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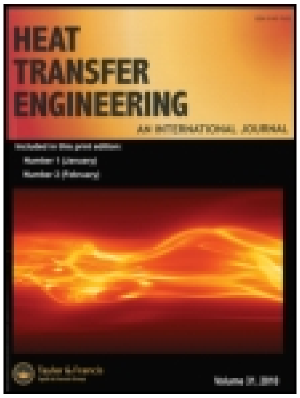

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Two-Phase Flow Across Small Sudden Expansions and Contractions

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Two-phase flow approaching singularities such as abrupt expansions and sudden contractions is widely encountered in typical industrial and heat exchanging devices. There have been some studies concerning this subject but they mostly are applicable for larger channels. In this study, the first attempt is made to review the existing efforts concerning two-phase flow across sudden expansions/contractions and to examine the applicability of the existing correlations with respect to the recent data in small channels. The second part of this study presents some newly measured pressure drops and observed flow patterns pertaining to some special flow phenomena by expansion/contraction. For an abrupt expansion, it is found that the existing correlations all fail to provide a reasonably predictive capability against the newly collected data. Furthermore, a unique flow pattern called "liquid jet-like flow pattern" occurs at a very low quality region of total mass flux of 100 $kg{\cdot}m^{-2}{\cdot}s^{-1}$*, and it raises a setback phenomenon of pressure drop. By contrast, an appreciable increase of pressure difference is seen when the liquid jet-like flow pattern is completely gone. A similar conclusion is drawn for the data of contractions. For the correlations/predictive models, the homogeneous model gives satisfactory prediction for conventional macro-channels but fails to do so when the channels become smaller. This is especially pronounced for a small-diameter tube with a Bond number being less than 1, in which the effect of surface tension dominates.*

INTRODUCTION

In recent years, utilization of mini- and micro-channels has become quite attractive in heat exchanging devices. However, the length of the mini-/micro-channel is normally shorter than that in a macro-channel to eliminate the accompanied high pressure loss. This eventually leads to another problem of accurate estimation of the total pressure drop in the mini/micro system. Flow of two-phase mixtures across sudden expansions and contractions is relevant in many applications, including heat-exchanging equipment and connection pipelines. As the two-phase mixture flows through the sudden area changes, the flow might form a separation region at a sharp corner and cause an irreversible pressure loss. For the macro system, this loss is usually regarded as a minor loss, but its role becomes more and more crucial as the channel size gets smaller and shorter. Accurate methods for predicting the pressure drop in such systems are therefore important. This is particularly true for modern cooling systems where numerous channels or micro-channels are exploited to distribute the working fluid. There have been some developed correlations applicable to the pressure loss for the conventional tubes; however, very few investigations have been reported for this abrupt pressure change with respect to mini-channels (Abdelall et al. [1], Chen et al. [2, 3]). Applicability of the existing correlations/models to this area is quite ambiguous. It is therefore essential to examine the corresponding applicability.

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LITERATURE REVIEW

For single-phase flow, by combining with the pressure balance equation, the static pressure difference subject to the sudden expansion is related to the kinetic energy of the flow:

$$\frac{\Delta P_e}{\frac{G^2}{2\rho}} = -2\sigma_A (1-\sigma_A) \tag{1}$$

where the mass flux density (G) and velocity (u) are calculated based on the smaller cross-sectional area of the inlet tube. The negative sign indicates a pressure recovery whose value varies from zero to a maximum of 0.5 at $\sigma_A = 0.5$. Analogously, the pressure loss subject to sudden contraction is normally in terms of the contraction loss coefficient (K) and multiplication of kinetic energy of the flow:

$$\frac{\Delta P_c}{\frac{G^2}{2\rho}} = K \tag{2}$$

where σ_A is the passage cross-sectional area ratio and $0 < \sigma_A < 1$, and total mass flux (G) and velocity (u) are calculated based on the smaller cross-sectional area of the outlet tube. The value of K is related to the Reynolds number and contraction ratio, and it is close to 0.5 and 1.0 with a very small cross-sectional area ratio ($\sigma_A < 0.05$) (Kays and London [4]).

For single-phase flow through the contraction, Chisholm [5] combined the static pressure drop to the vena contracta (ΔP_{iv}) and the pressure recovery downstream of the vena contracta (ΔP_{vc}) to give the total static pressure drop at the contraction (ΔP_c):

$$\Delta P_{iv} = \left(\frac{G^2}{2\rho_L}\right)\left[(\sigma_A C_C)^{-2} - 1\right] \tag{3}$$

$$\Delta P_{VC} = \frac{\left(\frac{G}{\sigma_A}\right)^2}{\rho_L}\left[C_c^{-1} - 1\right] \tag{4}$$

$$\begin{aligned} \Delta P_C &= \Delta P_{iv} - \Delta P_{vc} \\ &= \left(\frac{G^2}{2\rho_L}\right)\left[\frac{1}{(\sigma_A C_C)^2} - 1 - \frac{2\left(C_C^{-1}-1\right)}{\sigma_A^2}\right] \end{aligned} \tag{5}$$

By comparing Eq. (2) with Eq. (5), the contraction loss coefficient can be found as:

$$K = \frac{1}{(\sigma_A C_C)^2} - 1 - \frac{2\left(C_C^{-1}-1\right)}{\sigma_A^2} \tag{6}$$

The contraction coefficient C_C correlated by Chisholm [5] is given as follows:

$$C_C = \frac{1}{[0.639(1-\sigma_A)^{0.5} + 1]} \tag{7}$$

Another contraction coefficient C_C proposed by Geiger [6] is:

$$C_C = 1 - \frac{(1-\sigma_A)}{[2.08\,(1-\sigma_A) + 0.5371]} \tag{8}$$

Two-Phase Pressure Change Across Sudden Expansion

For the two-phase pressure change across sudden expansion, Romie [7] derived the following expression for sudden enlargement:

$$\Delta P_e = \frac{-G^2\sigma_A(1-\sigma_A)}{\rho_L}\left[\begin{array}{l}\left(\frac{(1-x)^2}{(1-\alpha_{in})} + \frac{(\rho_L/\rho_G)\,x^2}{\alpha_{in}}\right) - \\ \sigma_A\left(\frac{(1-x)^2}{(1-\alpha_{\exp})} + \frac{(\rho_L/\rho_G)\,x^2}{\alpha_{\exp}}\right)\end{array}\right] \tag{9}$$

where the subscript "in" denotes upstream of expansion whereas "exp" represents downstream of expansion. If the void fraction remains unchanged, Eq. (9) becomes

$$\Delta P_e = \frac{-G^2\sigma_A(1-\sigma_A)}{\rho_L}\left[\frac{(1-x)^2}{(1-\alpha)} + \frac{(\rho_L/\rho_G)\,x^2}{\alpha}\right] \tag{10}$$

Richardson [8] simplified the energy balance model and assumed that the pressure recovery is proportional to the kinetic energies of the phases, yielding:

$$\Delta P_e = -0.5G^2\left(1-\sigma_A^2\right)\left[\frac{\sigma_A\left(1-x^2\right)}{\rho_L(1-\alpha)}\right] \tag{11}$$

Assuming that a heterogeneous flow and a loss of dynamic pressure head takes place in the liquid phase, Lottes [9] derived:

$$\Delta P_e = \frac{-G^2\sigma_A(1-\sigma_A)}{\rho_L(1-\alpha)^2} \tag{12}$$

By considering the pressure rise in a sudden expansion with irreversible flow induced in a rough tube, Chisholm and Sutherland [10] also developed a heterogeneous model based on the momentum balance:

$$\begin{aligned} \Delta P_e &= -G^2\sigma_A(1-\sigma_A)(1-x)^2 \\ &\quad\times\left[\frac{\left(1 + C_h/X_{CH} + 1/X_{CH}^2\right)}{\rho_L}\right] \end{aligned} \tag{13}$$

where

$$X_{CH} = \left(\frac{\rho_G}{\rho_L}\right)^{0.5}\frac{(1-x)}{x} \tag{14}$$

$$C_h = \left\{1 + 0.5\left[1 - \left(\frac{\rho_G}{\rho_L}\right)\right]^{0.5}\right\}\left\{\left(\frac{\rho_L}{\rho_G}\right)^{0.5} + \left(\frac{\rho_G}{\rho_L}\right)^{0.5}\right\} \tag{15}$$

The model of Chisholm and Sutherland [10] was compared with the bubbly flow data ($\alpha < 0.35$) for an air–water mixture. Although their predictions remain reasonably good, their model slightly underestimated the data (Attou et al. [11]).

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Based on some test data, Wadle [12] proposed a formula to describe the pressure recovery in an abrupt diffuser. The model includes an artificial constant K in connection with different working fluids ($K = 0.667$ for steam-water, $K = 0.83$ for air–water):

$$\Delta P_e = -\left(1-\sigma_A^2\right)\frac{1}{2}G^2 K\left[\frac{(1-x)^2}{\rho_L}+\frac{x^2}{\rho_G}\right] \tag{16}$$

From the mechanical energy equation without friction dissipation, Collier and Thome [13] derived the following expression:

$$\Delta P_e = \frac{-G^2\left(1-\sigma_A^2\right)}{2\left(\frac{x}{\rho_G}+\frac{1-x}{\rho_L}\right)}\left[\frac{(1-x)^3}{(1-\alpha)^2\rho_L^2}+\frac{x^3}{\alpha^2\rho_G^2}\right] \tag{17}$$

For estimation of the void fraction α, a very simple correlation was recently proposed by Kawahara et al. [14] as

$$\alpha = \frac{0.03\beta^{0.5}}{\left(1-0.97\beta^{0.5}\right)} \tag{18}$$

where β is the gas volumetric flow ratio, given as $\beta = j_G/(j_G + j_L)$, j_L is the superficial liquid velocity, and j_G is the superficial gas velocity. For homogeneous flow, Eq. (10) becomes:

$$\Delta P_e = -G^2\sigma_A(1-\sigma_A)\left[\frac{(1-x)}{\rho_L}+\frac{x}{\rho_G}\right] \tag{19}$$

Since the gas phase and liquid phase have different velocities, particularly for a large density difference between the gas and liquid phases, more realistic models and additional parameters are needed for the pressure recovery calculation.

Based on an annular-mist flow model accompanied with mass and momentum balance, Schmidt and Friedel [15] proposed a rather complex formula:

$$\Delta P_e = \frac{G^2\left[\frac{\sigma_A}{\rho_{eff}}-\frac{\sigma_A^2}{\rho_{eff}}-f_e\rho_{eff}\left(\frac{x}{\rho_G\alpha}-\frac{(1-x)}{\rho_L(1-\alpha)}\right)\left(1-\sqrt{\sigma_A}\right)^2\right]}{1-\Gamma_e(1-\sigma_A)} \tag{20}$$

where

$$\frac{1}{\rho_{eff}} = \frac{x^2}{\rho_G\alpha}+\frac{(1-x)^2}{\rho_L(1-\alpha)}+\frac{\alpha_E\rho_L(1-\alpha)}{1-\alpha_E}\times\left[\frac{x}{\rho_G\alpha}-\frac{1-x}{\rho_L(1-\alpha)}\right]^2 \tag{21}$$

$$\alpha = 1-\frac{2(1-x)^2}{1-2x+\sqrt{1+4x(1-x)\left(\frac{\rho_L}{\rho_G}-1\right)}} \tag{22}$$

$$\alpha_E = \frac{1}{S}\left[1-\frac{1-x}{1-x(1-0.05We^{0.27}\mathrm{Re}^{0.05})}\right] \tag{23}$$

$$S = \frac{x}{1-x}\frac{(1-\alpha)}{\alpha}\frac{\rho_L}{\rho_G} \tag{24}$$

$$We = G^2x^2\frac{d}{\rho_G\sigma}\frac{(\rho_L-\rho_G)}{\rho_G} \tag{25}$$

$$\mathrm{Re} = \frac{G(1-x)d}{\mu_L} \tag{26}$$

$$\Gamma_e = 1-\sigma_A^{0.25} \tag{27}$$

$$f_e = 4.9\times10^{-3}x^2(1-x)^2\left(\frac{\mu_L}{\mu_G}\right)^{0.7} \tag{28}$$

By applying the momentum balance within the boundary of the conical jet for an incompressible and adiabatic flow, Attou and Bolle [16] obtained a correlation for the two-phase flow pressure recovery from a sudden expansion.

$$\Delta P_e = -\sigma_A(1-\sigma_A)\theta_\sigma^r G^2\Phi + \frac{\left(1-\theta_\sigma^r\right)\sigma_A(1-\sigma_A)G^2}{\rho_L} \tag{29}$$

where $\Phi = x^2/(\alpha\rho_G) + (1-x)^2/[(1-\alpha)\rho_L]$, $\theta_\sigma = 3/[1 + \sigma_A^{0.5} + \sigma_A]$, and r is a correction factor related to the physical properties of the mixture. For a gas quality $x = 0$, Eq. (29) can be reduced to $\Delta P_e = -\sigma_A(1-\sigma_A)G^2/\rho_L$. The best fitting to the correction factor is $r = 1$ for a steam–water mixture and $r = -1.4$ for an air–water mixture. The predictive ability of Eq. (29) associated with the air–water and steam–water data is around 23.4% or less (mean deviation). The correlation is particularly good for small mass velocities, but it is inapplicable to high quality flows.

More recently, Abdelall et al. [1] investigated air–water pressure drops caused by abrupt flow area expansions in two small tubes. The tube diameters were 1.6 and 0.84 mm, respectively. Their measured two-phase pressure difference indicated the occurrence of significant velocity slip. Assuming an ideal annular flow regime in accordance with minimum kinetic energy of the flowing mixture (slip ratio $S = (\rho_L/\rho_G)^{1/3}$) leads to a reasonable agreement between the data and the predictions. However, in practice the slip ratio is actually varying along the flow path. The pressure drop across the sudden expansion (ΔP_e) is the difference between the reversible pressure change (ΔP_{eR}) and the irreversible pressure changes (ΔP_{eI}), i.e., $\Delta P_e = \Delta P_{eR} - \Delta P_{eI}$. For an incompressible and adiabatic flow,

$$\Delta P_{eR} = \frac{-0.5\rho_h G^2\left(1-\sigma_A^2\right)}{\rho''^2} \tag{30}$$

$$\Delta P_{eI} = 0.5\rho_L G^2\left[\frac{2\rho_L\sigma_A(\sigma_A-1)}{\rho'}-\frac{\rho_h\rho_L(\sigma_A-1)}{\rho''^2}\right] \tag{31}$$

Where

$$\rho_h = \left[\frac{x}{\rho_G}+\frac{(1-x)}{\rho_L}\right]^{-1} \tag{32}$$

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$$\rho' = \left[\frac{(1-x^2)}{\{\rho_L (1-\alpha)\}} + \frac{x^2}{(\rho_G \alpha)} \right]^{-1} \quad (33)$$

$$\rho'' = \left[\frac{(1-x)^3}{\rho_L^2 (1-\alpha)^2} + \frac{x^3}{(\rho_G^2 \alpha^2)} \right]^{-1/2} \quad (34)$$

To apply Eqs. (30) and (31), an empirical relation between α and x is needed. The predictions of the Abdelall et al. slip flow model [1] are only slightly higher than the experimental data, but the simplified momentum balance model, Eq. (8), by Collier and Thome [13] significantly overpredicts the data.

Two-Phase Pressure Change Across Sudden Contraction

Geiger [6] measured pressure drops for steam–water mixtures flowing through sudden contraction with area ratios (σ_A) of 0.398, 0.253, and 0.144. His data were compared with the homogeneous model, momentum equation, and mechanical energy equation across the contractions. The homogeneous model gave the best predictions of the data.

McGee [17] had also measured the steam–water mixtures flowing through sudden contraction using the same test rig as Geiger [6], but with different test sections and conditions ($\sigma_A =$ 0.608, 0.546). The predictions by homogeneous model when compared with his test data are quite satisfactory. The predictions by the momentum and mechanical energy equations were much lower than the test data. This deviation is believed to be due to the assumption of no mechanical energy loss for the acceleration of the fluids at the downstream of the contraction.

For two-phase flow, the frictional pressure drop due to contraction can be estimated using a homogeneous flow model as recommended by Collier and Thome [13]:

$$\Delta P_c = \left(\frac{G^2}{2\rho_L} \right) [(C_C^{-1} - 1)^2 + (1 - \sigma_A^2)] \left[1 + x \left(\frac{\rho_L}{\rho_G} - 1 \right) \right] \quad (35)$$

The mass flux G is calculated based on the cross-sectional area of the outlet tube with smaller cross-sectional area.

Chisholm [5] also used his contraction loss coefficient K, Eq. (5), in association with the homogeneous model, i.e.,

$$\Delta P_c = \left(\frac{G^2}{2\rho_L} \right) \left[\frac{1}{(\sigma_A C_C)^2} - 1 - \frac{2(C_C^{-1} - 1)}{\sigma_A^2} \right] \times \left[1 + x \left(\frac{\rho_L}{\rho_G} - 1 \right) \right] \quad (36)$$

Furthermore, Chisholm [5] introduced a constant B coefficient for flow through a discrete interval in evaluating the contraction loss:

$$\Delta P_c = \Delta P_{cL} \left[1 + \left(\frac{\rho_L}{\rho_G} - 1 \right) (Bx(1-x) + x^2) \right] \quad (37)$$

$$B = \frac{\left\{ \frac{1}{K_O} \left(\frac{1}{(\sigma_A C_C)^2} - 1 \right) - \frac{2}{(K_O C_C \sigma_A^2)} + \frac{2}{(\sigma_A^2 K_O^{0.28})} \right\}}{\frac{1}{(\sigma_A C_C)^2} - 1 - \frac{2}{(C_C \sigma_A^2)} + \frac{2}{\sigma_A^2}} \quad (38)$$

where ΔP_{cL} is the contraction pressure drop for total flow assumed liquid across the same sudden contraction as shown in Eq. (5), contraction coefficient C_C is given in Eq. (7), and K_O is given as

$$K_o = \begin{cases} \left(1 + x \left(\frac{\rho_L}{\rho_G} - 1 \right) \right)^{0.5} & \text{for} \quad X > 1 \\ \left(\frac{\rho_L}{\rho_G} \right)^{0.25} & \text{for} \quad X \le 1 \end{cases} \quad (39)$$

where X is the Martinelli parameter.

Based on the momentum and mass transfer balance, Schmidt and Friedel [18]developed a new pressure drop model for sudden contraction that incorporates all of the relevant boundary conditions. In this model all the relevant physical parameters were also included in their sudden expansion paper [15]. In addition, the influence of liquid entrainment (α_E) in the gas stream is included, along with relevant parameters like area ratio (σ_A), mass flux (G), gas quality (x), mean void fraction (α), surface tension (σ), the viscosity and density of the gas and liquid phases ($\mu_G, \mu_L, \rho_G, \rho_L$), and slip ratio ($S$), to calculate the effective two-phase density (ρ_{eff}). Their test sections incorporated inlet tube diameters in the range of 44.2–72.2 mm and with outlet tubes in the range of 17.2–44.2 mm. The model predicts several experimental data sets with different physical properties. The comparison of the model and test results is fair, with 80% of the data sets being predicted within ±30%. For incompressible and adiabatic flow, the equation for pressure drop across a sudden contraction is given as:

$$\Delta p_c = \frac{G^2 \left[\frac{1}{\rho_{eff}} - \frac{\sigma_A}{\rho_{eff}} + f_{con} \rho_{eff} \left(\frac{x}{\rho_G \alpha} - \frac{1-x}{\rho_L (1-\alpha)} \right)^2 (1 - \sigma_A^{1/2})^2 \right]}{1 + \Gamma_{con} \left(\frac{1}{\sigma_A} - 1 \right)} \quad (40)$$

where f_{con} is the total fiction factor for the contraction, α is the void fraction, ρ_{eff} is the effective density, Γ_{con} is the base pressure coefficient for the contraction, Re is the Reynolds number, and *We* is the Weber number; ρ_{eff}, α, α_E, S, We, and Re are given in Eqs. (19)–(24) and

$$\Gamma_{con} = 0.77 \sigma_A (1 - \sigma_A^{0.306}) \quad (41)$$

$$f_{con} = 5.2 \times 10^{-3} x^{0.1} (1-x) \left(\sigma_A \frac{\mu_L}{\mu_G} \right)^{0.8} \quad (42)$$

Abdelall et al. [1] investigated air–water pressure drops caused by abrupt flow area expansion and contraction in a very small test section. The larger and small tube diameters were 1.6 and 0.84 mm, respectively. Assuming incompressible gas and liquid phases, and assuming x and α remained constant across

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the sudden contraction, the pressure drop across the sudden contraction was derived as

$$\Delta P_c = G^2 \left\{ \frac{\rho_h \left(\frac{1}{C_C^2} - \sigma_A^2 \right)}{2\rho''^2} + \frac{(1 - C_C)}{\rho'} \right\} \quad (43)$$

where ρ_h, ρ' *and* ρ'' are given in Eqs. (32)–(34).

The contraction coefficient C_C, Eq. (7), given by Geiger [6], is utilized in the Abdelall et al. correlation, as in Eq. (43). The two-phase flow pressure change across the sudden contraction data was found significantly lower than the predictions of the homogeneous model. It may be attributed to the significant velocity slip at the vicinity of the flow area change. As a consequence, Abdelall et al. reappraised the suitability of homogeneous flow and concluded that the slip ratio model may be more relevant to accelerating two-phase flow (Moody [19]). Using the Zivi [20] model with the assumption of an ideal annular flow regime, the slip ratio becomes:

$$S = \frac{u_G}{u_L} = \frac{(1-\alpha)\,\rho_L x}{[(1-x)\rho_G \alpha]} = \left(\frac{\rho_L}{\rho_G} \right)^{\frac{1}{3}} \quad (44)$$

where u_G and u_L are the actual gas and liquid velocities of gas and liquid phases, respectively. The void fraction (α) is calculated from Eq. (44) by using gas quality and gas and liquid densities. Subsequently Eq. (43) is combined with the slip flow model to give a relatively close agreement with Abdelall et al.'s experimental data. An attempt was also made to correlate the two-phase pressure change data at the contraction point in terms of the Martinelli parameter (X), yielding

$$X = \left(\frac{u_L}{u_G} \right)^{0.1} \left(\frac{1}{x} - 1 \right)^{0.9} \left(\frac{\rho_G}{\rho_L} \right)^{0.5} \quad (45)$$

$$\frac{\Phi_{cL}}{X} = 120\,(X\mathrm{Re}_L)^{-0.7} \quad (46)$$

where $\Phi_{cL} = \Delta P_c / \Delta P_{cL}$ is the two-phase multiplier with all liquid flow through the contraction, and ΔP_{cL} is the pressure drop assuming total flow to be liquid flow. Re_L is the Reynolds number in the smaller tube (considering total flow to be liquid). Thus, the two-phase pressure change at the sudden contraction can be predicted by

$$\Delta P_c = \Delta P_{cL} X \left[120\,(X\mathrm{Re}_L)^{-0.7} \right] \quad (47)$$

From the foregoing review of the two-phase pressure change across the sudden contraction, most of the investigations are related to the abrupt area change for the upstream and downstream having round tube configuration and the tube sizes are generally above 10 mm. However, it should be mentioned that configuration variation across singularity (e.g., from rectangular to round) is very common in practice. Hence the first objective of this study is to provide new test data regarding to this influence. Further, it will be shown in subsequent comparisons that most of the proposed correlations/models are only applicable to their own database. In addition, a flow visualization experiment is also carried out to link with certain special pressure drop phenomenon.

EXPERIMENTAL SETUP

The test rig shown in Figure 1 is designed to conduct tests with air–water mixtures. Air is supplied from an air compressor and then stored in a compressed-air storage tank. Air flows through a pressure reducer and, depending on the mass flux range, is measured by three Aalborg mass flow meters for different ranges of flow rates with measurement uncertainty ranging from 0.5 to 2%. The water flow loop consists of a variable-speed gear pump that delivers water, and the water volumetric flow rate is detected by a magnetic flow meter with 0.001 L/s resolution. A self-made Y-shape mixer having a spring insert that provides good mixing amid air and water is placed ahead of the test section. An extra calming test section with a length of 300 mm after the mixer is used before the mixture goes into the test section. The gas quality is calculated from the measured air flow rate and water flow rate, namely,

$$x = \frac{\dot{m}_G}{\dot{m}_G + \dot{m}_L}$$

The total mass flux (G) of air and water flow rate evaluated at the entrance of rectangular test section range from 100 to 700 $\mathrm{kg \cdot m^{-2} \cdot s^{-1}}$ with gas quality (x) varying from 0.001 to 0.8. The corresponding uncertainties for x are within $\pm 4.3\%$. The inlet temperatures of air and water are near 25°C. The pressure drops of the air–water mixtures are measured by three Yokogawa EJ110 differential pressure transducers having an adjustable span of 1300 to 13,000 Pa. Resolution of this pressure differential transducer is 0.3% of the measurements. The drilled holes of the pressure taps are perpendicular to the test sections with a diameter of 0.5 mm. Pressure measurements are made at nine locations along the inlet tube and along the rectangular channel as shown in Figure 1. For validation of the present test setup, measurements of the single-phase pressure drops for air and water alone are in terms of friction factors to compare with the friction factor equations for laminar and turbulent flows in rectangular channels. The results are in line with the known correlations having a deviation within $\pm 5\%$. Leaving the test section, the air–water mixture is separated by an open water tank in which the air is vented and the water is recirculated. The air and water temperatures are measured by resistance temperature device (Pt100Ω) having a calibrated accuracy of 0.1 K (calibrated by Hewlett-Packard quartz thermometer probe with quartz thermometer, models 18111A and 2804A). Observations of flow patterns are obtained from images produced by a high-speed camera of Redlake Motionscope PCI 8000s. The maximum camera shutter speed is 1/8000 s. The high-speed camera can be placed at any position along the rectangular channels or at the side view of the abrupt change of flow area.

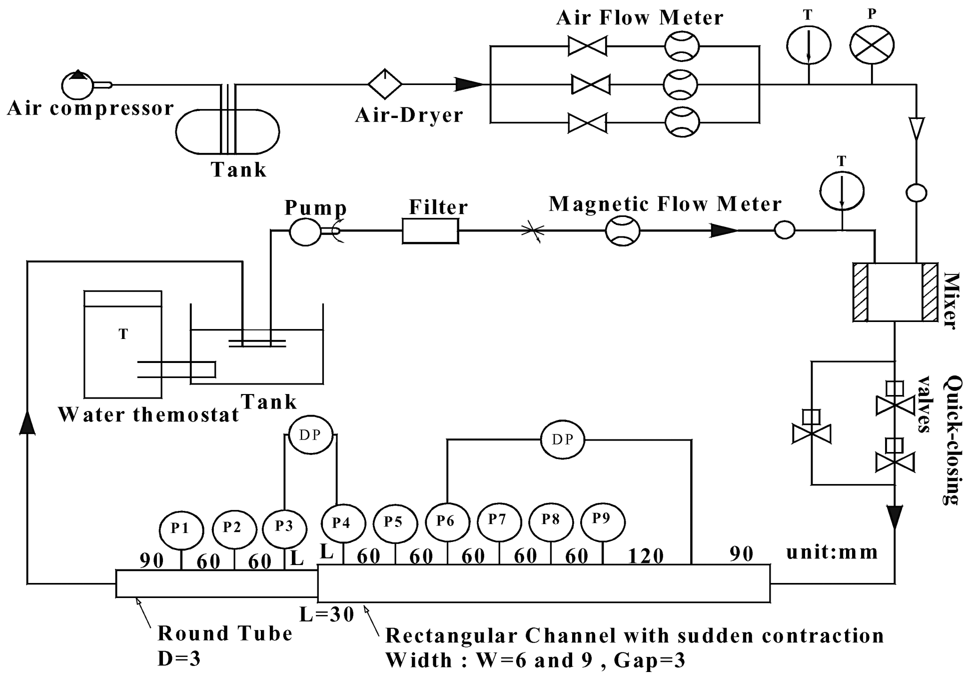

Figure 1 Schematic of test rig and test section.

The test sections are made of transparent acrylic resin, so that the flow pattern and flow structure at the vicinity of the abrupt cross-sectional area change can be visualized. For the expansion test sections, the test sections have the dimensions of gap $(G) \times$ width $(W) = 3 \times 6$ mm and 3×9 mm, and thus the corresponding aspect ratio $A = g/W = 0.333$–ß0.50. These test sections are also arranged in horizontal longitudinal (HL, the wide side is vertical) and the intersection between the rectangular and round tube is well fabricated to avoid any irregularity. The area ratio for the abrupt flow area change (σ_A) is in the range of $0.26 \sim 0.39$. The schematic and the dimensions of the test sections are also shown in Figure 1.

Figure 2a shows a typical change of static pressure along the axis for flow across the expansion. Due to the deceleration of the flow in the transitional region, the static pressure initially increases at the expansion area. After the pressure reaches the maximum, the pressure gradient merges with the downstream pressure gradient line. The pressure change at the sudden expansion is defined as the pressure difference for upstream and downstream fully developed pressure gradient lines extended to the expansion position, i.e., ΔP_{exp}, as shown in Figure 2a. Similarly, Figure 2b represents the variation of static pressure along the flow direction across singularity (for expansion and contraction). When flow approaches the contraction, due to the acceleration of the flow in the transitional region, the static pressure initially decreases to the contraction area. After the pressure reaches the minimum, the pressure increases to a downstream point and then merges with the downstream fully developed pressure gradient line. The pressure change at the sudden contraction is defined as the pressure difference at the interception of singularity evaluated using the fully developed pressure

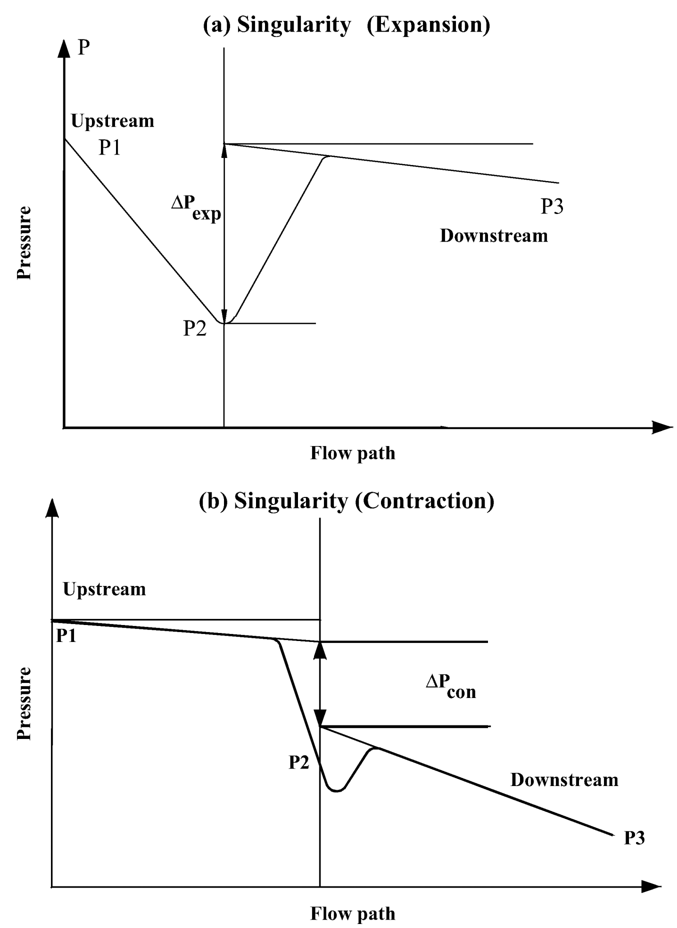

Figure 2 Definition of pressure drop subject to the influence of (a) sudden enlargement and (b) abrupt contraction.

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gradients from upstream and downstream, respectively, i.e., ΔP_{con}, as shown in Figure 2b.

Some previous investigations used a single pressure differential transducer to measure the difference across the sudden contraction. This kind of measurement cannot actually reflect the pressure loss and pressure recovery across the abrupt area change. In this study, several pressure transducers are utilized for measuring the local pressures in the upstream and downstream parts of the test sections as shown in Figure 1. The measured axial pressures versus the pressure tap positions are plotted in a figure to setup the fully developed pressure gradient lines in the upstream and downstream for further obtaining the corresponding pressure change at the sudden contraction/expansion.

RESULTS AND DISCUSSION

Results for Expansion

The corresponding measured pressure drops subject to the sudden enlargement is shown in Figure 3. In general, the pressure drops increase with the rise of vapor quality. However, a notable drop of pressure difference is seen for $G = 100$ kg·m^{-2}·s^{-1} near a vapor quality of 0.01. A similar phenomenon was also reported by Schmidt and Friedel [15]. They also found that the pressure difference occurs at the very low quality region (∼0.005–0.02, depending on the mass flux) and argued that this phenomenon is not due to experimental errors but is associated with the flow pattern changes. For better understanding about this unique phenomenon, one can examine the variation of the corresponding flow pattern. Photographs that are representative of the observed flow patterns for 9 × mm and 6 × mm rectangular channels with a mass flux density of 100 kg·m^{-2}·s^{-1} and vapor qualities ranging from 0.001 to 0.5 are displayed in Figure 4 to show the flow progress in the presence of abrupt enlargement. As seen in Figure 4, for $G = 100$ kg·m^{-2}·s^{-1} at $x = 0.001$ flowing into the 9 × mm enlarged section, the original flow pattern before flowing across the sudden enlargement is elongated bubbly flow, which turns into slug flow after the abrupt area change. This is due to the considerable increase of cross-sectional area, leading to a decline of velocity and to aggregation of elongated bubble.

Note that the original liquid flow gathers at the bottom due to gravity. Hence, as the vapor quality is increased further, the entering flow contains sufficient momentum to become a jet-like (Figure 4c, $x = 0.01$) pattern. However, the liquid jet front heads towards the bottom section since its accompanied liquid momentum is unable to sustain itself against the gravity. For a further increase of vapor quality (Figure 4d, $x = 0.05$), the liquid jet spreads more evenly around the periphery pertaining to the higher momentum. The liquid jet-like flow pattern is similar to the quasi-free-jet pattern observed by Attou et al. [11]. The confined free-jet pattern appears to be conical with a small opening angle. With a further increase of vapor quality, the inherited momentum can easily break up the liquid into droplets and a quite uniformly spread film around the whole rectangular section. The flow pattern is thus an annular flow pattern (e.g., Figure 4f, $x = 0.5$). The unique "liquid jet-like" flow pattern persists at the entrance of the 9 × mm test section. This flow pattern, however, is not seen for the 6 × mm test section as seen from Figures 4g–l. However, by comparing the foregoing observed flow patterns, one can see the unusual phenomenon

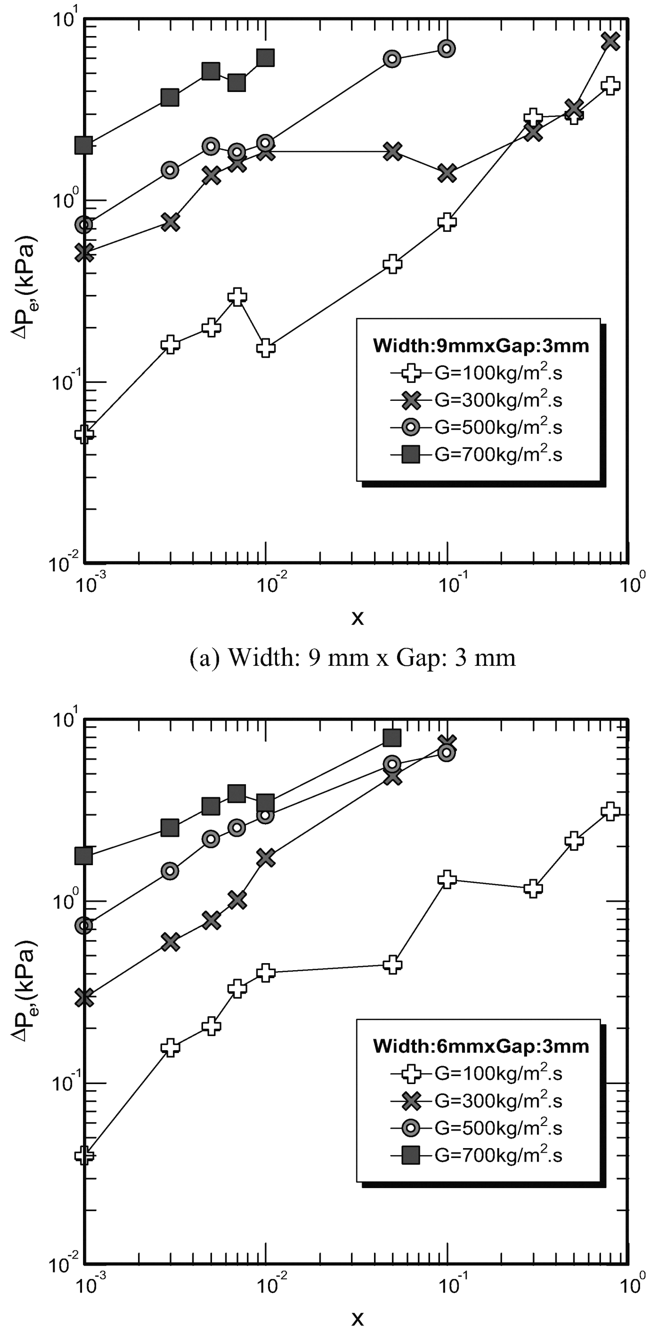

Figure 3 Measured pressure changes as a function of mass flux and gas quality.

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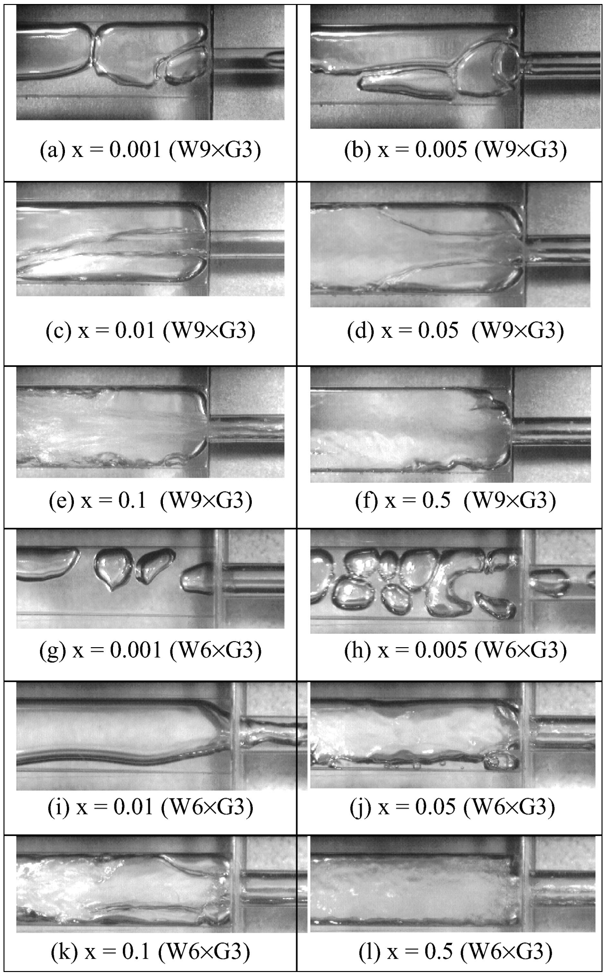

Figure 4 Progress of flow pattern for sudden enlargement vs. quality at $G = 100\ \text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$.

is actually related to the liquid jet-like flow pattern. When the liquid jet flows across the sudden enlargement, part of its momentum is conserved and the gas flow is alongside the liquid jet. In this regard, the corresponding expansion pressure change decreases.

In fact, some of the measured pressure difference data of this study also reveal a flat or slightly leveled-off phenomenon versus vapor quality (e.g., $G = 300\ \text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$). A close examination of the data suggests that this region is also related to the "liquid jet-like" flow pattern. The observations suggest that a conceivable reduction of the pressure difference is encountered when liquid jet-like flow pattern prevails, yet a flattening or leveling off of the pressure difference is seen when the liquid jet-like flow is less pronounced. By contrast, one can find an appreciable increase of pressure difference when the liquid jet-like flow pattern is completely gone. Apparently the confinement effect of this test section is the main cause for the disappearance of this flow pattern. Note that the distance from the brim of the small diameter tube at the intersection of the abrupt enlargement to the upper or lower surface of the rectangular tube for the 9 × 3 mm tube is two times larger than that of the 6 × 3 mm tube. Hence, the liquid flowing out of the smaller diameter tube touches the adjacent walls comparatively easily, thereby eliminating the "liquid jet-like" flow pattern accordingly. In addition, the size of air bubbles of the 6 × 3 mm tube (Figure 4g) is much smaller than that of the 9 × 3 mm tube at $x = 0.001$ (Figure 4a).

For a higher mass flux of 300 $\text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$, progress of flow pattern change is somewhat similar to that at $G = 100\ \text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$ except that the development of flow pattern moves toward a lower quality region. However, the liquid jet-like flow pattern is rather unclear at higher gas quality in both test sections. Therefore, the leveling-off phenomenon of the expansion pressure loss is almost unseen.

The measured pressure differences subject to abrupt enlargement are compared with the foregoing models/correlations described in the literature review. In total, nine models and correlations were used for comparison; detailed comparisons indicated that none of the models/correlations can predict the present test results with an acceptable accuracy. There are several possible causes for this significant departure. The first one is attributable to the evaluation of void fraction. Note that most of the models/correlations require information on a void fraction, except for the homogeneous model, the Chisholm and Sutherland [10] correlation, and the Wadle correlation [12]. This eventually involves suitable selection of the correlation of void fraction. In this study, the void fraction correlation being used is identical to those suggested in previous researchers. However, one still sees a tremendous digression from these comparison charts. The second cause of this deviation is related to the geometric difference. The previous geometries are mostly at a much larger scale, except those by Abdelall et al. [1]. Furthermore, previous studies are generally applicable for round/round sudden enlargement, while the present experiments are conducted for round/rectangular transformation. The best predictive abilities for the previous models are those by Wadle [12] with a mean deviation around 200%. The deviation is calculated as $\frac{1}{N}(\sum_{1}^{N} \frac{|\Delta P_{PRED}-\Delta P_{EXP}|}{\Delta P_{EXP}}) \times 100\%$, where N is the number of total data points. The results imply that further efforts should be made for seeking a better correlation.

Results for Contraction

Figure 5 shows the measured two-phase pressure drops subject to the sudden contraction for the 6 × 3 mm and 9 × 3 mm channels, respectively; each test section has 40 data points. In general, the contraction pressure drops increase with the quality and mass flux. However, a noticeable drop of the pressure loss is encountered at a low mass quality region ($x < 0.1$). This phenomenon was also observed by Schmidt and Friedel [18] for their air–water data at $G = 1000\ \text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$, where a local peak of pressure drop versus vapor quality is seen at a low mass flow quality. This phenomenon was characterized as a change of flow pattern in the inlet pipe or outlet pipe. In this study, a notable pressure drop is seen for $G = 100\ \text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$ and a gas quality

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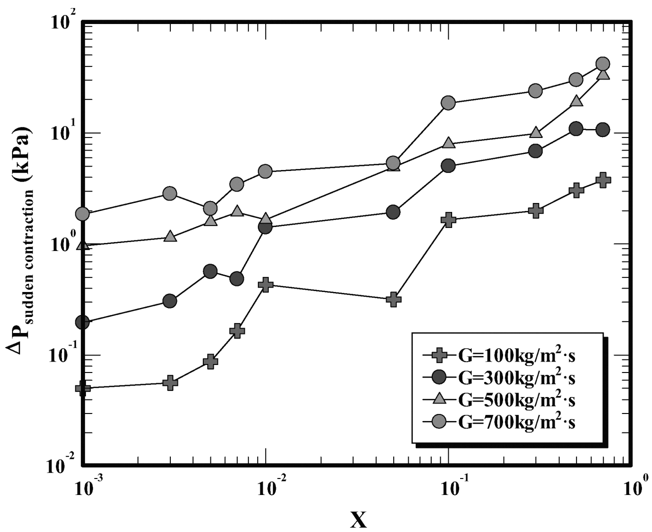
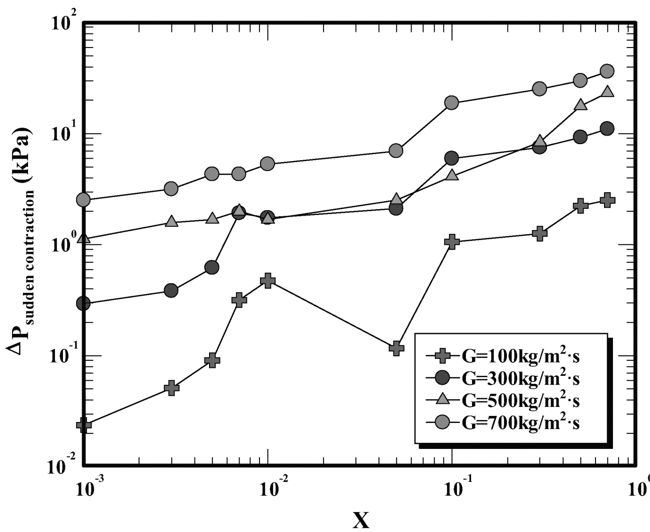


(a) Width: 6 mm x Gap: 3 mm



(b) Width: 9 mm x Gap: 3 mm

Figure 5 Measured pressure change as a function of mass flux and quality for (a) 6 mm × 3 mm and (b) 9 mm × 3 mm channel.

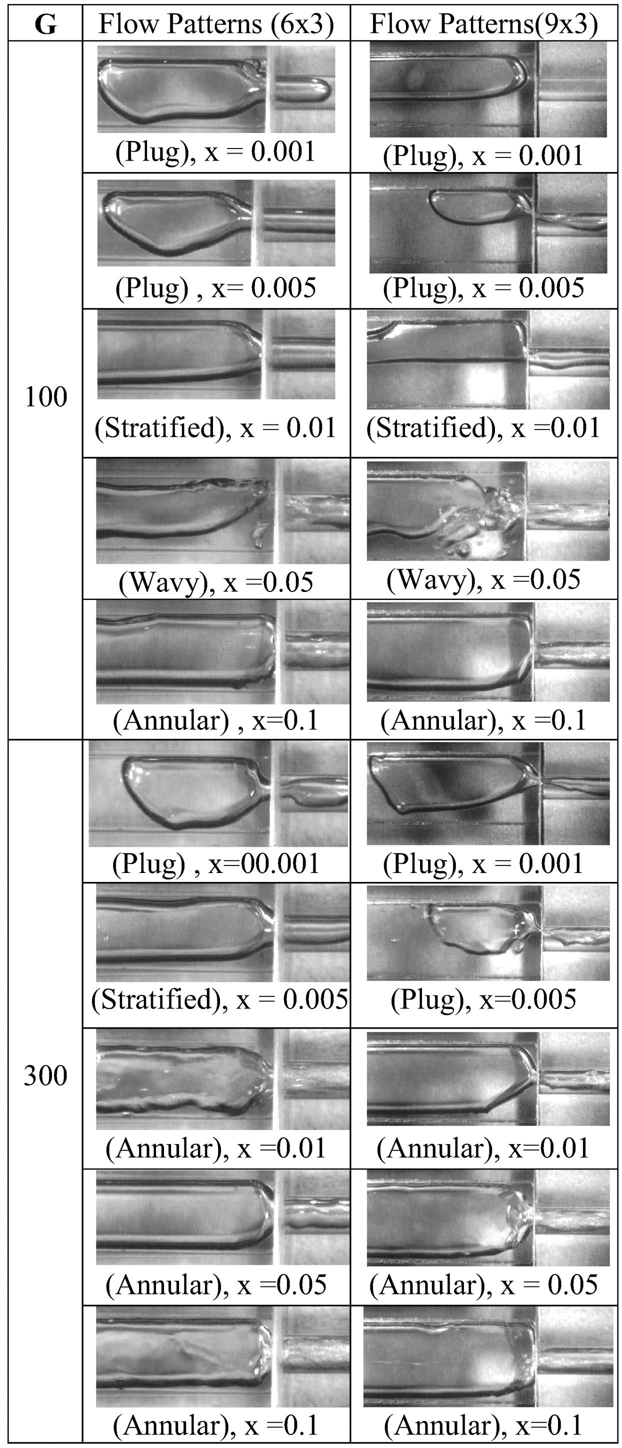


Figure 6 Progress of flow pattern vs. quality across the sudden contractions for $G = 100$ and 300 $kg{\cdot}m^{-2}{\cdot}s^{-1}$.

of 0.05 as shown in Figure 5. For further understanding about the phenomenon, one can see the progress of the flow pattern subject to the influence of contraction from Figure 6. As shown in the figure, for a mass flux of 100 $kg{\cdot}m^{-2}{\cdot}s^{-1}$ at a low quality less than 0.01, one can see that the intermittent flow pattern prevails before and after the contraction. Although the stratified flow might persist after the contraction, one can see that the vena contracta is clearly absent. Note that the mentioned vena contracta is slightly different from that in single-phase flow. For single-phase flow, the vena contracta represents the point in a fluid stream where the diameter of the jet stream is the least, such as in the case of a stream issuing out of a nozzle. Here the observed vena contracta denotes the gas stream surrounded by liquid with the smallest diameter of gas core.

With the rise of vapor quality ($x = 0.01$–0.1), the vena contracta is observable for the major flow pattern after the contraction turn into an annular flow pattern. As delineated by Collier and Thome [13], the presence of the vena contracta effectively converted the accelerated and pressure energy into kinetic energy with little or no frictional dissipation from upstream. This

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will certainly offset the irreversible pressure loss caused by the singularity. Beyond the vena contracta position, the conditions are similar to those of sudden enlargement through which considerable frictional dissipation occurs. In essence, the considerable deflection of the contraction pressure drop is associated with the transition of forming the vena contracta. For a larger mass flux like $G = 300$ kg·m^{-2}·s^{-1}, the vena contracta is still seen even at a very low quality range of 0.001–0.005. However, the rising frictional contribution with mass flux will counteract the occurrence of vena contracta, thereby leading to a comparatively small influence of vena contracta on the pressure drop. In addition, formation of vena contracta also shifts toward a smaller gas quality, giving rise to an early deflection of pressure drop at a lower gas quality. However, it should be mentioned that this phenomenon becomes less and less pronounced when the mass flux is increased. This is because the increasing frictional performance counterbalances the effect of the vena contracta.

To test the validity of the foregoing described models/correlations from the existing literatures, the measured two-phase pressure drop data subject to abrupt contraction in two test sections are compared with the previously described homogeneous model (Collier and Thome [13]), and correlations of Chisholm [5], Schmidt and Friedel [18], and Abdelall et al. [1]. The detailed comparisons and relevant mean deviations for the models/correlations give fair predictions of the present data, but none of them can accurately predict the entire database. Note that most of the existing data were conducted for the geometry from round/rectangular to round/rectangular configuration, while the present test section is from rectangular to round tube configuration. The average mean deviations of the relevant predictions are 49.15%, 54.12%, 50.8%, 64.5%, and 74.8% by the homogeneous model (Eq. 35), Chisholm homogeneous model (Eq. 36), Chisholm correlation (Eq. 37), Schmidt and Friedel correlation (Eq. 40), and Abdelall et al. correlation (Eq. 43), respectively. Also, the Abdelall et al. simplified correlation (Eq. 43) gave a mean deviation of more than 100%. The homogeneous model gives better predictive ability than the others, which had also been indicated by Geiger [6] and McGee [17]. For homogeneous predictions, the data with higher pressure drop give a good agreement with the predictions. Note that the flow patterns for these data with good predictions are actually annular flow. However, the data for all slug and stratified flows, where buoyancy force plays a significant role, are greatly over the predictions.

There are some pressure drop data for two-phase flow across sudden contractions available from the literature, as mentioned earlier. For comparison purposes, the present data and some available data along with some databases (Geiger [6]; McGee [17]; Schmidt and Friedel [18]; Abdellal et al. [1]) are compared with the relevant correlations and models. In general, the homogeneous predictions give a very good agreement with the two data sets reported by Geiger [6] and McGee [17], with corresponding mean deviations of 19.93% and 16.81%, respectively. The mean deviations for the data of Schmidt and Friedel [18] and the present data are 43.46% and 49.15%, respectively. Nevertheless, the homogeneous model significantly overpredicted the data of Abdelall et al. [1] with 468.79% mean deviation, which was also previously reported by these authors. The possible cause for this large deviation is associated with the micro-tubes (with one inner diameter of 1.60 mm and the other of 0.84 mm) being tested. For these micro-tubes, influence caused by surface tension takes control. For obtaining a better predictive ability, one should also take into account the influence of surface tension force (Tripplet et al. [21]). The balance of buoyancy and surface tension force can be represented by the Bond number (Bo) as:

$$Bo = \frac{g(\rho_L - \rho_G)(D/2)^2}{\sigma} \tag{48}$$

When the value of Bo is near or less than 1.0, the stratified flow pattern is not able to exist in most of the two-phase flow conditions. Chen et al. [2] had utilized the Bond number (Bo) to count for the force balance between buoyancy and surface tension in development of the two-phase frictional pressure drop correlation in small tubes. From the foregoing comparison, it is found that the homogeneous model gives good predictive ability but fails to predict the Abdelall et al. microchannel data. By examining the corresponding Bond number for these five databases, it is found that the departure of the predictive ability of homogeneous model is strongly related to the Bond number. In that regard, it is suggested that the influence of Bond number should be included in future development of the correlation.

CONCLUSIONS

Two-phase flow approaching singularities such as abrupt expansion and sudden contraction is widely encountered in typical industrial and heat-exchanging devices. For conventional channels where the flow path may be comparatively long, the associated pressure drops relative to the total pressure drops may be small and are often neglected. However, with smaller and smaller channels being introduced (such as mini- or microchannels) to the real world, one can no longer ignore their influence, and the need for more accurate estimation of the expansion/contraction loss becomes evident. There have been some studies concerning this subject but mostly applicable for larger channels. The major effort of this study is to review the existing efforts concerning two-phase flow across sudden expansions/contractions and to examine the applicability of the existing correlations with respect to the recent data in small channels. The second part of this study is to present some newly measured pressure drops and observed flow patterns pertaining to influence of expansion/contraction. For an abrupt expansion, it is found that the existing correlations all fail to provide a reasonably predictive capability versus the newly collected data. Furthermore, a unique flow pattern called a "liquid jet-like flow pattern" occurs at a very low quality region of $G = 100$ kg·m^{-2}·s^{-1}, and it raises a setback phenomenon of pressure drop. By contrast, an appreciable increase of pressure difference is seen when the liquid jet-like flow pattern is completely gone. Similar conclusion is drawn to the data of contractions. For the correlations/predictive models, the homogeneous model gives satisfactory prediction for conventional macrochannels but fails to do so when the channels become smaller. This is especially pronounced for a small-diameter tube with a Bond number being less than 1, in which the effect of surface tension dominates. It is also found that most existing correlations can only predict their own database, and extrapolation of their correlations outside their test range is questionable. Hence, it is suggested that the Bond number should be introduced to modify the existing homogeneous model.

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NOMENCLATURE

A	aspect ratio, gap/width, $0 \leq A \leq 1$
C_h	Chisholm's factor, defined in Eq. (15)
Bo	Bond number, defined in Eq. (48)
C_C	contraction coefficient
d	internal diameter of circular tube, m
D_h	hydraulic diameter, m
f	friction factor
f_{con}	total fiction factor for the contraction
g	rectangular channel gap, mm
G	total mass flux, $kg \cdot m^{-2} \cdot s^{-1}$
j_L	superficial liquid velocity, $m \cdot s^{-1}$
j_G	superficial gas velocity, $m \cdot s^{-1}$
K	loss coefficient
$\dot{m}$	mass flow rate, $kg \cdot s^{-1}$
P	pressure, Pa
ΔP_c	pressure change across the sudden contraction, Pa
ΔP_{cL}	pressure drop across the sudden contraction for total flow assumed liquid, Pa
ΔP_{iv}	static pressure drop to the vena contracta, Pa
ΔP_{vc}	pressure recovery downstream of the vena contracta, Pa
ΔP_e	pressure change across the sudden expansion, Pa
ΔP_{eI}	irreversible pressure changes across the sudden expansion, Pa
ΔP_{eR}	reversible pressure changes across the sudden expansion, Pa
dP/dz	frictional pressure gradient, $Pa \cdot m^{-1}$
Re	Reynolds number, $\rho ud/\mu$
S	slip ratio
u	mean axial velocity in the small tube, $m \cdot s^{-1}$
x	gas quality
W	rectangular channel width, mm
We	Weber number
Γ_{con}	base pressure coefficient for the contraction
X	Martinelli parameter
X_{CH}	correlation parameter for Chisholm and Sutherland [10]; see Eq. (14)

Greek Symbols

σ	surface tension, $N \cdot m^{-1}$
σ_A	flow cross-sectional area contraction ratio
α	mean void fraction
α_E	mean volumetric liquid entrained in the gas flow
μ	viscosity, $N \cdot s \cdot m^{-2}$
Φ_{cL}	two-phase multiplier with all liquid flow through the contraction
β	gas volumetric flow ratio, $= j_G/(j_G + j_L)$
ρ	density, $kg \cdot m^{-3}$
ρ'	fictitious mixture density defined in Eq. (33), $kg \cdot m^{-3}$
ρ''	fictitious mixture density defined in Eq. (34), $kg \cdot m^{-3}$
ρ_{eff}	effective density defined in Eq. (21), $kg \cdot m^{-3}$

Subscripts

con	contraction
EXP	experiments
G	gas-phase
h	homogeneous
L	liquid-phase
PRED	values for correlation or model predictions

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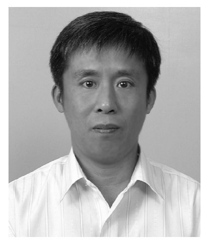

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