Counter-Current Air-Water Flow in Narrow Rectangular Channels With Offset Strip Fins

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Counter-current two-phase flows of air-water in narrow rectangular channels with offset-strip fins have been experimentally investigated in a 760 mm long and 100 mm wide test section with 3.0 and 5.0 mm gap widths. The two-phase flow regime, channel-average void fractions and two-phase pressure gradients were studied. Flow regime transition occurred at lower superficial velocities of air than in the channels without fins. In the bubbly and slug flow regimes, elongated bubbles rose along the subchannel formed by fins without lateral movement. The critical void fraction for the bubbly-to-slug transition was about 0.14 for the 3 mm gap channel and 0.2 for the 5 mm gap channel, respectively. Channel-average void fractions in the channels with fins were almost the same as those in the channels without fins. Void fractions increased as the gap width increased, especially at high superficial velocity of air. The presence of fins enhanced the two-phase distribution parameter significantly in the slug flow, where the effect of gap width was almost negligible. Superficial velocity of air dominated the two-phase pressure gradients. Liquid superficial velocity and channel gap width has only a minor effect on the pressure gradients.

Key Words: Counter-Current Two-phase Flow, Narrow Rectangular Channel, Offset Strip Fins, Flow Regime, Void Fraction, Two-Phase Pressure Gradient

Nomenclature

a_1, a_2, a_3	: Dimensionless parameters in Eq. (12)
B	: Constant in Eq. (19)
C	: Constant in Wallis's flooding correlation
C_o	: Distribution parameter
D	: Diameter [m]
D_h	: Hydraulic diameter [m]
f	: Friction factor
g	: Gravitational acceleration [m/s^2]
h	: Fin height [m]
j	: Superficial velocity [m/s]
K	: Constant in Eq. (3)
l	: Length [m]
m	: Constant in Wallis's flooding correlation
n	: Exponent in Eq. (3)
n_1, n_2	: Exponents in Eq. (19)
p	: Pressure [Pa]
Q	: Volumetric flow rate [m^3/s]
Re	: Reynolds number
s	: Lateral fin spacing [m]
t	: Fin thickness [m]
u	: Velocity [m/s]
u_{gj}	: Drift velocity [m/s]
X, Y, Z	: Dimensionless parameters
z	: Coordinate in flow direction [m]

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Greek Symbols

α : Void fraction
α_o : Void fraction at zero liquid flow rate
δ : Film thickness [m]
μ : Dynamic viscosity [Ns/m^2]
ρ : Density [kg/m^3]
σ : Surface tension [N/m]
τ : Shear stress [N/m^2]

Subscripts

a : Acceleration
b : Bubble
f : Liquid
g : Gas
i : Liquid-gas interface
TP : Two-phase

1. Introduction

Two-phase flow in round tubes has been extensively studied and a considerable body of information on flow regimes, void fractions, and pressure gradients exists in the literature. Application of these findings for round tubes to two-phase flow in non-circular geometries would seem to open to question. The behavior of a gas-liquid mixture confined in narrow channels differs from that in a tube due to the increased surface tension force and frictional pressure drop. Studies of co-current and counter-current two-phase flow in rectangular channels cover a period of more than 30 years. Recently published studies include Mishima et al. (1993), Wilmarth and Ishii (1994), Lee and Lee (1999), and Kim et al. (2001A). The effects of channel gap width and inclination angle on the void fractions and the pressure gradients were also investigated.

Finned flow passage geometries have been mainly used in single-phase compact heat exchangers. Over the past decade, compact high-performance heat exchanger surfaces have been used with increasing frequency for applications involving boiling and condensation. Carey and Mandrusiak (1986) and Mandrusiak et al. (1988) have developed flow pattern maps for co-current vertical upflow and co-current horizontal two-phase flow in a offset strip fin geometry for the conditions encountered during the flow boiling at low to moderate wall superheat levels. Xu and Carey (1987) have similarly developed flow regime maps for the flow boiling in a cross-ribbed channel geometry at moderate wall superheat levels. Souidi and Bontemps (2001) studied the countercurrent gas-liquid flow in narrow rectangular channels with plain and perforated fins. They observed different flow patterns dependent on flow rates and measured the flooding velocities and pressure drops. However a survey of literature is rarely found on counter-current two-phase flow in rectangular channels with offset strip fins, which are widely used in compact heat exchangers.

In this study, counter-current air-water flow in narrow rectangular channels with offset strip fins was investigated experimentally. The objective was to understand the flow regimes, void fractions, and pressure gradients, and take out the design data of plate-fin heat exchangers for the phase-change heat transfer.

2. Experimental Apparatus

The flow apparatus was set up to allow adiabatic flow experiments with air-water mixtures in a channel of small cross-sectional area as shown in Fig. 1. The narrow channel consisted of two acrylic resin plates whose gap was formed by sealing metal strips along the plate's outer boundaries. The width and the length of the test section were 100 mm and 760 mm, respectively. The gap widths were 3.0 and 5.0 mm. Offset strip fins of two kinds were tested and their dimensions are

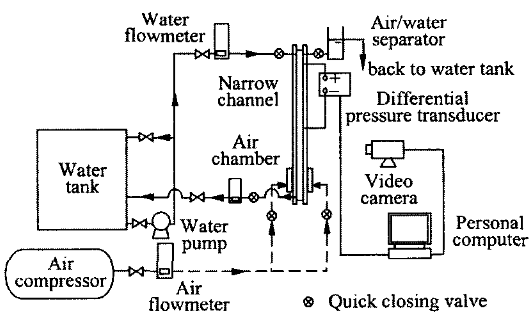
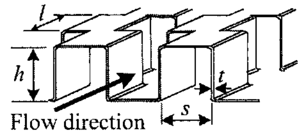


Fig. 1 A schematic of experimental apparatus

Table 1 Dimensions of offset strip fins



Fin height, h (mm)	2.8/4.8
Fin length, l (mm)	1.5/5.0
Lateral fin spacing, s (mm)	3.5/4.6
Fin thickness, t (mm)	0.2/0.2
Fin density, N	28/20.5
Hydraulic diameter, D_h (mm)	2.84/4.56

given in Table 1.

Two 10 mm-diameter holes were drilled 15 mm from the top and bottom edges to serve as water inlet and outlet. A 20-mm diameter hole, 25 mm from the top edge, was used as air outlet connected to an air-water separator.

The air-water separator was a rectangular box with a 50×100 mm base and 60 mm in height. It had 20 mm-diameter hole located 250 mm from the bottom edge, through which water overflowed to a water tank during experiments. To inject air into the channel 25 capillary tubes, with 0.25 mm inside diameter and 0.3 mm of outside diameter, were planted in the lower part of each plate. Two pressure taps, one 210 mm from the top and the other 350 mm from the bottom edge, were located at the center of the plate. The pressure taps were connected to a digital manometer to measure the pressure drop. The air/water inlets and outlets were equipped with quick-closing valves. These valves were simultaneously activated electrically.

Purified water was supplied by a pump and was regulated by float-type flow meter. The water flow entered the test section through the top water inlet. However, a portion of the water was carried over to the air-water separator by the air flow in the test section. Air was supplied by a compressor through a pressure regulator and was controlled by float-type flow meter. Air was first introduced into the air chambers adjacent to the rectangular channel and then injected into the channel through capillary tubes. These tubes produced uniformly sized bubbles. Bubbles were distributed evenly in the channel and mixed with the downward water flow as they rose. The air-water mixture flowed out of the rectangular channel and entered the air-water separator, where the air escaped to the atmosphere while the water flowed back to the water tank. The water flow rate in the test section was measured by a float-type flow meter. Water flow rate through the test scion was reconfirmed by diverting the water to a graduated cylinder at the exit of the test section and measuring the amount collected over a period of time. Superficial velocities of water and air ranged 0-0.2 and 0-3.5 m/s respectively.

Before each experiment the channel and the air-water separator were initially filled with water supplied at a constant flow rate. Water continuously flowed through the system with the flow rate controlled by the needle valve at the water exit. Air was then injected and the desired flow rate was set with float-type flow meter. When the steady state was reached for each flow rate, the pressure drop was measured. Also the channel-average void fraction was obtained by the collapsed height of water level in the channel using the quick-closing valves. For the unsteady or oscillatory flow regimes, instantaneous pressure drop and void fraction corresponding to the specific flow regime were measured.

The probability density function of the pressure drop in two-phase flows was analyzed in terms of the flatness and skewness factors to classify flow regimes. However, for all flow ranges tested, the factors were similar and seldom provide any means for the classification of flow regimes. Therefore, the flow regimes were classified purely according to visual observations. A shutter speed of 1/1000 for the capture of video images was used to identify the air-water interface and corresponding flow regime.

The uncertainty analysis has been performed according to the method proposed by Kline (1985). The estimated uncertainties on the volumetric flow rates are ±6% for air and ±2% for water. The uncertainty of the differential pressure measurements in air-water two-phase flow is ±3%.

3. Results and Discussion

3.1 Flow regimes

In the present study, the aspect ratio of the channels was so large and the lateral fin spacing was so small that spherical bubbles and slug bubbles looked as if they were crushed between the two walls and fins. Four flow regimes were recognized in counter-current air-water flow-bubbly, slug, churn, and annular flows. Some examples are shown in Fig. 2. Even though the air-water interface was noticeable with the naked eye during the experiments, it is very hard to identify in the video images. Therefore the interface is highlighted manually in Fig. 2.

In narrow channels with offset strip fins, the entrance length for flow regime development was about 30 mm from the air injection nozzle for the bubbly flow regimes. The entrance length seemed to be almost the same for the slug and churn-turbulent flow regimes. Therefore most of the test section showed identical flow patterns except in the vicinity of air inlet. In the annular flow regime very close to the counter-current flow limitations, frothy bubbles were observed near the air injection port, while a falling sheet of liquid formed on the prime surface discontinuously and distorted liquid droplets dangled on the fins.

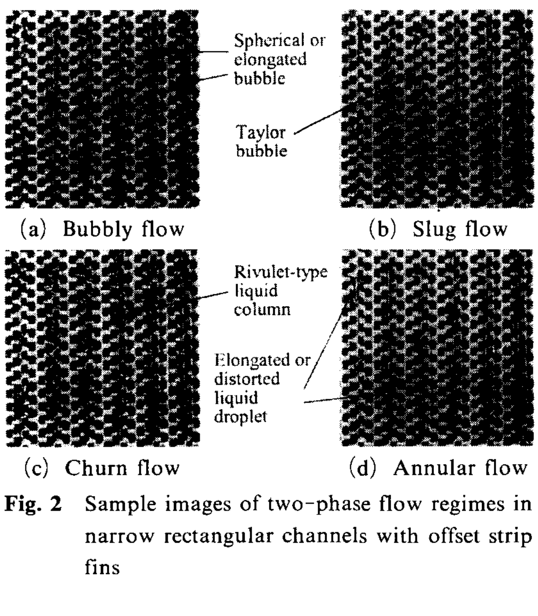


Fig. 2 Sample images of two-phase flow regimes in narrow rectangular channels with offset strip fins

Two-phase flow regimes were classified as follows.

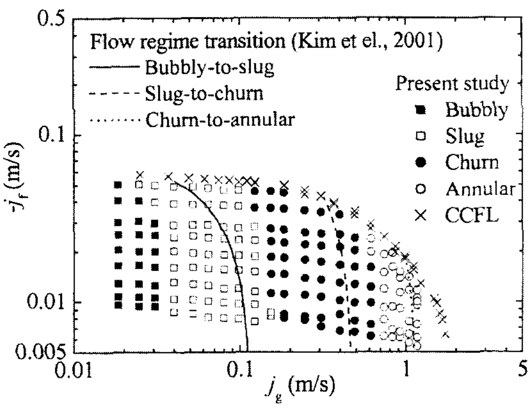
Bubbly flow - Liquid was flowing as a continuous phase while gas was distributed as discrete small bubbles, which were reasonably uniform in a continuous liquid phase. At a lower gas flow rate, spherical bubbles, whose diameter were less than the gap width of the channel, were formed and flowed upward vigorously without coalescence. At a higher gas flow rate, the confinement of the walls and fins caused the growing bubbles to be flattened and elongated. Due to the presence of fins, bubble rose vertically along the subchannel between fins and its lateral movement was depresed. Thus the formation of cap bubble due to bubble coalescence, which was very common in the channels without fins (Kim et al., 2001A), was hardly noticed. In bubbly flow regime, the diametr of elongated bubbles reached the gap width of rectangular channels.

Slug flow - This flow had two-dimensional 'Taylor bubbles' which contained an ellipsoidal nose and a flat rectangular body. Due to the interference of fins, the growth of Taylor bubbles was very limited. Taylor bubbles were never able to fill up the channel cross-section. The width of Taylor bubble reached almost 3 times of the lateral fin spacing. Taylor bubble was caught in the fins continuously as it rose. However, individual bubble rose vertically without any lateral movement and had no interaction with one another. Thus the smooth interface of slug was somewhat well maintained.

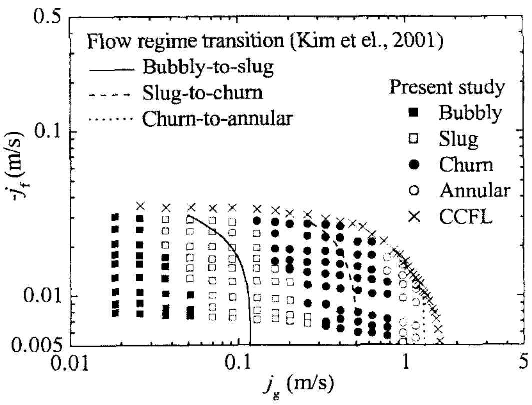
Churn-turbulent flow - This flow was formed by the breakdown of air bubbles in the slug flow. The bubbles were slug-like but were chaotic, frothy and disordered. The presence of the fins made the shape of bubble much more distorted than those in the channels without fins. The slug bubbles were lengthened and distorted until they were no longer discernible. The air-water interface was cut like the shape of saw. Water bridges reaching the width of the test section were repeatedly destroyed due to the high local gas concentration. Water flowed down as rivulets along the fin. Unlike the bubbly and slug flow, churn flow was characterized by the oscillatory motion of lumps of air and water mixture in the axial and lateral directions.

Annular flow - This flow was comprised of a solid gaseous core, continuous in the axial direction, with a liquid film surrounding the core. Due to the fins, continuous wavy falling film on the wall was seldom observed. Rather, small droplet formed by the breakup of the falling film were dangling on the fins and dripping down. The prime surface of channel walls was covered with liquid film discontinuously. Intermittent flooding type waves forming at the bottom of the test section and moving upwards resulted a lump of liquid entrained in the middle of the gas flow.

These experimental data on the flow regime were represented in Fig. 3, which showed superficial liquid velocity, $j_f = Q_f/A$, plotted versus



(a) Gap width 3 mm



(b) Gap width 5 mm

Fig. 3 Counter-current flow regimes for narrow rectangular channels

the superficial gas velocity, $j_g = Q_g/A$. Q is the volumetric flow rate, A is the actual flow area and the subscripts f and g denote liquid and gas, respectively. From the comparison of the present works to the flow regimes proposed by Kim et al. (2001A) for the narrow channels without fins, it is obvious that the presence of fins reduces the superficial gas velocities necessary for the flow regime transition. The liquid superficial velocity range was limited by the counter-current flow limitation, which decreased as the gap width increased. As the superficial gas velocities increased the flow regime changed consecutively from bubbly to slug, churn and annular flow. At higher liquid superficial velocities, the transition gas superficial velocities decreased. As the gap width increased the transition gas superficial velocities seemed to increase. However, the qualitative characteristics of flow regimes were similar for all gap widths.

Taitel and Barnea (1983) proposed a mechanistic set of models for flow regime transitions in counter-current gas-liquid vertical flows by dividing the entire flow map into bubbly, slug, and annular flow patterns. Even though their model was based on the criteria that bubbles tend to collapse and form Taylor bubbles at void fractions greater than 0.3, experimental observations in the present study showed that the critical void fraction for the transition from bubbly to slug flow was 0.14 for the channel of 3 mm gap width and 0.2 for the channel of 5 mm gap width, respectively. These critical void fractions for the bubbly-to-slug flow transition are smaller than that of channels without fins. Therefore the model of Taitel and Barnea predicts pretty higher superficial gas velocities for the bubbly-to-slug transition than those observed in the present study.

There is considerable difficulty in accurately identifying the slug-churn transition because there is confusion as to the description of the churn flow itself. A model proposed by Mishima and Ishii (1984) used the criterion that the mean void fraction be greater than the slug-bubble section mean void fraction. However Taylor bubble was never able to fill up the channels in the present study and the model of Mishima and Ishii

(1984) is not applicable to the case of two-phase flow in rectangular channels with fins. Channel-average void fraction data at various liquid superficial velocities are depicted in Fig. 4 and are compared with the empirical correlation developed by Yamaguchi and Yamazaki (1982) for all flow patterns.

$$\frac{\alpha}{\alpha_o}=1+(1.6\times10^4)Z \quad (1)$$

and

$$Z=\left(\frac{\rho_g}{\rho_f}\right)^{-0.4}\left(\frac{\mu_g}{\mu_f}\right)^{0.7}\left(\frac{\rho_f g D_h^2}{\sigma}\right)^{-0.45}\left(\frac{\mu_f j_g}{\sigma}\right)^{1.3}\left(1-\frac{j_f}{j_g}\right)^{1.8} \quad (2)$$

where α_o, representing the channel average void fraction in the limit of zero liquid flow rate, was obtained from the following empirical correlation (Sudo, 1980):

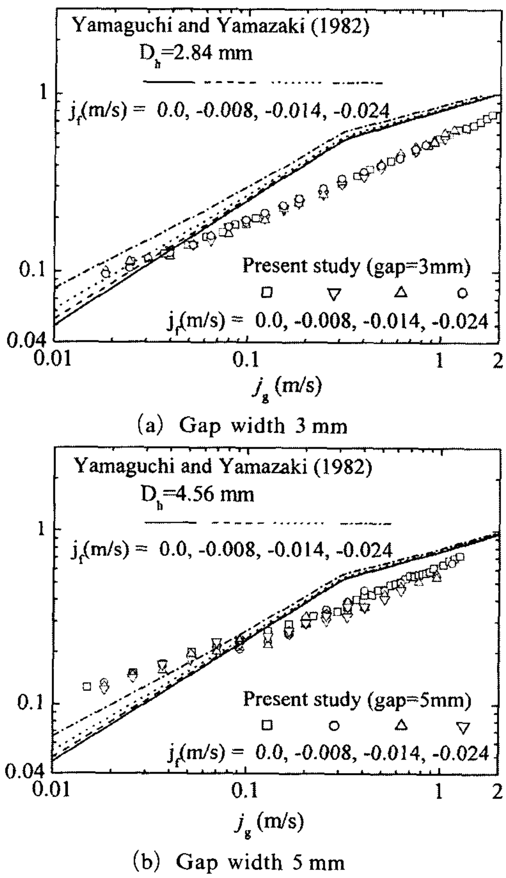


Fig. 4 Average void fractions in narrow rectangular channels with offset strip fins

$$\alpha_o=\frac{Y}{KX^m} \quad (3)$$

where

$$Y=\left(\frac{\sigma}{g\rho_f D_h^2}\right)^{0.064}\left(\frac{\mu_g}{\mu_f}\right)^{0.125} \quad (4)$$

$$X=\left[\left(\frac{\mu_f}{\mu_g}\right)^{0.82}\left(\frac{\rho_g}{\rho_f}\right)^{0.2}\right]\left(\frac{\mu_g j_g}{\sigma}\right) \quad (5)$$

$K=0.00523$, $n=-0.704$ when $X<0.0005$ (6.1)

$K=0.093$, $n=-0.325$ when $0.0005\leq X\leq0.004$ (6.2)

$K=0.54$, $n=0$ when $X>0.004$ (6.3)

Hydraulic diameter of the finned channel is defined as

$$D_h=\frac{4shl}{\{2(sl+hl+th)+ts\}} \quad (7)$$

where s, h, l and t are the lateral fin spacing, fin height, fin length and fin thickness, respectively.

Basically, channel-averge void fractions were not very different from that in the channels without fins (Kim et al., 2001B). Channel-average void fraction increased with the superficial velocity of air. However the effect of liquid superficial velocity on the void fraction was negligible. Also the void fraction values for the channel with 5 mm gap was larger than the corresponding values of the channel with 3 mm gap for the same water and air volumetric flux density especially at higher flux density, which was observed only when the superficial velocity of air was very small in the case of the round tubes of relatively large diameter. Unlike the case of round tubes, the effect of liquid superficial velocity on the void fractions was almost negligible in rectangular channels with fins. Fins in the rectangular channel behave as the flow obstacles and prohibit the formation of large bubbles. Even the lateral movement of bubble during its rise was very limited within the subchannel formed by the fins. High aspect ratio of narrow rectangular channel and the presence of fins resulted in two-phase flow pattern and void fraction significantly different from those in round tubes.

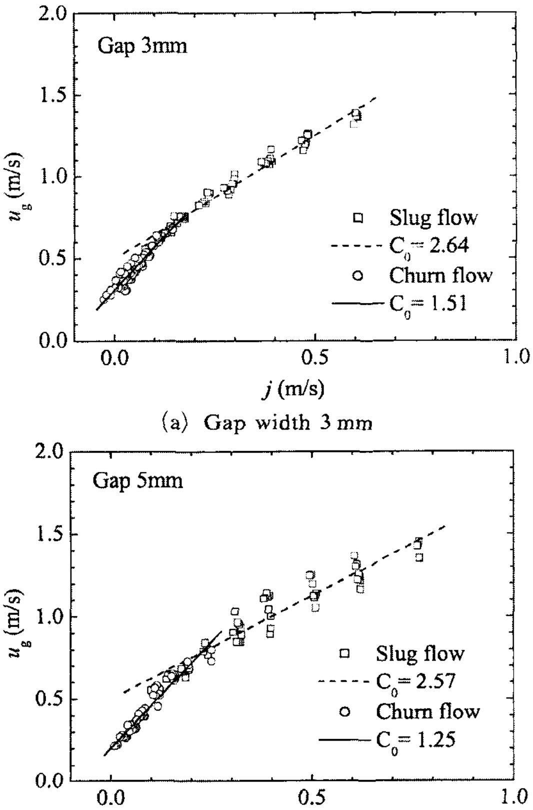
These features of two-phase flow in rectangular channels with fins could be also identified by the two-phase distribution parameter. The two-phase distribution parameter has strong effects on the prediction of flow regime transition. According to the drift-flux model (Ishii, 1977), the relationship between the gas velocity and the mixture volumetric flux can be expressed by Eq. (8):

$$u_g=\frac{j_g}{\alpha}=C_o j+u_{gj} \tag{8}$$

where the superficial velocity of two-phase mixture $j=j_g-j_f$ and C_o is the two-phase distribution parameter given by

$$C_o=1.35-0.35\sqrt{\rho_g/\rho_f} \tag{9}$$

for rectangular channels regardless of the flow regimes. The gas drift velocity, u_{gj}, is found from the rise velocity of the Taylor bubbles relative to the mean velocity of the liquid. The experimental results are shown in Fig. 5 for the slug and churn flows. The void fractions are well-correlated by the drift flux model. For the rectangular channels with fins, the distribution parameter for the slug flow were found to be unexpectedly bigger than the prediction of Ishii (1977) and also bigger than the case of rectangular channels without fins (Kim et al., 2001A). For the churn flow, the distribution parameter became smaller than that of slug flow but still bigger than that in round tube. As the gap width increased from 3 mm to 5 mm, the distribution parameter in slug flow remains almost the same, but decreased in churn flow.



(a) Gap width 3 mm

(b) Gap width 5 mm

Fig. 5 Drift-flux correlation of mean void fraction

Iida and Takahashi (1976) reported that values larger than 1.3 exist for the small test sections. Moriyama and Inoue (1991) also recently reported larger values for extremely narrow gaps. Present study shows that fins in the channel of narrow gaps abruptly increase the distribution parameter.

For high gas flow rates the flow becomes annular. Taitel and Barnea (1983) modeled the slug-to-annular transition in counter-current flow in vertical tubes. Transition was assumed to occur when the relative velocity between the Taylor bubble and the liquid film adjacent to it reached the condition of flooding. However their model can be hardly applicable to the two-phase flow in rectangular channels with fin since the growth of Taylor bubble is very limited and the large portion of liquid flows down as distorted liquid droplets frequently caught by the fins as shown in Fig. 2. The superficial velocity of air necessary for the transition from churn to annular flow was decreased as offset strip fins were introduced in narrow rectangular channels.

Since it is essential to understand the characteristics of counter-current flow limitation in the given geometry, counter-current flow limitation (CCFL) data were correlated by the flooding curve as shown in Fig. 6, where m and C are constants in Wallis' flooding correlation (Wallis, 1969)

$$j_g^{*1/2}+m j_f^{*1/2}=C \tag{10}$$

$$j_g^*=\frac{j_g\rho_g^{1/2}}{(gD\Delta\rho)^{1/2}} \tag{11.1}$$

$$j_f^*=\frac{j_f\rho_f^{1/2}}{(gD\Delta\rho)^{1/2}} \tag{11.2}$$

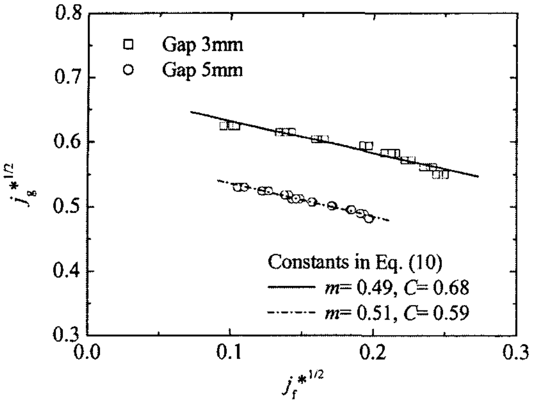


Fig. 6 Counter-current flow limitations in narrow rectangular channels with offset strip fins

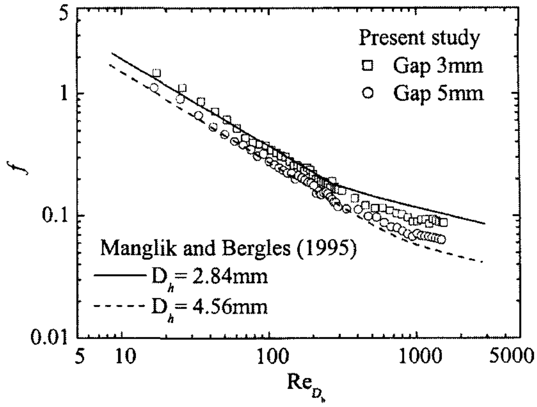


Fig. 7 Single-phase friction factors in narrow rectangular channels with offset strip fins

As the gap width increased C decreased but m increased, however m was still smaller than those predicted for simple vertical channels. For simple vertical channels m=0.8-1.0, and C=0.7-1.0 have been reported, with C mainly depending on the channel end conditions. Values for m and C significantly differ from the aforementioned ranges for more complex channel configurations as proposed by Osakabe and Kawasaki (1989) and Ghiaasiaan et al. (1995).

3.2 Two-phase pressure gradients

Prior to performing the experiments on two-phase pressure drop, the friction factor for the single-phase (water) flow was obtained to check the reliability of the experimental system. The friction factors for the water flow through the rectangular channels with fins were obtained by the correlation proposed by Manglik and Bergles (1995). As seen in Fig. 7, the experimental results agree very well with the correlation.

$$f=9.6243\mathrm{Re}^{-0.7422}a_1^{-0.1856}a_2^{0.3053}a_3^{-0.2659}[1+(7.669\times10^{-8}\mathrm{Re}^{4.429}a_1^{0.920}a_2^{3.7673}a_3^{0.236}]^{0.1} \quad (12)$$

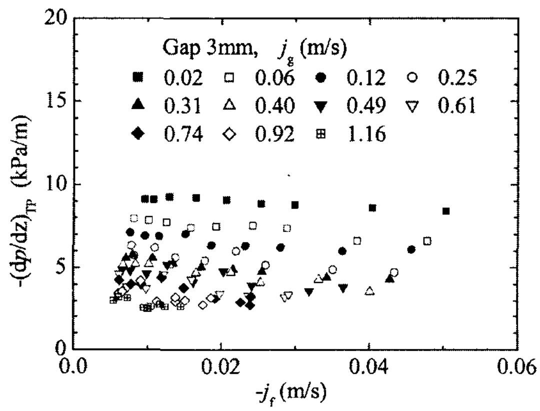
where a_1, a_2 and a_3 are defined as

$$a_1=s/l \quad (13.1)$$

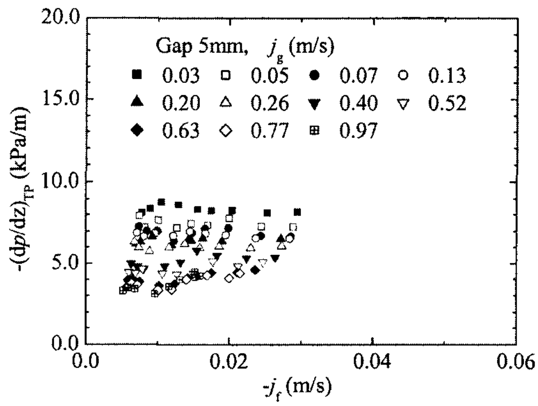
$$a_2=t/l \quad (13.2)$$

$$a_3=t/s \quad (13.3)$$

Figure 8 shows the two-phase pressure gradients in narrow rectangular channels with fins. As the superficial velocity of gas increases, counter-current two-phase pressure gradient decreases due to the increase of void fraction. The effect of liquid superficial velocity on two-phase



(a) Gap width 3 mm



(b) Gap width 5 mm

Fig. 8 Two-phase pressure gradients in narrow rectangular channels with offset strip fins pressure gradient seems to be negligible compared to that of gas. Basically two-phase pressure gradient is consisted of the frictional, gravitational and accelerational pressure gradients. In the present study of counter-current flow of air and water in vertical channels, two-phase pressure gradient was dominated by the gravitational pressure gradient which was determined by the void fraction. However, as shown in Fig. 4, liquid superficial velocity had negligible effect on the void fraction, and so did on two-phase pressure gradient. There are only a few studies presented on the counter-current two-phase pressure gradients. Taitel and Barnea (1983) proposed that the frictional pressure drop is so small in bubbly flow that the gravitational pressure gradient is dominant. In slug flow they calculated the total pressure gradient by

$$\frac{dp}{dz}=\left.\frac{dp}{dz}\right)_f+\left.\frac{dp}{dz}\right)_g+\left.\frac{dp}{dz}\right)_a \tag{14}$$

where the subscript f, g and a denote the liquid slug, the Taylor bubble and the acceleration, respectively.

The pressure gradient in the liquid slug consists of gravitational and frictional contributions.

$$\left.\frac{dp}{dz}\right)_f=\left[-\rho_f g-\frac{2}{D}f_f\rho_f\mid u_f\mid u_f\right]\left(1-\frac{l_g}{l}\right) \tag{15}$$

where l_g/l is the relative length of the gas bubble in slug flow. The friction factor f_f is calculated either by Eq. (16.1) for laminar or Eq. (16.2) for turbulent flow.

$$f_f=16\mathrm{Re}^{-1} \tag{16.1}$$

$$f_f=0.046\mathrm{Re}^{-0.2} \tag{16.2}$$

The pressure gradient in the Taylor bubble zone is usually very small. Still it can be calculated by

$$\left.\frac{dp}{dz}\right)_g=\left[-\rho_g g-\frac{4\tau_i}{D(1-2\delta/D)}\right]\frac{l_g}{l} \tag{17}$$

where τ_i is the frictional contribution caused by the interfacial shear and given as

$$\tau_i=\frac{1}{2}(0.005+1.5\delta/D)\,\rho_g u_g^2 \tag{18}$$

The liquid film thickness δ is estimated using the steady-state falling film momentum conservation equation :

$$\frac{\delta}{D}=B\left[\frac{\mu_f^2}{D^3g(\rho_f-\rho_g)\rho_f}\right]^{n_1}\left[\frac{\rho_f j_f D}{\mu_f}\right]^{n_2} \tag{19}$$

where the constants B, n_1 and n_2 are appropriately defined for laminar and turbulent films (Taitel and Barnea, 1983).

In addition, acceleration pressure drop is associated with the mixing zone of the liquid slug at which the liquid film decelerates to the slug velocity. However it is small enough to be neglected. In an annular flow the pressure gradient is consisted of the interfacial shear and the gravitational pressure gradient of gas core.

As mentioned earlier, unlike that in round tube, Taylor bubble hardly fills the channel flow area in rectangular channels with fins. Since the model of Taitel and Barnea (1983) was developed for the counter-current two-phase flow in round tubes, it is not reasonable to compare the model to the results of the present study. However they are depicted together in Fig. 9 just for the qualitative comparison. New model is necessary to predict the two-phase pressure gradient accurately in terms of the gap width and fin geometry, which is beyond the scope of the present study. Experimental results show that the two-phase pressure gradient decreases with the increase of the superficial velocity of air. Its reduction rate is bigger in slug flow regime than in bubbly flow regime. The increase of gap width results in the slight increase of two-phase pressure gradient

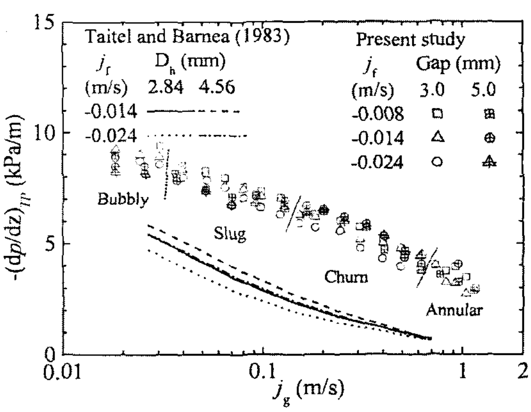


Fig. 9 Two-phase pressure gradients in narrow rectangular channels with offset strip fins

especially at high gas flow rate. However the effect of gap width on the two-phase pressure gradient is no more significant in rectangular channels with offset strip fins. Comparisons of the present study to the predictions of the model show that the development of new model is necessary to accurately estimate the two-phase pressure gradient in rectangular channels with fins.

4. Conclusions

A study of counter-current air-water two-phase flows in narrow rectangular channels with offset-strip fins has been performed. Two-phase flow regimes were experimentally investigated in a 760 mm long and 100 mm wide test section with 3.0 and 5.0 mm gap widths. Channel-average void fractions and two-phase pressure gradients were compared with the previous studies of the existing correlations.

In the narrow rectangular channels with fins, flow regime transition occurred at lower superficial velocities of air than in the channels without fins. The growth of Taylor bubble was very limited in the slug flow. Bubbles rose along the subchannel formed by fins and the lateral movement was depressed. The critical void fraction for the bubbly-to-slug transition was about 0.14 for the 3 mm gap channel and 0.2 for the 5 mm gap channel, respectively. In transition from churn to annular flows the effect of liquid superficial velocity was found to be insignificant. Channel average void fraction increased as the gap width increased, especially at high superficial velocity of air. The presence of fins in narrow rectangular channels enhanced the two-phase distribution parameter significantly in the slug flow, where the effect of gap width was almost negligible. Superficial velocity of air dominated the two-phase pressure gradients. Liquid superficial velocity and channel gap width has only a minor effect on the pressure gradients.

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