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# An experimental study of flow boiling in a rectangular channel with offset strip fins

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## Abstract

An experimental study on saturated flow boiling heat transfer of R113 was performed in a vertical rectangular channel with offset strip fins. Two-phase pressure gradients and boiling heat transfer coefficients in an electrically heated test section were measured for the quality range of 0–0.6, mass flux range of 17–43 $kg/m^2$ s and heat flux of 500–3000 $W/m^2$. Two-phase frictional multiplier was determined as a function of Martinelli parameter. The two-phase forced convective component of the local boiling heat transfer coefficient was found to be well correlated with the Reynolds number factor. A superposition method for the flow boiling heat transfer coefficient that included the contribution of saturated nucleate boiling was verified also for flow boiling in a channel with offset strip fins. The predictions of local flow boiling heat transfer coefficients were found to be in good agreement with experimental data.


*Keywords:* Flow boiling heat transfer; Rectangular channel; Offset strip fins; Two-phase frictional multiplier; Martinelli parameter; Reynolds number factor

## 1. Introduction

The reduced size and enhanced performance of plate-fin heat exchangers have made them to be used in a wide variety of single-phase applications. The fin geometry can be plain fins, offset strip fins, perforated fins, wavy fins, pin fins and louvered fins. Among these, offset strip fin is frequently adopted for its high heat transfer coefficients. Heat transfer enhancement is achieved by periodic growth of boundary layer on the fin length and their dissipation in the fin wakes, which accompanies an increase in pressure drop due to increased friction and forms drag. Recently, it has become increasingly apparent that plate-fin geometry is also applicable to the phase-change side of both evaporators and condensers in refrigeration, air-conditioning, chemical process and cryogenic systems. Flow boiling of coolant in a channel with fins also provides an effective means to cool electronic devices.

There has been considerable effort to understand the convection mechanisms and to predict the thermal hydraulic behavior of single-phase heat transfer in plate-fin heat exchangers with offset strip fins by London and Shah (1968), Wieting (1975), Joshi and Webb (1987), Manglik and Bergles (1990), and others. In particular, Manglik and Bergles (1995) have compiled and analyzed the heat transfer and friction factor data for 18 offset strip surfaces and shown that they were affected by the fin geometric parameters and the Reynolds number. They have also developed correlations that describe the asymptotic behavior of the data in the deep laminar and fully turbulent regions. However the validity of their correlations for liquids is open to doubt.

The flow conditions in plate-fin channel are quite dissimilar to those in round tubes. Flow passages are small rectangular finned channels with walls spaced approximately millimeters apart. To avoid excessive pressure drop, mass

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## Nomenclature

| | |
|---|---|
| $A_p$ | primary surface area of channel, m$^2$ |
| $A_s$ | surface area of fins in channel, m$^2$ |
| $C$ | constant in Martinelli parameter |
| $D_h$ | hydraulic diameter, m |
| $F$ | Reynolds number factor |
| $f$ | friction factor |
| $G$ | mass velocity, kg/m$^2$ s |
| $g$ | gravitational acceleration, m/s$^2$ |
| $H$ | fin height, m |
| $h$ | heat transfer coefficient, W/m$^2$ K |
| $h_{fg}$ | latent heat of vaporization, kJ/kg |
| $j$ | Colburn factor |
| $k_s$ | thermal conductivity of fin, W/m K |
| $L$ | length of test section, m |
| $l$ | fin length, m |
| $M$ | molecular weight |
| $p$ | pressure, kPa |
| $Q$ | heat transfer rate, W |
| $q''$ | heat flux, W/m$^2$ |
| $Re$ | Reynolds number |
| $S$ | nucleate boiling suppression factor |
| $s$ | lateral fin spacing, m |
| $T$ | temperature, °C |
| $t$ | fin thickness, m |
| $v$ | specific volume, m$^3$/kg |
| $W$ | mass flow rate, kg/s |
| $X$ | Martinelli parameter |
| $x$ | quality |
| $z$ | coordinate in flow direction |

*Greek symbols*

| | |
|---|---|
| $\alpha$ | void fraction |
| $\beta$, $\delta$, $\gamma$ | fin geometric parameters |
| $\eta$ | fin efficiency |
| $\rho$ | density, kg/m$^3$ |
| $\sigma$ | surface tension, N/m |
| $\phi$ | two-phase frictional multiplier |

*Subscripts*

| | |
|---|---|
| A | acceleration |
| F | friction |
| c | critical |
| cb | convective boiling |
| exp | experiment |
| e | exit of test section |
| f | liquid |
| G | gravitation |
| g | gas |
| nb | nucleate boiling |
| pre | prediction |
| pb | pool boiling |
| sat | saturated state |
| TP | two-phase |
| w | wall |

velocities in the finned passage are kept low, usually less than 100 kg/m$^2$ s. Heat fluxes are about 3 kW/m$^2$. Consequently, the results of experimental work using large-bore round tubes, usually at high mass fluxes and heat fluxes, may not be applicable to plate-fin heat exchangers. Moreover, the characteristics of phase-change in these small channels, where most of heat transfer area resides on fins or secondary surfaces, may show some unusual aspects.

The characteristics of boiling and condensation heat transfer in a channel with offset strip fins have not been widely studied, but a few analytical and experimental investigations were reported. Robertson and Lovegrove (1983) studied the flow boiling in a channel with offset strip fins at very low wall superheat levels. Measured local boiling heat transfer coefficients were obtained for R11 and compared with those for nitrogen by Robertson (1979). For the conditions considered in their study, the effects of nucleate boiling were found to be negligible and the dominant heat transfer mechanism was two-phase forced convection. A systematic approach by using a special test section, which enables the direct visual observation of the two-phase flow and simultaneous measurement of convective boiling heat transfer coefficient for methanol and water, was performed by Mandrusiak et al. (1988) and Mandrusiak and Carey (1988, 1989). A two-phase flow regimes map, theoretical model for predicting the two-phase frictional multiplier and empirical correlations for boiling heat transfer coefficients were developed. However the finned channel was partially heated for the visual observations, and the offset strip fins machined in the copper slab were too thick to represent the actual fins. In more recent studies, Thonon et al. (1997) and Feldman et al., 2000 conducted experiments with CFC114 flowing in an electrically heated test section of finned passage to measure the local boiling heat transfer coefficients. They incorporated the effects of quality, mass flux and heat flux individually and proposed a semi-empirical model for predicting the boiling heat transfer coefficients. Nevertheless the accuracy of the model seems to be inadequate for designing plate-fin heat exchangers.

The present study was carried out to accomplish two main objectives. The first objective was to propose a method for the prediction of pressure drop during flow boiling in a channel with offset strip fins. The second objective was to obtain the local boiling heat transfer coefficient and to explain the contribution of nucleate boiling and two-phase forced convection.

## 2. Experimental investigations

### 2.1. Experimental apparatus

Experimental studies on the flow boiling process in a channel with offset strip fins were conducted using an experimental setup shown in Fig. 1. It consisted of two circuits: a refrigerant circuit and a circuit of water with constant temperature.

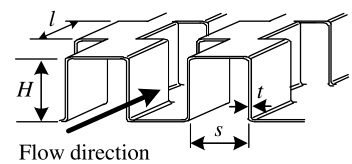
The test section was placed in the refrigerant circuit. R113 was selected as the refrigerant because of its low latent heat of vaporization as well as handling convenience. The gear pump circulated liquid through a mass flow meter (Coriolis type, 0–0.9 kg/min), a preheater, the test section and then a water-cooled condenser at which refrigerant vapor condensed before it returned to the storage tank. A control valve at the outlet of the gear pump enabled the regulation of the refrigerant flow rate.

The test section consisted of a single channel with offset strip fins sandwiched between two stainless steel side plates. It is a 750 mm long rectangular duct with internal dimensions of 3 mm × 100 mm. Ten sheets of silicone rubber electrical heater, each of 120 mm × 150 mm, were glued to plates to supply the heat necessary for the flow boiling. Stainless steel plates with 5 mm thickness were used to produce a reasonably uniform heat flux on their inner surfaces in contact with the fluid. Heat fluxes were available up to 3000 W/m$^2$ and were measured by the wattmeter. The test range and the dimensions of the offset strip fins are given in Table 1.

Two immersion heaters were installed in the storage tank for coarse regulation of liquid temperature. A 300 W preheater unit located near the inlet of the test section enabled fine control of the liquid temperature before it flowed into the test section. The preheater was a stainless-steel tube wrapped with a silicone rubber heater. The test section, the preheater and the pipe segment between the preheater and the test section were heavily insulated to minimize heat loss to the environment.

Table 1
Test range and dimensions of offset strip fins

| Variables | Range |
|---|---|
| System pressure, $p$ (kPa) | 102.4 |
| Mass flux, $G$ (kg/m$^2$ s) | 17–43 |
| Heat flux, $q''$ (W/m$^2$) | 500–3000 |
| Quality, $x$ | 0–0.6 |




| Variables | Range |
|---|---|
| Fin height, $H$ (mm) | 2.8 |
| Fin length, $l$ (mm) | 1.5 |
| Lateral fin spacing, $s$ (mm) | 3.5 |
| Fin thickness, $t$ (mm) | 0.2 |
| Hydraulic diameter, $D_h$ (mm) | 2.84 |

Liquid bulk temperatures at the inlet and outlet of the test section and wall temperatures at six locations along the length of test section were measured using $J$-type thermocouples with 1-mm-diameter stainless-steel sheath. To check out the uniformity of wall temperature along its width, five thermocouples were imbedded in the plate transverse to the flow direction. The thermocouples were 600 mm apart from the inlet of the test section. The vapor temperature in the condenser and the liquid temperature at the outlet of the storage tank were also recorded. The absolute pressure at the outlet of the test section and the differential pressure along the length of the test section were measured by the strain gauge transducers. The pressure of the liquid at the inlet of the test section is given by a simple sum of these two values. The location of differential pressure ports in the test section was carefully determined so that the opening should not clogged by the fins in a channel. The pressures of the condenser and the storage tank were also measured.

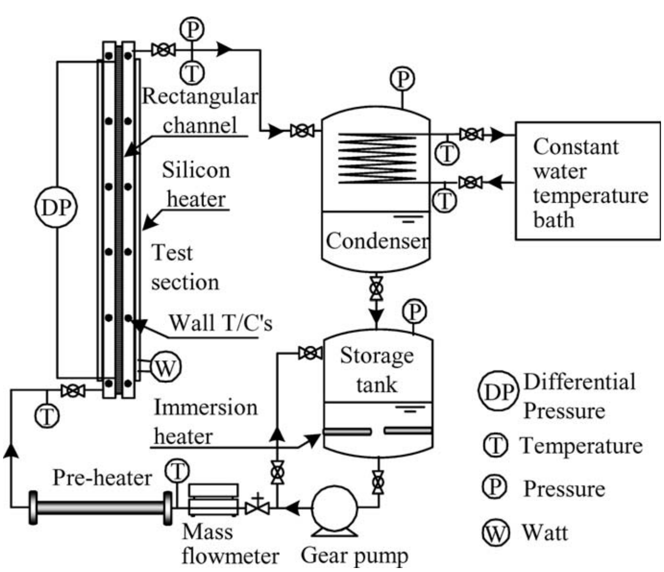



Fig. 1. Schematic of experimental apparatus.

### 2.2. Experimental procedure

All the heat transfer tests were carried out with saturated liquid R113 at the entry to the test section and boiling occurred within the heated length of the test section. For those tests investigating the single-phase heat transfer characteristics, R113 and water remained subcooled over the whole length of the test section.

Before each experiment the refrigerant circuit was evacuated with a mechanical vacuum pump to remove the air that might have entered from the ambient and accumulated in the circuit.

An experimental run was conducted as follows: the required flow rate of refrigerant liquid was obtained by adjusting the control valve at the outlet of the gear pump in the main circuit. The electrical heater power was then applied to the immersion heaters in the storage tank and to the preheater to control the liquid temperature at the inlet of the test section. Once the power was supplied to the test section, the boiling and evaporation of refrigerant liquid proceeded and the circuit pressure increased gradually. By continually adjusting the flow rate and temperature of the cooling water supplied to the condenser, it was possible to condense the incoming refrigerant vapor from the test section and to maintain the system pressure slightly above the ambient pressure. In this way the required experimental conditions have been attained at thermal equilibrium. The flow rate of the refrigerant liquid, the temperatures of the fluid and the wall, and the absolute and differential pressures were recorded for the later analysis.

An energy balance was examined by comparing the measured increase of liquid bulk temperature along the test section with the calculated value using measured mass flow rate, heat input and the appropriate fluid thermal properties in the single-phase heat transfer experiments. The difference between the calculated and experimental values was attributed mainly to heat loss from the test section to the ambient and the difference found in each run (approximately 7%) was taken into account in the analysis of the experimental data.

Based on the method suggested by Kline (1985), an uncertainty analysis was conducted to determine the maximum error for the local boiling heat transfer coefficients and the two-phase frictional multiplier. The uncertainty in the measurements was estimated to be ±3% for the mass velocity, ±3% for the pressure drop and ±4% for the local vapor quality. The uncertainty in two-phase frictional multiplier was estimated to be ±12.5%. The uncertainty in the boiling heat transfer coefficient for all the reported experimental data was ±8.7%, which was unavoidable since the temperature difference between the wall and the saturated liquid was small for the heat flux range tested.

## 3. Results and discussion

### 3.1. The single-phase flow and heat transfer

Before running the flow boiling experiments, the single-phase flow and heat transfer characteristics in a channel with offset strip fins were determined first to establish the fundamental data and to check the reliability of the experimental apparatus and procedure. The measured single-phase flow and heat transfer data, using water and R113 as the test fluid, are given in terms of the friction factor $f$ and the Colburn factor $j$ as a function of the Reynolds number in Fig. 2. Manglik and Bergles (1995) proposed

$$f = 9.6243Re_{D_h}^{-0.7422}\beta^{-0.1856}\delta^{-0.3053}\gamma^{-0.2659} \times [1 + 7.669 \times 10^{-8}Re_{D_h}^{4.429}\beta^{0.92}\delta^{3.767}\gamma^{0.236}]^{0.1} \quad (1)$$

$$j = 0.6522Re_{D_h}^{-0.5403}\beta^{-0.1541}\delta^{0.1499}\gamma^{0.0678} \times [1 + 5.269 \times 10^{-5}Re_{D_h}^{1.34}\beta^{0.504}\delta^{0.456}\gamma^{-1.055}]^{0.1} \quad (2)$$

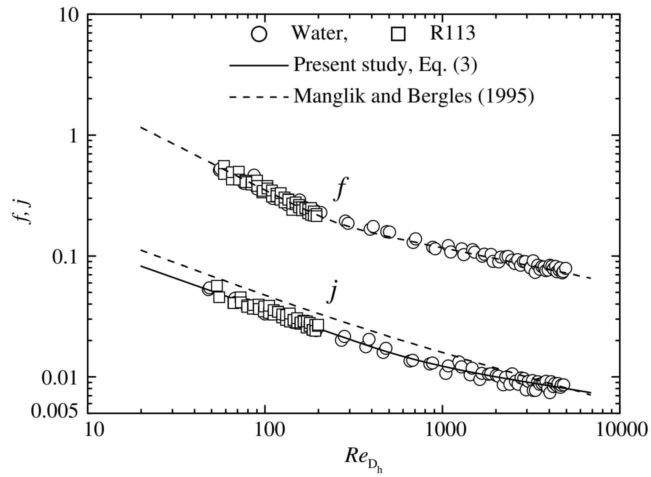



Fig. 2. Single-phase flow friction factor and Colburn factor in a channel with offset strip fins.

where the dimensionless parameters $\beta = s/H$, $\delta = t/l$ and $\gamma = t/s$ describe the offset strip fin geometry, and $s$, $t$, $l$ and $H$ denote lateral fin spacing, fin thickness, length and height, respectively. Good agreement of Eq. (1) with the results of the present study is evident with the relative error of within ±15%. However empirical data of Colburn factors show discrepancy from Eq. (2). Basically Eq. (2) was correlated for the flow of low Prandtl number fluids, such as air and gas. The flow for a larger Prandtl number has a longer thermal developing region on the channel wall and on each fin, which leads to a higher average heat transfer rate but a smaller Colburn factor. Hu and Herold (1995) have emphasized that the air-cooled model, such as Eq. (2), tends to overpredict the Colburn factor for liquids at a given Reynolds number. They proposed that the Colburn factors of the liquid and air differ by a factor of approximately two. However, the Colburn factor measured in the present study was about 25% smaller than the prediction of Eq. (2) for $Re < 1000$. For $Re > 1000$, the difference between the experimental data and the prediction of Manglik and Bergles (1995) reduced as the Reynolds number increased.

For the flow of water and R113 in a channel with offset strip fins adopted in the present study, experimental data on the Colburn factor is correlated as

$$j = 0.389Re_{D_h}^{-0.518}[1 + 1.2 \times 10^{-8}Re_{D_h}^{2.76}]^{0.1} \quad (3)$$

In fact, 92% of the Colburn factor data lies within ±12% of Eq. (3) and the root-mean-square error is 6.3%.

### 3.2. Pressure drop in flow boiling

The two-phase pressure gradient in the boiling channel consists of the frictional, accelerational and gravitational pressure gradients

$$-\left.\frac{dp}{dz}\right)_{TP} = -\left.\frac{dp}{dz}\right)_F - \left.\frac{dp}{dz}\right)_A - \left.\frac{dp}{dz}\right)_G \quad (4)$$

For the case where a liquid is evaporated from liquid at the saturation temperature to form a vapor–liquid mixture with a linear change of local quality $x$ over a length $L$, the overall pressure drop in the test section becomes

$$\Delta p_{\mathrm{TP}} = \frac{2G^2 v_{\mathrm{f}} L}{D_{\mathrm{h}}}\left[\frac{1}{x}\int_0^x f_{\mathrm{f}}\phi_{\mathrm{f}}^2\,\mathrm{d}x\right] + G^2\left[\frac{x^2 v_{\mathrm{g}}}{\alpha} + \frac{(1-x)^2 v_{\mathrm{f}}}{1-\alpha} - 1\right] + \frac{Lg}{x}\int_0^x [\alpha/v_{\mathrm{g}} + (1-\alpha)/v_{\mathrm{f}}]\,\mathrm{d}x \qquad (5)$$

where $G$ is the mass velocity, $v$ is the specific volume, $D_{\mathrm{h}}$ is the hydraulic diameter and $f_{\mathrm{f}}$ is the friction factor. The subscripts f and g denote the liquid and the vapor, respectively. The accelerational and gravitational pressure gradients depend upon the specific volumes of the both phases, the local quality $x$ and the local void fraction $\alpha$. $g$ denotes the gravitational acceleration.

The two-phase frictional pressure gradient is usually expressed in terms of single-phase pressure gradient for the liquid phase flowing alone in the channel. The two-phase frictional multiplier in round tubes, $\phi_{\mathrm{f}}^2$, was uniquely correlated as a function of a parameter $X$ by Lockhart and Martinelli (1949), where

$$\phi_{\mathrm{f}}^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \qquad (6)$$

$$X = \left[\frac{(\mathrm{d}p/\mathrm{d}z)_{\mathrm{F,f}}}{(\mathrm{d}p/\mathrm{d}z)_{\mathrm{F,g}}}\right]^{1/2} = \frac{(1-x)}{x}\left[\frac{f_{\mathrm{f}}}{f_{\mathrm{g}}}\frac{v_{\mathrm{f}}}{v_{\mathrm{g}}}\right]^{1/2} \qquad (7)$$

A constant $C$ depends upon the flow characteristics of each phase, i.e., laminar or turbulent.

Fig. 3 shows the measured two-phase flow pressure drop as a function of exit quality for flow boiling in the test section. Basically two-phase pressure drop in a channel with offset strip fins was larger than that in a channel without fins. At low qualities, the presence of the fins redirected the two-phase flow periodically and increased the two-phase pressure drop above the value caused by wall shear alone. At high qualities, the interaction between the high-velocity vapor flow with the fins produced an even larger pressure drop. The frictional pressure drop in the test section was determined by correcting the measured pressure drop for the effects of gravity and acceleration. The relation between the void fraction and the quality is necessary for the estimation of the accelerational and gravitational pressure drop. For the present study the homogenous flow model is adopted to calculate the void fraction from the quality since the two phases are distributed almost uniformly in the channel due to the presence of fins as mentioned by Sohn (2002) in his empirical study on an adiabatic two-phase flow in a channel with offset strip fins. Numerical integration was carried out to estimate the gravitational pressure drop in the channel.

The two-phase frictional pressure drop was plotted in terms of $\phi_{\mathrm{f}}$ and $1/X$ in Fig. 4. The inverse Martinelli parameter is calculated by substituting Eq. (1) into Eq. (7) for the liquid and the vapor phases. From an energy balance, the local quality of two-phase fluid is estimated by

$$x(z) = \frac{Qz}{WLh_{\mathrm{fg}}} \qquad (8)$$

where $W$ denotes the mass flow rate and $i_{\mathrm{fg}}$ is the latent heat of vaporization. The predictions of the Lockhart and Martinelli equation for flow in round tubes, Eq. (6), are also given for comparison. The disagreement between the experimental data and the predictions for round tubes is quite pronounced especially at high values of $1/X$. At low qualities, the periodic redirection of two-phase flow by the offset fins appears to increase the two-phase pressure drop slightly above the value caused by wall friction alone. At high qualities, the vapor flow of high velocity interacts with the fins to produce a larger pressure drop than that observed in round tubes.

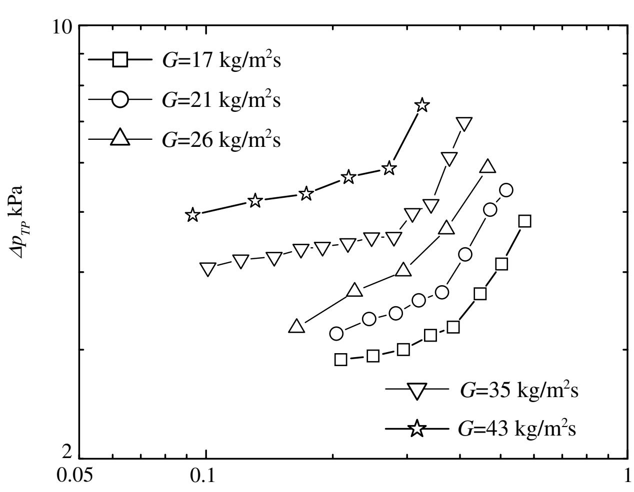



Fig. 3. Measured two-phase flow pressure drop as a function of exit quality for flow boiling in a channel with offset strip fins.

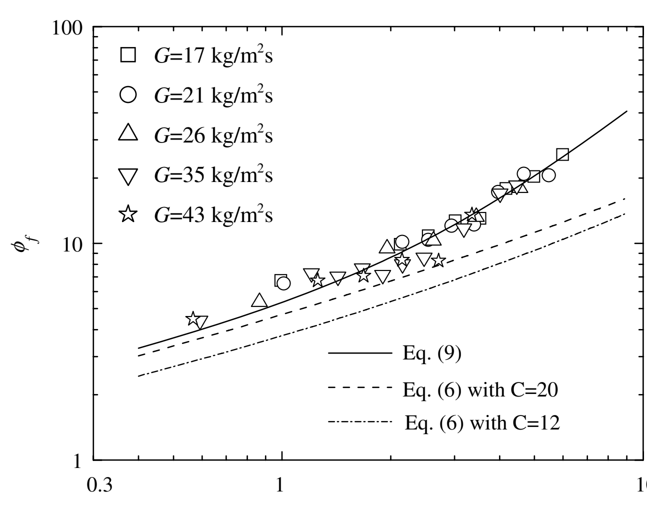



Fig. 4. Experimental data of two-phase frictional multipliers in comparison with the prediction of Lockhart and Martinelli for round tubes.

Mandrusiak and Carey (1988) mentioned that the effects of fin form drag become more significant at higher qualities and the two-phase frictional multiplier for flow through the passage with the offset strip fins is consistently about 50% higher than that for round tubes under comparable flow conditions. However the present study shows a stronger dependence of $\phi_f$ on $1/X$ than Mandrusiak and Carey have found. The difference in two-phase frictional multiplier between the experimental findings of the present study and the prediction for round tubes is even bigger than those suggested by Mandrusiak and Carey, especially at higher qualities. The fins in their experiments were machined on the copper slab and were 3–5 times bigger in thickness, length and height than the actual ones. For instance, the hydraulic diameter was 8.84 mm, which is much greater than that of the present study at 2.84 mm. These geometrical differences seem to result in a serious discrepancy of the fin form drag.

Experimental data on the two-phase frictional multiplier in a channel with offset strip were correlated as Eq. (9) with an error bound of ±20%

$$\phi_f^2 = 1 + \frac{23.4}{X} + \frac{4.17}{X^{2.66}} \tag{9}$$

Two-phase frictional multiplier was originally conceived to quantify the effect of two-phase flow on the wall friction. For a channel with fins, two-phase pressure drop depends on the fin form drag as well as on the wall friction. Then Martinelli parameter alone is not sufficient to predict the two-phase pressure drop in a channel with fins. Two-phase multiplier, as attempted by Mandrusiak and Carey (1988), has to account for the simultaneous effects of wall friction and fin form drag, which is beyond the scope of the present study.

### 3.3. Flow boiling heat transfer

In flow boiling, heat is transferred to the refrigerant liquid by nucleate boiling as well as by two-phase forced convection mechanisms from both the primary surface and the secondary surface, i.e., fins. By analogy to single-phase convection, heat transfer rate is related to the overall heat transfer coefficient $h_{TP}$ by

$$h_{TP} = \frac{Q}{(A_p + \eta A_s)(T_w - T_{sat})} \tag{10}$$

where $A_p$ and $A_s$ are the primary and the secondary surface area in a channel, respectively. Subscripts w and sat denote the wall and the saturated state. The local saturation temperature of the liquid is estimated from the local pressure in the test section. The fin efficiency $\eta$ is given by

$$\eta = \tanh(mH)/mH \tag{11}$$

$$m = [2h_{TP}(t + l)/(k_s t l)] \tag{12}$$

where $k_s$ is the thermal conductivity of the fins.

Measured values of the local boiling heat transfer coefficient with quality at selected heat inputs and mass velocities are presented in Fig. 5. At low to moderate quality, trends of increasing heat transfer coefficient with increasing mass velocity or quality are easily observed. The dependence of heat transfer coefficient on heat input was not so strong as that in round tubes at low qualities, which was also noticed by Robertson and Lovegrove (1983) and Mandrusiak et al. (1988). It is quite possible that the nucleate boiling on the primary surface occurs simultaneously with the forced convection of liquid phase on the fins. Therefore the distinction between a nucleate boiling region and a convective boiling region is not very clear in a channel with fins. Besides, the condition of uniform heat flux is not attainable in a channel with offset strip fins since the fin efficiency varies along the channel. The fin efficiency decreases as the two-phase forced convective heat transfer coefficient increases, which results in the decrease of effective heat transfer area in the secondary surface and also results in the increase of heat flux.

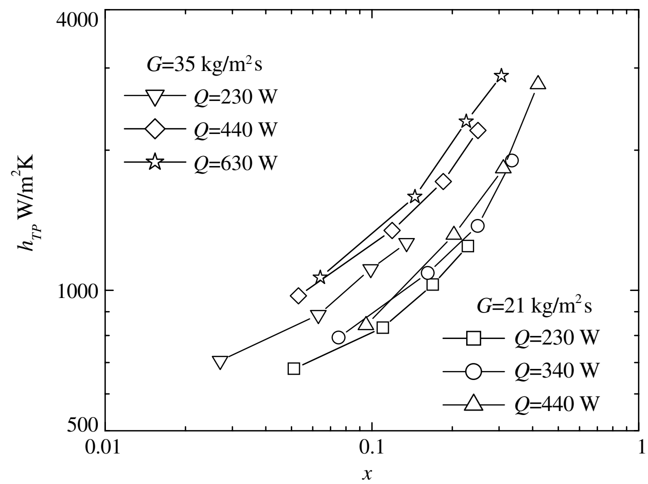



Fig. 5. Measured local heat transfer coefficients as a function of local quality for flow boiling in a channel with offset strip fins.

The local boiling heat transfer coefficient is commonly represented by

$$h_{TP} = h_{nb} + h_{cb} \tag{13}$$

as proposed by Chen (1966). He correlated the two-phase forced convective heat transfer coefficient as a function of the single-phase heat transfer coefficient and the Reynolds number factor $F$

$$F = \frac{h_{cb}}{h_f} \tag{14}$$

where the single-phase heat transfer coefficient $h_f$ can be calculated from Eq. (3). The Reynolds number factor has generally been postulated to be a function of the Martinelli parameter $X$. The measured two-phase forced convective heat transfer data are presented in Fig. 6 by plotting $F$ as a function of $1/X$. All data are shown in the plot regardless of the dominant heat transfer mechanism. Obviously the Reynolds number factor undergoes a transition at $1/X$ slightly greater than 1.0, which implies that the dominant

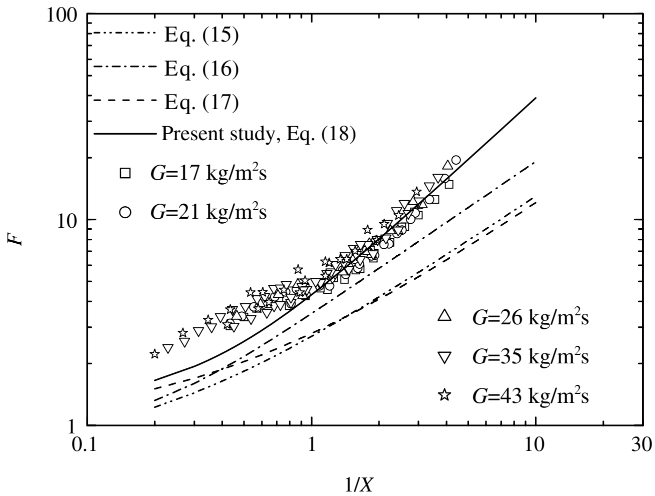



Fig. 6. Experimental data on the Reynolds number factor in comparison with the predictions of correlations reported.

heat transfer mechanism switches from the nucleate boiling to the two-phase forced convection.

The predictions of Eq. (15) by Chen (1966) for round tubes, Eq. (16) proposed by Mandrusiak and Carey (1989) for the flow passage with enlarged offset strip fins and Eq. (17) by Feldman et al. (2000) for the plate-fin channel with serrated or perforated fins are also shown for the comparison

$$F = 2.35\left[0.213 + \frac{1}{X}\right]^{0.736} \tag{15}$$

$$F = \left[1 + \frac{28}{X}\right]^{0.372} \tag{16}$$

$$F = 1 + \frac{1.8}{X^{0.79}} \tag{17}$$

The disagreement between the experimental data of the present study and the predictions of referred correlations is not surprising in view of a disparity between the geometries of the flow passage and the fin. Certainly the prediction of Chen (1966) for round tubes is not applicable to the flow boiling in the plate-fin channels. The correlation of Mandrusiak and Carey (1989) underpredicts $F$ of the present study as their experimental values of the two-phase frictional multiplier are smaller than those of the present study. It is surprising that the prediction of Eq. (17), given by Feldman et al., is similar to that of Chen. It is not clear whether Feldman et al. considered the effect of the fin efficiency on the effective heat transfer area.

The Reynolds number factor in the present study is correlated as

$$F = \left[1 + \frac{2.52}{X^{1/2}} + \frac{15.1}{X^2}\right]^{0.5} \tag{18}$$

for the case of $1/X > 1.0$. With the correlation, all the two-phase forced convection data are predicted within ±9.8% error. Certainly those data for which the nucleate boiling was dominant, $1/X < 1.0$, or the partial dryout of heat transfer surface was suspected have been excluded in the development of empirical correlations of $F$.

Even though the convective boiling heat transfer coefficient is independent of the wall superheat, the nucleate boiling heat transfer coefficient is strongly governed by the local wall superheat. Since the temperature of the fin is not uniform, the individual contributions of the primary and the secondary surface areas to the nucleate boiling heat transfer must be computed separately. The effective nucleate boiling contribution to the local boiling heat transfer coefficient for the primary and the secondary surface area is given by

$$h_{\mathrm{nb}} = \frac{h_{\mathrm{nb,p}}A_{\mathrm{p}} + h_{\mathrm{nb,s}}\eta A_{\mathrm{s}}}{A_{\mathrm{p}} + \eta A_{\mathrm{s}}} \tag{19}$$

where $h_{\mathrm{nb,p}}$ is the nucleate boiling heat transfer coefficient in the primary surface and $h_{\mathrm{nb,s}}$ is calculated based on the wall superheat of $\eta(T_{\mathrm{w}} - T_{\mathrm{sat}})$. The nucleate boiling component of the local boiling heat transfer coefficient is predicted using the suppression factor $S$ and the pool boiling heat transfer coefficient proposed by Nishikawa et al. (1982)

$$h_{\mathrm{nb}} = Sh_{\mathrm{pb}} \tag{20}$$

$$h_{\mathrm{pb}} = 31.4\left[\frac{p_{\mathrm{c}}^{0.2}F_{\mathrm{p}}}{M^{0.1}T_{\mathrm{c}}^{0.9}}\right]q''^{0.8} \tag{21}$$

$$F_{\mathrm{p}} = \frac{(p/p_{\mathrm{c}})^{0.23}}{[1.0 - 0.99(p/p_{\mathrm{c}})]^{0.9}} \tag{22}$$

where $M$ is the molecular weight, $q''$ is the local heat flux and the subscript c denotes the critical constant. The suppression factor for flow boiling in round tubes, proposed by Bennett et al. (1980), is directly extended to the flow boiling in a channel with offset strip fins in the present study

$$S = \frac{24.4}{N_{\mathrm{B}}}[1 - \exp(-0.041N_{\mathrm{B}})] \tag{23}$$

$$N_{\mathrm{B}} = \frac{h_{\mathrm{f}}}{k_{\mathrm{f}}}\left[\frac{\sigma}{g(\rho_{\mathrm{f}} - \rho_{\mathrm{g}})}\right]^{0.5} \tag{24}$$

where $\sigma$ is the surface tension and $\rho$ is the density. The suppression factor could be correlated as a function of the bubble growth factor, $N_{\mathrm{B}}$, by manipulating the experimental data. For the present study, however, empirical correlation for the suppression factor shows considerable scatter and is no more accurate than that of Bennett et al.

Experimentally measured flow boiling heat transfer coefficients are compared with the values predicted by Eq. (13) in Fig. 7. For the predictions, Eq. (18) is used for the estimation of the Reynolds number factor $F$ and Eq. (23) proposed by Bennett et al. is employed to calculate the suppression factor $S$. Predicted values of the local boiling heat transfer coefficient are in good agreement with the experimental data. The correlation has a tendency to overpredict the contribution of nucleate boiling to the local

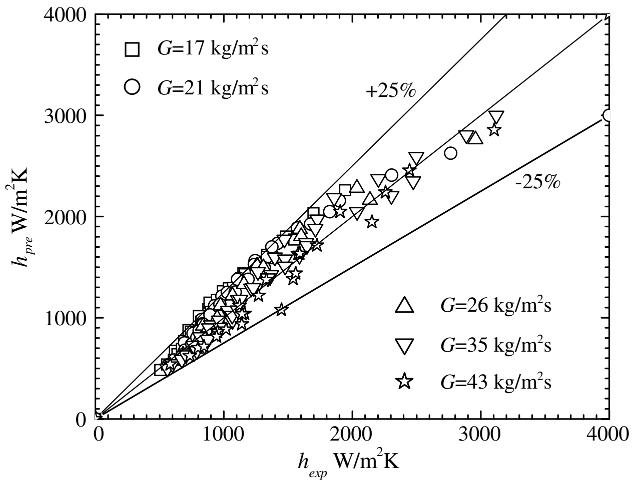



Fig. 7. Comparison of measured boiling heat transfer coefficients with values predicted by the correlation proposed in the present study.

boiling heat transfer coefficient, especially at low quality. The discrepancy was unavoidable since the contribution of the primary and the secondary surface area to the nucleate boiling heat transfer coefficient was hardly computed separately and the suppression factor was unable to include the effect of the fins on the nucleate boiling heat transfer. However, all the data of local boiling heat transfer coefficient lies within ±25% of the predictions. The mean difference between the predicted local boiling heat transfer coefficients and the experimental measurements is 13%.

## 4. Conclusions

In the present study, an experimental investigation was performed to analyze the flow boiling heat transfer of R113 in a vertical rectangular channel with offset strip fins. Two-phase frictional multiplier and the local boiling heat transfer coefficients were measure and correlated.

Measured values of two-phase frictional multiplier in a channel with offset strip fins were higher by as much as 80% than those in round tubes within the test range of present study. The deviation from the results for a round tube is even larger at higher qualities, at which the contribution of the fin form drag becomes more significant due to the periodic redirection of two-phase flow of higher mass velocity.

Heat transfer data for the two-phase forced convection were well equated by the Reynolds number factor used for convective boiling in round tubes. The Reynolds number factor, in turn, was correlated by the Martinelli parameter with reasonable accuracy. The difference in the Reynolds number factor between the flow in the finned passage and the flow in round tubes increases with the inverse Martinelli parameter, up to 100% at the upper limit of quality considered in the present study. An superposition method that includes the saturated nucleate boiling component in the flow boiling heat transfer coefficient was also valid for the flow boiling in a rectangular channel with offset strip fins. Measured values of local flow boiling heat transfer coefficients could be predicted to within ±25% of the correlation proposed in the present study.

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