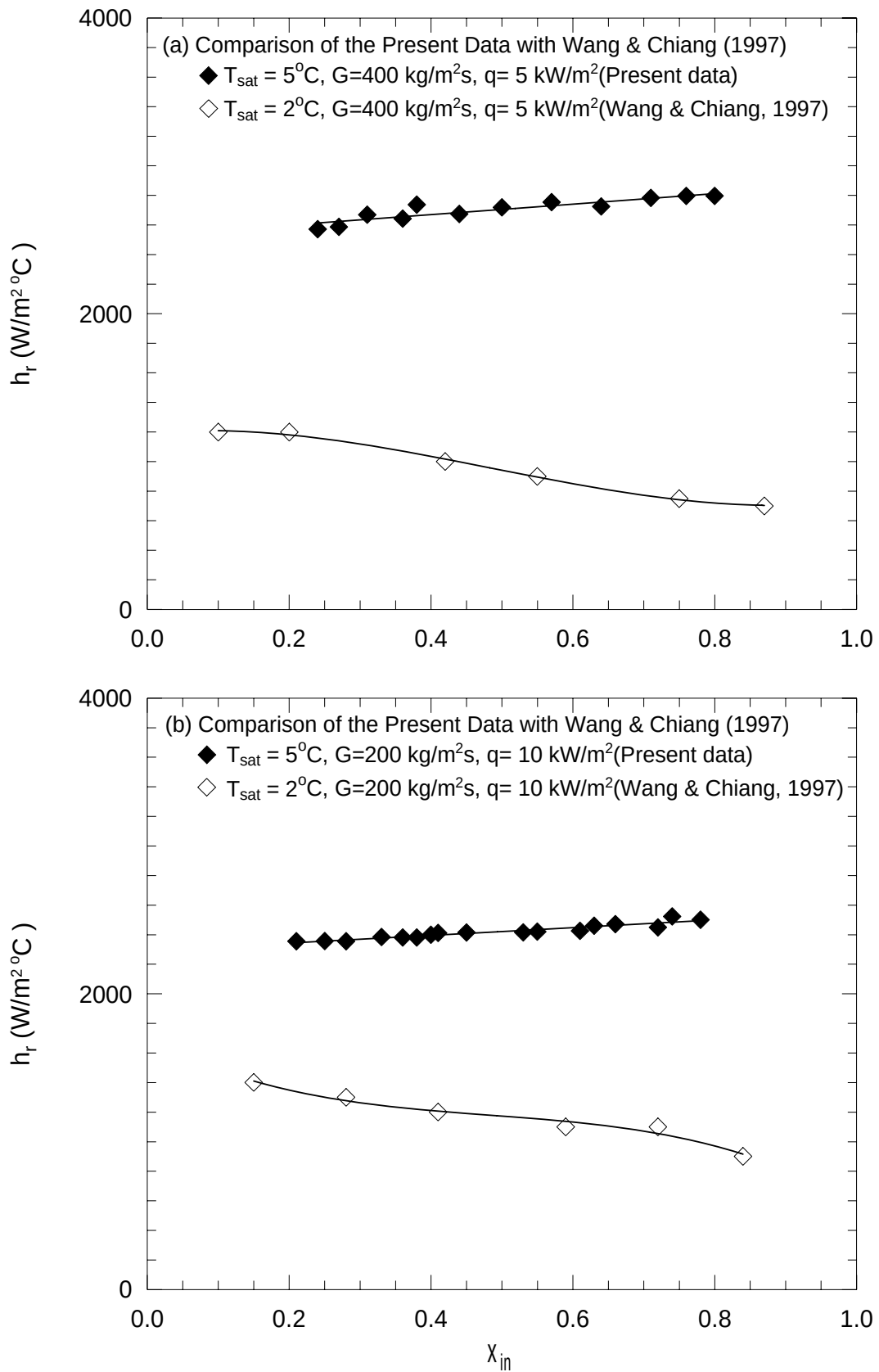


Fig. 5.12 Comparison of the R-407C evaporation heat transfer coefficient for small tubes ($T_{\text{sat}}=5^{\circ}\text{C}$) with those for smooth tube ($T_{\text{sat}}=2^{\circ}\text{C}$) from Wang and Chiang (1997) for (a) $q=5 \text{ kW/m}^2$ & $G= 400 \text{ kg/m}^2\text{s}$ and (b) $q=10 \text{ kW/m}^2$ & $G= 200 \text{ kg/m}^2\text{s}$.

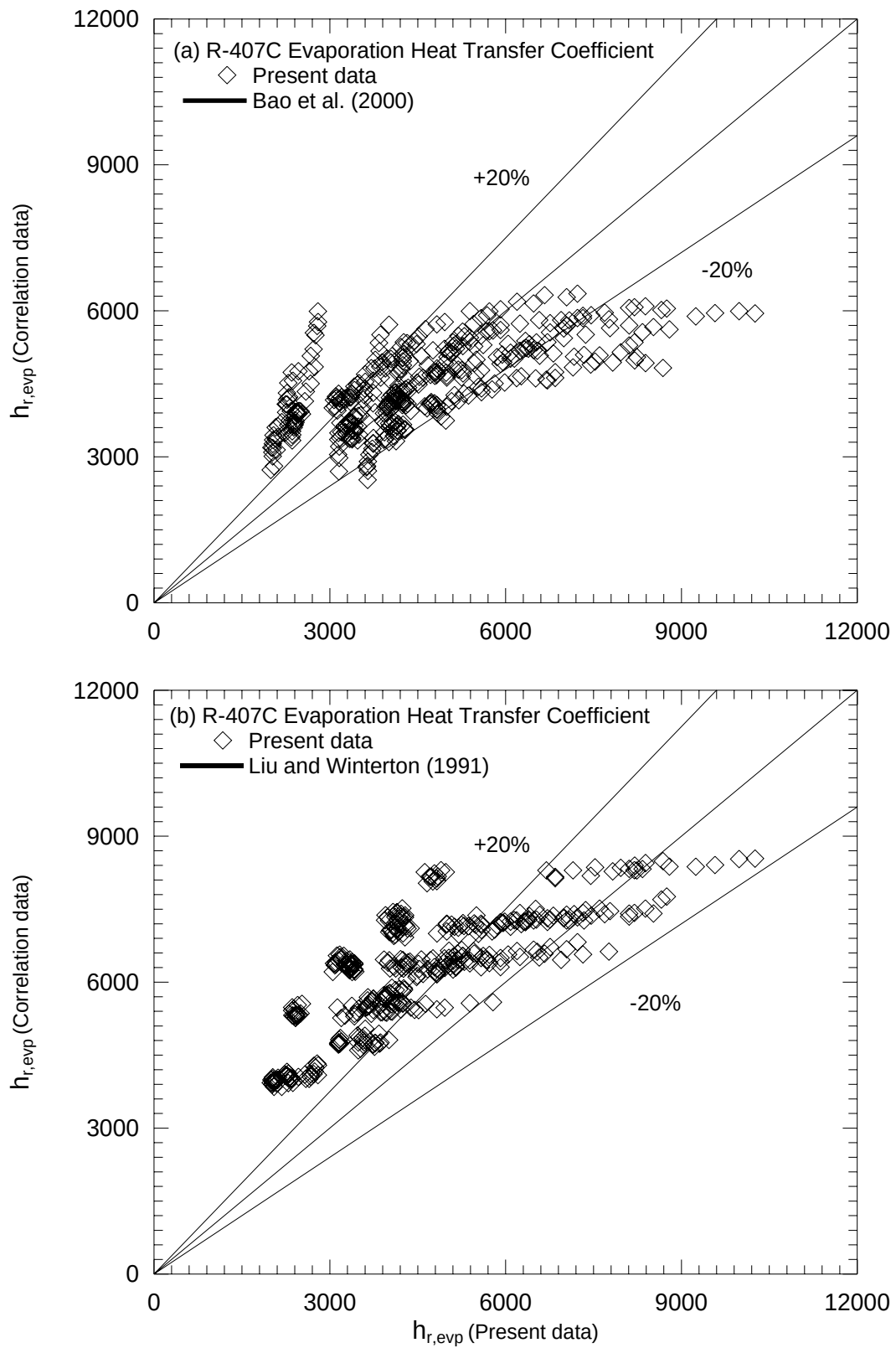


Fig. 5.13 Comparison of the measured data for heat transfer coefficient in the evaporation of R-407C with the proposed correlations from (a) Bao et al. (2000) and (b) Liu and Winterton (1991).

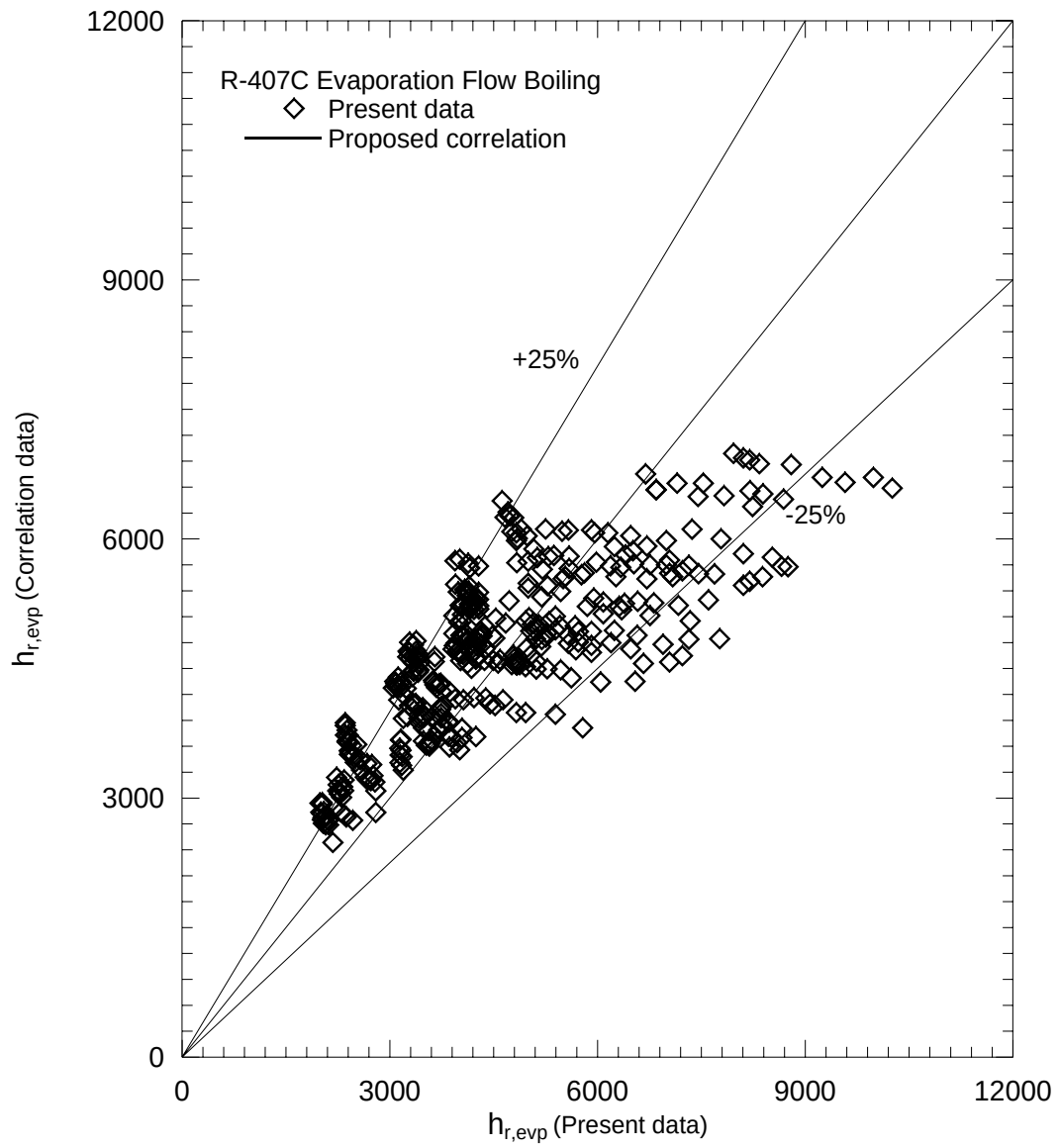


Fig. 5.14 Comparison of the measured data for heat transfer coefficient in the evaporation of R-407C with the proposed correlation.

CHAPTER 6

TWO-PHASE PRESSURE DROP

IN SMALL CIRCULAR TUBES

The measured data for the pressure drop of R-407C evaporation in the horizontal small circular tubes are presented in this chapter. Based on the estimation calculated from equations (3.15-3.17) the acceleration pressure drop and the pressure losses at the inlet and exit ports are rather small and the frictional pressure drop ranges from 94 % to 97 % of the total pressure drop measured. In what follows the effects of the imposed heat flux, refrigerant mass flux and system pressure on the frictional pressure drop are examined in detail. Besides, the frictional pressure drops of R-407C evaporation in the horizontal small circular tubes are also compared with those of Yan and Lin [12] for R-134a and the proposed correlation of Tran et al. [26].

6-1 Two-Phase Frictional Pressure Drop

The frictional pressure drops for the refrigerant R-407C evaporation in the small circular tubes under various flow and thermal conditions are presented in Figs. 6.1-6.9. First, Figs. 6.1-6.6 show the effects of the refrigerant mass flux and saturated temperature on the frictional pressure drop of the R-407C. The results in Figs. 6.1-6.6 indicate that at a given mass flux the frictional pressure drop increases significantly with the inlet vapor quality. The increase is almost linear. In addition, the frictional pressure drop increase with the inlet vapor quality is larger for a higher mass flux. Moreover, a substantial increase in ΔP_f results for the increase of the refrigerant mass flux. For instance in Fig. 6.1(c), at the high refrigerant mass flux of $G = 400 \text{ kg/m}^2\text{s}$ the frictional pressure drop is about 60% and 200% on average higher than those for

$G=300 \text{ kg/m}^2\text{s}$ and $200 \text{ kg/m}^2\text{s}$, respectively. The higher frictional pressure drop for a high mass flux is attributed to the fact that both the liquid and vapor speeds in the refrigerant flow are directly proportional to the refrigerant mass flux. We also note that an increase in the refrigerant saturated temperature causes a large reduction in the frictional pressure drops (Figs. 6.2, 6.4 and 6.6). It is clear from the results in Fig. 6.2(c) that at $G = 400 \text{ kg/m}^2\text{s}$ and $q = 5 \text{ kW/m}^2$ the frictional pressure drop at the saturated temperature of 5°C is about 44 % and 60 % higher than those at $T_{\text{sat}} = 10^\circ\text{C}$ and 15°C , respectively. The higher ΔP_f for a lower refrigerant saturated temperature is attributed to the fact that for a lower saturated temperature (that is for lower system pressure) the specific volume of the R-407C vapor is larger and the viscosity of the liquid R-407C is higher.

It is of interest to note that the frictional pressure drop is not affected to a noticeable degree by the imposed heat flux, as is evident from the results in Figs. 6.7-6.9. For example in Fig. 6.7(b), at $G = 300 \text{ kg/m}^2\text{s}$ and $T_{\text{sat}} = 5^\circ\text{C}$ the frictional pressure drop at the given heat flux of $q = 5 \text{ kW/m}^2$ is only about 6.1 % and 6.8 % higher than those at $q = 10$ and 15 kW/m^2 , respectively. This change of ΔP_f with the imposed heat flux is within the experimental uncertainty. We further note that for most cases the rise in the evaporation heat transfer coefficient with the imposed heat flux is more prominent than that in the frictional pressure drop.

The comparison of the present data of ΔP_f for the R-407C evaporation in the small circular tubes with those for R-134a in the same tubes is shown in Fig. 6.10. The results indicate that at $x_{in} > 0.4$ the frictional pressure drop of the R-407C is higher than that of R-134a. But at $x_{in} < 0.4$ the opposite is the case. Moreover, the comparison of the present data for the frictional pressure drop for the evaporation of R-407C with the correlation proposed by Tran et al. [26] is shown in Fig. 6.11. The results show that most of the present data fall within $\pm 30\%$ of their correlation.

6-2 Correlation Equations

Correlation equations for the frictional pressure drop associated with the R-407C evaporation in the small circular tubes are important for thermal design of the evaporators in various air conditioning and refrigeration systems. Based on the present data, an empirical correlation for the dimensionless frictional pressure drop is proposed in terms of the friction factor. It is expressed as

$$f_{tp} = 1100 \cdot \text{Re}_{eq}^{-1.125} \quad (6.1)$$

where Re_{eq} is the equivalent Reynolds number for the evaporating flow and is defined as

$$\text{Re}_{eq} = \frac{G_{eq} \cdot D_i}{\mu_f} \quad (6.2)$$

in which

$$G_{eq} = G \cdot \left[(1 - x_{in}) + x_{in} \cdot \left(\frac{\rho_f}{\rho_g} \right)^{0.5} \right] \quad (6.3)$$

Here G_{eq} is an equivalent mass flux which is a function of the R-407C mass flux, mean vapor quality and density at the test condition. Besides, x_{in} is the inlet vapor quality of the flow. Figure 6.12 shows that more than 63% of the present data for f_{tp} fall within $\pm 25\%$ of Eq. (6-1), and the mean deviation between the present data for f_{tp} and the proposed correlation is about 19%.

6-3 Concluding Remarks

Experimental measurement has been carried out here to investigate the frictional pressure drop of R-407C evaporation in the small circular tubes. Results for the frictional pressure drop have been examined in detail. The results reveal that the increase of the R-407C frictional pressure drop with the refrigerant vapor quality is

almost linear and the increase is larger for a higher refrigerant mass flux. Moreover, a larger R-407C frictional pressure drop results for a higher refrigerant mass flux. But an opposite trend is noted for a rise in the refrigerant saturated temperature. Furthermore, the imposed heat flux exhibits a negligible effect on the frictional pressure drop. Finally, an empirical correlation is proposed to correlate the present data for the frictional pressure drop.

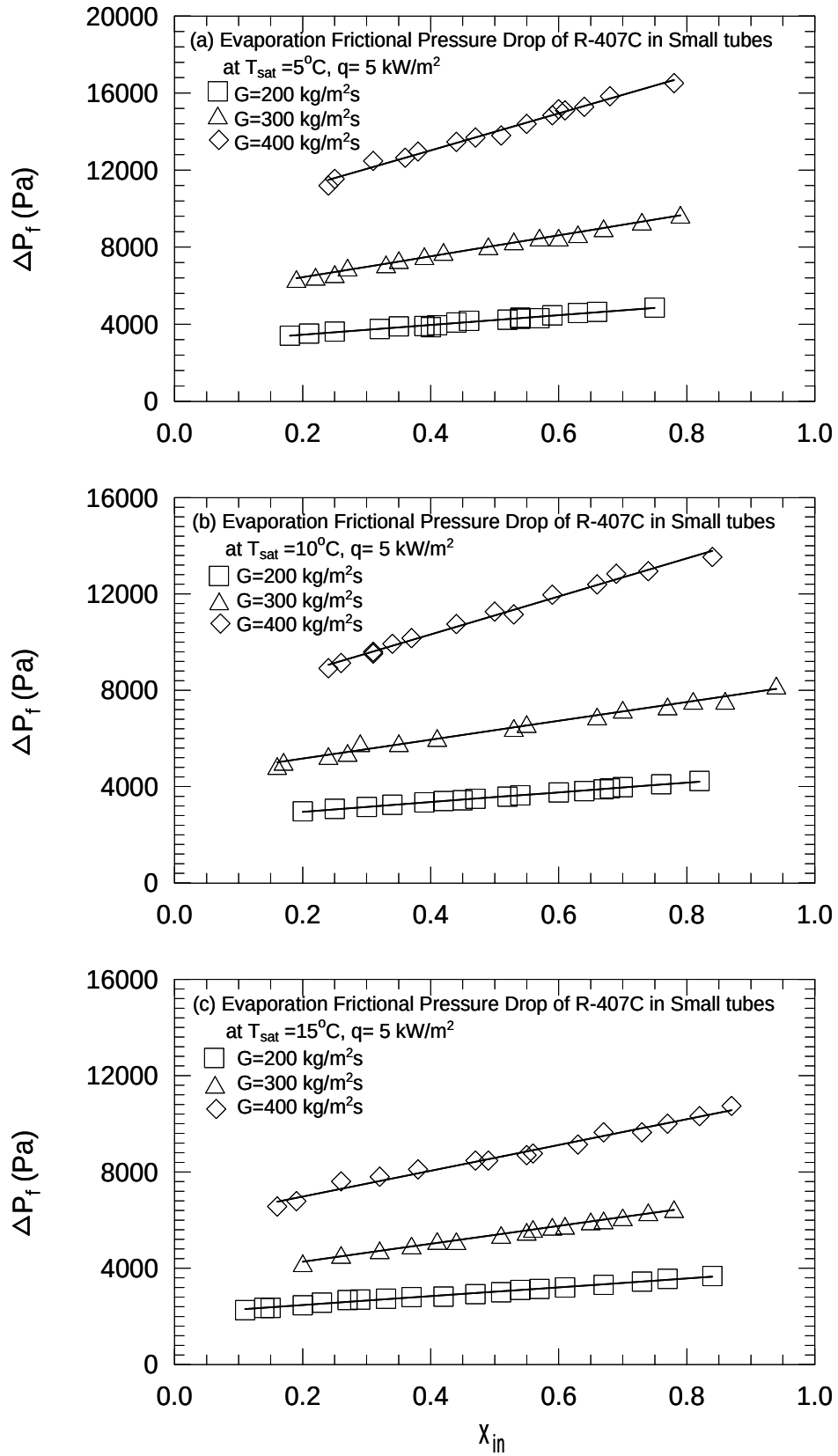


Fig. 6.1 Variations of frictional pressure drop with inlet vapor quality at $q = 5 \text{ kW/m}^2$ for various mass fluxes for (a) $T_{\text{sat}} = 5^\circ\text{C}$, (b) $T_{\text{sat}} = 10^\circ\text{C}$ and (c) $T_{\text{sat}} = 15^\circ\text{C}$.

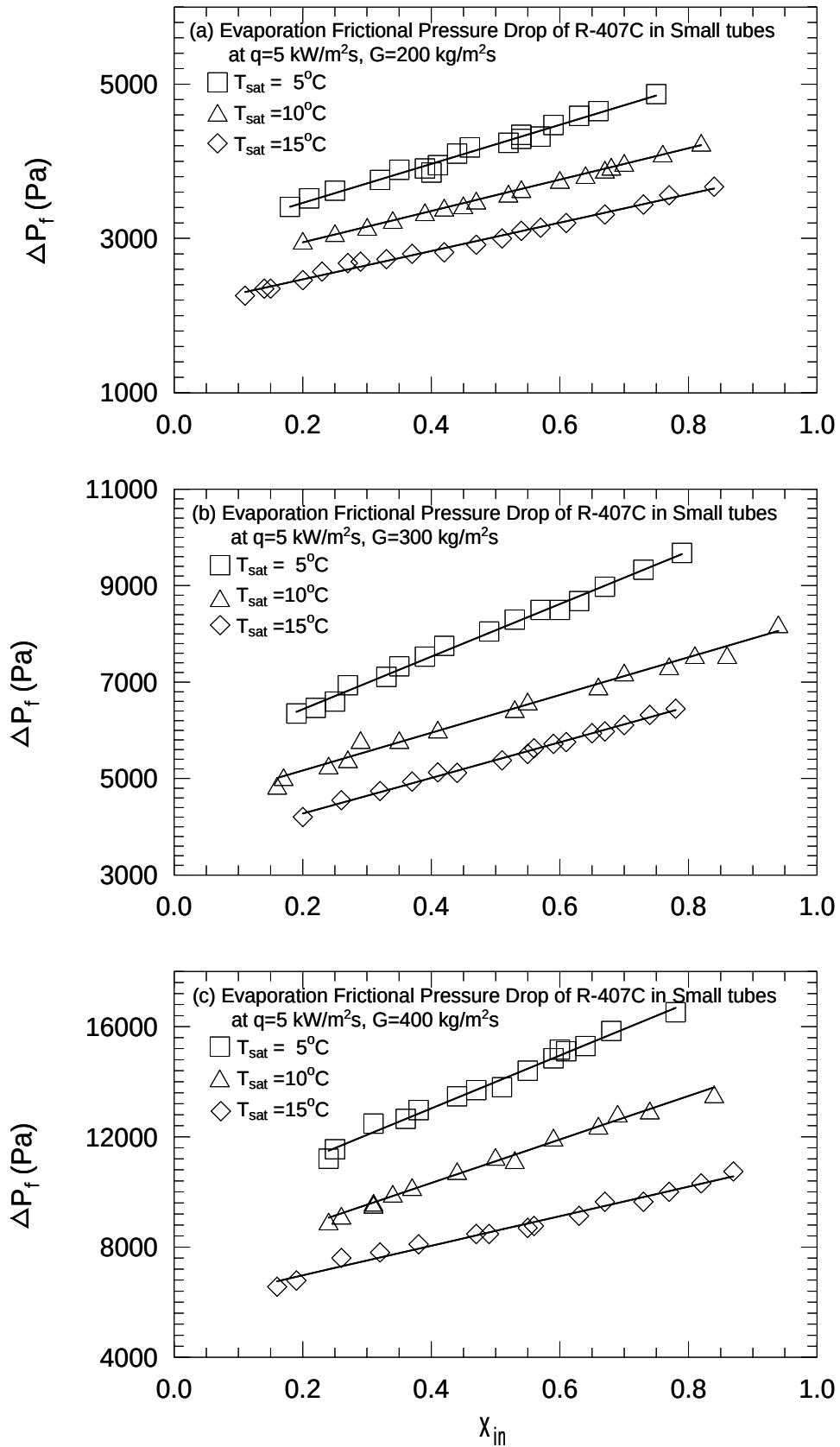


Fig. 6.2 Variations of frictional pressure drop with inlet vapor quality at $q=5 \text{ kW/m}^2$ for various saturated temperatures for (a) $G= 200 \text{ kg/m}^2\text{s}$, (b) $G= 300 \text{ kg/m}^2\text{s}$ and (c) $G= 400 \text{ kg/m}^2\text{s}$.

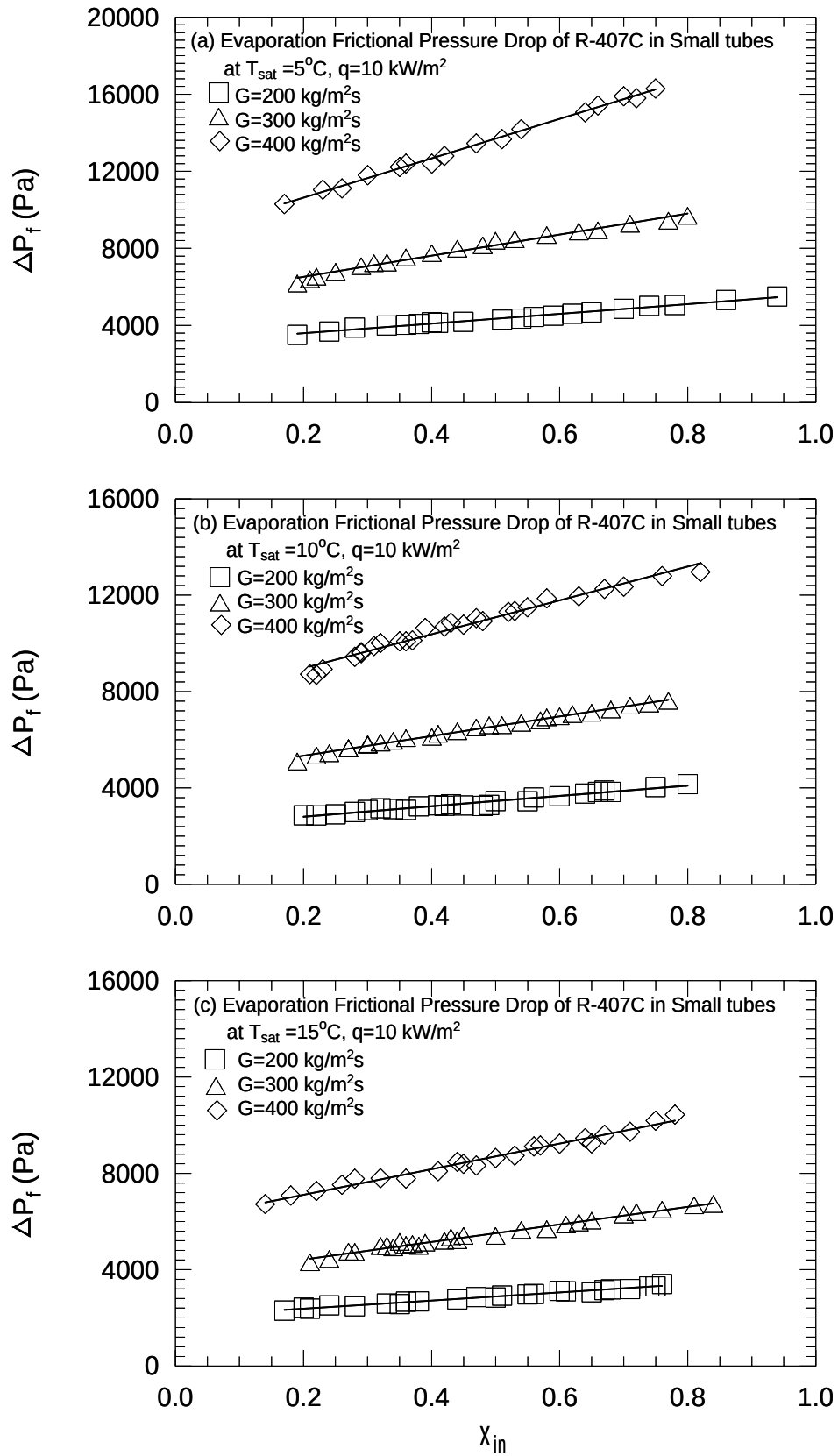


Fig. 6.3 Variations of frictional pressure drop with inlet vapor quality at $q = 10 \text{ kW/m}^2$ for various mass fluxes for (a) $T_{\text{sat}} = 5^\circ\text{C}$, (b) $T_{\text{sat}} = 10^\circ\text{C}$ and (c) $T_{\text{sat}} = 15^\circ\text{C}$.

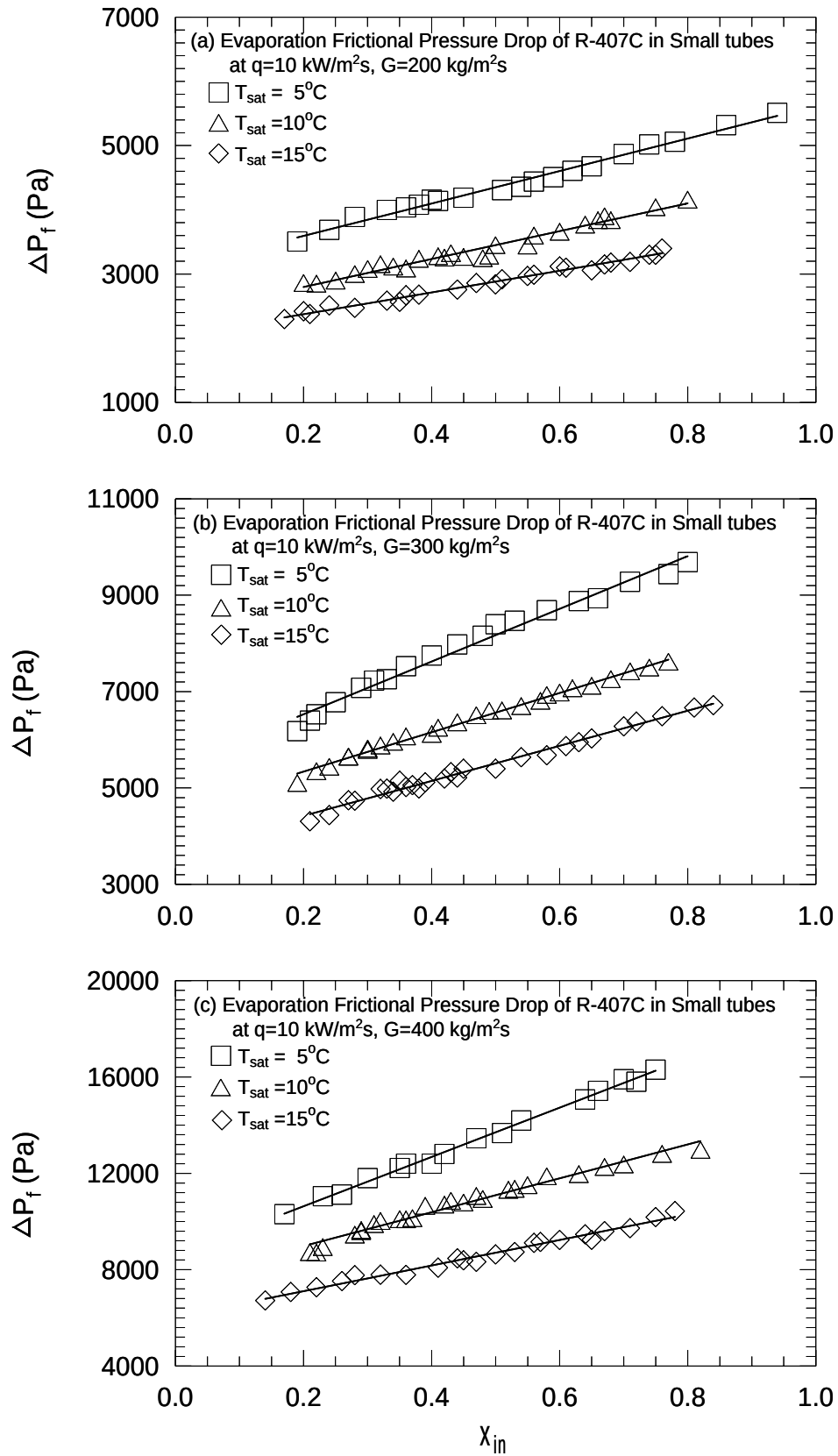


Fig. 6.4 Variations of frictional pressure drop with inlet vapor quality at $q=10 \text{ kW/m}^2$ for various saturated temperatures for (a) $G= 200 \text{ kg/m}^2\text{s}$, (b) $G= 300 \text{ kg/m}^2\text{s}$ and (c) $G= 400 \text{ kg/m}^2\text{s}$.

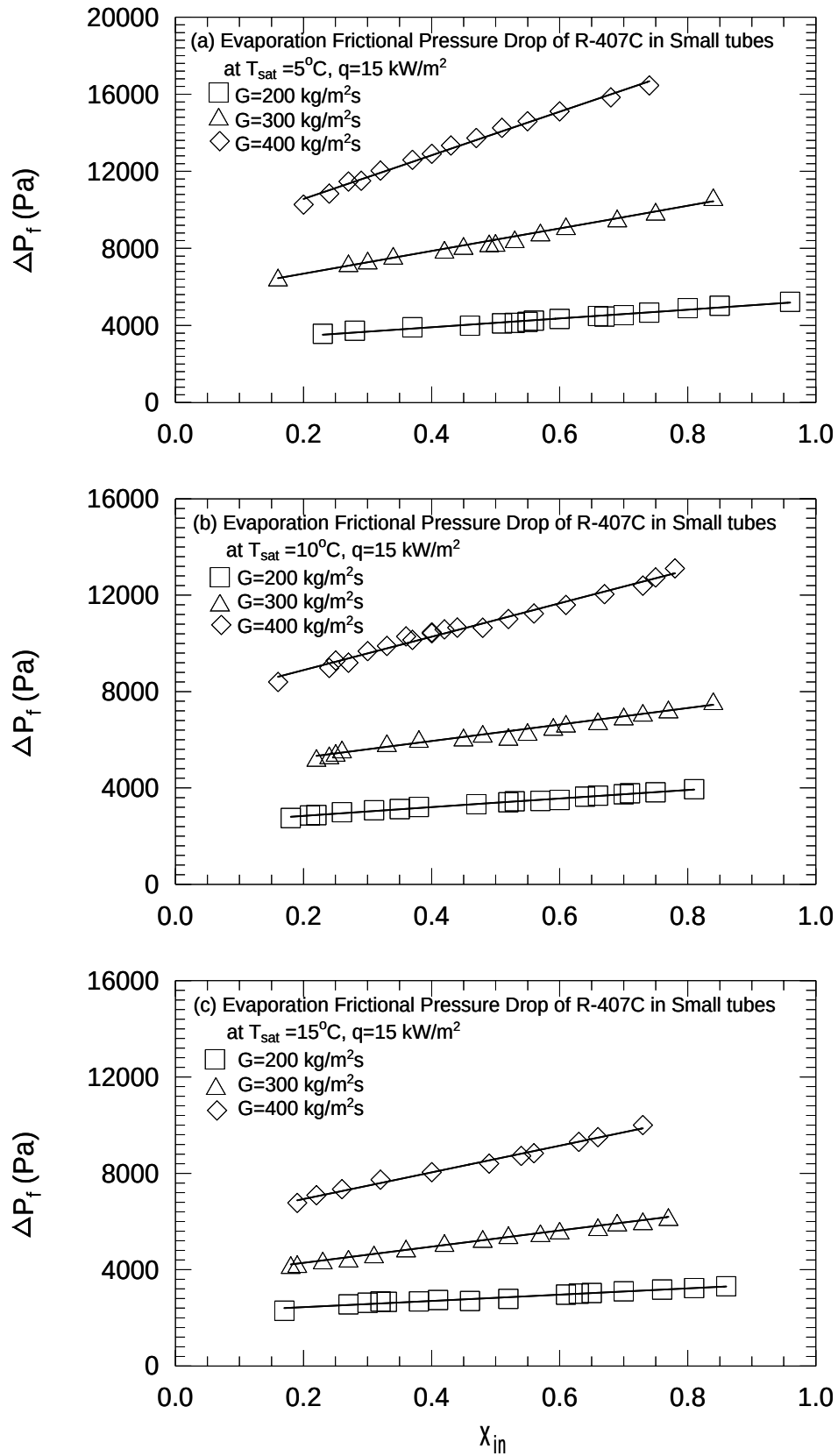


Fig. 6.5 Variations of frictional pressure drop with inlet vapor quality at $q = 15 \text{ kW/m}^2$ for various mass fluxes for (a) $T_{\text{sat}} = 5^\circ\text{C}$, (b) $T_{\text{sat}} = 10^\circ\text{C}$ and (c) $T_{\text{sat}} = 15^\circ\text{C}$.

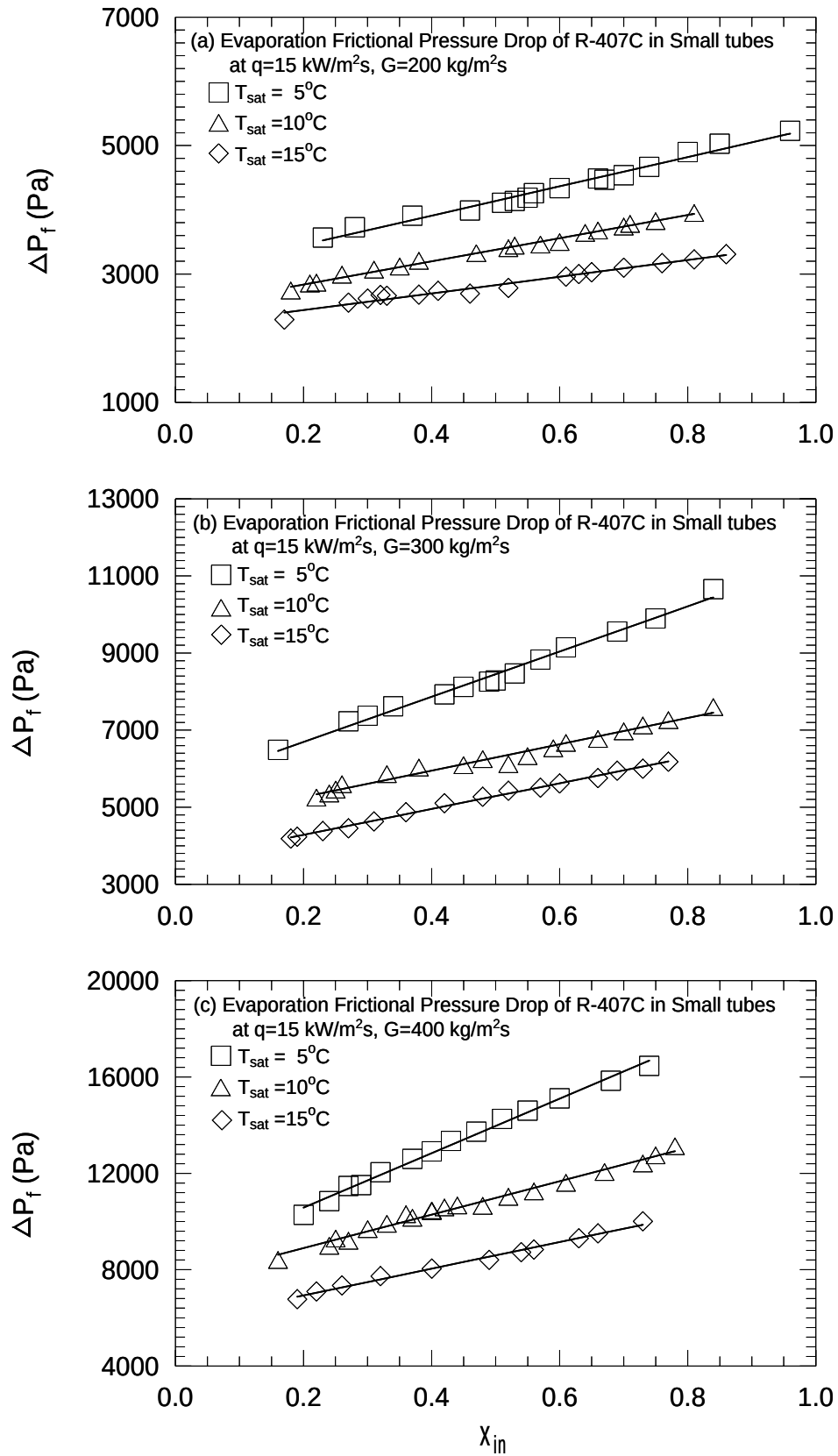


Fig. 6.6 Variations of frictional pressure drop with inlet vapor quality at $q=15 \text{ kW/m}^2$ for various saturated temperatures for (a) $G= 200 \text{ kg/m}^2\text{s}$, (b) $G= 300 \text{ kg/m}^2\text{s}$ and (c) $G= 400 \text{ kg/m}^2\text{s}$.

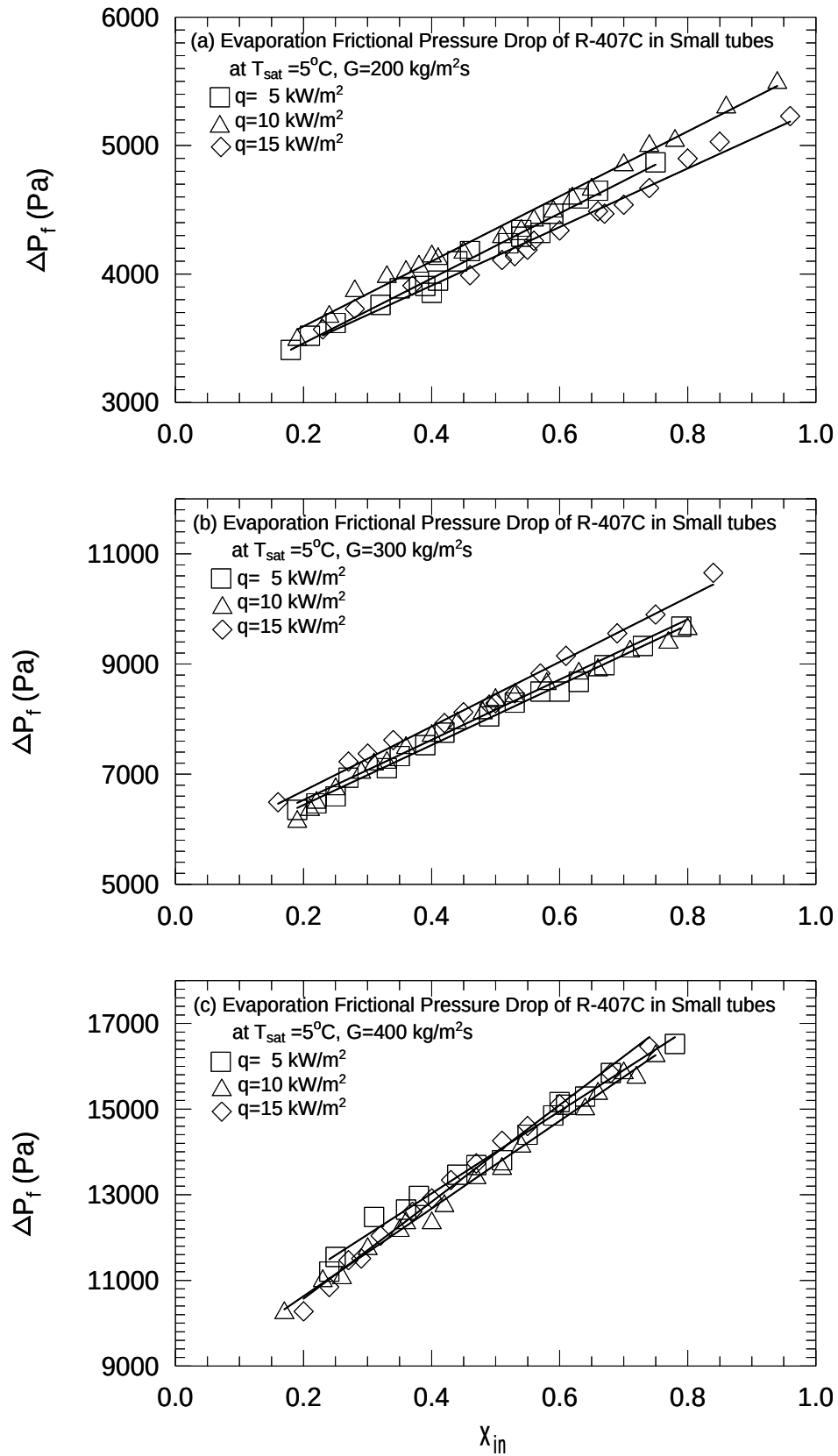


Fig. 6.7 Variations of frictional pressure drop with inlet vapor quality at $T_{\text{sat}} = 5^{\circ}\text{C}$ for various heat fluxes for (a) $G = 200 \text{ kg/m}^2\text{s}$, (b) $G = 300 \text{ kg/m}^2\text{s}$ and (c) $G = 400 \text{ kg/m}^2\text{s}$.

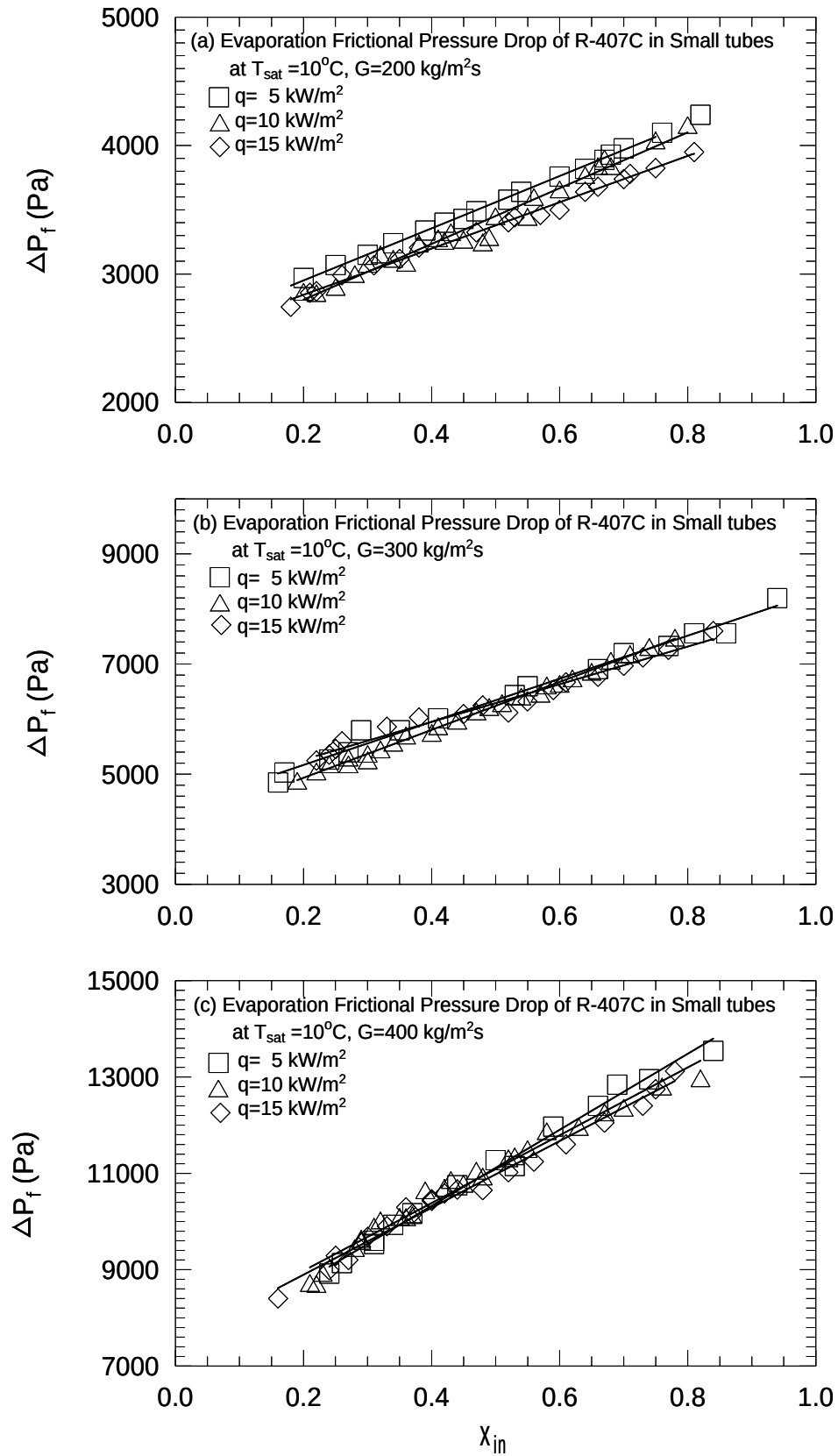


Fig. 6.8 Variations of frictional pressure drop with inlet vapor quality at $T_{\text{sat}} = 10^\circ\text{C}$ for various heat fluxes for (a) $G = 200 \text{ kg/m}^2\text{s}$, (b) $G = 300 \text{ kg/m}^2\text{s}$ and (c) $G = 400 \text{ kg/m}^2\text{s}$.

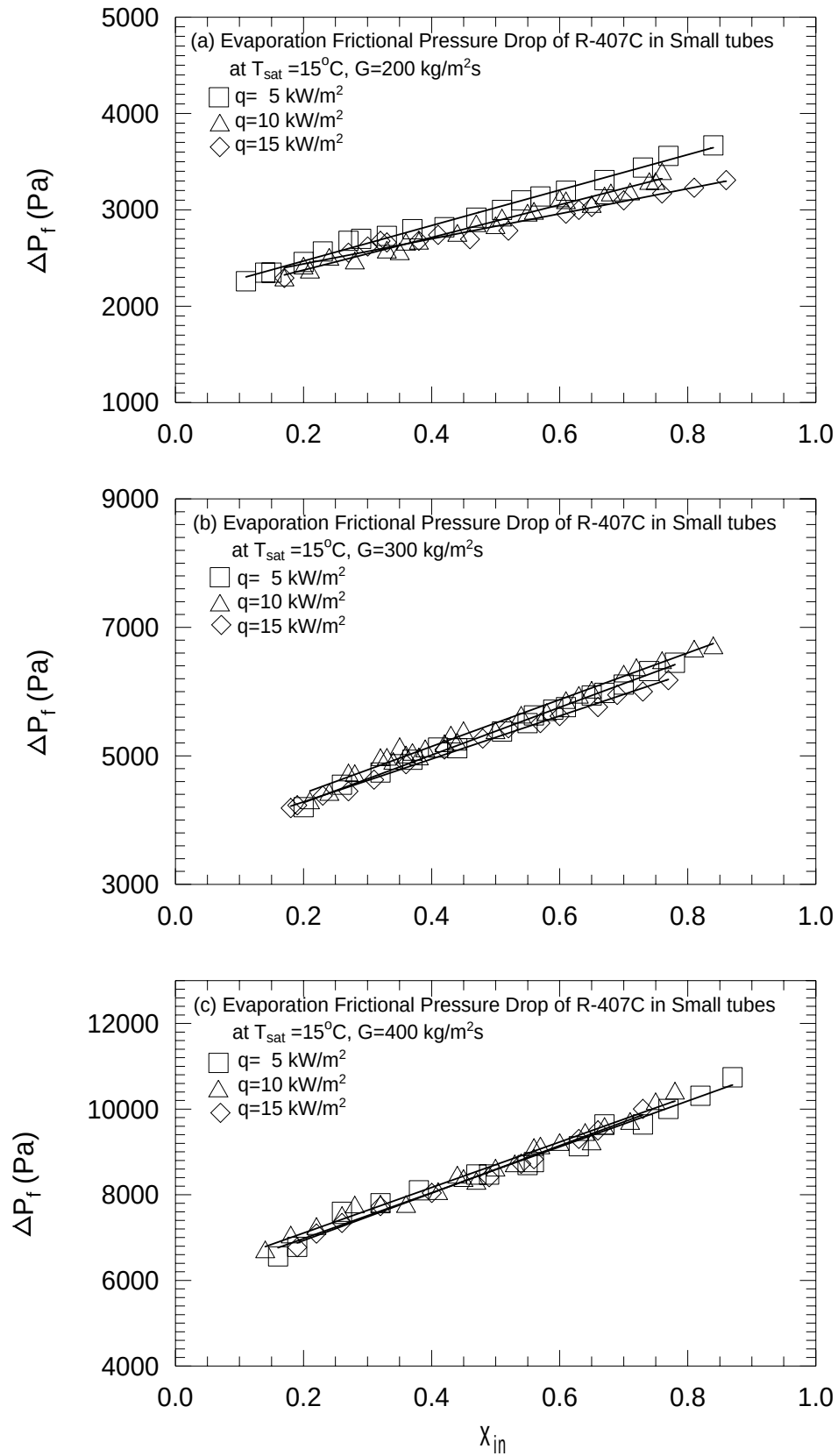


Fig. 6.9 Variations of frictional pressure drop with inlet vapor quality at $T_{\text{sat}} = 15^\circ\text{C}$ for various heat fluxes for (a) $G = 200 \text{ kg/m}^2\text{s}$, (b) $G = 300 \text{ kg/m}^2\text{s}$ and (c) $G = 400 \text{ kg/m}^2\text{s}$.

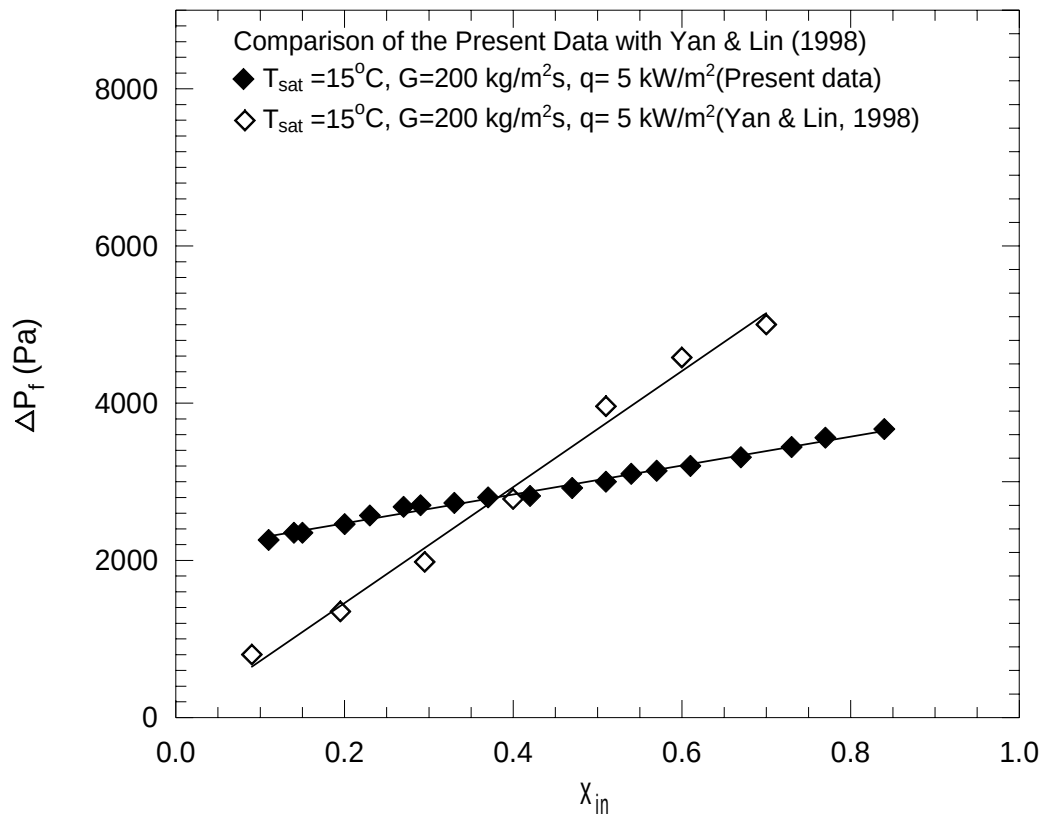


Fig. 6.10 Comparison of the R-407C frictional pressure drop for small tubes with those for R-134a in the same small tubes from Yan and Lin (1998).

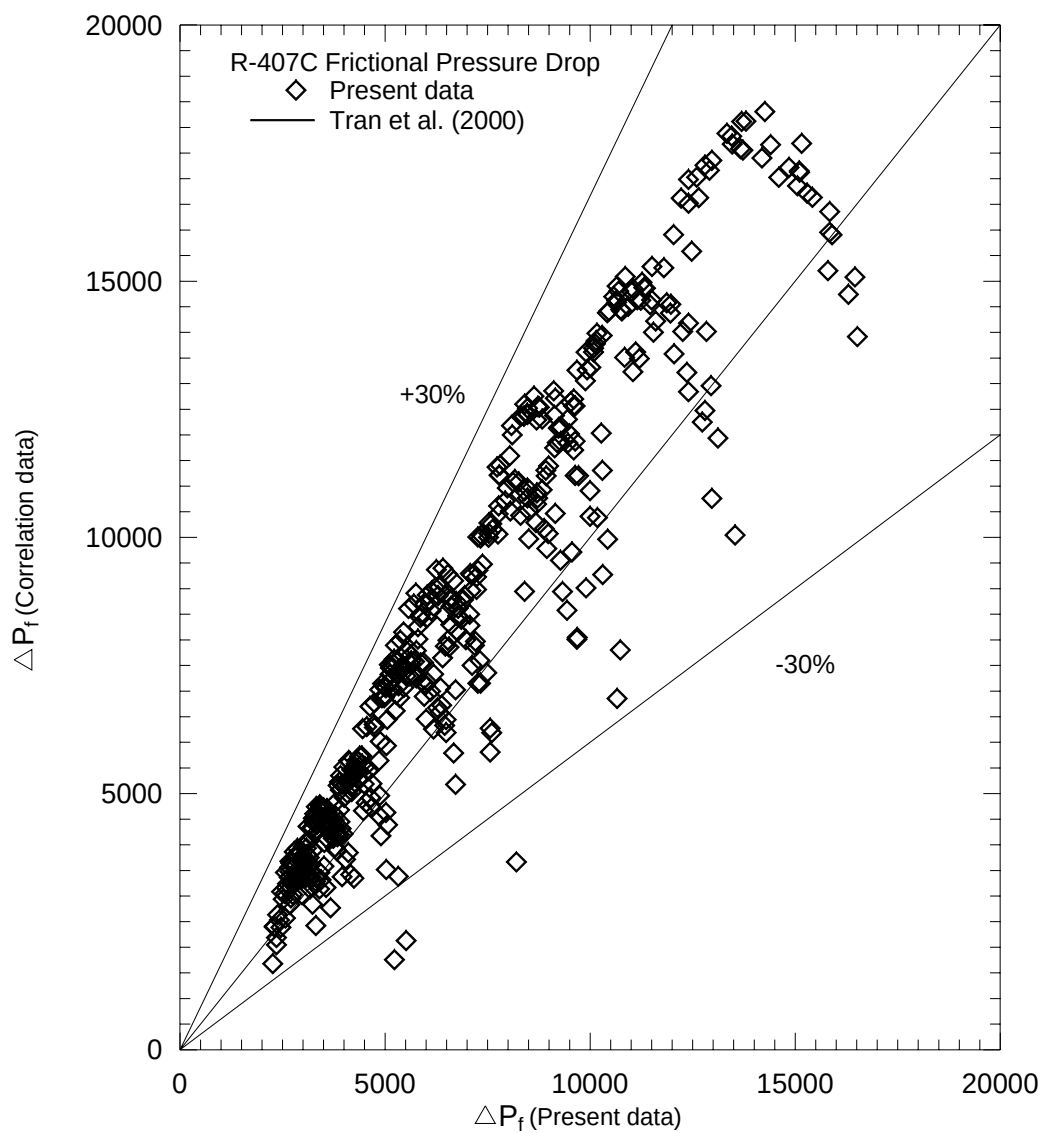


Fig. 6.11 Comparison of the measured data for frictional pressure drop in the evaporation of R-407C with the proposed correlation from Tran et al. (2000).

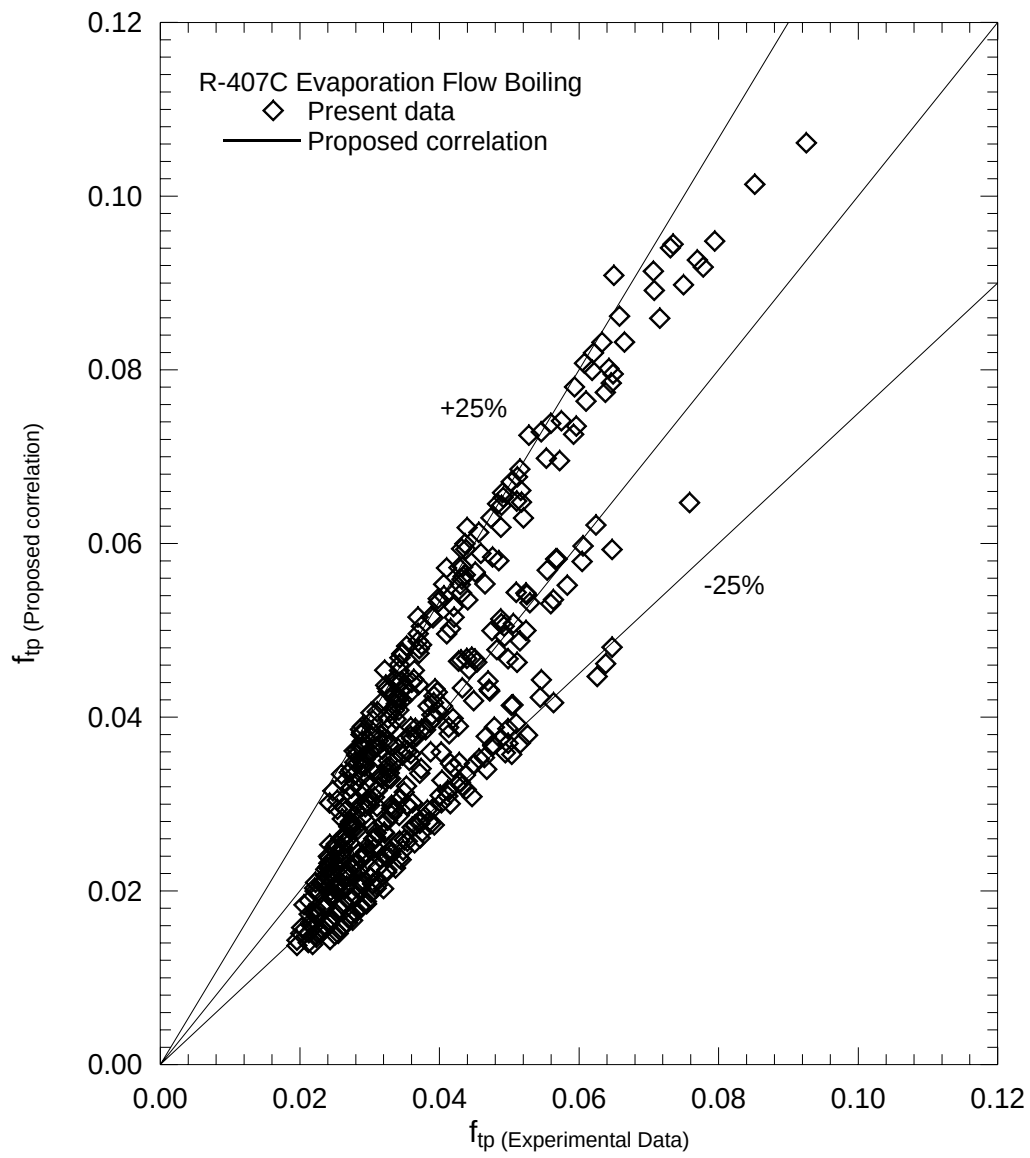


Fig. 6.12 Comparison of the proposed correlations with the present data for the friction factor.

CHAPTER 7

SATURATED FLOW BOILING OF R-407C and R-410A IN a HORIZONTAL NARROW ANNULAR DUCT

In this chapter the measured heat transfer data and observed bubble characteristics for the saturated flow boiling of R-407C and R-410A in the narrow annular duct affected by the refrigerant mass flux, system pressure and duct size are presented. In the tests we have saturated liquid refrigerant at the inlet of the duct ($x_{in} = 0$). The experiments are performed for the refrigerant mass flux varying from 500 to 600 kg/m²s, imposed heat flux q from 1 to 45 kW/m² and the system pressure set at 776 kPa and 900 kPa (corresponding to the saturation temperature $T_{sat}=10^{\circ}\text{C}$ and 15°C) for R-407C and set at 1100 kPa and 1250 kPa (corresponding to the saturation temperature $T_{sat}=10^{\circ}\text{C}$ and 15°C) for R-410A, for the annular gap of the duct fixed at $\delta=1.0$ mm. The measured boiling heat transfer data are expressed in terms of the boiling curves and boiling heat transfer coefficient. Moreover, the close view flow photos taken at a small region around the middle axial station $z=80$ mm are presented to illustrate the bubble characteristics in the boiling flow.

7.1 Single-phase Heat Transfer

Before beginning the two-phase flow boiling experiments, the single-phase convective heat transfer experiments are conducted for liquid R-407C and for liquid R-410A. The measured single-phase convection heat transfer coefficients are compared with the correlations proposed by Dittus-Boelter [47] and Gnielinski [48] for $Re_1 > 2,300$. In the single-phase heat transfer tests the refrigerant mass flux is varied from 300 to 1,050 kg/m²s for the annular gap of the duct $\delta=1.0$ mm, refrigerant saturate temperature $T_{sat}=15^{\circ}\text{C}$ and inlet liquid subcooling $\Delta T_{sub}=3^{\circ}\text{C}$. Selected results from these tests are plotted in Figure 7.1.

The Dittus-Boelter correlation is

$$Nu_{1\phi} = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \quad \text{for } Re > 10^4 \quad (7.1)$$

and the Gnielinski correlation is

$$Nu_{1\phi} = \frac{(f_f/2)(Re-1000)Pr}{1.07 + 12.7\sqrt{f_f/2}(Pr^{2/3}-1)} \quad \text{for } 2,300 < Re < 10^5 \quad (7.2)$$

$$\text{where } f_f = (1.58 \ln Re - 3.28)^{-2} \quad (7.3)$$

The results in Figure 7.1 manifest that the present data for h_1 can be well correlated with the Gnielinski correlations.

7.2 Saturated Flow Boiling Curves

The effects of the refrigerant mass flux, refrigerant saturated temperature and refrigerants on the saturated flow boiling characteristics at the middle axial location ($z = 80$ mm) of the narrow annular duct are shown in Figures 7.2-7.4 by presenting the boiling curves for various G , T_{sat} and refrigerants.

First, the effects of the refrigerant mass flux and refrigerant saturated temperature on the saturated flow boiling curves of R-407C and R-410A are shown in Figures 7.2 and 7.3. The results indicate that at a low imposed heat flux the wall superheat is lower than that for the onset of nucleate boiling (ONB) and no bubble nucleates from the heating surface. Hence heat transfer in the flow results completely from the single-phase forced convection. As the imposed wall heat flux is raised gradually, the wall superheat increases correspondingly. At a certain wall superheat bubbles start to nucleate from the heating surface and we have onset of nucleate boiling in the flow. Beyond the ONB there is a significant increase in the slope of the boiling curves, implying that a small rise in the wall superheat causes a large increase in the heat transfer rate from the wall to refrigerant. Note that the effects of refrigerant mass flux and refrigerant saturated temperature of R-407C are not significant. But the smaller refrigerant mass flux results in the smaller wall superheat and smaller imposed heat flux. This mainly results from the fact that the axial temperature rise of the refrigerant flow is larger for a smaller refrigerant mass flux. Moreover, the lower wall superheat is needed to achieve ONB for a higher saturated temperature for a given

mass flux. This is attributed to the fact that at a lower T_{sat} the surface tension and enthalpy of vaporization of R-407C is higher. At the higher surface tension and enthalpy of vaporization the liquid refrigerant is more difficult to completely flood the cavities, which in turn retards the bubbles to nucleate from the cavities on the heated surface. Note that ΔT_{ONB} and q_{ONB} for the R-410A flow are less affected by the refrigerant mass flux and saturated temperature. Besides, at increasing refrigerant saturated temperature the boiling curves shift slightly to the left, which indicates that at a higher refrigerant saturated temperature the heat transfer in the saturated boiling is slightly better. No apparent temperature undershoot occurs during the onset of boiling of R-407C and R-410A when the flow is saturated.

Then the effects of the refrigerants on the saturated flow boiling curves are shown in Figure 7.4. It is noted that the boiling curve shifts significantly to the left from R-134a to the R-407C and finally to R-410A indicating that the boiling heat transfer of R-410A is better than R-407C and R-407C boiling heat transfer is better than R-134a. It is also evident from the data that the required imposed heat flux and wall superheat for the boiling incipience is lowest for R-410A and highest for R-134a.

7.3 Saturated Flow Boiling Heat Transfer Coefficient

The effects of the refrigerant mass flux, refrigerant saturated temperature and refrigerants on the saturated flow boiling heat transfer at the middle axial location ($z = 80$ mm) in the narrow annular duct are shown in Figures 7.5-7.7 by presenting the saturated flow boiling heat transfer coefficient against the imposed heat flux for various G , T_{sat} and refrigerants. The results indicate that at given G and T_{sat} the saturated boiling heat transfer coefficient increases substantially with the imposed heat flux. This large increase in the saturated boiling heat transfer coefficient is ascribed to the higher active nucleation site density on the heating surface and higher bubble departure frequency for a higher imposed heat flux. The effects of the refrigerant mass flux on the saturated flow boiling heat transfer coefficient of R-407C and R-410A are insignificant, as shown in Figure 7.5. The effects of the refrigerant saturated temperature on the saturated flow boiling heat transfer coefficient of R-407C and R-410A shown in Figure 7.6 manifest that for R-410A increasing the saturated temperature results in higher saturated heat transfer coefficient at higher imposed heat flux.

Then, the effects of the refrigerants on the saturated flow boiling heat transfer coefficient are shown in Figure 7.7. The data show that the saturated boiling heat transfer coefficient of R-410A is much higher than R-134a but R-407C only has mildly higher h_{sat} than R-134a. For example, at $q = 45 \text{ kW/m}^2$, $T_{\text{sat}} = 15^\circ\text{C}$ and $G = 600 \text{ kg/m}^2\text{s}$ the saturated boiling heat transfer coefficient of R-410A is about 66% higher than that of R-407C. At $q = 45 \text{ kW/m}^2$, $T_{\text{sat}} = 15^\circ\text{C}$ and $G = 600 \text{ kg/m}^2\text{s}$ the saturated boiling heat transfer coefficient of R-407C is about 47% higher than that of R-134a. At the higher surface tension, the cavities on the heating surface can be activated easier. Besides, for a higher enthalpy of vaporization, the heat transfer rate between two phases is larger. These effects are thought to be the main reasons that the heat transfer performance of R-410A is better than R-407C and R-134a.

7.4 Bubble Characteristics in Saturated Flow Boiling

The bubble characteristics of R-407C in the narrow duct around the middle axial location for $\delta = 1.0 \text{ mm}$ affected by the refrigerant mass flux and saturated temperature are illustrated by the photos in Figure 7.8. The results in Figures 7.8(a)-(f) indicate that at a higher mass flux the active nucleation site density is lower. Note that at the low mass flux the bubble coalescence is more important and a number of bigger bubbles form in the duct. Then, the effects of the refrigerant saturation temperature on the bubble characteristics are illustrated by comparing the photos in Figures 7.8(a)~(c) with Figures 7.8(g)~(i). The results indicate that at a lower saturation temperature the bubbles grow bigger due to the higher surface tension, and the active nucleation site density is lower due to the higher surface tension and enthalpy of vaporization. The bubble characteristics of R-410A in the narrow duct around the middle axial location for $\delta = 1.0 \text{ mm}$ affected by the refrigerant mass flux and saturated temperature are illustrated by the photos in Figure 4.9. The results in Figure 7.9 show a similar trend to that for R-407C. But the bubbles are much smaller for R-410A.

The bubble characteristics in the narrow duct around the middle axial location for $\delta = 1.0 \text{ mm}$ at $T_{\text{sat}} = 15^\circ\text{C}$ and $G = 600 \text{ kg/m}^2\text{s}$ affected by the refrigerants are illustrated by the photos in Figure 7.10. At the imposed heat flux $q = 15 \text{ kW/m}^2$ the active nucleation sites on the heated surface for R-407C is slightly more than R-134a. The active nucleation sites on

the heated surface for R-410A is significantly more than R-407C and R-134a but the bubbles in R-410A flow is smaller than R-407C and R-134a due to its smaller surface tension force. As the imposed heat flux is increased to $q=25 \text{ kW/m}^2$, the active bubble nucleation density increases especially for R-410A and a lot more bubbles appear and they move faster. Many coalescence bubbles are seen as the heat flux is raised to $q=35 \text{ kW/m}^2$. The coalescence bubbles in the R-410A flow are bigger than R-407C and R-134a.

To be more quantitative on the bubble characteristics, we move further to estimate the average bubble departure diameter and frequency and the average active bubble nucleation site density on the heating surface from the images of the boiling flow stored in the video tapes. The results from this estimation are examined in the following. The effects of the three parameters, namely, the refrigerant mass flux, refrigerant saturated temperature and refrigerants, on the mean bubble departure diameter for the saturated flow boiling at the middle axial location ($z = 80 \text{ mm}$) in the annular duct are shown in Figures 7.11-7.13 by presenting the average bubble departure diameter against the imposed heat flux for various G , T_{sat} and refrigerants. First, the effects of the refrigerant mass flux on the average bubble departure diameters for both R-407C and R-410A shown in Figure 7.11 indicate that the average departing bubble is only slightly larger for a lower refrigerant mass flux. The results in Figure 7.12 indicate that the average bubble departure diameter is smaller for a higher refrigerant saturated temperature. Then, the departing bubbles are substantially larger for R-134a, according to the data given in Figure 7.13. Moreover, the bubble departure diameter for R-407C is slightly larger than R-410A since R-407C has a higher surface tension.

How the bubble departure frequency is affected by the three parameters for the saturated flow boiling at the middle axial location ($z = 80 \text{ mm}$) in the annular duct are shown in Figures 7.14-7.16 by presenting the average bubble departure frequency against the heat flux for various G , T_{sat} and refrigerants. Note that the increase of the bubble departure frequency with the imposed heat flux is rather significant for all cases presented here. First, the effects of the refrigerant mass flux on the saturated flow boiling average bubble departure frequency for both R-407C and R-410A are shown in Figures 7.14. The results indicate that the average bubble departure frequency is somewhat higher for a higher refrigerant mass flux especially at high imposed heat flux. The data given Figure 7.15 indicate that the average bubble departure frequency is significantly higher for a

higher saturated temperature. Then, the saturated flow boiling average bubble departure frequency for various refrigerants shown in Figure 7.16 manifests that the average bubble departure frequencies for both R-410A and R-407C are much higher than R-134a. This is attributed to the much higher surface tension for R-134a.

The number density of the active nucleation sites for ONB affected by the three parameters for the saturated flow boiling at the middle axial location ($z = 80$ mm) in the annular duct are shown in Figures 7.17-7.19 by presenting the average active nucleation site density against the imposed heat flux for various G , T_{sat} and refrigerants. The data clearly show the substantial increase of the active nucleation site density with the imposed heat flux for all cases examined here. First, the effects of the refrigerant mass flux on the saturated flow boiling average active nucleation site density are shown in Figures 7.17. The results in Figure 7.17(a) indicate that the average active nucleation site density is significantly higher for a smaller refrigerant mass flux especially at high imposed heat flux for R-407C. The data shown in Figure 7.18 suggest that the average active nucleation site density increases significantly with T_{sat} especially at high heat flux. Then, the saturated flow boiling average active nucleation site density for the three refrigerants shown in Figure 7.19 manifest that the average active nucleation site density of R-410A is significantly higher than R-407C and R-134a, at high imposed heat flux due to its smaller surface tension.

7.5 Concluding Remarks

The experimental heat transfer data for the saturated flow boiling of R-407C and R-410A in the narrow annular duct have been presented here. Meanwhile, the bubble behavior in the boiling flow is examined. The effects of the imposed heat flux, refrigerant mass flux, system pressure and refrigerants on the saturated flow boiling heat transfer coefficient and associated bubble characteristics have been investigated in detail. The major results obtained here can be summarized in the following.

- (1). The temperature undershoot at ONB are insignificant for the saturated flow boiling for the three refrigerants used in the horizontal narrow annular duct.
- (2). The saturated boiling heat transfer coefficients of R-410A are better than R-407C and R-134a. Besides, raising the imposed heat flux can cause a significant increase in the boiling heat transfer coefficient. However, the effects of the refrigerant mass flux and

saturated temperature on the boiling heat transfer coefficient are small.

- (3). The results from the flow visualization show that the mean diameter of the bubbles departing from the heating surface decreases slightly with increasing refrigerant mass flux. Besides, at a high imposed heat flux many bubbles generated from the cavities in the heating surface tend to merge together to form big bubbles. The bubble departure frequency increases with the increasing refrigerant mass flux and saturated temperature. The active nucleation site density is much lower at a higher refrigerant mass flux and lower saturation temperature. The higher active nucleation site density on the heating surface for the R-410A flow is due to its smaller surface tension and is the main reason that the flow boiling heat transfer performance of R-410A is better than R-407C and R-134a.

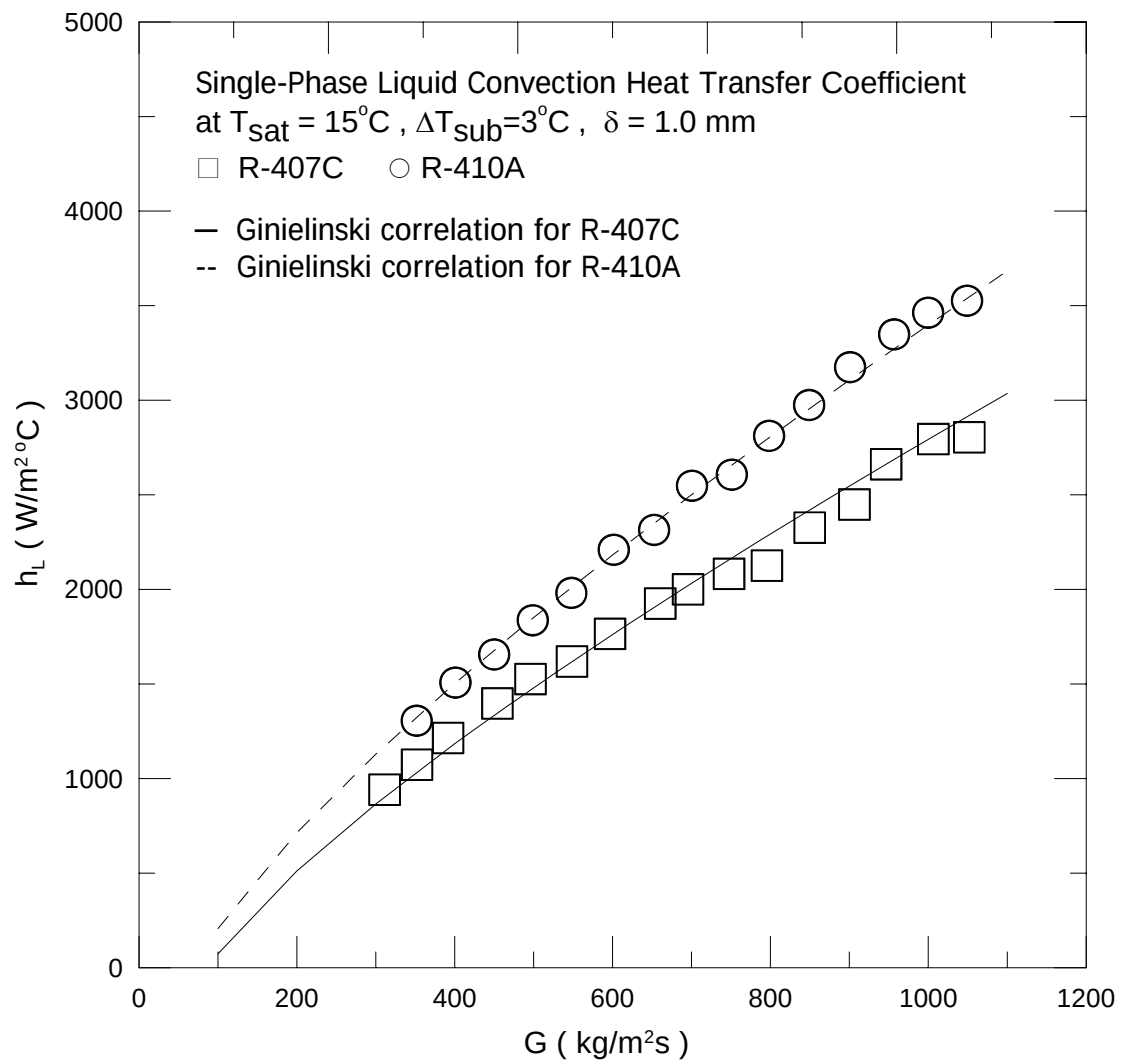


Fig. 7.1 Comparison of the present single-phase liquid convection heat transfer data for R-407C and R-410A with the correlation of Gnielinski.

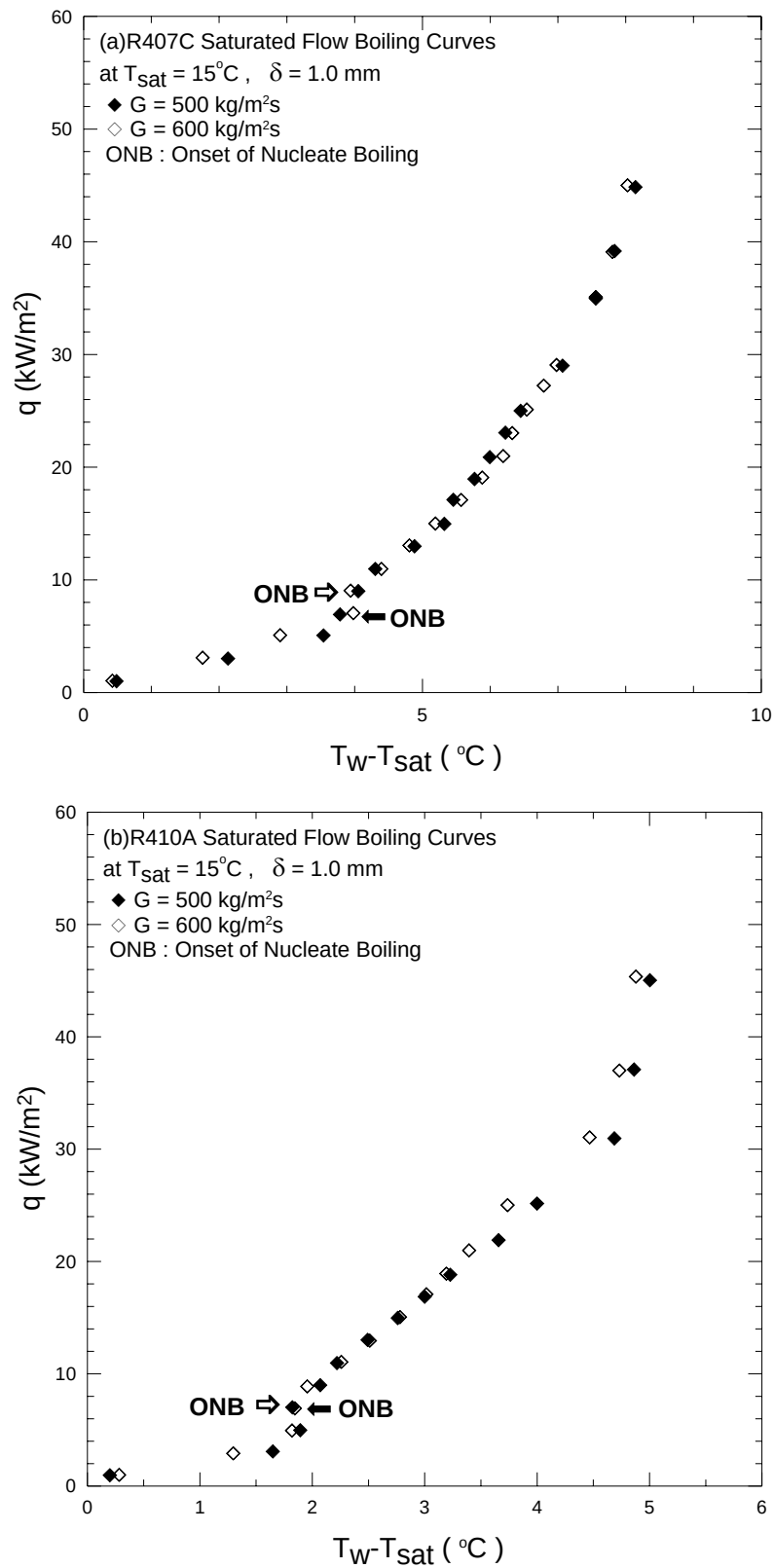


Fig. 7.2 Saturated flow boiling curves for various refrigerant mass fluxes for refrigerants (a) R-407C and (b) R-410A.

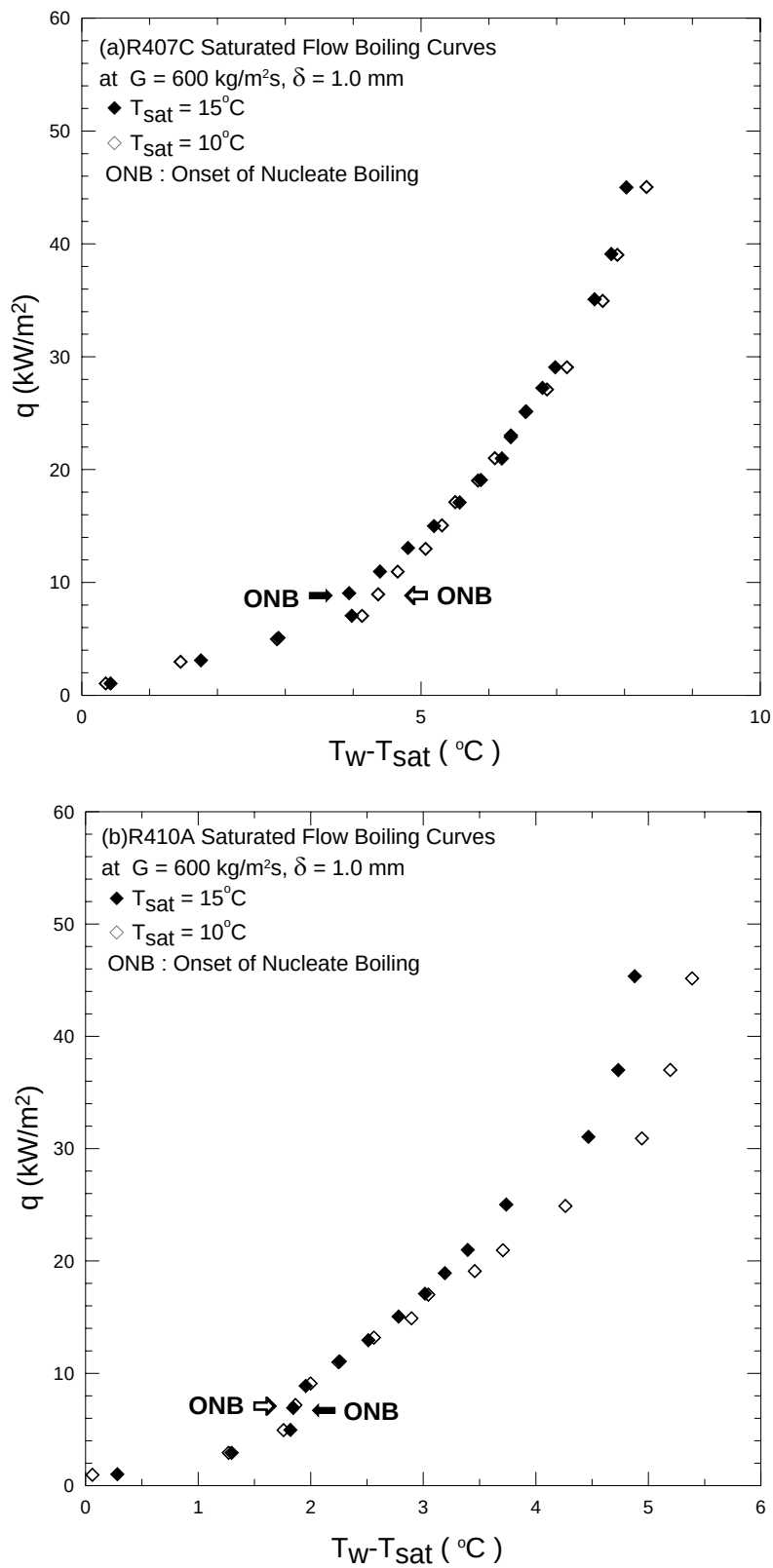


Fig. 7.3 Saturated flow boiling curves for various refrigerant saturated temperatures for refrigerants (a) R-407C and (b) R-410A.

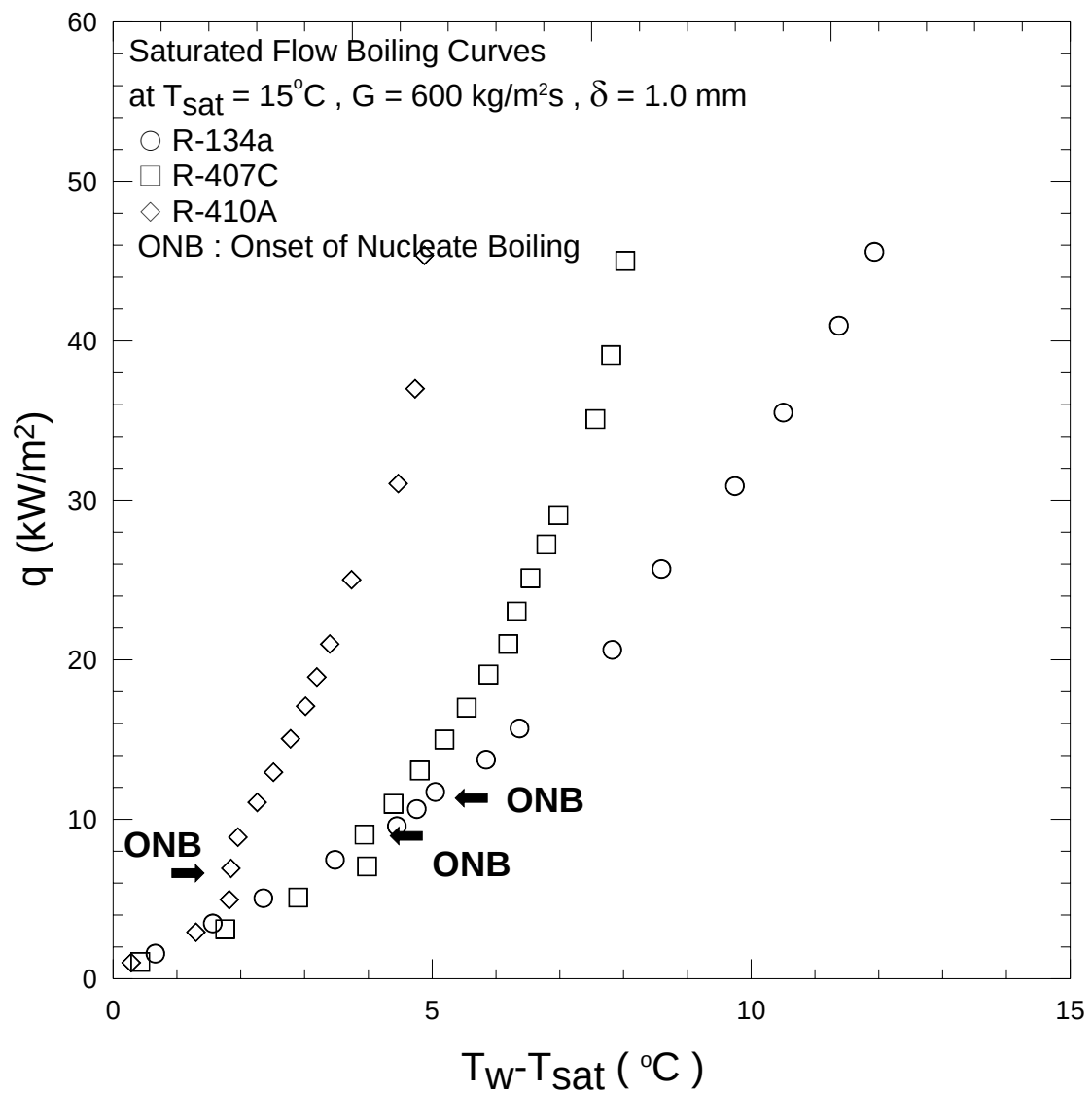


Fig. 7.4 Saturated flow boiling curves for various refrigerants at $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 1.0 \text{ mm}$.

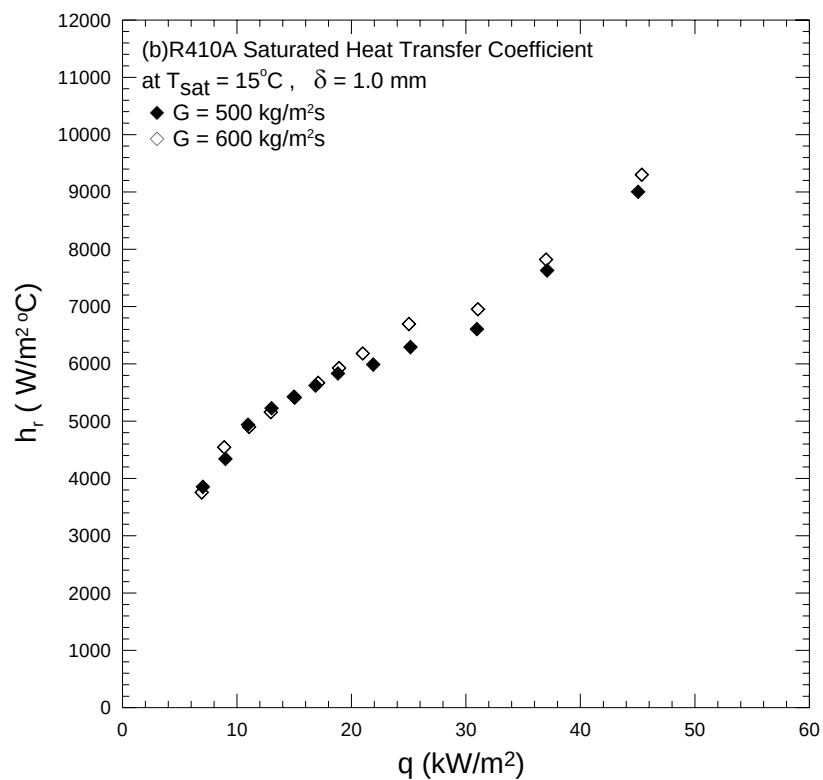
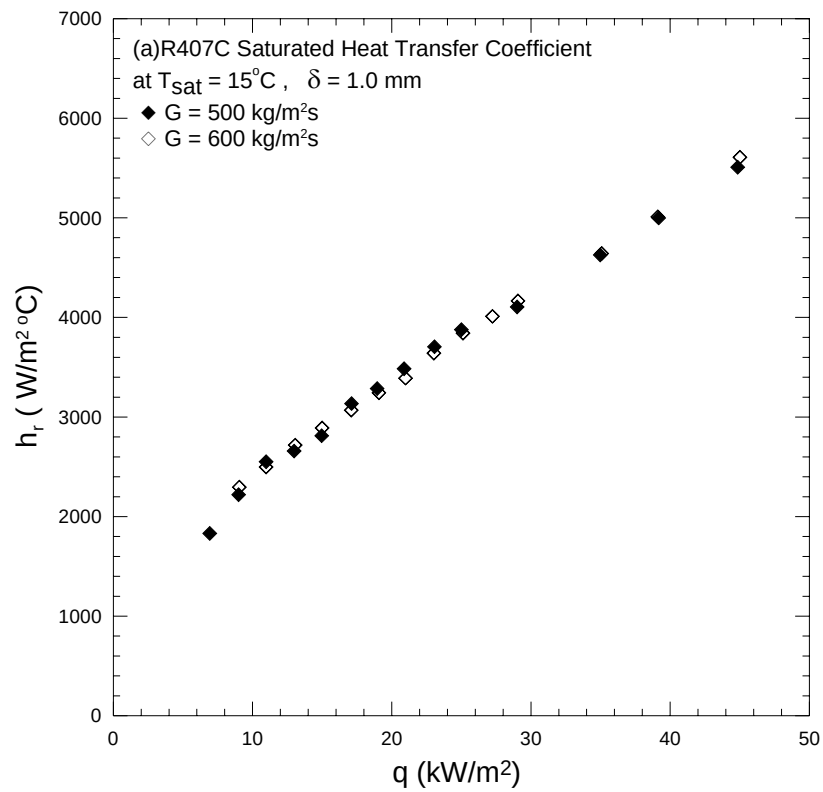


Fig. 7.5 Saturated flow boiling heat transfer coefficients for various refrigerant mass fluxes for refrigerants (a) R-407C and (b) R-410A.

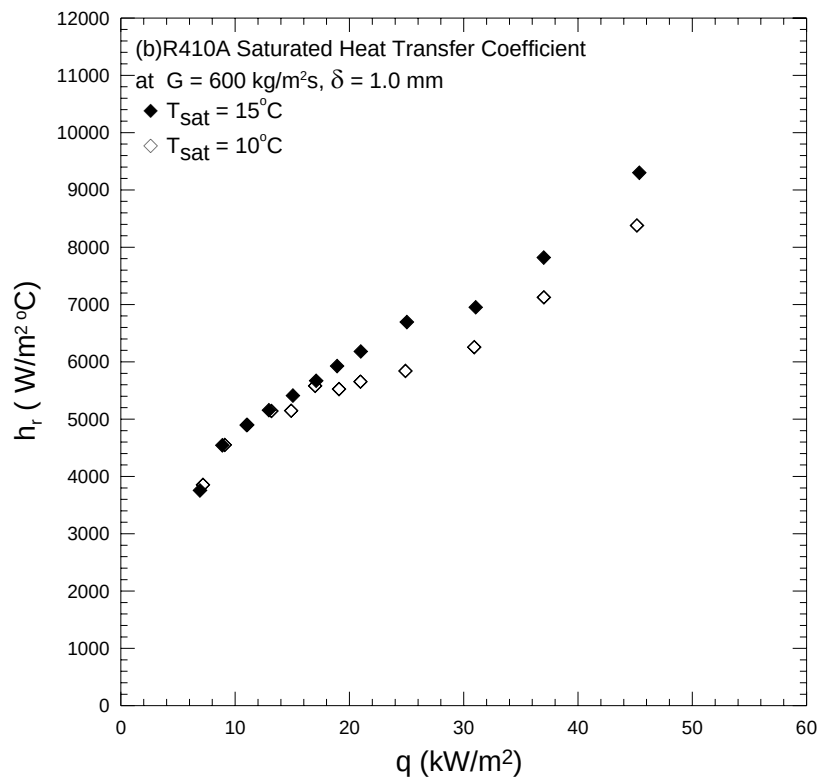
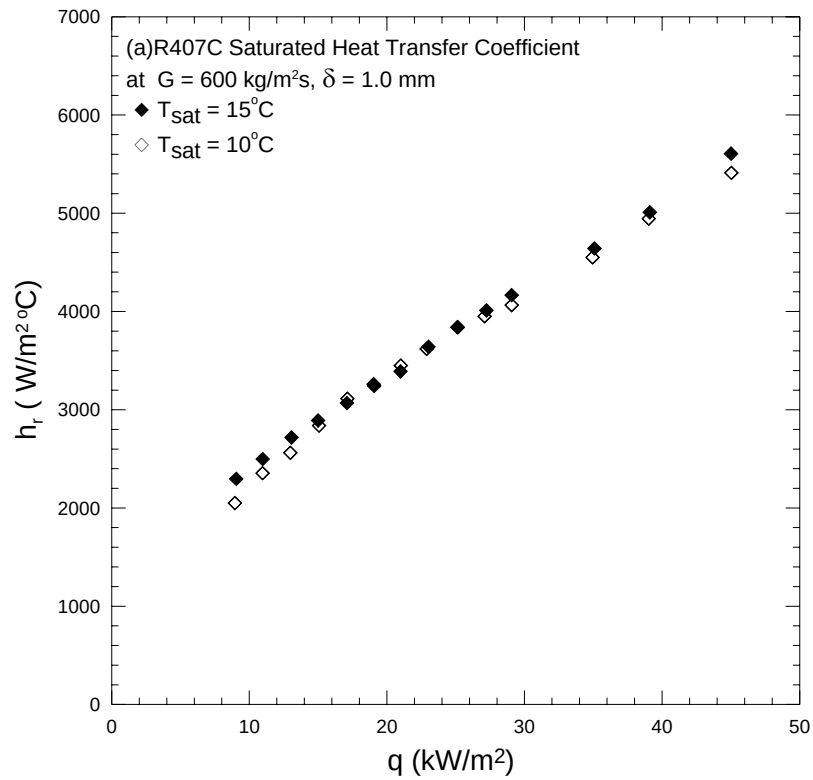


Fig. 7.6 Saturated flow boiling heat transfer coefficients for various refrigerant saturated temperatures for refrigerants (a) R-407C and (b) R-410A.

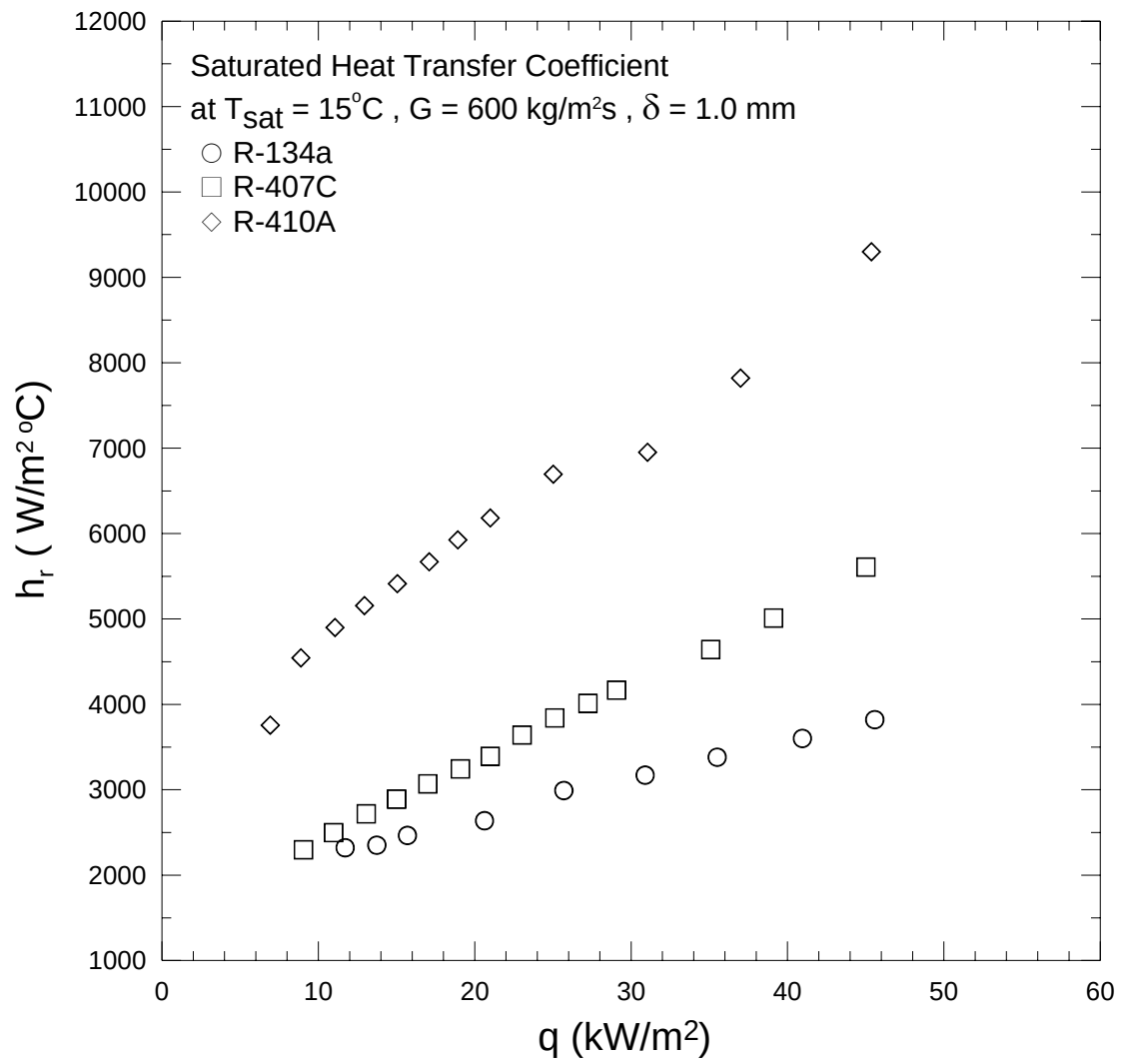


Fig. 7.7 Saturated flow boiling heat transfer coefficients for various refrigerants at $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 1.0 \text{ mm}$.

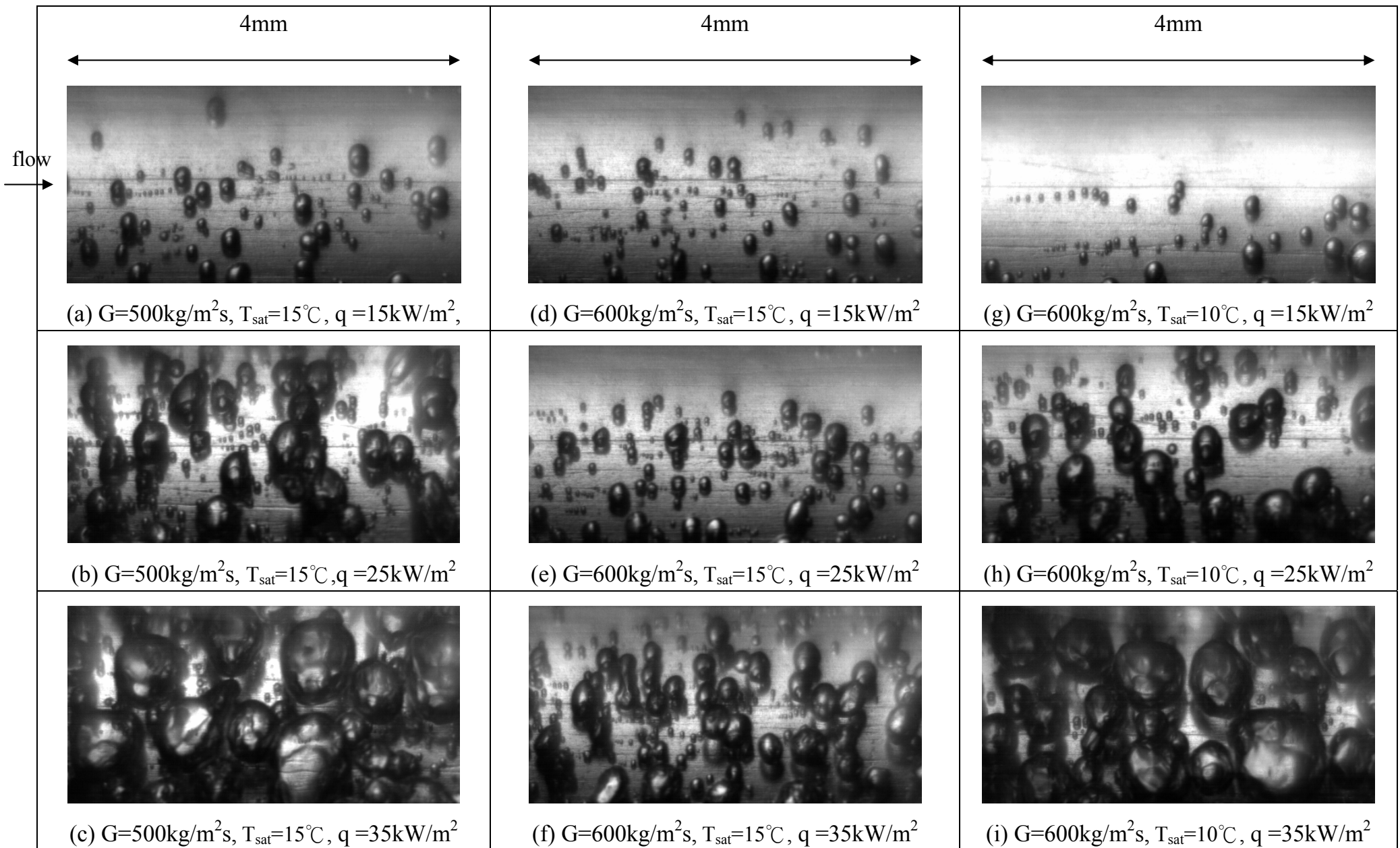


Fig. 7.8 Photos of bubbles in the saturated flow boiling of R-407C in a small region around middle axial location at $\delta=1\text{ mm}$ for various imposed heat flux, mass fluxes and saturated temperatures.

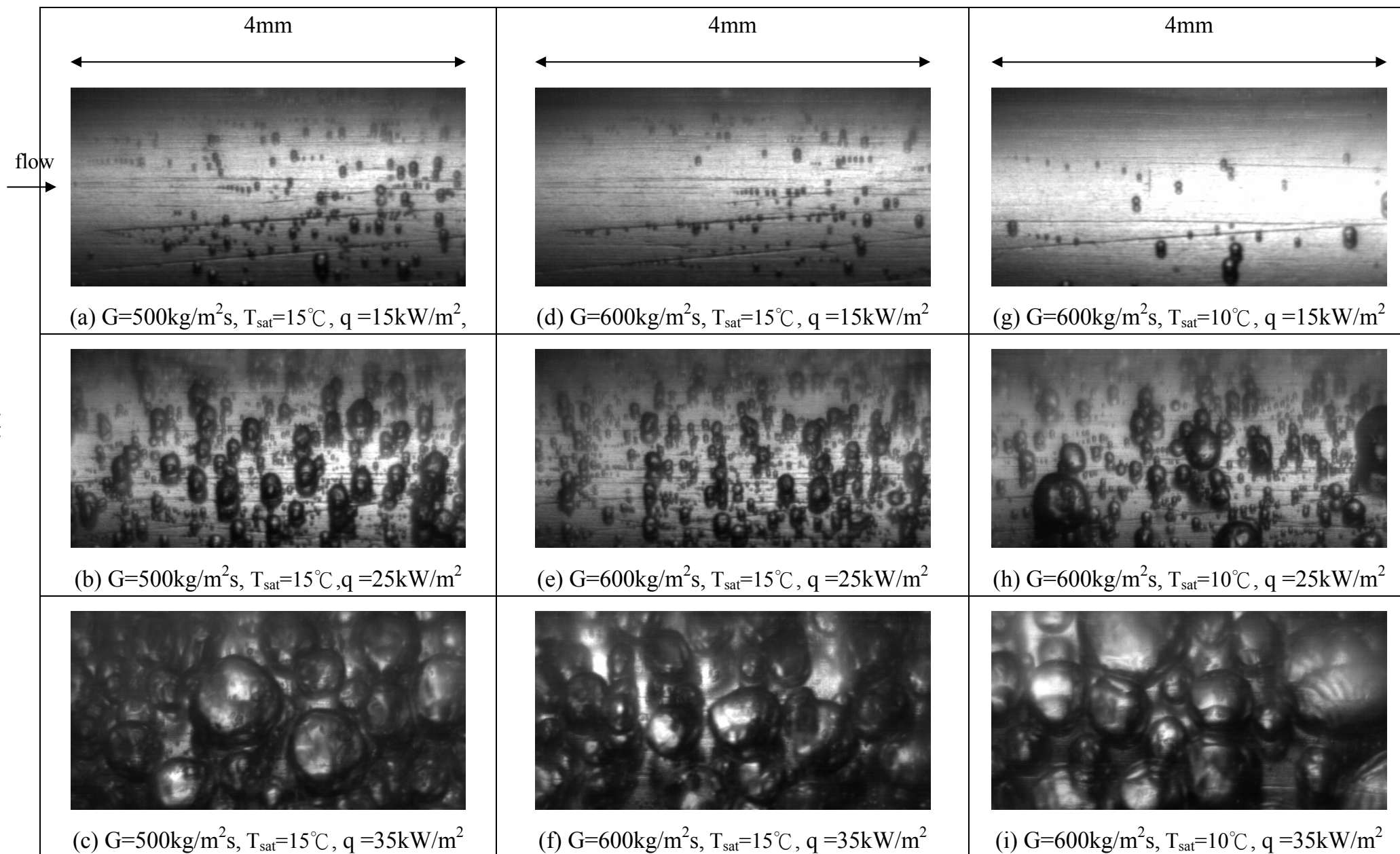


Fig. 7.9 Photos of bubbles in the saturated flow boiling of R-410A in a small region around middle axial location at $\delta=1\text{ mm}$ for various imposed heat flux, mass fluxes and saturated temperatures.

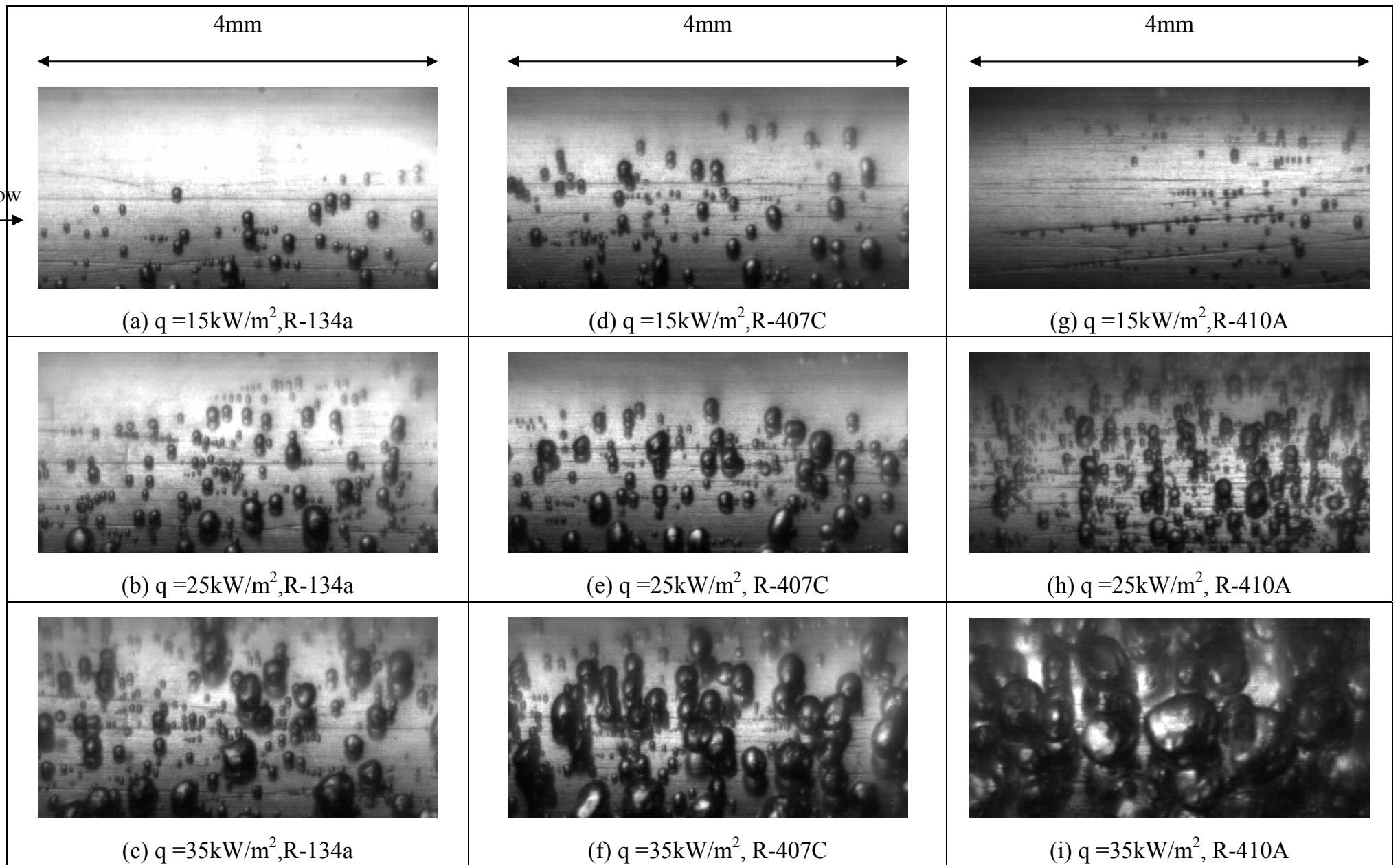


Fig. 7.10 Photos of bubbles in the saturated flow boiling in a small region around middle axial location at $T_{\text{sat}} = 15^\circ\text{C}$, $\delta = 1.0 \text{ mm}$ and $G = 600 \text{ kg/m}^2\text{s}$ for various imposed heat flux and refrigerants.

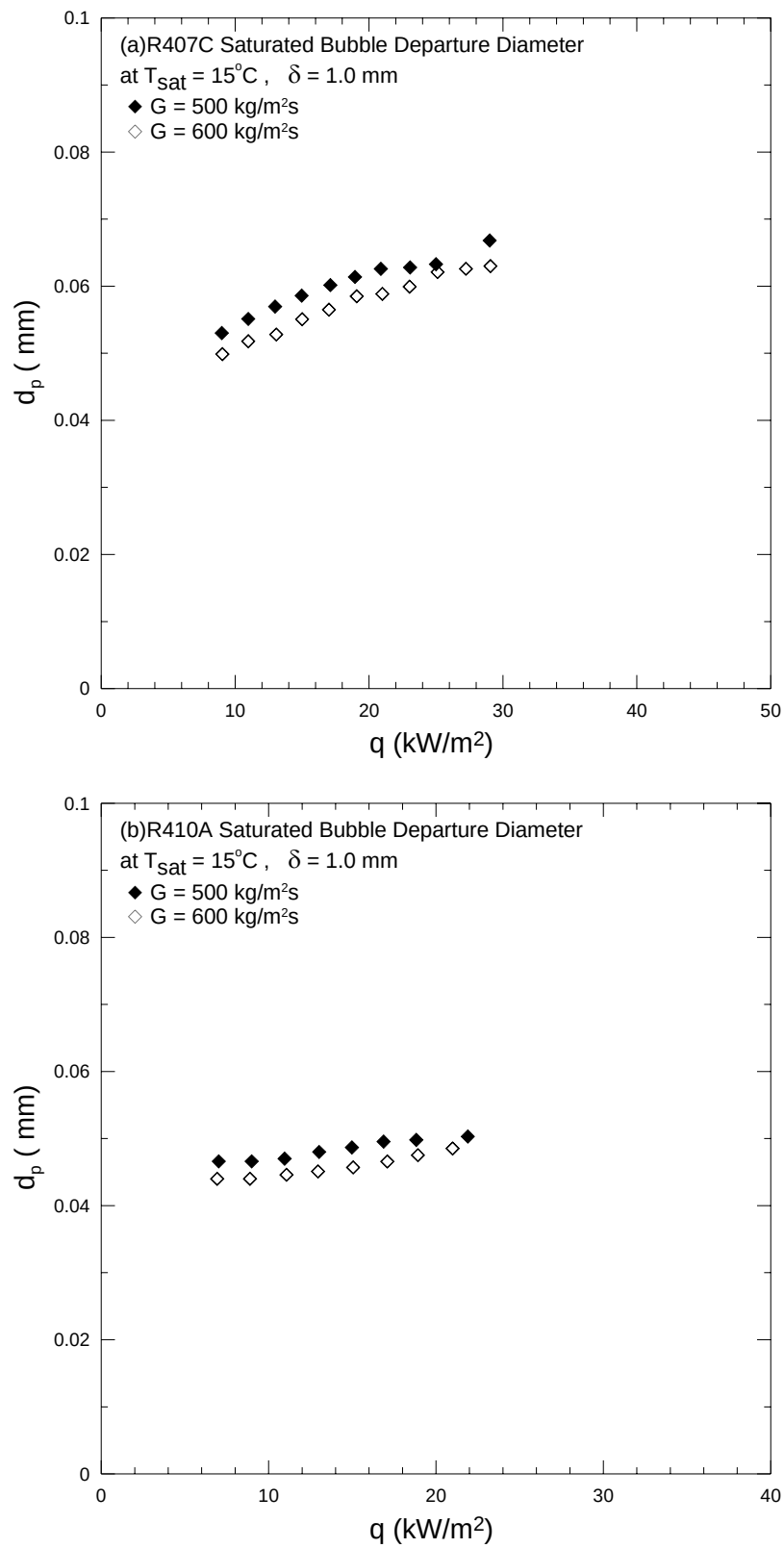


Fig. 7.11 Saturated flow boiling mean bubble departure diameter for various refrigerant mass fluxes for refrigerants (a) R-407C and (b) R-410A.

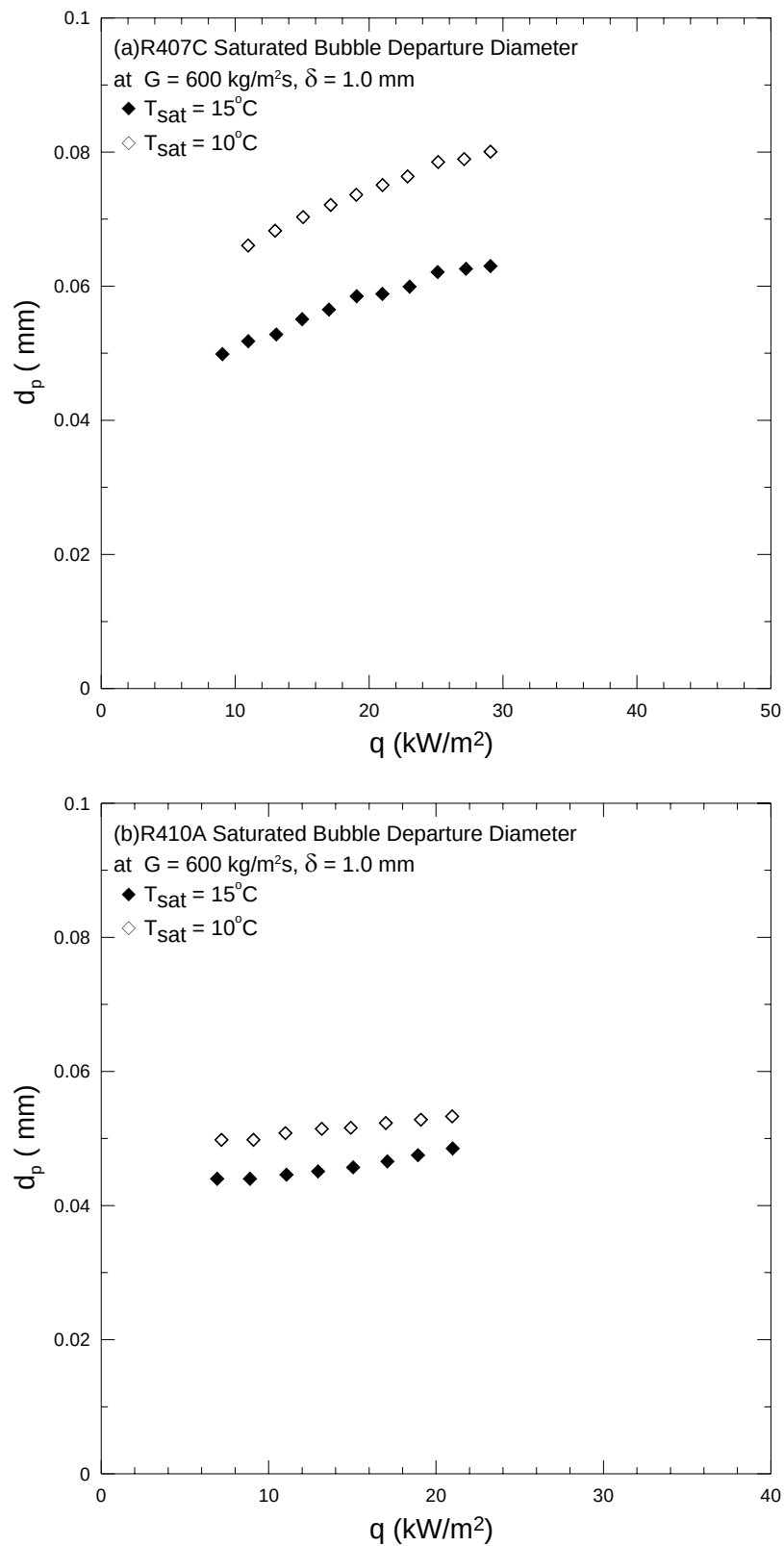


Fig. 7.12 Saturated flow boiling mean bubble departure diameter for various refrigerant saturated temperatures for refrigerants (a) R-407C and (b) R-410A.

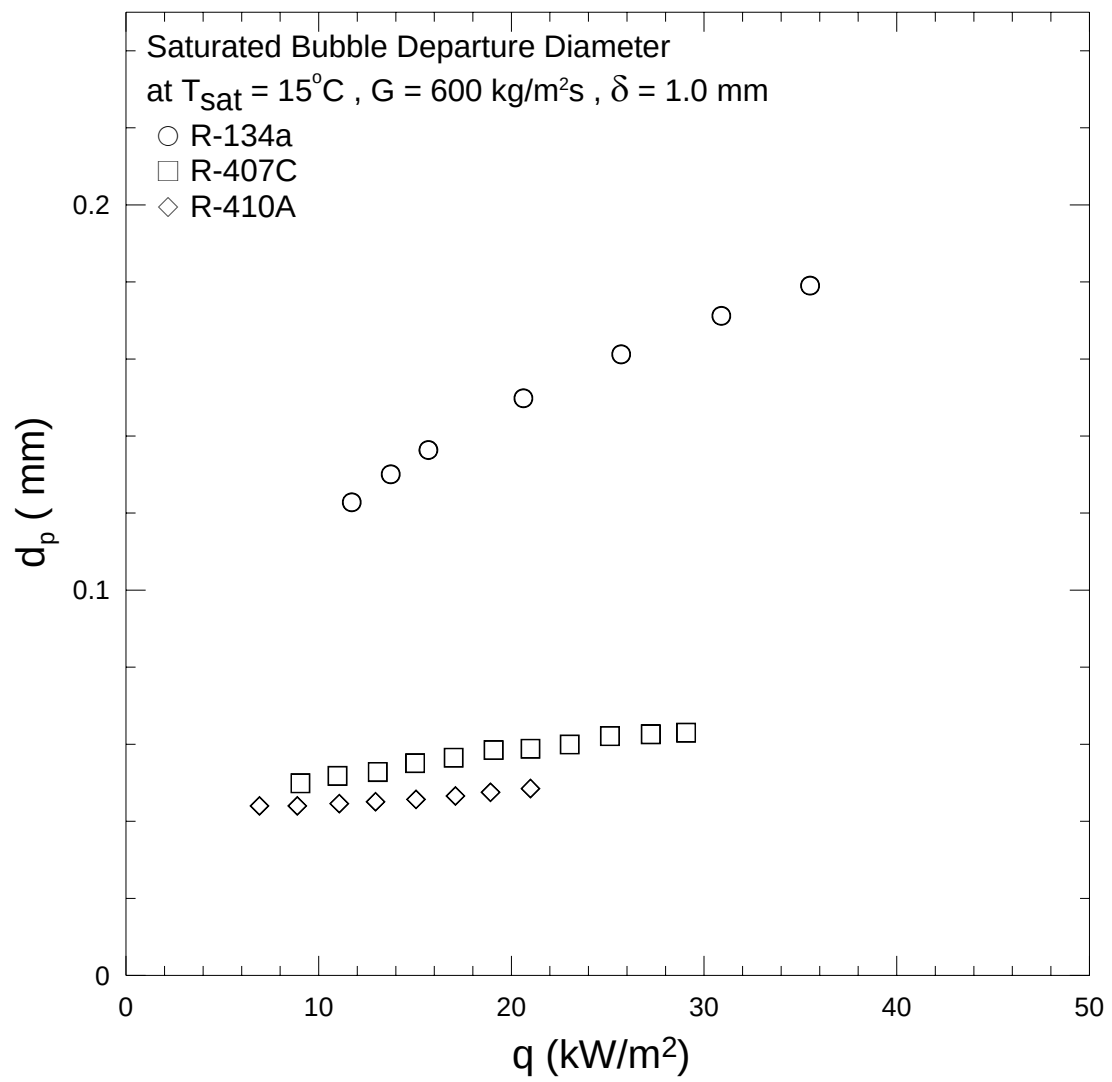


Fig. 7.13 Saturated flow boiling mean bubble departure diameter for various refrigerants at $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 1.0 \text{ mm}$.

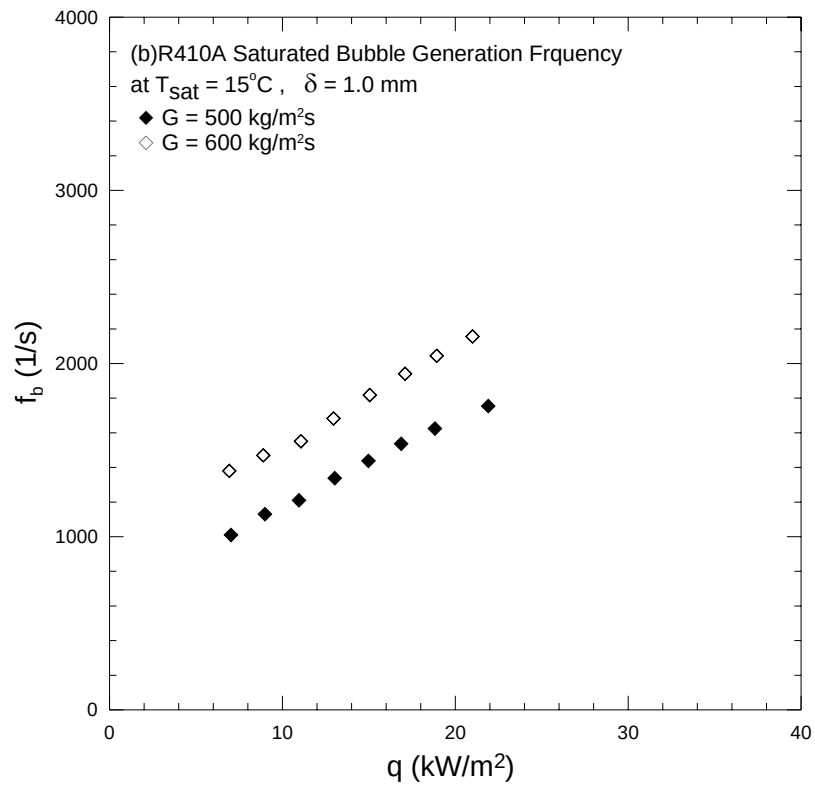
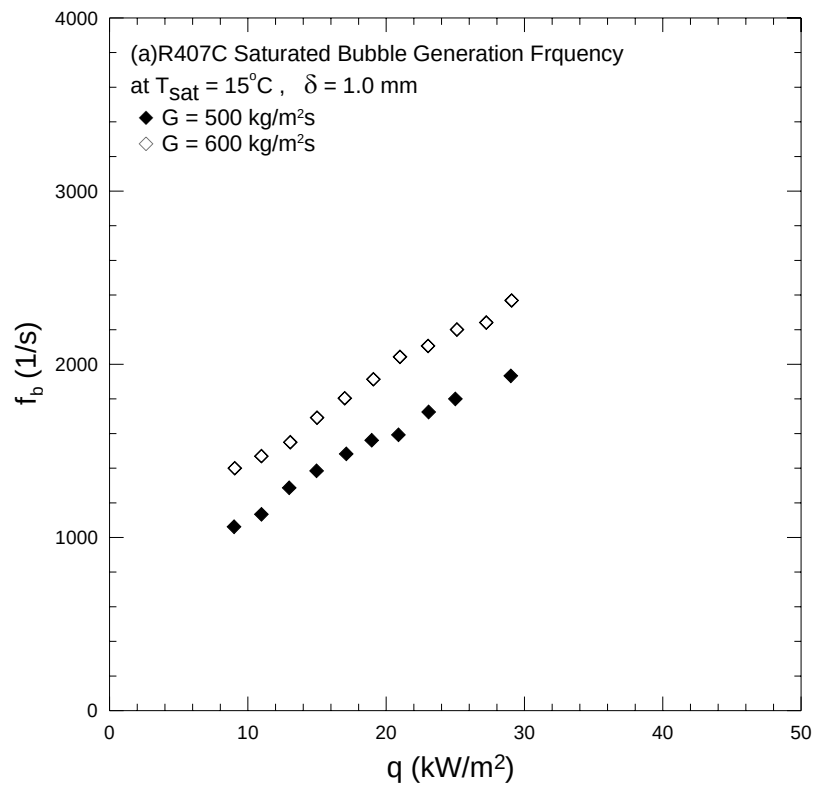


Fig. 7.14 Saturated flow boiling mean bubble departure frequency for various refrigerant mass fluxes for refrigerants (a) R-407C and (b) R-410A.

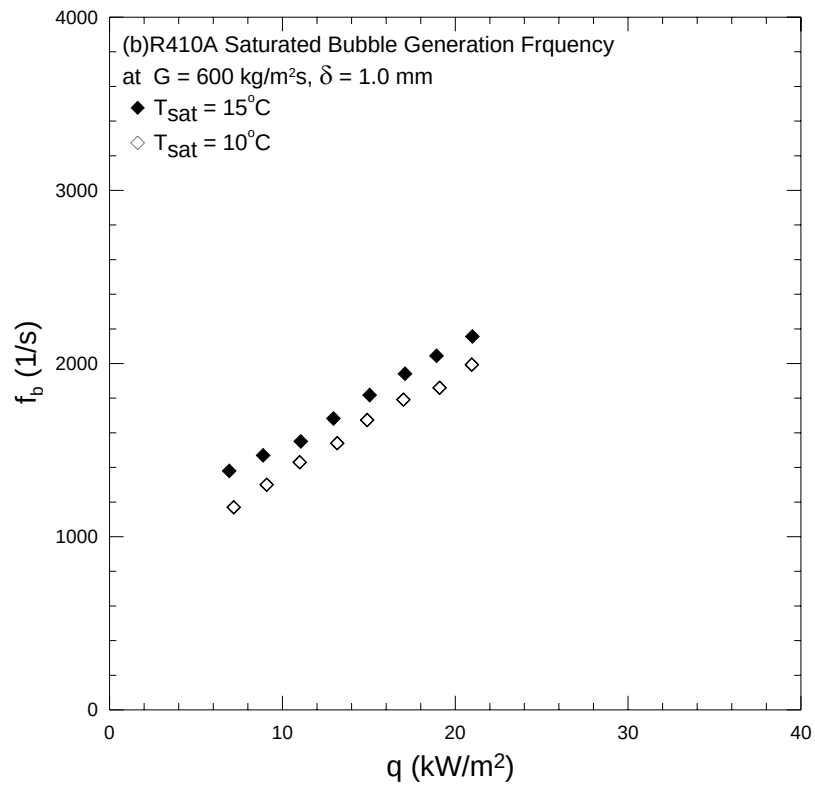
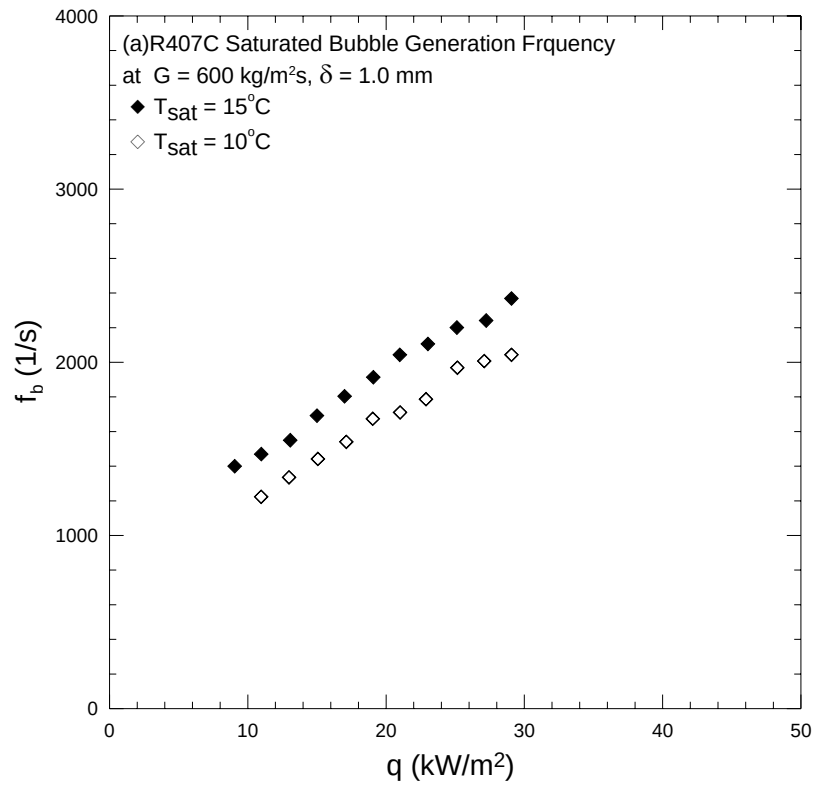


Fig. 7.15 Saturated flow boiling mean bubble departure frequency for various refrigerant saturated temperatures for refrigerants (a) R-407C and (b) R-410A.

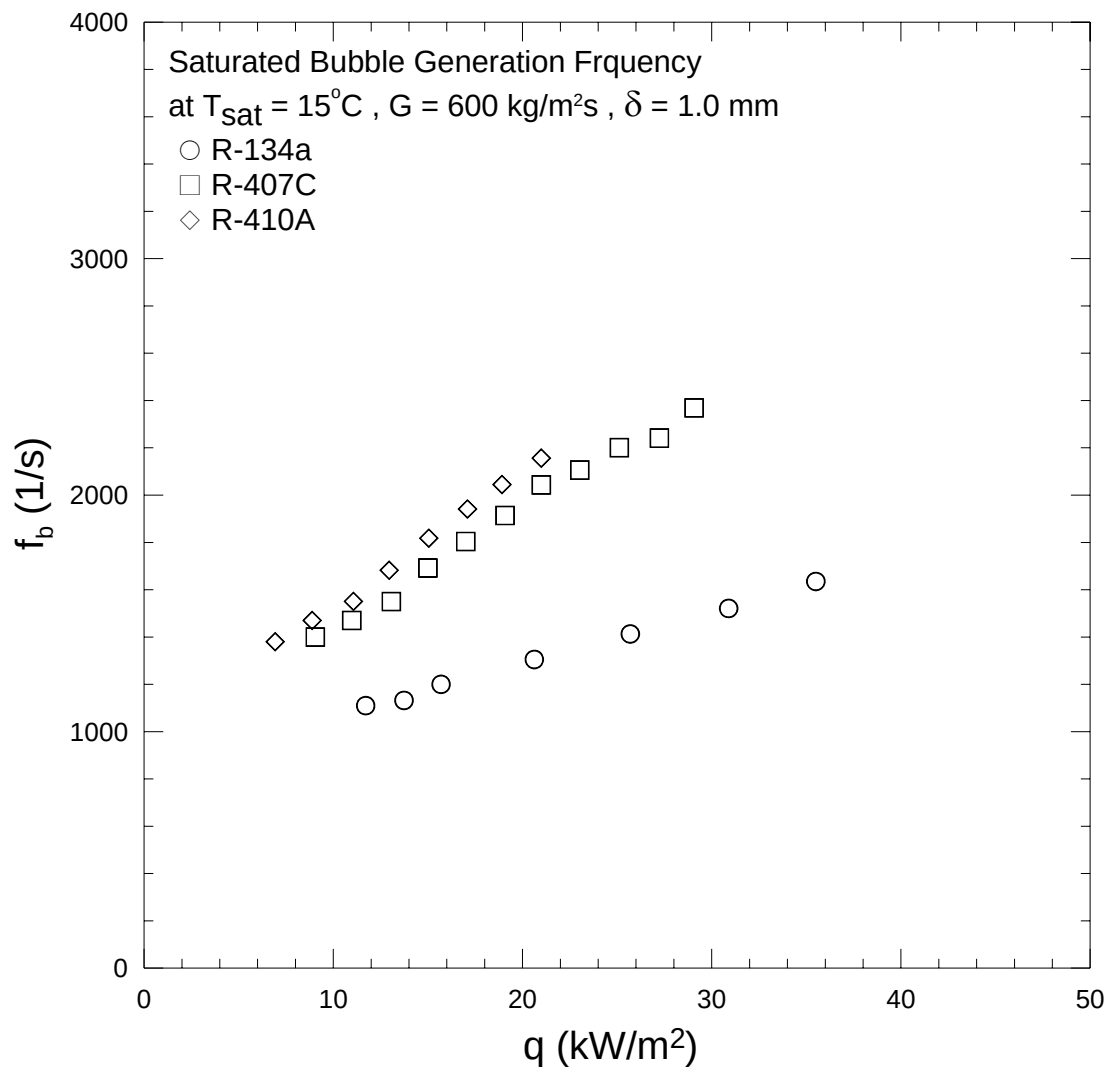


Fig. 7.16 Saturated flow boiling mean bubble departure frequency for various refrigerants at $T_{\text{sat}} = 15^{\circ}\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 1.0 \text{ mm}$.

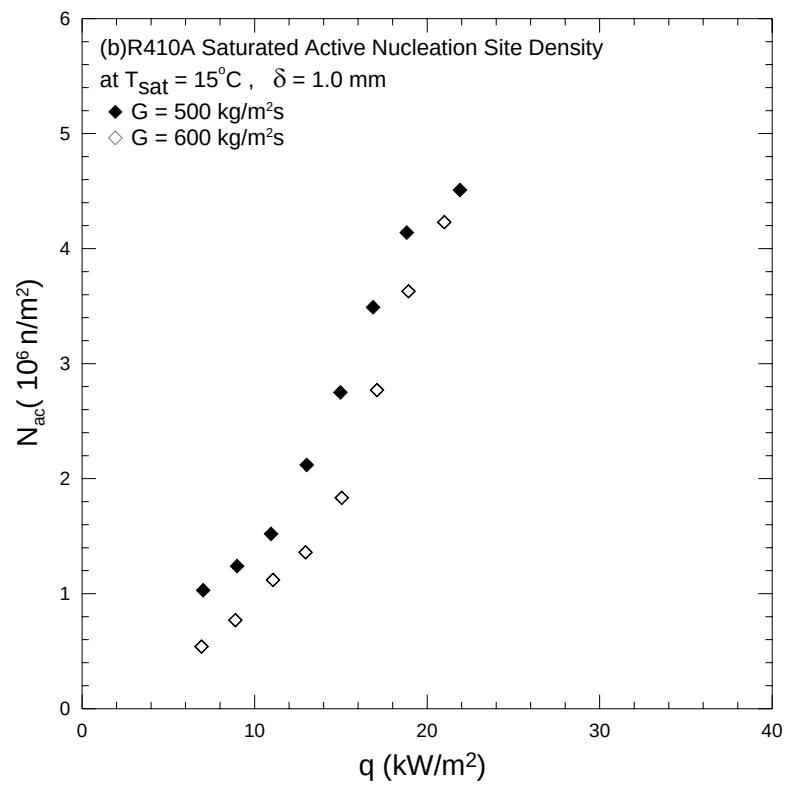
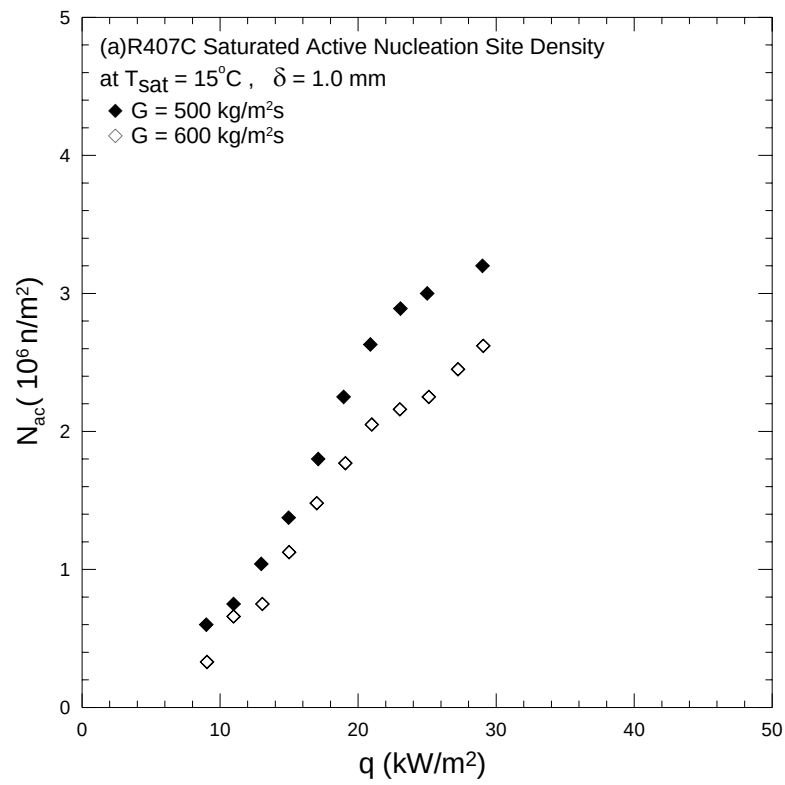


Fig. 7.17 Saturated flow boiling mean active nucleation site density for various refrigerant mass fluxes for refrigerants (a) R-407C and (b) R-410A.

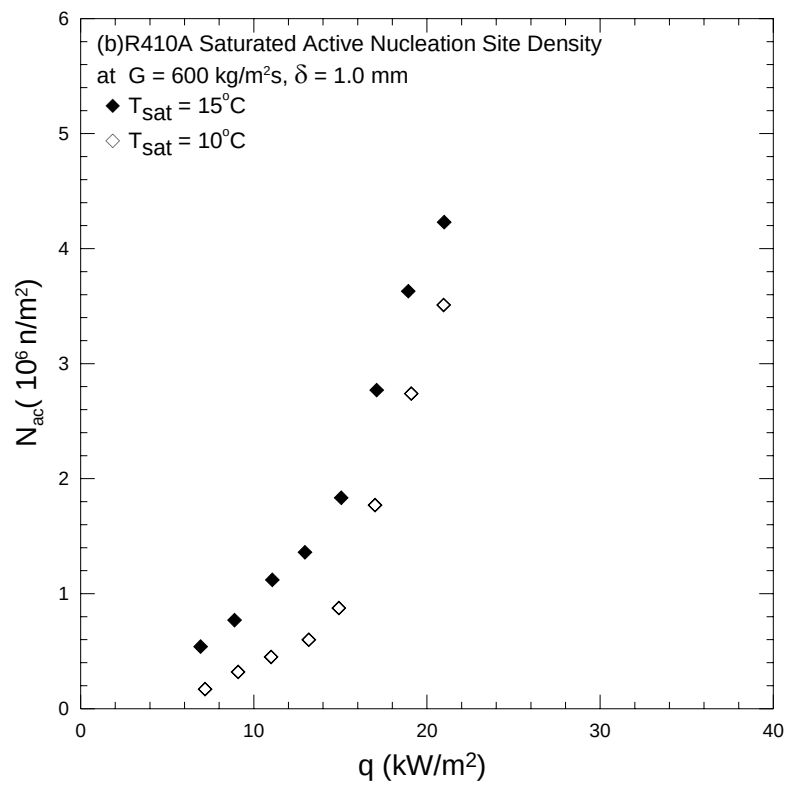
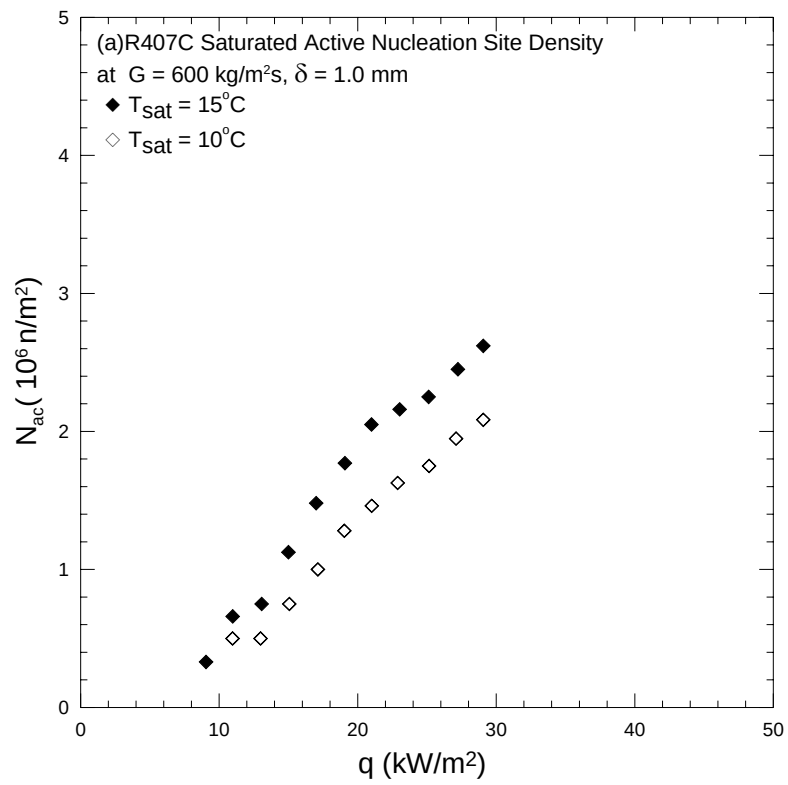


Fig. 7.18 Saturated flow boiling mean active nucleation site density for various refrigerant saturated temperatures for refrigerants (a) R-407C and (b) R-410A.

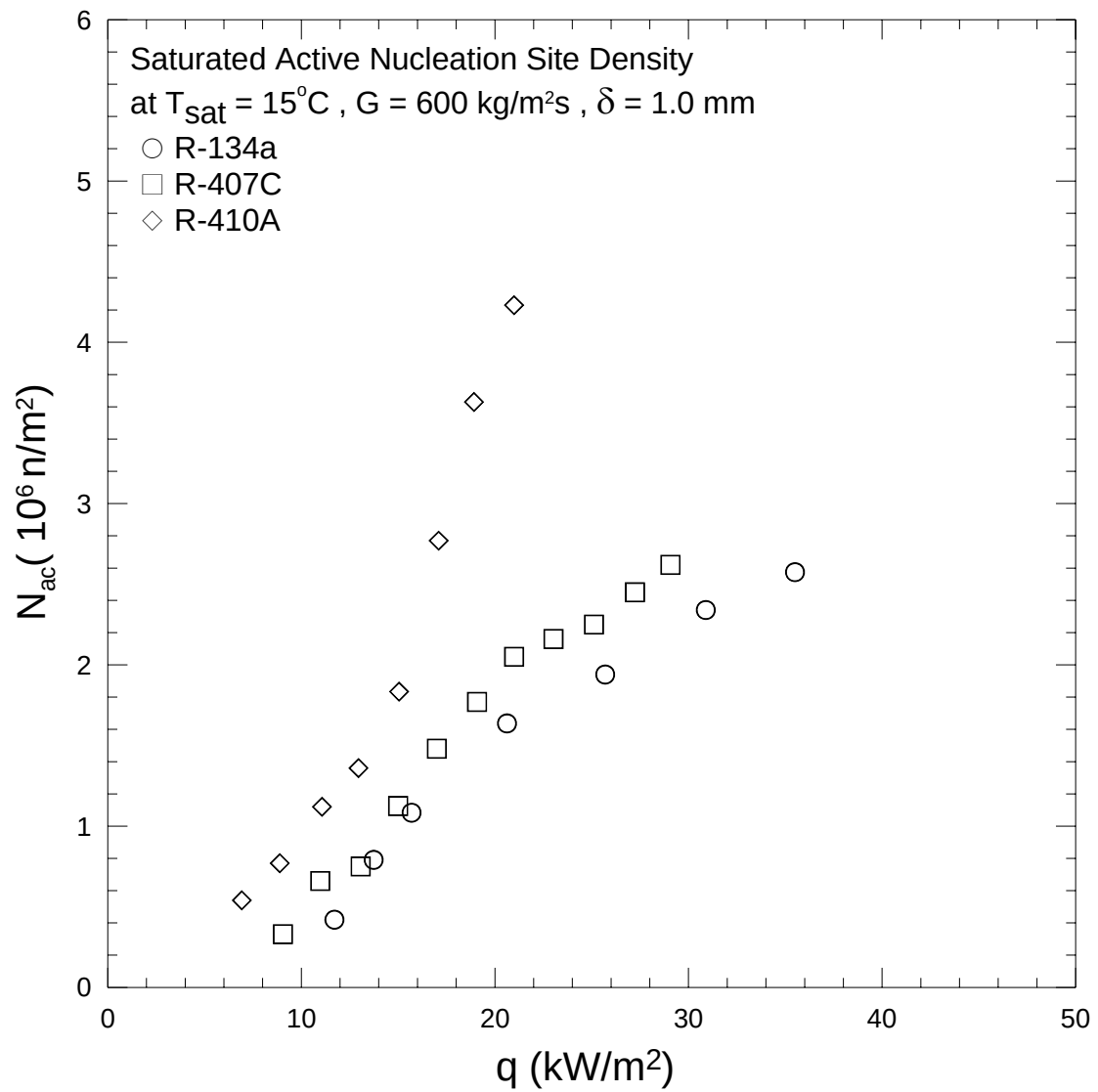


Fig. 7.19 Saturated flow boiling mean active nucleation site density for various refrigerants at $T_{\text{sat}} = 15^\circ\text{C}$ & $G = 600 \text{ kg/m}^2\text{s}$ & $\delta = 1.0 \text{ mm}$.

CHAPTER 8

CONCLUDING REMARKS

8.1 Concluding Remarks

First, we have carried out experiments to unravel the saturated flow boiling heat transfer characteristics and associated bubble characteristics of R-134a in a horizontal narrow annular duct. The effects of the refrigerant mass flux, imposed heat flux, and refrigerant saturated temperature have been examined in detail. A summary of the major findings from the present results is given in the following.

- (1). The temperature undershoot at ONB are insignificant for the saturated flow boiling.
- (2). The saturated boiling heat transfer coefficient of R-134a is found to increase with a decrease in the gap size. Besides, raising the imposed heat flux can also cause a significant increase in the saturated flow boiling heat transfer coefficient. However, the effects of the refrigerant mass flux and saturated temperature on the boiling heat transfer coefficient are also small.
- (3). The results from the flow visualization of the bubble behavior in the saturated flow boiling show that the mean diameter of the bubbles departing from the heating surface decreases slightly at increasing refrigerant mass flux. Besides, at a high imposed heat flux many bubbles generated from the cavities in the heating surface tend to merge together to form a big bubbles. The mean bubble departure frequency is higher in the smaller duct.
- (4). Empirical correlations for the flow boiling heat transfer coefficients for R-134a in the narrow annular duct has been provided. Correlating equations have also been proposed for some bubble characteristics in the saturated boiling of R-134a in the narrow annular duct.

Then, in the small circular tubes we have experimentally measured the heat transfer coefficient and pressure drop for the R-407C evaporation in the small circular tubes ($D_i=2\text{mm}$). The influences of the refrigerant mass flux, heat flux, saturated temperature and vapor quality on the evaporation heat transfer and frictional pressure drop have been

investigate in detail. The major results obtained can be briefly summarized as follows.

- (1). Both the evaporation heat transfer coefficient and frictional pressure drop increase with the mass flux and vapor quality except at low mass fluxes. A rise in the imposed heat flux results in a significant increase in the evaporation heat transfer coefficient except at low heat fluxes. Nevertheless the influence of the imposed heat flux on the frictional pressure drop is rather slight. The increase in the saturated temperature of R-407C results in a smaller increase in the evaporation heat transfer coefficient when compared that due to a rise in the mass flux. But the trend in the frictional pressure drop is the opposite. The decrease in the R-407C saturated temperature results in a noticeable increase in the frictional pressure drop.
- (2). Empirical correlations for the heat transfer coefficient and friction factor for the R-407C evaporation in the small tubes were provided.

We have also carried out experiments to unravel the saturated flow boiling heat transfer characteristics and associated bubble characteristics of R-407C and R-410A in a horizontal narrow annular duct. The effects of the refrigerant mass flux, imposed heat flux, refrigerant saturated temperature and refrigerants have been examined in detail. A summary of the major findings from the present results is given in the following.

- (1). The temperature undershoot at ONB are insignificant for the saturated flow boiling for all refrigerants used in the horizontal narrow annular duct.
- (2). The saturated flow boiling heat transfer coefficients of R-410A are better than R-407C and R-134a. Besides, raising the imposed heat flux can cause a significant increase in the boiling heat transfer coefficient. However, the effects of the refrigerant mass flux and saturated temperature on the boiling heat transfer coefficient are small.
- (3). The results from the flow visualization show that the mean diameter of the bubbles departing from the heating surface decreases slightly with increasing refrigerant mass flux. Besides, at a high imposed heat flux many bubbles generated from the cavities in the heating surface tend to merge together to form big bubbles. The bubble departure frequency increases with the increasing refrigerant mass flux and saturated temperature. The active nucleation site density is much lower at a higher refrigerant mass flux and lower saturation temperature. The higher active nucleation site density on the heating surface for the R-410A flow is due to its smaller surface tension and is

the main reason that the flow boiling heat transfer performance of R-410A is better than R-407C and R-134a.

Finally during the present study three articles have been published in the International Journal of Heat and Mass Transfer up to this point. They are included here for more detailed reference of the boiling of R-134a, R-407C and R-410A in small channels.

1. Y.M. Lie and T.F. Lin, Saturated flow boiling heat transfer and associated bubble characteristics of R-134a in a narrow annular duct, International Journal of Heat and Mass Transfer 48(25-26)(2005)5602-5615.
2. Y.M. Lie and T.F. Lin, Subcooled flow boiling heat transfer and associated bubble characteristics of R-134a in a narrow annular duct, International Journal of Heat and Mass Transfer 49(13-14)(2006)2077-2089.
3. Y.M. Lie, F.Q. Su, R.L. Lai and T.F. Lin, Experimental study of evaporation heat transfer characteristics of refrigerants R-134a and R-407C in horizontal small tubes , International Journal of Heat and Mass Transfer 49(1-2)(2006)207-218.

REFERENCES

1. J. P. Meyer, Experimental evaluation of five refrigerants as replacements for R-22, ASHRAE Transactions: Symposia 106 (2000) 583-588.
2. S. Devotta, A. V. Waghmare, N. N. Sawant, B. M. Domkundwar, Alternatives to HCFC-22 for air conditioners, Applied Thermal Engineering 21 (2001) 703-715.
3. P. A. Newman, Preserving earth's stratosphere, Mechanical Engineering (1998) 88-91.
4. S. Gopalnaryanan, Choosing the right refrigerant, Mechanical Engineering (1998) 92-95.
5. C. C. Wang, C. S. Kuo, Y. J. Chang, D. C. Lu, Two-phase flow heat transfer and friction characteristics of R-22 and R-407C, ASHRAE Transactions: Symposia 102 (1996) 830-838.
6. C. S. Kuo, C. C. Wang, Horizontal flow boiling of R-22 and R-407C in a 9.52mm micro-fin tube, Applied Thermal Engineering 16 (1996) 719-731.
7. C. C. Wang, C. S. Chiang, Two-phase heat transfer characteristics for R-22/R-407C in a 6.5 mm smooth tube, International Journal of Heat and Fluid Flow 18 (1997) 550-558.
8. J. Y. Shin, M. S. Kim, S. T. Ro, Experimental study on forced convective boiling heat transfer of pure refrigerants and refrigerant mixtures in a horizontal tube, International Journal of Refrigeration 20 (1997) 267-275.
9. G. M. Lazarek, S. H. Black, Evaporative heat transfer, pressure drop and critical heat flux in a small vertical tube with R-113, International Journal of Heat and Mass Transfer 25 (7) (1982) 945-960.
10. M. W. Wambsganss, D. M. France, J. A. Jendrzeczyk, T. N. Tran, Boiling heat transfer in a horizontal small-diameter tube, ASME Journal of Heat transfer 115 (1993) 963-972.
11. T. N. Tran, M. W. Wambsganss, D. M. France, Small circular and rectangular channel boiling with two refrigerants, International Journal of Multiphase flow 22 (1996) 485-498.
12. Y. Y. Yan, T. F. Lin, Evaporation heat transfer and pressure drop of refrigerant R-134a in a small pipe, International Journal of Heat and Mass Transfer 41 (1998)

4183-4194.

13. Z. Y. Bao, D. F. Fletcher, B. S. Haynes, Flow boiling heat transfer of Freon R-11 and HCFC123 in narrow passages, *International Journal of Heat and Mass Transfer* 43 (2000) 3347-3358.
14. S. Lin, P. A. Kew, K. Cornwell, Two-phase heat transfer to a refrigerant in a 1 mm diameter tube, *International Journal of Refrigeration* 24 (2001) 51-56.
15. W. Yu, D. M. France, M. W. Wambsganss, J. R. Hull, Two-phase pressure drop, boiling heat transfer, and critical heat flux to water in a small-diameter horizontal tube, *International Journal of Multiphase flow* 28 (2002) 927-941.
16. G. R. Warrier, V. K. Dhir, L. A. Momoda, Heat transfer and pressure drop in narrow rectangular channels, *Experimental Thermal and Fluid Science* 26 (2002) 53-64.
17. Y. Fujita, Y. Yang, N. Fujita, Flow boiling heat transfer and pressure drop in uniformly heated small tubes, *Proceedings of the Twelfth International Heat Transfer Conference* 3 (2002) 743-748.
18. S. G. Kandlikar, Fundamental issues related to flow boiling in minichannels and microchannels, *Experimental Thermal and Fluid Science* 26 (2000) 389-407.
19. S. G. Kandlikar, Masahiro Shoji, Vijay K. Dhir, *Handbook of phase change: boiling and condensation*, TAYLOR & FRANCIS, 1999, Chapter 16, p. 414.
20. S. G. Kandlikar, M. E. Steinke, Flow boiling heat transfer coefficient in minichannels – correlation and trends, *Proceedings of the Twelfth International Heat Transfer Conference*, vol. 3 (2002) 785-790.
21. D. Brutin, F. Topin, L. Tadrist, Experimental study on two-phase flow and heat transfer in capillaries, *Proceedings of the Twelfth International Heat Transfer Conference* 3 (2002) 779-784.
22. C. Y. Yang, R. L. Webb, Friction pressure drop of R-12 in small hydraulic diameter extruded aluminum tubes with and without micro-fins, *International Journal of Heat and Mass Transfer* 39 (4) (1996) 801-809.
23. C. Y. Yang, H. T. Chien, S. R. Lu, R. J. Shyu, Friction characteristics of water, R-134a and air in small tubes, Accepted for Publication in the *Journal of Microscale Thermophysical Engineering* 1-7.
24. S. D. Chang, S. T. Ro, Pressure drop of pure HFC refrigerants and their mixtures

- flowing in capillary tubes, *International Journal of Multiphase Flow* 22 (3) (1996) 551-561.
25. W. Tong, A. E. Bergles, M. K. Jensen, Pressure drop with highly subcooled flow boiling in small-diameter tubes, *Experimental Thermal and Fluid Science* 15 (1997) 202-212.
 26. T. N. Tran, M. C. Chyu, M. W. Wambsganss, D. M. France, Two-phase pressure drop of refrigerants during flow boiling in small channels an experimental investigation and correlation development, *International Journal of Multiphase Flow* 26 (2000) 1739-1754.
 27. C. P. Yin, Y. Y. Yan, T. F. Lin and B. C. Yang, Subcooled flow boiling heat transfer of R-134a and associated bubble characteristics in a horizontal annular channel, *Int. J. Heat Mass Transfer*, 43 (2000) 1885-1896.
 28. P. A. Kew and K. Cornwell, Correlations for the prediction of boiling heat transfer in small-diameter channels, *Applied Thermal Engineering* 17 (8-10) (1997) 705-715.
 29. K. Cornwell and P. A. Kew, Boiling in small parallel channels, in: *Proc. of the Internat. Conf. on Energy Efficiency in Process Technology*, Elsevier Applied Science, Athens, Greece, 1992, pp. 624-638.
 30. C. H. Sheng and B. Palm, The visualization of boiling in small diameter tubes, in: *Proc. of International Conference on Heat Transfer and Transport Phenomena in Microscale*, 2000, pp. 204-208.
 31. C. Bang, W. P. Baek, and S. H. Chang, A digital photographic study on nucleate boiling in subcooled flow for water and refrigerant 134a fluids, in: *Proc. of 10th International Conference on Nuclear Engineering*, Vol. 3, 2002, pp. 155-162.
 32. E. L. Bibeau and M. Salcudean, A study of bubble ebullition in forced-convective subcooled nucleate boiling at low pressure, *International Journal of Heat and Mass Transfer* 37 (15) (1994) 2245-2259.
 33. G. E. Thorncroft, J. F. Klausner and R. Mei, An experimental investigation of bubble growth and detachment in vertical upflow and downflow boiling, *Int. J. Heat and Mass Transfer*, 41 (23)(1998) 3857-3871.
 34. G. E. Thorncroft and J. F. Klausner, The influence of vapor bubble sliding on forced convection boiling heat transfer, *ASME J. Heat Transfer*, 121 (1999) 73-79.

35. O. Zeitoun and M. Shoukri, Bubble behavior and mean diameter in subcooled flow boiling, *ASME J. Heat Transfer*, 118 (1996) 110-116.
36. J. F. Klausner, R. Mei, D. M. Bernhard, and L. Z. Zeng, Vapor bubble departure in forced convection boiling, *Int. J. Heat and Mass Transfer*, 36(3) (1993) 651-662.
37. L. H. Chien and R. L. Webb, Measurement of bubble dynamics on an enhanced boiling surface, *Experimental Thermal and Fluid Science*, 16 (1998) 177-186.
38. S. R. Yang and R. H. Kim, A Mathematical Model of Pool Boiling Nucleation Site Density in Terms of Surface Characteristics, *Int. J. Heat Mass Transfer*, 31(1988)1127–1135.
39. R. F. Gaertner, Distribution of Active Sites in the Nucleate Boiling of Liquids, *Chem. Eng. Prog., Symp. Ser.*, 59(1963)52–61.
40. M. Sultan and R. L. Judd, Spatial Distribution of Active Sites and Bubble Flux Density, *ASME J. Heat Transfer*, 100(1978)56–62.
41. L. Z. Zeng and J. F. Klausner, Nucleation Site Density in Forced Convection Boiling, *ASME J. Heat Transfer*, 115,(1993) 215–221.
42. G. Kocamustafaogullari and M. Ishii, Interfacial Area and Nucleation Site Density in Boiling Systems, *Int. J. Heat Mass Transf.*, 26(9)(1983)1377–1387.
43. N. Basu, G.R. Warrier, V.K. Dhir, Onset of nucleate boiling and active nucleation site density during subcooled flow boiling, *J. Heat Transfer* 124 (2002) 717–728.
44. J. G. Collier, Convective boiling and condensation, McGraw-Hill International Book Company, 2nd ed (1982) p.32, pp. 90-93 and p.341
45. S. M. Zivi, Estimation of steady state steam void-fraction by means of principle of minimum entropy production, *Transaction of the ASME Journal of Heat Transfer* 86 (1964) 237-252.
46. S. J. Kline, F. A. McClintock, Describing uncertainties in single-sample experiments, *Mechanical Engineering* 75 (1953) 3-8.
47. F. W. Dittus, L. M. K. Boelter, Heat transfer in automobile radiator of the tube type, *Publication in Engineering*, University of California, Berkley, 2 (1930) 250.
48. V. Gnielinski, New equations for heat and mass transfer in turbulent pipe and channel flow, *International Chemical Engineering* 16 (1976) 359-368.

49. Z. Liu, R. H. S. Winterton, A general correlation for saturated and subcooled boiling in tube and annuli, based on a nucleate pool boiling equation, *International Journal of Heat and Mass Transfer* 34 (1991) 2759-2766.
50. K. E. Gungor, R. H. S. Winterton, A general correlation for flow boiling in tubes and annuli, *International Journal of Heat and Mass Transfer* 29 (1986) 351-358.
51. S.B. Choi, R.F. Barron, and R.O. Warrington, Fluid flow and heat transfer in microtubes, *Micromech. Sensors Actuat. Syst. ASME DSC* 32 (1991) 123–134.
52. M. Ciampi, S. Faggiani, W. Grassi, and G. Tuoni, Mixed convection heat transfer in horizontal, concentric annuli for transitional flow conditions, *International Journal of Heat and Mass Transfer* 30 (1987) 833-841.