Two-phase Flow Distribution in Plate Heat Exchanger and Bifurcation Channel

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Two-phase Flow Distribution in Plate Heat Exchanger and Bifurcation Channel

By

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*By (the Token of) Time (through the ages)*
*Verily Man is in loss*
*Except such as have Faith, and do righteous deeds, and (join together) in the mutual teaching of Truth, and of Patience and Constancy.*
*(Quran, Chapter 103 surat Al-Asr (The Declining Day), verse 1-3)*
ABSTRACT

Distributor is an essential section for fluid distribution, moreover for heat exchanger, reactor, fuel cell etc. Uniformity distribution must be reached, in order to get better heat and mass transfer as well as hydraulic performance. For single-phase flow is easier, rather than two-phase flow, due to complicated interaction between phases. There is anomaly in two-phase flow. Usually, the when uniformity is enhanced, better heat transfer performance is achieved. Unfortunately, this achievement followed by decreasing hydraulic performance, specially increasing pressure loss. This problem comes to severer, because the behavior of two-phase flow distribution has not been understood clearly.

Here, the investigation is directed to explore more detail about two-phase flow behavior using both experimental and modelling approached. The works is emphasized on bifurcation channel distributor and expanded to brazed plate heat exchanger. The bifurcation channel distributor is the basic form of most of distributors including plate heat exchanger. In the bifurcation channel distributor the phase not only passes straight along channel but also some phases impacts to the wall as well as devises in different channel, some phase devises without impacting the wall, in one region, junction section. Different from bifurcation channel distributor, in the T-junction all phases impact and devise to channels.

The bifurcation channel distributor, experiments have been carried out upon the distributor constructed by acryllics resembling merged triple pipe, 8 mm in diameter of main channel and two set 5 mm in diameter of each outlet channel. The test section had been tested in three positions; horizontal, 45° inclined and vertical. Three flow patterns were fed i.e.; bubble, slug and stratified flow and the phase as well as their distributions were observed via high speed video camera. The pressure distribution was measured by series of U-tube water gauge manometer. The flow patterns, phase distribution and pressure drop were analyzed by CFD software, validated by experimental data and compared by existing correlation, analytically.
The plate heat exchanger, experiments have been done on acrylic corrugated channels as mockup of brazed plate heat exchanger channel, having chevron angle 30° and corrugation depth 1.5 mm. The stratified flow is fed to test section in similar total mass flux $G=10.0$ kg.s$^{-1}$.m$^{-2}$, in different mass quality from 0.1 to 0.6. The void fraction is weighed carefully using digital scale, the phase distribution also are tracked by using same high speed video camera.

For modelling, the boundary condition for both of bifurcation channel distributor and plate heat exchanger were set up similar to experiment setup condition, nevertheless there is some simplification in corrugated channel geometry, in order to attain convergence in iteration. Another modeling work also was done. There are comparison studies of hydraulic and thermal performance for corrugated and dimple/protrusion plate of plate heat exchanger on single phase mode.

There are revealed that; the uniformity of two phase flow distribution depends on flow pattern, channel geometry and its orientation. The some flow patterns can transform to another form while they impact and devise in junction distributor. There is oscillation and damping effect in between two phases in flow field, caused by successive changing kinetic energy and pressure energy when the phase impact the wall and devise in junction distributor. The existing correlations of two-phase flow should be reconstructed, to provide complicated problem in distributor.

For plate heat exchanger, it were discovered that; similar to the bifurcation channel distributor for uniformity behavior and also the uniformity is depend on flow orientation. The upward flow tends to has more lack uniformity of phase distribution than downward flow, due to bigger slip factor and effect of damping of liquid and geometry as well as buoyancy. The increasing gas quality drives increasing void fraction in certain trend line quite different from existing correlations. The better expected correlation of void fraction for plate heat exchanger is formulated based on drift flux model.

To support understanding to this study, here also supported relevant basic science of two-phase flow and modelling for single-and two-phase flow.
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Nomenclature

$A$  cross sectional area ($m^2$)

$m$  mass flow rate (kg/s or kg/min)

$C$  Martinelli coefficient, diverging ratio of venturi tube

$c_p$  specific heat (kJ/kg K)

$d$  differential operator

$D$  tube diameter (m), substantial derivative

$F$  frictional, usually for two phase pressure drop

$Fr$  Froude number, dimensionless

$G$  Mass flux or mass velocity (kg/m$^2$s)

$g$  acceleration due to gravity (m/s$^2$)

$h$  height (m)

$i$  enthalpy (kJ/kg)

$j$  superficial velocity (m/s)

$K$  slip ratio, dimensionless

$L$  length (m)

$p$  pressure (Pa)

$Pr$  Prandtl number, dimensionless

$Q$  volume flow rate ($m^3$/s)

$R$  result, uniformity ratio, deviation ratio, dimensionless

$Re$  Reynolds number, dimensionless

$R_L$  liquid holdup or liquid fraction, dimensionless

$S$  slip ratio, slip factor, source term momentum in CFD, dimensionless

$T$  temperature (K)

$t$  time, instant time (s)

$u$  velocity (m/s), superficial velocity (m/s), velocity in x direction (m/s), uncertainty (%)

$U$  drift velocity
specific volume \( (m^3/kg) \), velocity in \( y \) direction \( (m/s) \)
mass flow rate \( (kg/s) \)
velocity in \( z \) direction \( (m/s) \)
Martinelli parameter, dimensionless
quality or dryness fraction, dimensionless
length of element in the finite-difference grid, m

Greeks symbols

\begin{align*}
\alpha & \quad \text{void fraction, dimensionless} \\
\beta & \quad \text{volumetric quality, dimensionless} \\
\partial & \quad \text{partial derivative} \\
\nabla & \quad \text{del operator} \\
\rho & \quad \text{density (kg/m}^3)\text{)} \\
\nu & \quad \text{specific volume (m}^3/kg\text{)} \\
\mu & \quad \text{dynamic viscosity (Pa.s)} \\
\nu & \quad \text{kinematic viscosity (Stoke, cm}^2/s\text{)} \\
\zeta & \quad \text{phase compressibility factor, dimensionless} \\
\omega & \quad \text{Beattie and Whalley correlation, dimensionless} \\
\phi & \quad \text{two-phase frictional multipliers, dimensionless} \\
\psi & \quad \text{parameter in Baker’s flow regime map} \\
\theta & \quad \text{inclination angle of pipe} \\
\lambda & \quad \text{parameter in Baker’s flow regime map} \\
\sigma & \quad \text{surface tension (N/m), standard deviation, converging ratio of} \\
& \quad \text{nozzle/venturi} \\
\tau & \quad \text{shear stress (N/m}^2)\text{)} \\
\Gamma & \quad \text{The rate of production of phase mass from the phase changes at the} \\
& \quad \text{interfaces per unit volume} \\
\varepsilon & \quad \text{percentage error}
\end{align*}
Subscripts

$v$  viscous
$a$  air, momentum component in pressure gradient
$ave$  average
$b$  based side
$E$  energy
$f$  frictional component in pressure gradient, liquid phase
$fg$  gas liquid fraction different
$G$  gas phase
$g$  gravitational component in pressure gradient or gas phase
$h$  homogenous
$Hom$  Homogeneous model
$i$  number of a variable
$L$  liquid phase
$M$  Momentum component
$m$  mixture
$o$  based on Fanning equation
$q$  phase variable (gas or liquid)
$R$  relative
$SI$  area change effect for Tapucu correlation
$t$  turbulent
$TP$  two-phase
$u$  upper side
$W$  water
$x$  x direction in Cartesian coordinate
$y$  y direction in Cartesian coordinate
$z$  gravitational component in pressure gradient
$z$  z direction in Cartesian coordinate
Chapter 1
Introduction

1.1 Background

Point of view this work is represented to word ‘distribution’. There are many lexical meanings for word ‘distribution’, depend on users. However the closed meanings associated with this work is; ‘the way in which the fuel-air mixture is supplied to each cylinder of a multi cylinder internal-combustion engine, used for engineering, Licker (2003). Here, distribution is the way how the fluid is distributed to individual or collective devices. In heat exchanger, distribution of fluid evenly to each tube of arranged tube banks, is the most important one. The ‘distributor’ plays major role for this purpose. For refrigeration system containing multiple condenser and or evaporators, that is used to distribute fluid evenly or properly to each of them. Special case is in plate heat exchanger, Fig. 1. There are two distributors; header distributor which distributes fluid from main conduit to each channel distributor and channel distributor which distributes fluid from header distributor to downstream channel, weaved channel. Both channel and its conjugate distributor are formed by stacked plates.

![Fig. 1.1 Gasket plate heat exchanger and its distributors, courtesy of Teplotex®](image)
Also, the distributors play major role for enhancing hydraulic and thermal performance. Uniform flow distribution and the low pressure loss in them are two main parameters which shall be become target. They are quite complicated, since mostly both parameters have anomaly effect. The increasing uniformity usually is followed by increasing pressure loss and vice versa. The dense is more serious, when the working fluid is two-phase flow, due to existence of vapor-liquid interface momentum exchange, interfacial tension forces and phase-change phenomena, Winkler et al. [2012]. Accordingly, understanding behavior of flow distribution in distributor is the most fundamentals requirement, in order to succeed both of target parameters.

The complicated investigation of two-phase flow for authentic distributors, promote to generalize and to simply them on to become bifurcation channel distributor form. Bifurcation channel is the basic form of distributor, distributing fluid from single pipe main channel to double pipe outlet. They are merged in certain configuration, as shown on Fig. 1.2. Meanwhile, the authentic distributors, having; double outlet channels or numerous outlet channels is depicted on Fig. 1.3 or having complex and various channels of the distributor such as in plate heat exchanger, Fig. 1.1. As shown on Figure 1.3, the bifurcation channel distributor could represent as basic profile for the most of distributors junction.

![Fig. 1.2 Bifurcation channel distributor](image)

(a) acrylic test section, (b) 3d modeling and (c) 2d plot
Fig. 1.3 Authentic distributors; (a) double outlet port (b) multiple outlet port (c) multiple port header (d) shell and tube (e) binary arborescence (f) constructal design (g) multiple structural bifurcation

The authentic distributors of (a) to (d) have been commercially implemented, while (f) to (h) still have been developed for particular purpose, such as; fuel cell, reactor, heat exchanger, etc. Some of their works is summarized in table 1.1. There is the fact that, mostly, their uniformity flow distribution has not been achieved if the working fluid is substituted by adiabatic two-phase flow instead of single phase flow. Moreover, by non-adiabatic two phase flow, their uniformity shall be absence. Conversely, also, if the
distributor is designed for two-phase flow, their uniformity goes to lack when is implemented for single phase flow. Accordingly, the distributor should be designed referring to their working fluid.

Table 1.1 Previous works related to distributor

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Zang et al. (2012) | heat exchanger, catalytic, packed bed, array of test tubes | to minimize the global power requirement subject to the total volume and channel-material constraints | Single-phase (air)

Sukas et al. (2013) | Solid micro fabricated column structures for capillary electrochromatography | Evaluation of injector/distributor structures for solid micro fabricated column structures for CEC | Single-phase of chemical solution

Hoang et al. (2014) | multiphase microreactors | To design and to operate a multi-junction bubble distributor | Two-phase flow (HFE-7500 and air)

As described on forgoing table 1.1, Most of them play on single-phase flow. The study regarding two-phase flow distributor are only the Hoang’s (2014) and Tseng’s (2012) works. Both of them direct to particular purpose; micro reactor and fuel cell. The basic fundamental of two-phase flow distribution on them has not discovered, clearly. Instead, the rather clear phenomenon come from tee-junction studies, as briefly described on following table 1.2

Table 1.2 The works of flow through tee-junction

<table>
<thead>
<tr>
<th>Work</th>
<th>Experiment</th>
<th>Results</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ji et al. (2009)</td>
<td><img src="image" alt="" /></td>
<td>Obtain momentum correction coefficient; $K = 1 + 0.256M^{0.221}$ $M = \frac{\rho_1u_1^2}{\rho_2u_2}$</td>
<td>$d_1 = d_2 = d_{out}$ =16mm Horizontal</td>
</tr>
<tr>
<td>Azzi et al. (2010)</td>
<td><img src="image" alt="" /></td>
<td>• The phase superficial velocities influence the misdistributions • the diameter and pressure do not affect the phase maldistribution</td>
<td>$d_{in} = d_1 = d_2$ =1mm Horizontal</td>
</tr>
<tr>
<td>Work</td>
<td>Experiment</td>
<td>Results</td>
<td>Remarks</td>
</tr>
<tr>
<td>-----------------------</td>
<td>------------</td>
<td>--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------</td>
<td>--------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------</td>
</tr>
</tbody>
</table>
| He et al. (2011)      | ![Diagram](image1.png) | - Phase split characteristic highly depends on flow patterns of inlet  
- Slightly depend on cross section of channel  
- The phase distribution is remarkably influenced by the surface tension | Working fluid; nitrogen-water                                                                                                                                  |
| Wang et al. (2011)    | ![Diagram](image2.png) | The properties of non-Newtonian fluid have little effect on phase split for slug flow, while for annular flow the influence is significant | Working fluid; nitrogen-aqueous solutions of carboxymethyl cellulose (CMC)                                                                                                                                 |
| Choi et al. (2011)    | ![Diagram](image3.png) | In the hydrophilic, the major flow patterns were bubbly, elongated bubble and liquid ring flow.  
In the hydrophobic, the major flow pattern was stratified flow, which was governed by capillary force and hydrophobicity. | microchannel;  
- Hydrophobic 608µmX410µm  
- Hydrophobic 617µmX430µm                                                                                                                                          |
| Mohamed et al. (2011) | ![Diagram](image4.png) | The degree of mal-distribution of the phases depends on the inlet conditions and the mass split ratio at the junction and the inclination angle of the outlets.  
The phase distribution in stratified flow is very sensitive to the outlet angle | $d_{in} = d_1 = d_2 = 13.5$ mm  
Inclination, $\theta$: 2.5°, 7.5°, 15°, 30°, 60°, 75°, and 90°                                                                                       |
| Mohamed et al. (2012) | ![Diagram](image5.png) | Decreasing the diameter of T-junctions increases the fraction of liquid taken off | $d_{in} = d_1 = d_2 = 0.5$ mm  
Horizontal                                                                                                                                         |
| Elazhary et al. (2012)| ![Diagram](image6.png) | New correlations developed for the pressure drop at the junction under single- and two-phase flow conditions  
Obtaining models for predicting the average gas and liquid velocities and the void distribution in the fully-developed region of the inlet side of the junction | Rectangular Channel T-Junction  
20mmx1.87mm                                                                                                                                                |
| Mohamed et al. (2012)| ![Diagram](image7.png) | simple correlation, in terms of magnitude and trend, based on the similarity between these flow phenomena and the phenomenon of liquid entrainment in small upward branches was developed | $d_{in} = d_1 = d_2 = 13.5$ mm  
Inclination, $\theta$: 2.5°, 7.5°, 15°, 30°, 60°, 75°, and 90°                                                                                       |
<table>
<thead>
<tr>
<th>Work</th>
<th>Experiment</th>
<th>Results</th>
<th>Remarks</th>
</tr>
</thead>
</table>
| Reis et al. (2013)   | ![Image](image1.png) | The slug flow splitting phenomenon was showed as having a high complexity and its future modeling will require consideration of it. More efforts are requested for studying the non-linear and time-variant characteristics of these multiphase flow systems. | \(d_{in} = d_1 = d_2 = 34 \text{ mm} \)  
Horizontal Pressure measurement in transient mode |
| Chen et al. (2013)   | ![Image](image2.png) | The liquid taken off from the branch did not decrease with increasing branch angle for all kinds of inlet flow patterns. More liquid was taken off from the branch at microchannel junctions. | Rectangular T-Junction 0.5x0.5mm² |
| Tuo et al. (2014)    | ![Image](image3.png) | Liquid separation efficiency strongly depends on the flow pattern right above the impact region (junction). The refrigeration efficiency deteriorates dramatically when mist turns into churn flow regime, with increasing inlet flow rate and/or quality. An empirical correlation to predict churn flow transition is proposed as a function of Fr and \(X_{tt}\). | \(d_{in}: 8.7 \text{mm} \)  
\(d_{up}: 18.3 \text{mm} \)  
\(d_{dn}: 13.4 \text{mm} \)  
Vertical Working fluid; R134a and R410A |
| Tuo et al. (2014)    | ![Image](image4.png) | Larger diameter of the inlet tube improves the separation efficiency, (within 5%) T-junction with flat (rectangular) inlet shows superior performance. This is because flat tube confines incoming liquid flow in a thin jet, thus leaving more free flow area for the upward vapor flow and resulting lesser vapor-liquid interaction. | Inclination, \(\theta: 0°, 15°, 30°, 45°\)  
(Di/Do : 0.475, 0.563, 0.656)  
Round and Rectangular T  
Vertical Working fluid; R134a and R410A |
| Mohamed et al. (2014)| ![Image](image5.png) | The pipe diameter has a small effect on phase redistribution for the whole tested range. System pressure was found to have a significant effect on phase redistribution at small inlet velocities and this effect was found to decrease as the inlet velocities increased. The phase redistribution model shall be developed. | \(d_{in} = d_1 = d_2 = 13.5 \text{ mm and } 37.8 \text{mm} \)  
Pressure 150kPa and 200kPa (absolute) |
1.2 Problems

As described in Table 1.2, the uniformity of two-phases flow distribution in tee-junction depends on; superficial velocities, inlet flow pattern, surface tension, fluid properties, and channel orientation/angle of outlet channel. It slightly depends on; channel diameter, cross section shape of channel, branch angle, and system pressure. Also, there are some phenomena; in microchannel the flow pattern is strongly dependent on the wetness and capillarity, the phase split is very complicated during slug flow due to high uncertainty of flow pattern and fluctuating pressure.

Although many scientific proofs were discovered, the remaining questions have been disclosed yet. These are:

- Does outlet channel length affect uniformity?
- How does phase separation work?
- Why does the flow pattern strongly affect uniformity?
- Why in slug flow the phase separation comes to complex with time variance?
- If the channel diameter/cross section area varies along channel, what shall be? etc.

The answer for all the questions is absolutely required for exploring two-phase flow distribution phenomena in all types of distributors, in which have not been revealed by others.

1.3 Simplification of Problems

To investigate two-phase flow behavior in plate heat exchangers is very complicated, due to the complicated geometry of plates. Moreover, the basic fundamental of flow distribution have not clearly disclosed yet. The bifurcation channel distributor is very important as fundamental form of universal distributor instead of tee-junction. There is big difference in between tee-junction and bifurcation channel distributor. In tee-junction, all phase impact and separate to two channel or phase separate promoted by sucking phase
through branch channel. Meanwhile, the bifurcation channel distributor, the part of phase/not all impact to junction and devise to two channels, downstream junction. Basically, this impacting mechanism occurs in all type distributors from simple distributor to complex distributor. Accordingly, the bifurcation channel distributor can represent as basic fundamental form of universal distributor, Fig. 1.3, appropriate implemented for; heat exchanger, reactor, fuel cell etc. The research regarding it is limited for both single and two-phase flow.

1.4 Objective

Recent work firstly investigates briefly two-phase flow phenomena through bifurcation channel distributor including effect of; flow pattern, shape of junction, length channels, orientation, pressure distribution, fluctuating velocity and pressure. Also, here was found the new phenomena in two-phase internal flow impingement such as; phase transformation, pressure gain and loss, dumping and oscillation. Potentially, the new phenomena shall complete establish three basic two phase flow model; homogenous, separate and drift flux.

Secondly, here investigates single and two phase flow distribution in plate heat exchanger channel, covering; thermal and flow distribution of single-phase through corrugated channel and dimple-protrusion channel, two-phase flow distribution in corrugated channel including; phases, quality and void fraction. Here is proofed, that two-phase flow distribution phenomena in bifurcation channel distributor have better agreement with those in plate heat exchanger.
References


Ji, L., Wua, B., Chena, K., Zhua, J. and Liu, H., (2009), Momentum correction coefficient for two jet flows mixing in a tee junction, Chemical Engineering research and design 87, p. 1065–1068


Chapter 2
Two-Phase Flow Model and Modeling

2.1 Flow pattern

Two-phase flow is flow of pair immiscible fluid. The flow can be encountered external devices or internally. The pair immiscible fluid may content; liquid-liquid (refrigerant-oil in refrigeration system), liquid-solid (water-slag in engine cooling system), gas-solid (air-coal in pneumatic conveyor) or gas-liquid (steam-water in steam power cycle). Here, the study is emphasized on two-phase flow of gas-liquid. Mostly, in the authentic gas-liquid two-phase flow process sink or release heat. If the heat is sunk, called evaporation and another one is condensation, in which the heat is discarded.

The evaporation and condensation is very complicated processes, since the fluid transform from liquid to gas or vice versa. Therefore, the basic fundamentals two-phase flow theory is derived from adiabatic two-phase flow. Air-water is the most pair immiscible fluid, used as experiment since decades, due to their unique. Air and water have contras in properties, appendix 1. Therefore, this combination promotes stability two-phase form, resulting better investigation.

If the air and water pass through a tube in different velocity, than they form unique gas-liquid configuration. It is termed; flow-pattern. It has been investigated since several decades.

2.2 Flow Patterns in Vertical Tubes

For co-current up flow of gas and liquid in a vertical tube, the liquid and gas phases distribute themselves into several recognizable flow structures. These are referred to as flow patterns and they are depicted in Figure 2.1 and can be described as follows:

- Bubbly flow. Numerous bubbles are observable as the gas is dispersed in the form of discrete bubbles in the continuous liquid phase. The bubbles may vary widely in size and shape but they are typically nearly spherical and are much smaller than the diameter of the tube itself.
• Slug flow. With increasing gas void fraction, the proximity of the bubbles is very close such that bubbles collide and coalesce to form larger bubbles, which are similar in dimension to the tube diameter. These bubbles have a characteristic shape similar to a bullet with a hemispherical nose with a blunt tail end. They are commonly referred to as Taylor bubbles after the instability of that name. Taylor bubbles are separated from one another by slugs of liquid, which may include small bubbles. Taylor bubbles are surrounded by a thin liquid film between them and the tube wall, which may flow downward due to the force of gravity, even though the net flow of fluid is upward.

• Churn flow. Increasing the velocity of the flow, the structure of the flow becomes unstable with the fluid traveling up and down in an oscillatory fashion but with a net upward flow. The instability is the result of the relative parity of the gravity and shear forces acting in opposing directions on the thin film of liquid of Taylor bubbles. This flow pattern is in fact an intermediate regime between the slug flow and annular flow regimes. In small diameter tubes, churn flow may not develop at all and the flow passes directly from slug flow to annular flow. Churn flow is typically a flow regime to be avoided in two-phase transfer lines, such as those from a reboiler back to a distillation column or in refrigerant piping networks, because the mass of the slugs may have a destructive consequence on the piping system.

• Annular flow. Once the interfacial shear of the high velocity gas on the liquid film becomes dominant over gravity, the liquid is expelled from the center of the tube and flows as a thin film on the wall (forming an annular ring of liquid) while the gas flows as a continuous phase up the center of the tube. The interface is disturbed by high frequency waves and ripples. In addition, liquid may be entrained in the gas core as small droplets, so much so that the fraction of liquid entrained may become similar to that in the film. This flow regime is particularly stable and is the desired flow pattern for two-phase pipe flows.

• Wispy annular flow. When the flow rate is further increased, the entrained droplets may form transient coherent structures as clouds or wisps of liquid in the central vapor core.
• Mist flow. At very high gas flow rates, the annular film is thinned by the shear of the gas core on the interface until it becomes unstable and is destroyed, such that all the liquid in entrained as droplets in the continuous gas phase, analogous to the inverse of the bubbly flow regime. Impinging liquid droplets intermittently wet the tube wall locally. The droplets in the mist are often too small to be seen without special lighting and/or magnification.

Fig. 2.1 Two-phase flow pattern in vertical upward flow, courtesy of Thermopedia™

2.3 Flow Patterns in Horizontal Tubes
Two-phase flow patterns in horizontal tubes are similar to those in vertical flows but the distribution of the liquid is influenced by gravity that acts to stratify the liquid to the bottom of the tube and the gas to the top. Flow patterns for co-current flow of gas and liquid in a horizontal tube are shown in Fig. 2.2 and are categorized as follows:

- Bubbly flow. The gas bubbles are dispersed in the liquid with a high concentration of bubbles in the upper half of the tube due to their buoyancy. When shear forces are dominant, the bubbles tend to disperse uniformly in the tube. In horizontal flows, the regime typically only occurs at high mass flow rates.
- Stratified flow. At low liquid and gas velocities, complete separation of the two phases occurs. The gas goes to the top and the liquid to the bottom of the tube, separated by an undisturbed horizontal interface. Hence the liquid and gas are fully stratified in this regime.
• Stratified-wavy flow. Increasing the gas velocity in a stratified flow, waves are formed on the interface and travel in the direction of flow. The amplitude of the waves is notable and depends on the relative velocity of the two phases; however, their crests do not reach the top of the tube. The waves climb up the sides of the tube, leaving thin films of liquid on the wall after the passage of the wave.

• Intermittent flow. Further increasing the gas velocity, these interfacial waves become large enough to wash the top of the tube. This regime is characterized by large amplitude waves intermittently washing the top of the tube with smaller amplitude waves in between. Large amplitude waves often contain entrained bubbles. The top wall is nearly continuously wetted by the large amplitude waves and the thin liquid films left behind. Intermittent flow is also a composite of the plug and slug flow regimes. These subcategories are characterized as follows:
  o Plug flow. This flow regime has liquid plugs that are separated by elongated gas bubbles. The diameters of the elongated bubbles are smaller than the tube such that the liquid phase is continuous along the bottom of the tube below the elongated bubbles. Plug flow is also sometimes referred to as elongated bubble flow.
  o Slug flow. At higher gas velocities, the diameters of elongated bubbles become similar in size to the channel height. The liquid slugs separating such elongated bubbles can also be described as large amplitude waves.

• Annular flow. At even larger gas flow rates, the liquid forms a continuous annular film around the perimeter of the tube, similar to that in vertical flow but the liquid film is thicker at the bottom than the top. The interface between the liquid annulus and the vapor core is disturbed by small amplitude waves and droplets may be dispersed in the gas core. At high gas fractions, the top of the tube with its thinner film becomes dry first, so that the annular film covers only part of the tube perimeter and thus this is then classified as stratified-wavy flow.

• Mist flow. Similar to vertical flow, at very high gas velocities, all the liquid may be stripped from the wall and entrained as small droplets in the now continuous gas phase.
2.4 Flow Patterns Map

It is necessary to predict regimes as a basis for carrying out calculations on two-phase flow, and the usual procedure is to plot the information in terms of a flow regime map. Many of these maps are plotted in terms of primary variables (superficial velocity of the phases or mass flux and quantity, for instance), but there has been a great deal of work aimed at generalizing the plots, so that they can be applied to a wide range of channel geometries and physical properties of the fluids. A generalized map for vertical flows is shown in Fig. 2.3 and is due to Hewitt and Roberts (1969) (see Hewitt, 1982).

This map is plotted in terms of the superficial momentum fluxes of the two-phase $\rho_f U_f^2$ and $\rho_g U_g^2$. A generalized flow pattern map for horizontal flow is that of Taitel and Dukler (1976) (see Dukler and Taitel, 1986), and is illustrated in Fig. 2.4. This is plotted in terms of the following parameters:

\[ X^2 = \left( \frac{dp}{dz} \right)_f \left( \frac{dp}{dz} \right)_g \]  

\[ F = \frac{\rho_g}{\rho_f - \rho_g} \frac{u_g}{\sqrt{D} \cdot g \cdot \cos \alpha} \]
\[ K^2 = \frac{\rho_g \mu_g^2}{(\rho_f - \rho_g) D g \cos \theta} \frac{D \mu_f}{v_f} \]  

(2.3)

\[ T = \left( \frac{\frac{dp}{dz} F_f}{(\rho_f - \rho_g) g \cos \theta} \right)^{1/2} \]  

(2.4)

Fig. 2.3 Flow pattern map obtained by Hewitt and Roberts (1969) for vertical two-phase co-current upwards flow in a vertical tube, courtesy of Thermopedia™
Fig. 2.4 Flow pattern map for horizontal co-current flow obtained by Taitel and Dukler (1976). (See Dukler and Taitel, 1986), courtesy of Thermopedia™

where \((dp/dz)_f\) and \((dp/dz)_g\) are the pressure gradients for the liquid phase and gas phase respectively, flowing alone in the channel, \(\rho_f\) and \(\rho_g\) are the phase densities, \(u_f\) and \(u_g\) are the superficial velocities of the phases, \(D\) the tube diameter, \(\nu\) the liquid kinematic viscosity, \(g\) the acceleration due to gravity, and \(\theta\) the angle of inclination of the channel.

Taitel et al. (1980) also produced a flow pattern map for vertical flow, but this has met with less widespread use. Following similar approaches, Barnea (1987) has produced a unified model for flow pattern transitions for the whole range of pipe inclinations.

2.5 Two-phase flow model

The complicated two-phase flow patterns can be approached as a simple model as illustrated on Fig 2.5. A gas and a liquid pass through channel having constant \(A\) cross section area. They form gas phase velocity \(u_g\) normal to gas area \(A_g\) and liquid velocity \(u_f\) normal to liquid area \(A_f\). Therefore total cross section area \(A=A_g+A_f\). If the point of view is in instant channel length \(dz\) at instant time \(dt\).
Thus the $u_g, u_f, A_g$ and $A_f$ are approached to constant. Hence following set equations can be determined;

- The **void fraction** $\alpha$, ratio of gas cross section area $A_g$ to total area $A$

\[
\alpha = \frac{A_g}{A}, \text{ so } (1 - \alpha) = \frac{A_f}{A} \tag{2.5}
\]

Void fraction is an essential dimensionless for two-phase flow parameters calculation. Since, in the real case $A_g$ is not always constant along $z$, than the equation 1.1 is valid for very limited incident only. Therefore, most of void fraction is not defined based on area, but based on volume, termed as **volume void fraction**. Later, several void fraction correlations, volume based, shall be presented soon.

- The **mass quality** $x$, ratio of gas mass flow rate $W_g$ to total mass flow rate $W$

\[
x = \frac{W_g}{W_g + W_f}, \text{ so } (1 - x) = \frac{W_f}{W_g + W_f} \tag{2.6}
\]

It is should be remarks, that the mass quality or some time called as ‘quality’ only is very different from void fraction. Because, quality is related to mass which strongly depend
on density $\rho$. However, both of quality and void fraction have particular proportionality, which will be discussed, later.

- **The mass velocity**

$$G = \frac{W}{A} = \rho u = \frac{u}{v} \quad (2.7)$$

- **The mass flow rate**

$$W_g = GAx \quad \text{and} \quad W_f = GA(1 - x) \quad (2.8)$$

- **The phase velocity**

$$u_g = \frac{W_g}{\rho_g A_g} \quad \text{and} \quad u_f = \frac{W_f}{\rho_f A_f} \quad (2.9)$$

Where, the mass flow $W$ is proportional to quantity, volume flow rate $Q$ than;

$$u_g = \frac{Q_g}{A_g} \quad \text{and} \quad u_f = \frac{Q_f}{A_f} \quad (2.10)$$

Therefore, the phase velocity can be formed as function of void fraction and quality, $u=f(\alpha, x)$;

$$u_g = \frac{Gx}{\rho_g \alpha} \quad \text{and} \quad u_f = \frac{G(1-x)}{\rho_f (1-\alpha)} \quad (2.11)$$

- **The volumetric quality**

$$\beta = \frac{Q_g}{Q_g + Q_f} \quad \text{so} \quad (1 - \beta) = \frac{Q_f}{Q_g + Q_f} \quad (2.12)$$

All of forgoing equations are based on phase area ($A_g$ and $A_f$), in which is vary along channel length $z$ and time $t$. Accordingly, it is urgent to simply the equation based on total cross section area $A$ which is equal to tube cross section area, constant. This is superficial velocity parameter $j$.

- **The volumetric flux or the superficial velocity, $j$**

$$j = \frac{Q}{A}, \quad \text{so} \quad j_g = \frac{Q_g}{A} \quad \text{and} \quad j_f = \frac{Q_f}{A} \quad (2.13)$$

$$j_g = u_g \alpha = j\beta = \frac{Gx}{\rho_g}\quad \text{and} \quad j_f = u_f (1 - \alpha) = j(1-\beta) = \frac{G(1-x)}{\rho_f} \quad (2.14)$$

$$G_g = j_g \rho_g = Gx, \quad G_f = j_f \rho_f = G(1-x) \quad \text{in which} \quad G = G_g + G_f \quad (2.15)$$
Superficial velocity is very important parameter for defining the phases velocity \( u_g \) and \( u_f \), by condition of which void fraction \( \alpha \) is known. Also, superficial velocity is easy parameter to calculate, since the variables are easy to measure, as following measured variables, quantity \( Q \). This work, quantity of each phase was measured carefully, by means, the phase is separated in separator tube, followed by quantifying liquid and gas volume \( V_g \) and \( V_f \) in certain time interval and the last phase quantity is determined by;

\[
Q_g = \frac{V_g}{t_g} \quad \text{and} \quad Q_f = \frac{V_f}{t_f}
\]  

(2.16)

Since the gas and liquid velocity is different. It is very important to define ratio in between gas velocity \( u_g \) and liquid velocity \( u_f \), termed as the slip factor \( S \)

- The slip ratio

\[
S = \frac{u_g}{u_f} = \frac{W_g \rho_f A_f}{W_f \rho_g A_g} = \left( \frac{x}{1-x} \right) \left( \frac{\rho_f}{\rho_g} \right) \left( \frac{1-\alpha}{\alpha} \right)
\]  

(2.17)

Later, the control volume of instant two-phase flow in Fig. 2.5 can be solved analytically.

- Total pressure drop (overall static pressure gradient) is :

\[
\left( \frac{dp}{dz} \right) = \left( \frac{dp}{dz} F \right) + \left( \frac{dp}{dz} a \right) + \left( \frac{dp}{dz} z \right)
\]  

(2.18)

The term \( \left( \frac{dp}{dz} F \right) \) represent frictional pressure drop, while

The net frictional force acting on each phase;

\[
(dF_g + S) = -A_g \left( \frac{dp}{dz} gF \right) dz ; \quad (dF_f - S) = -A_f \left( \frac{dp}{dz} fF \right) dz
\]  

(2.19)

\[
(dF_g + dF_f) = -A \left( \frac{dp}{dz} F \right) dz
\]  

(2.20)
And \(-\left(\frac{dp}{dz}\right)\) is momentum pressure drop, promoted by quality change;

\[
-\left(\frac{dp}{dz}\right) = \frac{1}{A} \frac{d}{dz} (W_g u_g + W_f u_f) = G^2 \frac{d}{dz} \left[ \frac{x^2 v_g}{\alpha} + \frac{(1-x)^2 v_f}{(1-\alpha)} \right]
\]

(2.21)

And \(-\left(\frac{dp}{dz}\right)\) is the gravitational pressure drop

\[
-\left(\frac{dp}{dz}\right) = g \sin \theta \left[ \frac{A_g}{A} \rho_g + \frac{A_f}{A} \rho_f \right] = g \sin \theta \left[ \rho_g (1-\alpha) \rho_f \right]
\]

(2.22)

It should be emphasized at this point that the frictional component has been defined in terms of the force \((dF_g + dF_f)\)

To solve all of forgoing equation is hard due to existing two unknown differential variable \(u_g\) and \(u_f\). Therefore, approach solving is required. There are two approaching models i.e. homogeneous model and separated model. Homogeneous model assumes that both of phase gas and liquid pass in equal velocity, in its mean velocity. So as, the two phase flow problem considers as single phase flow and all properties are determined based on mean properties of both phases. Meanwhile, the separated model assumes that the phases is artificially segregated into stream; one of is liquid and another one is gas, and each phase velocity is the mean velocity of each phase, so that is constant. If both of phases have equal mean velocity, the equation reduces to those of homogeneous model.

### 2.6 The Homogeneous Model for pressure drop

#### 2.6.1 The approaching homogeneous model;

a. Equal vapor and liquid velocities

b. The attainment of thermodynamic equilibrium between the phases

c. The use of a suitably defined single-phase friction factor for two-phase flow

Accordingly, general equation of pressure gradient along \(z\) for homogeneous modeling is;

\[
-\left(\frac{dp}{dz}\right) = \frac{2 f T g^2 v_f}{D} \left[ 1 + x \left( \frac{v_{fg}}{v_f} \right) \right] + G^2 v_f \left( \frac{v_{fg}}{v_f} \right) \frac{dx}{dz} + \frac{g \sin \theta}{v_f} \left[ 1 + x \left( \frac{v_{fg}}{v_f} \right) \right]
\]

(2.23)
2.6.2 The Two-Phase Friction Factor

All the terms in eq. (2.47) are definable, except one \( f_{TP} \);

(a) \( f_{TP} \) with assumption all the fluid is liquid, an denote as \( f_{fo} \) as function of Reynolds number \((GD/\mu_f)\) and the pipe relative roughness \((\varepsilon/D)\). so;

\[
-\left(\frac{dp}{dz} F\right) = \frac{2f_{fo} G^2 \nu_f}{D} \left[ 1 + x \left( \frac{\nu_f}{\nu_f} \right) \right] = -\left(\frac{dp}{dz} F\right)_{fo} \left[ 1 + x \left( \frac{\nu_f}{\nu_f} \right) \right]
\]

(2.24)

Where \(-\left(\frac{dp}{dz} F\right)\) is frictional pressure gradient calculated from the Fanning equation for total flow (liquid plus vapor) assumed to flow as liquid, so

\[
-\left(\frac{dp}{dz} F\right) = \frac{2f_{fo} G^2 \nu_f}{D}
\]

(2.25)

(b) the viscosity using mean viscosity \( \mu \) of liquid and gas, where

\[x=0, \quad \mu = \mu_f \; \text{and} \; x=1, \quad \mu = \mu_g \]

(2.26)

and the \( \mu \) correlation by

McAdam, et.al. \[ \frac{1}{\mu} = \frac{x}{\mu_g} + \frac{1-x}{\mu_f} \] (2.27)

Cicchiti, et.al. \[ \mu = x\mu_g + (1-x)\mu_f \] (2.28)

Dukler et.al. \[ \mu = \frac{x\nu_g \mu_g + (1-x)\nu_f \mu_f}{x\nu_g + (1-x)\nu_f} \] (2.29)

Akers et al. \[ \mu_{tp} = \mu_f \] (2.30)

Owens \[ \mu_{tp} = \mu_f \] (2.31)

Beattie and Whalley \[ \mu_{tp} = \omega \mu_g + (1-\omega)(1+2.5\omega)\mu_f \] , \( \omega = \frac{x\nu_g}{\nu_f + x\nu_{fg}} \) (2.32)
Lin et al.  \[ \mu_p = \frac{\mu_f \mu_g}{\mu_g + x^{1,5}(\mu_f - \mu_g)} \] (2.34)

Assuming that the friction factor may be expressed in term of the Reynolds number by the

**Blasius equation**

\[ f_{TP} = 0.079 \left( \frac{GD}{\mu_{TP}} \right)^{1/4} = 0.079 \left( \frac{GD}{\mu} \right)^{1/4} \] (2.35)

For equation (2.51) the

\[-\left( \frac{dp}{dz} \right)_F = -\left( \frac{dp}{dz} \right)_f \left[ 1 + x \left( \frac{\nu_{fg}}{\nu_f} \right) ] 1 + x \left( \frac{\mu_{fg}}{\mu_g} \right) \right] \] (2.36)

**In general equation,**

\[-\left( \frac{dp}{dz} \right)_F = -\left( \frac{dp}{dz} \right)_f \phi_{fo}^2 \] (2.37)

\( \phi_{fo}^2 \), known as the **two-phase frictional multiplier**;

\[ \phi_{fo}^2 = \left[ 1 + x \left( \frac{\nu_{fg}}{\nu_f} \right) ] 1 + x \left( \frac{\mu_{fg}}{\mu_g} \right) \right]^{1/4} \] (2.38)

### 2.7 The Separated Flow Model

**Derivation of Model and Assumption;**

a. Each phase velocity is constant

b. The attainment of thermodynamic equilibrium between the phases
c. The use of empirical correlations or simplified concepts to relate \( \phi^2 \) and \( \alpha \) to an independent variable of the flow

So that;

\[-\left( \frac{dp}{dz} \right)_F = \frac{2f_{pg} G^2 \nu_f}{D} \phi_{fo}^2 + G^2 dx \left[ \frac{2x \nu_s}{\alpha} - \frac{2(1-x) \nu_j}{(1-\alpha)} \right] + \frac{d\alpha}{dx} \left[ \frac{(1-x)^2 \nu_f}{(1-\alpha)^2} - \frac{x^2 \nu_s}{\alpha^2} \right] + g \sin \theta \left[ \rho_f \alpha + \rho_j (1-\alpha) \right] \] (2.39)

There are many correlations of the Two-phase Multiplier \( \phi_{fo}^2 \) and void fraction, \( \alpha \);
2.7.1 The Lockhart-Martinelli correlation

- flow regime were defined on the basis of the behavior of the flow (viscous or turbulent) when the phases were considered to pass alone through the channel
- The liquid and gas phase pressure drop were considered equal irrespective of the detail of the particular flow pattern.

\[
\left( \frac{dp}{dz} \right)_F = \left( \frac{dp}{dz} \right)_f \phi_f^2 \quad \text{or} \quad \left( \frac{dp}{dz} \right)_G = \left( \frac{dp}{dz} \right)_g \phi_g^2
\]  

(2.40)

Empirical multiplier as function of \(X\)

\[
X^2 = \frac{\left( \frac{dp}{dz} F \right)_f}{\left( \frac{dp}{dz} F \right)_g}
\]  

(2.41)

Where;

\[
- \left( \frac{dp}{dz} F \right)_f = \frac{2f_f \nu_f G^2(1-x)^2}{D_h} \quad \text{;} \quad - \left( \frac{dp}{dz} F \right)_g = \frac{2f_g \nu_g G^2 x^2}{D_h}
\]  

(2.42)

\(f_f = 16 \text{Re}_k^{-1}\), for \(\text{Re}_k < 2000\)

\(f_f = 0.079 \text{Re}_k^{-0.25}\), for \(\text{Re}_k > 2001\)

\[
\phi_f^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \quad \text{or} \quad \phi_g^2 = 1 + CX + X^2
\]  

(2.43)

<table>
<thead>
<tr>
<th>Liquid</th>
<th>Gas</th>
<th>(C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulent</td>
<td>Turbulent</td>
<td>tt</td>
</tr>
<tr>
<td>Viscous</td>
<td>Turbulent</td>
<td>vt</td>
</tr>
<tr>
<td>Turbulent</td>
<td>Viscous</td>
<td>tv</td>
</tr>
<tr>
<td>Viscous</td>
<td>Viscous</td>
<td>vv</td>
</tr>
</tbody>
</table>


Kim et al. correlations are derived based on Lockhart-Martinelli correlation with some modification as follow;

\(f_k = 16 \text{Re}_k^{-1}\), for \(\text{Re}_k < 2000\)
For laminar flow in rectangular channel

\[ f_k = 0.079 \text{Re}_k^{-0.25}, \text{ for } 2000 \leq \text{Re}_k < 20,000 \]

\[ f_k = 0.046 \text{Re}_k^{-0.2}, \text{ for } \text{Re}_k > 20,000 \]

Where subscript \( k \) denotes \( f \) or \( g \) for liquid and vapor phases, respectively

\[ \text{Re}_f = \frac{G(1-x) D_h}{\mu_f}, \text{Re}_g = \frac{G x D_h}{\mu_g}, \text{Re}_{fo} = \frac{G D_h}{\mu_f}, \text{Suratman number } Su_{go} = \frac{\rho_g \sigma D_h}{\mu_g^2} \] (2.45)

<table>
<thead>
<tr>
<th>Liquid</th>
<th>Gas(Vapor)</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulent</td>
<td>Turbulent</td>
<td>0.39 \text{Re}<em>{fo}^{0.03} Su</em>{go}^{0.10} \left( \frac{\rho_f}{\rho_g} \right)^{0.35}</td>
</tr>
<tr>
<td>Turbulent</td>
<td>Laminar</td>
<td>8.7 \times 10^{-4} \text{Re}<em>{fo}^{0.17} Su</em>{go}^{0.50} \left( \frac{\rho_f}{\rho_g} \right)^{0.14}</td>
</tr>
<tr>
<td>Laminar</td>
<td>Turbulent</td>
<td>0.0015 \text{Re}<em>{fo}^{0.59} Su</em>{go}^{0.19} \left( \frac{\rho_f}{\rho_g} \right)^{0.36}</td>
</tr>
<tr>
<td>Laminar</td>
<td>Laminar</td>
<td>3.5 \times 10^{-5} \text{Re}<em>{fo}^{0.44} Su</em>{go}^{0.50} \left( \frac{\rho_f}{\rho_g} \right)^{0.48}</td>
</tr>
</tbody>
</table>

2.7.3 Other proper correlations

1. Friedel (D>4mm, air-water, air-oil, R-12)

\[ \left( \frac{dp}{dz} \right)_f = \left( \frac{dp}{dz} \right)_{fo} \phi_{fo}^2 \] (2.50)

\[ \phi_{fo}^2 = (1-x)^2 + x^2 \left( \frac{v_g}{v_f} \right) \left( \frac{f_{go}}{f_{fo}} \right) + 3.24x^{0.78}(1-x)^{0.224} \left( \frac{v_g}{v_f} \right)^{0.91} \left( \frac{\mu_g}{\mu_f} \right)^{0.19} \left( 1 - \frac{\mu_g}{\mu_f} \right)^{0.7} Fr_{ip}^{-0.045} We_{ip}^{-0.035} \] (2.51)

\[ Fr_{ip} = \frac{G^2}{gD_h \rho_H^{0.2}}, We_{ip} = \frac{G^2 D_h}{\sigma \rho_H}, \rho_H = \frac{1}{x \nu_g + (1-x) \nu_f}, \text{Re}_{go} = \frac{G D_h}{\mu_g} \] (2.52)

2. Muller-Steinhagen and Heck
(D=4-392 mm, air-water, water, hydrocarbon, refrigerant)

\[
\frac{dp}{dz}_F = \left[ \frac{dp}{dz}_{fo} \right] + 2 \left[ \frac{dp}{dz}_{go} \right] \left( 1 - x \right)^{1/3} + \left( \frac{dp}{dz} \right)_{go} x^3
\]  
(2.53)

3. Lee and Lee (Dh=0.78-6.67mm, Air-water)

\[
\frac{dp}{dz}_F = \left( \frac{dp}{dz} \right)_{fo} \phi_f^2
\]  
(2.54)

\[\phi_f^2 = 1 + \frac{C}{X} + \frac{1}{X^2}, \quad \psi = \frac{\mu_f j_f}{\sigma}, \quad \lambda = \frac{\mu_f^2}{\rho_f g D_h}\]  
(2.55)

\[C_w = 6.833 \times 10^{-8} \lambda^{-1.317} \psi^{0.719} \text{Re}_{go}^{0.557}, \quad C_v = 3.627 \text{Re}_{go}^{0.174}\]  
(2.56)

\[C_v = 6.185 \times 10^{-2} \text{Re}_{go}^{0.726}, \quad C_n = 0.048 \text{Re}_{go}^{0.451}\]  
(2.57)

4. Chen (D=1.02-9mm, adiabatic, air-water, R410A, Ammonia)

\[
\frac{dp}{dz}_F = \left( \frac{dp}{dz} \right)_{fo,Friedel} \Omega, \quad \text{Bo}^* = g \left( \rho_f - \rho_g \right) \left( D_h / 2 \right)^2 \sigma
\]  
(2.58)

For \(\text{Bo}^* < 2.5 \Rightarrow \Omega = \frac{0.0333 \text{Re}_{go}^{0.45}}{\text{Re}_g^{0.09} \left( 1 + 0.4 e^{-\text{Bo}^*} \right)}\]  
(2.59)

For \(\text{Bo}^* \geq 2.5 \Rightarrow \Omega = \frac{\text{We}_{go}^{0.2}}{(2.5 + 0.06 \text{Bo}^*)}\)  
(2.60)

5. Sun and Mishima (Dh=0.506-12mm, Air-water, Refrigerant, CO₂)

\[
\frac{dp}{dz}_F = \left( \frac{dp}{dz} \right) \phi_f^2, \quad \phi_f^2 = 1 + \frac{C}{X} + \frac{1}{X^2}, \quad C = 1.79 \left( \frac{\text{Re}_g}{\text{Re}_f} \right)^{0.4} \left( \frac{1 - x}{x} \right)^{0.5}
\]  
(2.61)

\[
\text{Re}_f = \frac{G(1 - x) D_h}{\mu_f}, \quad \text{Re}_g = \frac{G_x D_h}{\mu_g}
\]  
(2.62)

2.8 Void Fraction

1. Homogeneous Model

\[
\alpha = \frac{1}{1 + \left( \frac{1 - x}{x} \right) \frac{\rho_g}{\rho_f}}
\]  
(2.63)
2 Zivi Void fraction
\[ \alpha = \frac{1}{1 + \left( \frac{1 - x}{x} \right) \left( \frac{\rho_g}{\rho_l} \right)^{2/3}} \]  

(2.64)

3 Smith Void fraction
\[ \alpha = \left[ 1 + \frac{\rho_g}{\rho_l} \left( \frac{1 - x}{x} \right) \left( \frac{0.4 + 0.6}{x} \frac{\rho_l + 0.4 \frac{1 - x}{x}}{1 + 0.4 \frac{1 - x}{x}} \right) \right]^{-1} \]  

(2.65)

Homogenous model suitable for Bubbly and Disperse models

4. Local Void Fraction Using Drift Model
\[ \langle \varepsilon \rangle = \frac{x}{\rho G} \left[ C_o \left( \frac{x}{\rho G} + \frac{1 - x}{\rho L} \right) + \frac{U_{GU}}{\dot{m}} \right]^{-1} \]  

(2.66)

Valuable only if \( U_{GU} \geq 0.05 \langle U \rangle \)

At elevated pressure, Zuber (1967);

\[ C_o = 1.13 \] \quad With \( U_{GU} = 1.41 \left[ \frac{\sigma G (\rho_l - \rho_G)}{\rho_l^2} \right]^{1/4} \]  

(2.67)

Regardless flow regime.

This also can be implemented for bubbly flow, vertical up flow, with particular value of \( C_o \)

<table>
<thead>
<tr>
<th>Geometry</th>
<th>In-dim (mm)</th>
<th>( C_o )</th>
<th>( P_r ), Reduced pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube</td>
<td>&gt; 50</td>
<td>( C_o=1-0.5P_r )</td>
<td>Except for ( P_r&lt;0.5 ), where ( C_o=1.2 )</td>
</tr>
<tr>
<td>Tube</td>
<td>&lt; 50</td>
<td>( C_o=1.2 ) for ( P_r )</td>
<td></td>
</tr>
<tr>
<td>Tube</td>
<td>&lt; 50</td>
<td>( C_o=1.2-0.4(P_r-0.5) ) for ( P_r&gt;0.5 )</td>
<td></td>
</tr>
<tr>
<td>Rectangular</td>
<td></td>
<td>( C_o=1.4-0.4(P_r-0.5) )</td>
<td></td>
</tr>
</tbody>
</table>

For bubbly flow, vertical up flow, Wallis (1969);

\[ C_o = 1.0 ; \quad U_{GU} = 1.53 \left[ \frac{\sigma G (\rho_l - \rho_G)}{\rho_l^2} \right]^{1/4} \]  

(2.68)

\[ A_{mc} = \frac{1}{4} \pi d_{mc}^2 , \quad A_c = \frac{1}{4} \pi d_c^2 , \quad W_g = \rho_g Q_g , \quad W_f = \rho_f Q_f \]
2.9 Uniformity Distribution

Uniformity distribution in between two outlet channels is equated

\[ R_g = \frac{Q_{gcu} - Q_{ged}}{Q_{gcu} + Q_{ged}}, \text{ and } R_f = \frac{Q_{fcu} - Q_{fcd}}{Q_{fcu} + Q_{fcd}} \]  

(2.69)

\( R_g \) and \( R_f \) is dimensionless. If \( R_g > 0 \), gas phase tend to go to upper channel and vice versa. If \( R_g = 0 \), the gas is uniform. Similarly, for \( R_f \) is. If \( R_f > 0 \), liquid phase tend to go to upper channel and vice versa. If \( R_f = 0 \), the liquid is uniform.

2.10 Pressure Losses through Bifurcation Channel Distributor

Bifurcation channel distributor have inlet diameter 8 mm and 2 pair 5 mm outlet diameter, as shown in Fig. 2.6 (a).

\[ A1 = \frac{1}{2}\left(\frac{1}{4}\pi D^2_s\right) \]
\[ A2 = \frac{1}{2}\left(S_b + S_d\right)l \]
\[ A3 = \frac{1}{4}\pi D^2_c \]

(b)

Fig. 2.6 Bifurcation channel distributor; (a) 2d sketch and Simplified as Incline straight tube, (b) simplified as converging diverging nozzle and (c) 3d sketch

Because there are no correlations, related to bifurcation channel distributor. Three approach correlations are tested; area changes by Tapucu, 1989, straight incline pipe and Tee-junction.

Assumption:

Contraction and expansion cannot be separated in the case of short insert.

- Janssen & Kervinen (1964), assuming that the contraction losses are small compared to the expansion losses.

\[ \Delta p_{SI} = -\frac{G_1^2}{2\rho_L} \left( \frac{1}{\alpha^2 C^2} \right) \left[ \frac{\rho_L}{\rho_G} x^2 \bar{\alpha} \left( \frac{1}{\alpha_3} \right)^2 - \left( \frac{\sigma C}{\alpha_4} \right)^2 \right] + (1-x)^2 \left( 1 - \bar{\alpha} \left( \frac{1}{1-\alpha_3} \right)^2 - \left( \frac{\sigma C}{1-\alpha_4} \right)^2 \right) \]

\[ -2\sigma C \left[ \frac{\rho_L}{\rho_G} x^2 \left( \frac{1}{\alpha_3} - \frac{\sigma C}{\alpha_4} \right) + (1-x)^2 \left( \frac{1}{1-\alpha_3} - \frac{\sigma C}{1-\alpha_4} \right) \right] \]

(2.70)

Where; \( \bar{\alpha} = \frac{1}{2}(\alpha_3 + \alpha_4) \); \( \sigma = \frac{A_2}{A_1} = \frac{A_3}{A_4} \) and \( C = \frac{A_3}{A_2} \)

Assumed as constant void fraction

\[ \Delta p_{SI} = -\frac{1}{2} \frac{G_1^2}{\rho'} \left( \frac{1}{\sigma C} - 1 \right)^2 \]

(2.71)
• If use Momentum Energy Equation of Hewitt & Hall Taylor (1970) based on Jansen assumption

\[
\Delta p_{sl} = G_i^2 \left[ \frac{1}{\sigma C} \left( \frac{1}{\rho_3^e} - \frac{1}{\rho_4^e} \right) \right] - G_i^2 \rho_H \left[ \frac{1}{2(\sigma C)^2} \left( \frac{1}{\rho_3^e} - \frac{1}{\rho_4^e} \right) \right]
\]  

(2.72)

Where, \( \sigma = \frac{A_2}{A_1} \), and \( C = \frac{A_3}{A_2} \)

Assumed as constant void fraction

\[
\Delta p_{sl} = \frac{G_i^2}{\rho'} \left[ \frac{2}{\rho'} \left( \frac{1}{\sigma C} - 1 \right) - \frac{\rho_H}{\rho^*} \left[ \frac{1}{(\sigma C)^2} - 1 \right] \right]
\]  

(2.73)

Where \( \nu' \) momentum specific volume and is defined as

\[
\nu' = \frac{x^2}{\alpha \rho_g} + \frac{(1-x)^2}{(1-\alpha) \rho_L}
\]  

(2.74)

And \( \nu'^* = \frac{x^3}{\alpha^2 \rho_g^2} + \frac{(1-x)^3}{(1-\alpha)^2 \rho_L^2} \)

(2.75)

\[
\frac{1}{\rho_H} = \frac{x}{\rho_g} + \frac{(1-x)}{\rho_L}
\]  

(2.76)

\( \rho' = \sqrt[3]{\nu'} \), is the momentum density

Assuming a constant void fraction along the duct,

\[2.10.2 \text{ Pressure drop by Energy equation as T-Junction, Hwang et al. (1988)}\]

\[
\Delta p_{1-i,TP} = \frac{\rho_{Hom,i}}{2} \left[ \left( \frac{G_i}{\rho_{E,i}} \right)^2 - \left( \frac{G_i}{\rho_{E,1}} \right)^2 \right] + K_{1-i,TP} \frac{G_1^2}{2 \rho_L}
\]  

(2.77)

Where \( \rho_E \) is the energy density, defined as

\[
\rho_E = \left[ \frac{(1-x)^3}{\rho_L^2 (1-\alpha)} + \frac{x^3}{\rho_g^2 \alpha^2} \right]^{-\frac{1}{2}}
\]  

(2.78)

And \( K_{1-i,TP} \) is a two-phase pressure loss coefficient formulated as

\[
K_{1-i,TP} = \zeta K_{1-i,SP}
\]  

(2.79)
For Annular and churn flows, Rectangular Channel \( \zeta = 1.60 \left( \frac{\rho_L \rho_{Hom,i}}{\rho_{Hom,1}^2} \right)^{0.586} \) (2.80a)

For plug and bubbly flow, Rectangular Channel \( \zeta = 2.57 \left( \frac{\rho_L \rho_{Hom,i}}{\rho_{Hom,1}^2} \right)^{0.146} \) (2.80b)

For circular channel, \( \zeta = \left( \frac{\rho_L \rho_{Hom,i}}{\rho_{Hom,1}^2} \right) \) (2.81)

For \( \text{Re} \geq 5000 \), Rectangular channel

\[
k_{i-i} = 0.477 + 0.21 \left( \frac{W_i}{W_i} \right) + 0.744 \left( \frac{W_i}{W_i} \right)^2
\]

(2.82)

For low \( \text{Re} \), round channel;

\[
k_{i-i} = 1 - 0.8285 \left( \frac{W_i}{W_i} \right) + 0.6924 \left( \frac{W_i}{W_i} \right)^2
\]

(2.83)

.i:2,3: Upper channel, lower channel

\[
\rho_{Hom} = \left( \frac{x}{\rho_G} + \frac{(1-x)}{\rho_L} \right)^{-1}
\]

(2.84)

### 2.11 Modeling for Internal Flow

#### 2.11.1 Single Phase Flow Modeling

Summarized general conservative equation for single phase flow modeling is described on Table 2.1;

<table>
<thead>
<tr>
<th>Table 2.1 General conservative equation for fluid flow</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Continuity</strong></td>
</tr>
<tr>
<td><strong>x-momentum</strong></td>
</tr>
<tr>
<td><strong>y-momentum</strong></td>
</tr>
</tbody>
</table>
As presented on Table 2.1, all conservative equations can be derived general form as;

\[
\frac{\partial \rho \phi}{\partial t} + \text{div}(\rho \phi \vec{V}) = \text{div}(\Phi \text{ grad } \phi) + S_\phi
\]  

(2.90)

Where, \( \phi \) are fluid properties, correspond to; \( l, u, v, w, i \) or \( T \). The Eq. 3.39 is called as **Transport Equation**; which contain four terms: rate of change, convective, diffusive and source. Each term has specific characteristics of differential equations, promoting different way to solve. Accordingly, in the Computational Fluid Dynamics (CFD) each term shall be solved using different **solver**. Moreover, each **solver** contains several options of method in order to enhance accuracy and convergence as well as to accelerate speed of calculation.

This modeling, the solver for transient formulation used first order implicit, convective term used second order upwind, volume fraction used geo-reconstruct model, coupling pressure-velocity used SIMPLE scheme.
2.11.2 Generalizing Conservative Equation for Two-phase Flow

For two-phase flow, the conservative equation can be generalized as;

1. Continuity

The two-phase flow model is characterized by two independent velocity fields which specify the motions of each phase $q$. The most natural choice of velocity fields is obviously the mass-weighted mean phase velocities, $\hat{v}_q$

$$\alpha_q \frac{D_q \rho_q}{Dt} + \bar{\rho}_q \frac{D_q \alpha_q}{Dt} + \alpha_q \bar{\rho}_q \nabla \cdot \hat{v}_q = \Gamma_q.$$  \hspace{1cm} (2.91)

Where;

The first two term of Eq. 3.60 are obtained from partial derivation of phase volume fraction and average phase density.

$\Gamma_q$: The rate of production of $q^{th}$ phase mass from the phase changes at the
2. Momentum equation

In the two-fluid model formulation, the conservation of momentum is expressed by two momentum equations with the interfacial momentum transfer condition. As it was mentioned before, the appropriate field equations should be expressed by the center of mass or the barycentric velocity of each phase, $\dot{v}_q$. The scalar form of momentum equations are described below;

**x-component**

$$
\alpha_q \rho_q \left( \frac{\partial \dot{v}_{sq}}{\partial t} + \dot{v}_{sq} \frac{\partial \dot{v}_{sq}}{\partial x} + \dot{v}_{yz} \frac{\partial \dot{v}_{sq}}{\partial y} + \dot{v}_{zq} \frac{\partial \dot{v}_{sq}}{\partial z} \right) = -\alpha_q \frac{\partial \bar{p}_q}{\partial x} + \alpha_q \rho_q \ddot{g}_q + \left\{ \frac{\partial}{\partial x} \alpha_q \left( \bar{\tau}_{ixq} + \tau_{ixq} \right) \right\} 
$$

**y-component**

$$
\alpha_q \rho_q \left( \frac{\partial \dot{v}_{sq}}{\partial t} + \dot{v}_{sq} \frac{\partial \dot{v}_{sq}}{\partial y} + \dot{v}_{yz} \frac{\partial \dot{v}_{sq}}{\partial y} + \dot{v}_{zq} \frac{\partial \dot{v}_{sq}}{\partial z} \right) = -\alpha_q \frac{\partial \bar{p}_q}{\partial y} + \alpha_q \rho_q \ddot{g}_q + \left\{ \frac{\partial}{\partial y} \alpha_q \left( \bar{\tau}_{iyq} + \tau_{iyq} \right) \right\} 
$$

**z-component**

$$
\alpha_q \rho_q \left( \frac{\partial \dot{v}_{sq}}{\partial t} + \dot{v}_{sq} \frac{\partial \dot{v}_{sq}}{\partial z} + \dot{v}_{yz} \frac{\partial \dot{v}_{sq}}{\partial z} + \dot{v}_{zq} \frac{\partial \dot{v}_{sq}}{\partial z} \right) = -\alpha_q \frac{\partial \bar{p}_q}{\partial z} + \alpha_q \rho_q \ddot{g}_q + \left\{ \frac{\partial}{\partial z} \alpha_q \left( \bar{\tau}_{izq} + \tau_{izq} \right) \right\} 
$$
2.11.3 The VOF (Volume of Fluid) Model for Two-phase Flow

This work implements the VOF model, because it can model two or more immiscible fluids by solving a single set of momentum equations and tracking the volume fraction of each of the fluids throughout the domain.

The VOF formulation relies on the fact that two or more fluids (or phases) are not interpenetrating. For each additional phase that added to model, a variable is introduced: the volume fraction of the phase in the computational cell. In each control volume, the volume fractions of all phases sum to unity. The fields for all variables and properties are shared by the phases and represent volume-averaged values, as long as the volume fraction of each of the phases is known at each location. Thus the variables and properties in any given cell are either purely representative of one of the phases, or representative of a mixture of the phases, depending upon the volume fraction values. In other words, if the \( q^{th} \) fluid's volume fraction in the cell is denoted as \( \alpha_q \) then the following three conditions are possible:

- \( \alpha_q = 0 \); the cell is empty (of the \( q^{th} \) fluid)
- \( \alpha_q = 1 \); the cell is full (of the \( q^{th} \) fluid)
- \( 0 < \alpha_q < 1 \); the cell contains the interface between the \( q^{th} \) fluid and one or more other fluids

Based on the local value of \( \alpha_q \), the appropriate properties and variables will be assigned to each control volume within the domain, such as:

\[
\rho = \alpha_2 \rho_2 + (1-\alpha_2) \rho_1
\]  

(2.95)
2.11.5 Volume Fraction Equation

The tracking of the interface(s) between the phases is accomplished by the solution of a continuity equation for the volume fraction of one (or more) of the phases. For the \( q \)\(^{th} \) phase, this equation has the following form:

\[
\frac{1}{\rho_q} \left[ \frac{\partial}{\partial t} \left( \alpha_q \rho_q \right) + \nabla \cdot \left( \alpha_q \rho_q \vec{v}_q \right) \right] = S \alpha_q + \sum_{p=1}^{n} (\dot{m}_{pq} - \dot{m}_{qp}) \tag{2.96}
\]

where \( \dot{m}_{qp} \) is the mass transfer from phase \( q \) to phase \( p \) and \( \dot{m}_{pq} \) is the mass transfer from phase \( p \) to phase \( q \). By default, the source term on the right-hand side of Eq. 3.64, \( S \alpha_q \), is zero, but can be specified a constant or user-defined mass source for each phase, by using UDF menu.

The volume fraction equation will not be solved for the primary phase; the primary-phase volume fraction will be computed based on the following constraint:

\[
\sum_{q=1}^{n} \alpha_q = 1 \tag{2.97}
\]

The volume fraction equation may be solved either through implicit or explicit time discretization.

The Implicit Scheme

When the implicit scheme is used for time discretization, \textsc{Fluent}'s standard finite-difference interpolation schemes, \textsc{Quick}, \textsc{Second Order Upwind} and \textsc{First Order Upwind}, and the \textsc{Modified HRIC} schemes, are used to obtain the face fluxes for all cells, including those near the interface.

\[
\frac{\alpha_q^{n+1} \rho_q^{n+1} - \alpha_q^n \rho_q^n}{\Delta t} V + \sum_f \left( \rho_q^{n+1} U_f^{n+1} \alpha_{q,f}^{n+1} \right) = \left[ S \alpha_q + \sum_{p=1}^{n} (\dot{m}_{pq} - \dot{m}_{qp}) \right] V \tag{2.98}
\]

Since this equation requires the volume fraction values at the current time step (rather than at the previous step, as for the explicit scheme), a standard scalar transport equation is solved iteratively for each of the secondary-phase volume fractions at each time step.

The implicit scheme can be used for both time-dependent and steady-state calculations.
The Explicit Scheme

In the explicit approach, FLUENT’s standard finite-difference interpolation schemes are applied to the volume fraction values that were computed at the previous time step.

\[
\frac{\alpha_{q}^{n+1} \rho_{q}^{n+1} - \alpha_{q}^{n} \rho_{q}^{n}}{\Delta t} V + \sum_{f} \left( \rho_{q} U_{f}^{n} \alpha_{q,f}^{n} \right) = \left[ \sum_{p=1}^{n} (\dot{m}_{pq} - \dot{m}_{qp}) + S_{\alpha_{q}} \right] V
\]  

(2.99)

Where;

\( n+1 \) : index for new (current) time step
\( n \) : index for previous time step
\( \alpha_{q,f} \) : face value of the \( q^{th} \) volume fraction, computed from the first- or second-order upwind, QUICK, modified HRIC, or CICSAM scheme
\( V \) : volume of cell, infinitesimal control volume
\( U_{f} \) : volume flux through the face, based on normal velocity

This formulation does not require iterative solution of the transport equation during each time step, as is needed for the implicit scheme. When the explicit scheme is used, a time-dependent solution must be computed.

When the explicit scheme is used for time discretization, the face fluxes can be interpolated either using interface reconstruction or using a finite volume discretization scheme. The reconstruction based schemes available in FLUENT are Geo-Reconstruct and Donor-Acceptor. The discretization schemes available with explicit scheme for VOF are First Order Upwind, Second Order Upwind, CICSAM, Modified HRIC, and QUICK.
References


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Chapter 3
Two-phase Flow Distribution in Bifurcation Channel

Distributor plays a major role on uniformly of the fluid flow distribution through multiple channels, particularly in two phase flow. Uniformity and low pressure drop are the major issue which shall be solved. The other hand, two-phase flow distribution in the bifurcation channel distributor has still remained mal-uniformity problem and the causes have not clearly discovered yet. Therefore, the enhancement study is needed, absolutely.

The study is initiated by comparing the bifurcation channel distributor with curved round distributor models with three types of reducer namely; curve reducer, T-reducer and straight reducer, 2d modeling based by Fluent® software. Second is modeling and experimental study of the flow distribution in the bifurcation channel distributor, constructed by acrylics resembling merged triple pipe, 8 mm in diameter of inlet channel and two set 5 mm in diameter of each outlet channel, set horizontally sideways with three positions; horizontal, 45° inclined and vertical. Three flow patterns were fed i.e.; bubble, slug and stratified flow, observed via high speed video camera. Third is pressure loss study on sideways bifurcation channel distributor experimentally and the results are compared with several existing correlation. The pressure distribution was measured by series of U-tube water gauge manometer. Forth is study of length channel effect on bifurcation channel distributor by integrating flow patterns, phase distribution and pressure drop analysis. The experiment is extended by modeling vary three outlet channel lengths with length ratio l/dc; 3.2, 10 and 70. The results are validated by experimental data and compared by existing correlation, analytically.

It was revealed that the bifurcation channel distributor has tended to lack uniformity of phase distribution and slightly lower pressure drop than curved reducer distributor. While the curve round distributor models with T-reducer has exhibited better performance due to smaller pressure drop and optimum phase uniformity. The two-phase flow distribution tends to be mal-uniform and to transform to different flow pattern in outlet channels. These are promoted by different; outlet channel length, feeding two-phase flow pattern in inlet distributor and inclination angle. The changing of flow pattern is driven by fluctuating velocity in both upper and lower outlet channel. There is only limited correlation have good agreement. They are Friedel (1979) for all domain and Hwang correlation for bubble flow in junction distributor. Accordingly, new correlation shall be developed based on their correlation. It was shown that there is oscillation and dumping effect when the two-phase flow impacts to an object.

3.1 Introduction

The lack of two-phase flow uniformity on the distributor is a major problem especially in the multiple channels. If the mal-distribution occurred, the performance of heat exchangers and other industrial devices tend to decreases. In order to achieve even distribution, several works have been done and developed since constructal theory for fluid flow was constructed by Bejan (1997), adopted from natural phenomenon. The fluid flow path between one point and a finite-size volume (an infinite number of points) is optimized by minimizing the overall flow resistance when the flow rate and the duct volume are fixed. The optimizing step begin from the smallest block which have array of small duct to bigger
block which have bigger ones and continue toward bigger block ones. Optimizing couldn’t be done inversely.

Based on constructal theory, such distributors design and model had been derived. The early was ramified fluid distributor by Tondeur et al. (2004). Lou et al. (2005) optimized multi-scale structures of distributors which dichotomic tree configuration. Lou et al. (2007) had coupled constructal distributors/collectors with a mini cross flow heat exchanger (MCHE) to solve the problem of flow mal-distribution. Fan et al. (2008) proved that blocking one or two outlets of one branch will generally decrease the uniformity of the flow distribution.

Miguel (2010) developed constructal equation of dendritic structures distributor for fluid flow both Laminar and turbulent. Vascular design of constructal structures with low flow resistance and non-uniformity was designed by Cho et al. (2010). Ramos-Alvarado et al. (2011) investigated constructal flow distributor as a bipolar plate for proton exchange membrane fuel cells. The uniformity in flow distributors having novel flow channel bifurcation structures has been studied by Liu et al. (2010).

Almost of all foregoing studies have not involved two-phase flow in particular, whereas is not a few systems involve it, as in evaporator and condenser. The uniformity of phase distribution in them is absolutely urgent. If the phase distribution is uneven then the performance of it will decrease significantly, moreover in a T-junction bifurcation, where the gas phase tends to occupy the upper space while the liquid occupy lower channel space. T-junction bifurcation must be designed accurately to prevent phase separation and to minimize pressure drop, it will be supported by this work which is still little researcher studying on it.

Al-Rawashdeh et al. (2012) had designed and tested the barrier-type distributor for two-phase flow. His work was intended to mix gas and liquid flows at the entrance of each of the parallel channels uniformly, before enter distributor. Accordingly, it was only be implemented in the chemical process reactor. In the evaporator and condenser, the phase was complicated to maintain and control. Therefore, it must be gained the breakthrough to maintain uniformly of phase throughout the channels downstream of distributor. There isn’t an easy matter since two-phase flow pattern vary uncertainly.
Aziz et al. (2012) have conducted a research by designing bifurcation channel distributor with lower-upper parallel row configuration of channels bifurcation tested horizontally, incline and vertically. The experiments were done by various gas and liquid superficial velocity. They concluded that the uniformity phase distribution is increase parallel to increasing superficial velocity of gas and or liquid. Likely, at high superficial velocity the phase pattern develops to annular flow in which easily divided evenly. There is the remaining problem which has not been solved yet mal-uniformity on low superficial velocity on both gas and liquid phase. Here is very important to develop other forms of distributors to overcome the problems.

The other hand, in order to achieve the best performance of ordinary heat exchanger and chemical reactor, the uniformity of two-phase flow distribution through pipes are absolutely required. The two-phase distribution in the manifolds of compact heat exchangers with parallel flow technology has great importance for optimizing design Wen et al. (2008). In heat exchanger, mal-uniform distribution significantly reduces the thermal and hydraulic performance He et al. (2011). Mal-uniform distribution of flow in parallel evaporating channels promotes channel dry-out and the subsequent hot spots and possible device failure Zhang et al. (2011). In an extreme case of steam generator system, the mal-distribution is sufficient to produce thermally induced failures as temperature-dependent material strength properties are exceeded or excessive thermal stresses are created Gandhi et al. (2012).

Several mal-uniformity of two phase flow studies had been done. Chin et al. (2011) had derived mathematical modeling based on Taylor series expansion to investigate causes of mal-uniformity and its effect on heat transfer and hydraulic performance. He recommended that to reduce mal-uniformity the flow distribution profile should be directed to modify. Marchitto et al. (2012) mentioned that to obtain optimum distribution, a proper consideration must be given to flow behavior in the distributor, flow conditions of upstream and downstream of the distributor, and the distribution requirements of the equipment. Both of these analyses were based on specific experimental setup and the detail analysis has not done. However, their results were very important to promote new investigation.

Aziz et al. (2012) have conducted a research by a bifurcation channel distributor with lower-upper parallel row configuration where it was tested horizontally, incline and
vertically. The experiments were done by various gas and liquid superficial velocity. They concluded that the uniform phase distribution is increasing parallel to increase of superficial velocity of gas and liquid. For example, at high gas superficial velocity the flow pattern develops to annular flow in which gas and liquid is easily divided evenly. It was very important to investigate the two-phase flow behaviour in detail in by integrating three approaches; experimental, modelling and analytical correlation.

Remarks, the bifurcation channel distributor is also as a fundamentals form for almost all distributors. Discussion about two-phase flow distributions and their phenomenon have been done. Another important parameter is pressure drop. There are two general correlation approach related to pressure drop model; homogeneous and separated model, which must be supported by parameters; void fraction, absolute viscosity and or two-phase flow multiplier $\phi$. Each supporting parameter has several correlations. Each correlation has limited proportionality for certain case only. That are depend on; fluid combination, flow pattern and channel geometry as well as its orientation. Consequently, there is plenty amount of correlation for supporting parameter, due to huge amount of two-phase flow case. There is no exception for this case, bifurcation channel distributor.

For bifurcation channel distributor, particularly set sideways, the correlation for pressure drop has not been established yet. Therefore, this work evaluates some correlation which is appropriate to it. The selected correlations are compared with experimental data. The best agreement is chosen as frame to develop new correlation for bifurcation channel distributor. This also can be used to review establish correlation, in order to complete understanding two-phase flow distribution phenomenon.

For homogeneous model, the some correlations of two-phase viscosity were tried. These are Adam (1942), Akers (1958), Chicchitti (1960), Owens (1961), Dukler (1964), Beattie and Whalley (1982) as well as Lin (1991). For separated model, two-phase multiplier correlations, which were tested, are Lockhart and Martinelli (1949), Friedel (1979), Muller-Steinhagen and Heck (1986), Chen et al. (2001), Lee and Lee (2001), Sun and Mishima (2009), Kim and Mudawar (2012). Appropriate correlations were supported by Smith (1969) correlation for void fraction. All correlations are addressed to round tube for main channel and both distributor channels.
Different from channels in which the void fraction is constant, in the distributor junction, the phase is devised to two channels, upper and lower in different volume fraction. Consequently, there is momentum pressure loss, in junction due to, devising volume fraction. On the other hand, the distributor junction has not had specific correlation for pressure drop. Therefore, the forgoing correlation is extended to determine pressure drop in junction, using equivalent length factor as correction factor. Also, the other equivalent correlations were tested, they are; T-junction equation by Hwang et al. (1988) and converging-diverging nozzle by Tapuccu et al. (1989) as well as momentum energy equation of Hewitt & Hall Taylor (1970) based on Jansen assumption. Regarding to the all forgoing correlations was discussed briefly in chapter 2.

The objects of initial study are to investigate mal-uniformity of phase distribution in bifurcation channel distributor, to design new distributors enhancing uniformity and to analyse their performances as well as to find the best design.

Second work investigated mal-uniformity of two phase flow along distributor through 3D numerical modeling by CFD software and was validated by experimental data. It was revealed that the mal-uniformity is driven by two phase flow pattern in inlet distributor, channel length and inclination angle. The numerical analysis covers phase distribution along distributor, static pressure distribution and velocity distribution of test section model especially in the junction.

Third work investigated mal-uniformity of two phase flow along distributor through 3D numerical modeling by CFD software, validated by experimental data and compared by existing correlation, analytically. The analyses cover phase distribution along distributor, pressure and magnitude velocity distribution as well as pressure drop of test section. It was revealed that the two-phase flow distribution tends to be mal-uniform and to transform to different flow pattern in both of outlet channels, downstream distributor junction.

Forth work, all correlations for pressure drop shall be compared to experimental data. Some new evidence related to two-phase impingement and separation is revealed. The better agreement of correlation to experiment is obtained, used to construct new correlation.
3.2. Uniformity of Two-phase Flow Distribution in Curved Round Distributor Design and Analysis

3.2.1 Design

Three types of distributor are developed in this study, namely curved round distributor with; curve distributor reducer, T-reducer and straight reducer, Figure 3.1. The development was inspired by the extending research of multiple structural bifurcation channel based on constructal theory, Bejan (1997) which solely leads to organizing structural of bifurcation. There are few researchers who focus on the shape of branching points which greatly affects the mal-uniformity of phase in the downstream of the distributor.

![Types of distributor](image1)

(a) Bifurcation channel  (b) Curved reducer  (c) T-reducer  (d) Straight reducer

Fig. 3.1 Types of distributor

![Two dimensional bifurcation channel distributor](image2)

Fig. 3.2 Two dimensional bifurcation channel distributor

The study was begun from analyzing mal-uniformity of phase distribution along bifurcation channel distributor, in which it was simplifed to two dimensional on centre plane devising distributor two part symmetrically, Figure 3.2. The trend of flow characteristic on the main channel cap was similar to characteristic of flow over rectangular aerofoil, Mannini et al. (2010), as shown on Figure 3.3. Despite the both of Figure 3.2 and 3.3 are different flow condition, in which the first is internal flow and the others are external flow, but the stream line is tend to similar. The similarity assumption has been done on analyzing compressor blade aerodynamically, Gresh (2001). As sown on Figure
3.2, that the stream line change from symmetrically into asymmetrically by a little increasing rake angle $\alpha$ (equal to increasing angle of entrance velocity vector).

It can be deduced that the stream line around symmetrical body is strongly influenced by entrance velocity angle. Similarly, if the velocity vector angle reaching main channel cap surface is fully parallel, the flow will be divided to upper and lower channel equally. Unfortunately, it is easy for single phase flow not for two-phase flow. Both of fluids in two-phase flow have different density and velocity as well as phase distribution. The velocity vector of each fluid varies along channel. However, division of phase around lower and upper channel tend to uniform in annular and disperse flow pattern since the phase distribution spread quit evenly along channel and the stagnation area on the cap surface promotes to rearranging phase uniformity. Even though uniformity could be enhance by increasing stagnation area, the sharp edge of cap surface promotes stream line separation in which spoil phase uniformity. Accordingly, the phase uniformity shall be enhanced by appending stagnation area and curvature shape along distributor aerodynamically as aerofoil profile.

![Stream line on $\alpha=0^\circ$](image1) ![Stream line on $\alpha=4^\circ$](image2)

Fig. 3.3 Flow around rectangular aerofoil, Mannini et al. (2010)

The last three of designed distributors was varied by large, narrow and moderate of stagnation area respectively. The study was performed through CFD modelling by using Fluent® software, the 2d approach for iterating efficiency and high superficial velocity as well as bifurcation channel distributor as base line. The performance parameters which have been investigated were distribution of; phase, velocity and pressure through distributor. The boundary conditions are; superficial velocity of water inlet and air inlet are $U_L= 1m/sec$, $U_G= 10 m/sec$ respectively and both pressure outlet, $P_O= 100 Pa$. 
3.2.2 Analysis

All distributors are mostly occupied by air, Figure 3.4, while the water occupy in near wall area, because phase supply to distributor is mostly vapour. The curve reducer distributor type exhibit the highest uniformity followed by T-reducer, straight reducer and bifurcation channel respectively. The uniformity level is characterized by maintaining of annular flow pattern with thin water layer around wall, the pipe circumference. This underlines that increasing stagnation area drives phase rearranging further preserve annular flow pattern and uniformity distribution. Unfortunately, almost of all of fluid particle passing through stagnation area is decelerated reducing the outlet phase velocity. Consequently, the most of fluid particle on curved reducer distributor convert their kinetic energy to pressure energy. This produces problem while the distributor connected to channels of heat exchanger or others.

Conversely, the minimum stagnation area keeps phase velocity on high level so as the highest phase velocity is on T-reducer distributor followed by straight reducer, bifurcation channel and curved reducer, respectively, Figure 3.5. The converted kinetic energy to pressure energy is proportional to velocity difference along distributor as well as channel. While T-reducer distributor performance tends to have better performance, but if the model is expanded to the three dimensional analysis, the performance shall drop due to significantly additional stagnation area.

The attractive velocity distribution is depicted on bifurcation channel distributor, in which the phase velocity upper channel is higher than lower channel and the stream line separated on both upper and lower edge of main channel cap. Perhaps, the distinction of phase velocity is caused by buoyancy effect and sharp edge cap resulting separation. The separation promotes turbulence and vortex. Furthermore, the uniformity of flow distribution is fall off. Since the analysis is based on laminar flow approach, the effect of turbulence was not appearance. Instead of turbulence, the oscillating gas flow through channel was exhibited. Accordingly, this shall be appended viscous modelling on CFD parameter. It is inferred that the curvature shape distributor could shift separation back, as shown on velocity distribution of curved round distributor which almost no phase oscillates within channels. It is evidence that the curved round distributor designed aerodynamically as aerofoil contributes to increase uniformity of two flow distribution along distributor.
Furthermore, the static pressure distribution along junction, Figure 3.6, approves forgoing analysis. The curved reducer distributor has the highest average static pressure followed by bifurcation channel, straight reducer and T-reducer, respectively. In the curved reducer distributor the most of kinetic energy is converted to pressure energy. The fluid
particle/phase relaxes to uniform indicated by uniform pressure distribution and pumped to channels by its pressure energy. The process generates high pressure drop, destroying its performance. The bifurcation channel distributor has static pressure slightly lower than curve reducer distributor. The uniformity pressure distribution is disturbed by sharp edge of main channel cap raising the air to upper channel and oscillating it as well as the water come down to lower channel promoting static pressure jump to high level at 4 mm under centre line. The straight reducer distributor indicates low static pressure, in which kinetic energy loss is reduced significantly and the phase uniformity is better than the second one. There is negative static pressure in both lower and upper side of junction, in which the fluid is accelerated from stagnation condition to channel. In this case, it has to be carefully to design straight reducer, if the slope of reducer is big enough \((m>0.2)\), the pressure loss could drop dramatically. The best performance is exhibited by T-reducer distributor in which the phase distribution is more uniform than others due to the uniformity of phase distribution, lower energy kinetic loss and lower pressure drop. However, the design must be enhanced the owing to existing static pressure jump as well as the large stagnation area while expanded to 3d model.

Fig. 3.6 Pressure distribution through junction.
3.2.3 Summary

The curved reducer distributor has the best uniformity of phase distribution, conversely has the highest pressure drop. While the bifurcation channel distributor has the lack-uniformity of phase distribution due to stream line separation on sharp edge of main channel cap, but has lower pressure drop than curve reducer distributor. The straight reducer and T-reducer exhibit the optimum condition but both of them have to be enhanced their design. The straight distributor must be reduced stagnation area and optimized reducer slope. The T-reducer shall be narrowed the stagnation area while developed to 3d design.

The design development should refer to aerofoil contour by using aerodynamics or hydrodynamics theory. The CFD modelling is recommended to append viscous effect to detail turbulence phenomenon. The best modelling have to expand to 3d modelling analysis for perfection design.

3.3 Mal-uniformity of Two-phase Flow through Bifurcation channel Distributor

3.3.1 Experimental Apparatus

To justify numerical investigation, the well experimental setup is an essential requirement, shown on Fig.3.7. Air is compressed by compressor unit 1 to tee-type two-phase flow generator 5 through shut off globe valve 2a, flow control valve 3a and flow measuring device 4a. Water is also fed to two-phase flow generator through similar devices. Air and water are mixed and supplied to distributor test section 6 via 400mm acrylic pipe. The two-phase flow pattern behaviors are recorded accurately by high speed video camera 8 and the pressure distribution along distributor are measured by series of U-tube water gauge manometer 7. The downstream channels of distributor are hooked to air-water separator 9 to separate each phase. Each quantity of air and water exiting the separator is measured by individual positive displacement flow meter, 10 and 11 respectively. The distributor inclination \( \alpha \) is varied to 0°, 45° and 90° from horizontal plane.

The test section of bifurcation channel distributor consists of an inlet channel with diameter of 8 mm, bifurcation channel shape of junction and a pair outlet channels, upper and lower with equal diameter of 5 mm, Fig. 3.8c. The main channel, including inlet
channel as inlet distributor has 400 mm tube, to ensure a fully developed flow, Fig. 3.8a. The test section is formed from a transparent rectangular block of acrylic resin, drilled and polished carefully to assure precision of channels dimension, Fig. 3.8a. The main channel is fed by the two phase flow generator supplied by varied specific quantity of water and air in own inlet, Fig. 3.8b. The flow pattern was observed clearly through transparent test section by using high speed video camera (Keyence Motion Analyzing VW-6000) at shutter speed 1/30.000 second, frame rate per second (fps) at 500 fps, and reproduction speed at 15 fps.

Fig. 3.7 Experimental setup for bifurcation channel distributor.

Fig. 3.8 Test section
The detail of pressure measurement is presented on following Fig. 3.9. The acrylic test section is drilled 8 holes 1mm in diameter at certain point as shown in Fig. 3.9. The 8 (number 1 to 8) holes are connected to 8 U-tube manometers, using acrylic hub. The 8 flexible tubes are hosted to hub and Pyrex tube, attached to scaled board U-tube manometer. Both are filled by water to measure the pressure static pressure each point. All U-tube manometers are varied at least in three different elevations (I, II and III), in order to protect the filled water release out from manometer at high static pressure and to avoid the bubble trap to the U-tube manometer that can disturb measurement. In U-tube manometer, the elevation of water $\Delta h$ is measured.

Fig. 3.9 Pressure measurement setup.
The static pressure is determined by hydrostatic equation 3.1

\[ p = \rho_{\text{water}} g \Delta h \]  

(3.1)

To achieve consistency, each measured point was repeated ten times and the measuring uncertainty was analyzed by using a Moffat (1988) method. The maximum error of pressure measurement is 6.27%, for bubble flow and slug flow, but for stratified flow the pressure measurement error is larger, due to low static pressure and closing to minimum measuring limit of instrument. The air and water flow rate is measured using positive displacement bath flow meter. The maximum errors of the air and water flow rate measurements are 0.393% and 0.106%, respectively.

The experimental pressure loss is investigated on five zones or region; distributor entrance, junction to upper and lower as well as upper and lower outlet channel. They are denoted as \( \Delta p_e \), \( \Delta p_{ju} \), \( \Delta p_{jl} \), \( \Delta p_{cu} \) and \( \Delta p_{cl} \), respectively, formulated as following equation;

\[ \Delta p_e = p_1 - p_2 \]  

(3.2)

\[ \Delta p_{ju} = p_2 - p_3 \]  

(3.3)

\[ \Delta p_{jl} = p_2 - p_6 \]  

(3.4)

\[ \Delta p_{cu1} = p_3 - p_4 \]  

(3.5)

\[ \Delta p_{cu2} = p_4 - p_5 \]  

(3.6)

\[ \Delta p_{cl1} = p_6 - p_7 \]  

(3.7)

\[ \Delta p_{cl2} = p_7 - p_8 \]  

(3.8)

### 3.3.2 Volume of Fluid Model

Based on work by Aziz et al. (2012), that in high superficial velocity for liquid or gas the flow distribution tends to uniform in bifurcation channel distributor. Further, this work is emphasized on low to moderate superficial velocity for both liquid and gas.
Accordingly, turbulence modeling is not necessary. Therefore, the VOF two-phase flow model and unsteady solver could be applied. The domain model contains 218,476 T-grid cells in which minimum volume of cell is 0.0258 mm$^3$ and the maximum one is 0.5 mm$^3$. The variables and properties of each cell are either representative of one phase or representative of the mixture of the two phases, depending on the volume fraction value Ahmad et al. (2007). The tracking of the interface between the two phases is accomplished by the solution of the continuity equation for the volume fraction of one of the phases.

$$\frac{\partial \alpha_q}{\partial t} + \mathbf{v} \cdot \nabla \alpha_q = 0$$  \hspace{1cm} (3.9)

A single momentum conservation equation is solved throughout the domain and the resulting velocity field is shared among the two phases. The momentum equation, shown below, is dependent on the volume fraction of all phases through the properties $\rho$ and $\mu$.

$$\frac{\partial}{\partial t} (\rho \mathbf{v}) + \nabla \cdot (\rho \mathbf{v} \mathbf{v}) = -\nabla P + \nabla \cdot \left[ \mu (\nabla \mathbf{v} + \nabla \mathbf{v}^T) \right] + \rho g + \mathbf{F}$$  \hspace{1cm} (3.10)

where $\rho$ is the average mass density,

$$\rho = \sum \alpha_q \rho_q$$  \hspace{1cm} (3.11)

and $\mu$ is the average dynamic viscosity

$$\mu = \sum \alpha_q \mu_q$$  \hspace{1cm} (3.12)

3.3.2.1 Two-phase flow generator

Based on continuity law, the total measured quantity (volume flow rate) of water by devices 10, $Q_{Lo}$, is equal to water quantity entering the two-phase flow generator, $Q_{Lin}$. Similarly, total measured quantity of air by devices 11, $Q_{Go}$, is equal to air quantity entering the two-phase flow generator, $Q_{Gin}$.

Since the two-phase flow generator shown in Fig. 3.8(b) was simplified to model Fig. 3.10. The superficial gas velocity of air, $v_{Gin}$ enters center pipe by inlet diameter, $d_{Gin}$=4mm, while the superficial liquid velocity of water, $v_{Lin}$ enters coaxial surface with
outer diameter, $d_m=8$ mm and inner diameter, $d_{Gin}$. The inlet velocity boundary condition of gas and liquid in each inlet, should be derived a set of following equations.

![Fig. 3.10 Simplified model of two phase flow generator](image)

\[ v_{Gin} = \frac{4Q_{Gin}}{\pi d_{Gin}} \]  \hspace{1cm} (3.13)

\[ v_{Lin} = \frac{4Q_{Lin}}{\pi (d_m^2 - d_{Gin}^2)} \]  \hspace{1cm} (3.14)

The distributor was varied to three different size models of channel length which are long, medium and short with aspect ratio, $l_c/d_c$; 3.2, 10 and 70, respectively. In which $l_c$ is channel length downstream of distributor junction and $d_c$ is channel diameter in constant 5 mm length.

### 3.3.2.2 Pressure drop

In two-phase flow, the total pressure drop is contributed by three sources. There are; the static pressure drop $\Delta p_{\text{static}}$ affected by different elevation, the momentum pressure drop $\Delta p_{\text{mom}}$ contributed by transformation of kinetic energy entire phase and the frictional pressure drop $\Delta p_{\text{fric}}$ due to presence friction between fluid and wall.

\[ \Delta p_{\text{tot}} = \Delta p_{\text{static}} + \Delta p_{\text{mom}} + \Delta p_{\text{fric}} \]  \hspace{1cm} (3.15)

This work investigated the total pressure drop of two phase flow through merge-pipe distributor in different flow pattern, channel length and distributor inclination, modeling based as well as validated by measured data on certain condition.
3.3.3 Result and Discussion

3.3.3.1 Phase Distribution

There are several two phase flow patterns, but here are emphasized on low superficial velocity generating three patterns namely; bubble flow, slug flow and stratified flow. In the other patterns, specially generated by high superficial velocity for gas and liquid, the uneven distribution is rare. The models were validated at appropriate point as shown in Fig. 3.11 and Fig. 3.12. Flow patterns of experiments and simulation results give good agreement.

Fig. 3.11 Various flow pattern in various channel length of horizontal distributor.
The unevenly distributed flow pattern in main channel drives mal-uniformity of phase distribution. If the phase is evenly distributed in upstream distributor, the phase uniformity will be resulted in downstream channel distributor. Accordingly, the mal-uniformity depends strongly on the flow pattern in inlet channel of the distributor. Consequently, as shown in Fig. 3.11, the bubble flow is worse uniformity class than slug flow.

There are interesting behaviors that the reducing channel length promotes increase of mal-uniformity and change of flow pattern. In the slug flow, the long channel (\(l_c/d_c=70\)) has more uniform flow distribution, followed by medium channel and short channel. The flow pattern tends to transform from slug flow to plug flow in upper channel and to bubble in lower channel. On the other hand, in Fig. 3.12, the stratified flow tends to transform to plug flow in lower channel of medium channel length (\(l_c/d_c=10\)) and bubble flow in lower of short channel (\(l_c/d_c=3.2\)), while the flow pattern in upper channel is kept in similar as main channel.

\[v_G=0.71\text{m/s}, \quad v_L=0.03\text{m/s}\]

Fig. 3.12 Stratified flow in various channel length of horizontal distributor.
In spite of the mal-uniformity shown in such two phase flow pattern, the uniformity shall enhance by changing distributor inclination. While the distributor reoriented from horizontal to vertical, the uniformity is improved, as shown in Fig. 3.13. Obviously, the uniformity is strongly influenced by distributor inclination, wherein horizontal configuration promotes more mal-uniformity than others.

3.3.3.2 Static Pressure Distribution

The static pressure distribution was measured experimentally by using water gauge of U-tube manometer. The eight measured points were taken, 2 point in the inlet zone and 3 pair points in channel zone of the distributor. The start of channel zone in which the flow is separated, was made as reference point axis (x=0). All point before reference point is negative x axis, otherwise is positive. The measured points is sited on; -28,-8,0,16 and 30 mm. the last of three point is a pair of point was installed on each pair of channel.

As shown in Fig. 3.14, there are differences on static pressure distribution between experiment and simulation result except in stratified flow having low static pressure.
distribution. In the inlet zone, the static pressure gradient is small for all flow patterns. Total pressure drop is mainly contributed by the junction zone and inlet part of the channel zone. The pressure is decrease significantly, when the flow is devised in junction zone. Three classes of pressure drop exist in this region; static pressure drop, momentum pressure drop and frictional pressure drop. These differs each other, depending on different elevation of channel, fed pattern of two phase flow, junction surface shape. The experimental pressure gradient is similar for the bubble flow. However, in the case of the slug flow, the pressure gradient of simulation is smaller than the experimental result.

![Pressure Distribution](image)

**Fig. 3.14** Static pressure distribution in various two phase flow pattern.

### 3.3.3.3 Effect of Channel Length

Fig. 8 shows instantaneous pressure distributions for each flow patterns with different channel length. As shown in Fig. 3.15, the pressure gradient tend to be small in slug flow and stratified flow, while in bubble flow tend to be big. The instantaneous velocity distributions of air, water or mixture at the center line of the channels exhibit different trend line in juction zone for each flow pattern as shown in Fig. 3.16. In bubble flow, the velocities increase significantly and similar for both upper and lower channel as well as
depend on channel length size. In slug flow and stratified, the velocity between upper and lower channel are different and it depends on channel length.

Fig. 3.15 Static pressure distribution in different channel length.
Fig. 3.16 Magnitude of velocity of mixture distribution in different channel length.
The behavior of the bubble flow, slug flow and stratified flow is different. In the bubble flow, the channel is mostly occupied by liquid whereas the stratified and slug flow by gas. In slug flow, there is fluctuating velocity. The fluctuating velocity in junction zone still remains in channel zone with different wave function, depended on the channel length and resulting change of flow pattern along channel. In long channel, velocity modulates in high wave amplitude only in the end of junction then followed by small amplitude along channel.

In the stratified flow, the entire channel is inhibited by gas. The effect of gravity is exhibited, contributing to velocity distribution between upper and lower distributor. The medium and short channel has similar velocity distribution, but the long channel tends to reduce velocity difference.

Fig. 3.17 shows the static pressure distribution of the bubble flow at three different distributor slope position: 0° (horizontal), 45° (inclined) and 90° (vertical). The pressure gradient in the distributor is quite similar for distributor position at horizontal, 45° inclined and vertical. The pressure distribution increase with the changing of distributor slope position from the horizontal to 45° inclined and increase again in a vertical position, this is because of gravity effect.

Fig. 3. 17 Static pressure distribution in different inclination.
3.3.4 Summary

The Numerical simulations of two-phase flow in a distributor were carried out and following conclusions are obtained though the simulation conditions are very limited and further discussion is required.

The mal-uniformity of two phase flow distribution in bifurcation channel distributor is originated by two-phase flow pattern entering distributor, channel length and the inclination. The bubble flow tends to give mal-uniform phase distribution and the bubble pattern does not transform along channel in various length channel. The slug flow has better phase distribution. The flow pattern changes in both lower and upper channel, driven by fluctuating velocity in both upper and lower channel. All of two phase flow pattern become uniform, when the distributor set in vertically.

3.4 Effect of Outlet Channel Length to Pressure Loss and Gain of Two-phase Flow Distribution in Bifurcation channel Distributor

3.4.1 Entrance Region of Distributor

As shown in Fig. 3.18, there are differences on pressure loss or gain resulted by experiment, modeling on CFD and analytical calculation based on Sun-Mishima correlation (2009). However, the modeling exhibits similar trend like experiment, moreover the investigation is focused on flow pattern transformation, not in interface region, so as the modeling is valid and acceptable. In this work, the experimental results are made as base line or reference because supported by small measurement error.

In bubble flow, modeling based, the pressure loss of distributor entrance of short \((l_{c}/d_{c}=3.2)\) and medium \((l_{c}/d_{c}=10)\) outlet channel is lower than experimental, while in the long outlet channel \((l_{c}/d_{c}=70)\) is higher. On the other hand, the analytical calculation based, it is higher than experimental too. In the downstream of distributor entrance region, there is obstacle, base of distributor junction. The two phase flow does not enable flow freely as in straight channel. The flowing phase impacts to the wall of junction base resulting increasing static pressure as pressure gain, than reduces pressure drop. Increasing pressure gain is generated by converting kinetic energy of phase to pressure energy, due to impacting
process at end of junction distributor. The phase flow velocity is decelerated, Fig. 3.16, reducing dynamic pressure energy and converting to static pressure energy.

\[
\begin{align*}
  j_G &= 0.063 \text{ m s}^{-1} \\
  j_L &= 0.8256 \text{ m s}^{-1}
\end{align*}
\]

Fig. 3.18 Pressure loss or gain in entrance region of distributor at different flow patterns.

Despite, the pressure loss of modeling based in entrance region of distributor will equal to analytical based at certain length of outlet channel, but the meaning is difference. In analytical based, using Sun-Mishima correlation, the correlation is built based on straight channel without any restriction or obstacle. Therefore, there is no pressure gain effect.
Meanwhile, in the entrance region of long outlet channel distributor, the higher pressure drop is resulted by multiple effects of pressure gain and loss. If the outlet channel length is added, in the bubble flow, the pressure gain in entrance region increases, reducing pressure loss. Thus, the pressure drop for short outlet channel distributor is higher than medium one. When the length outlet channel is again added, the pressure gain will increase and continue to reduce pressure drop up to minimum point at certain outlet channel length. The phase flow will stagnant, further the static pressure entering distributor (operational pressure) increase as depicted on Fig. 3.15, the new balance occurs with high pressure drop is resulted. Hence, the pressure drop of long outlet channel distributor is the highest. The ‘modulating pressure drop’ phenomenon may be happen in transient mode, before reaching steady state.

Also, if the outlet channel length is extended carefully, pressure loss of entrance region, modeling based, will equal to experimental based may be at channel length slightly higher than medium outlet channel, \( l_c/d_c = 10 \). Perhaps, in this condition, the flow becomes ‘quasi’ balance between upstream and downstream junction distributor. This occurs in different length of outlet channel depend on flow pattern as shown in Fig. 3.18 and each junction, Fig. 3.19 and 3.20. Though, in the quasi balance condition, the pressure drop of modeling based will equal to experimental based, but it cannot represent equality. Because, the modeling based disregards shear stress of interface between gas phase and liquid phase.

The analytical pressure drop tends to higher than experimental ones, because it assumes the channel is straight and is not any restriction or obstacle as well as ignored interface shear stress. Hence, here is no pressure gain. There have not been, proper correlation for this case and limited researches emphasizing on it. In order to develop new universal correlation of pressure drop in future. It is a good challenge to enhance knowledge through investigating phenomenon of impacting wall of two-phase internal flow by mean integrating between experimental and modeling based with considering interface shear stress.

The behavior of *slug flow* in entrance distributor is similar to bubble flow both of modeling based and correlation based, but in the bubble flow the ‘quasi balance’ region comes earlier, at length ratio \( l_c/d_c < 10 \). The slug flow is formed by increasing gas superficial velocity at the same liquid superficial velocity. The gas adds ‘obstacle’ flowing phase
through distributor junction. Consequently, the pressure gain in the distributor entrance region increases more. The characteristic of obstacle by gas is different from obstacle by length of outlet channel. The more quantity of gas, distributed around limited liquid, contributes gas oscillates in the junction and affecting to modulate magnitude velocity in the outlet channels at different amplitude of wave depend on outlet channel length of distributor as shown in Fig. 3.16. The oscillating velocity effect is subtle in the entrance region because damped by existing sufficient quantity of liquid. In the long outlet channel \( (l_c/d_c=70) \), the dumping effect by slug flow stronger than by bubble flow, indicated by big gap pressure loss between model and experimental. This fact matches with Beguin’s works (2009), because his experimental setup, the end of channel is filtered. Thus, it closes to long outlet channel mode. Accordingly, the existing pressure loss and gain promoted by existing oscillation and damping among phase is proven. The using combination analysis between experimental and modeling is valid.

Different from other flow pattern, the stratified flow has very low liquid quantity. In spite of low liquid quantity, but the effect of damping is bigger. Because, there is significant difference of superficial velocity between gas and liquid, in which liquid superficial velocity is lower than gas superficial velocity, so that the air is damped by water, resulting over pressure gain or negative pressure loss, as shown in Fig. 3.18. In the other word, the damping effect can be increased by increasing air superficial velocity and decreasing liquid superficial velocity. Indeed, this condition is unstable. Moreover it is in the low pressure, so that the flow through distributor looks like intermittent flow as well as the pressure measurements is disturbed resulting bigger errors.

3.4.2 Junction Region of Distributor

In junction region, the phases is devised to two parts; to upper outlet channel and to lower outlet channel with different behavior of velocity, pressure drop or gain as well as phase distribution. These are influenced by fed flow pattern, outlet channel length, Fig. 3.12 and 3.13, and inclination, Fig. 3.14.

The pressure drop or gain in junction to upper outlet channel of distributor is depicted on Fig. 3.19. In bubble flow, in short outlet channel, the magnitude velocity, Fig. 3.16, is bigger than others and either in lower cannels wherein almost free from the ‘obstacle’ of gas, is bigger than in upper outlet channel. In bubble flow, though it has more
liquid, but in this case, the superficial velocity of gas is limited. Therefore, the effect of damping also reduces to minimum. Instead of minimum damping effect, the pressure loss is increases significantly, closing to single phase liquid flow. Accordingly, both of upper and lower outlet channel have similar magnitude velocity, Fig. 3.16.

For the slug flow, although in upper and lower outlet channel is damped by liquid, but it is insufficient, due to similar superficial velocity in between gas and liquid. Still, the damping process is supported by outlet channel length. In the long outlet channel distributor, the magnitude velocity of junction to upper outlet channel is smaller than to lower outlet channel due to more ‘obstacle’ of gas (more damping effect) in upper outlet channel than others, Fig. 3.16. However, in the medium outlet channel, the magnitude velocity of junction to upper outlet channel is higher than to lower outlet channel. Therefore, the damping effect by outlet channel length is reduced significantly promoting unbalance damping by liquid between upper and lower outlet channel, wherein upper outlet channel is under damp and lower is over damp. Consequently, the different flow pattern is formed between upper and lower outlet channel.

In the junction of short outlet channel, the gas phase tend to flow freely to upper outlet channel due to small gravity effect, forming plug flow and the liquid flow to lower outlet channel owing to more gravity effect, forming bubble flow, Fig. 3.12, so that the damping effect is unbalance in between upper and lower outlet channel. The damping effect in upper outlet channel is bigger due to more ‘obstacle of gas’ than in lower outlet channel. Until the pressure loss in junction to upper outlet channel, Fig. 3.19 is smaller than it is in junction to lower outlet channel, Fig. 3.20. Consequently, in the upper outlet channel the magnitude velocity is smaller, followed by oscillating effect in high amplitude in upper outlet channel, due to the minimum dumping by outlet channel. But, in lower outlet channel, the magnitude velocity is high, while the oscillating effect in the lower outlet channel is disappeared, Fig. 3.16.

The stratified flow in the junction is different from it in the entrance region, in which impacting phase effect is dominant. While, in the junction the phase is devised and expanded through downstream of ‘orifice’, producing high pressure loss. Ultimately, the pressure gain is disappeared, covered by high pressure loss, Fig. 3.19-3.20. Although, the damping effect in junction is limited, but the damping effect in outlet channel is significant,
especially in lower outlet channel, Fig. 3.21-3.22. The damping effects, is meanly supported by outlet channel length, so that in the junction of long outlet channel distributor, the magnitude velocity is devised in little difference between lower and upper outlet channel, wherein to the upper outlet channel is higher than in to the lower outlet channel, due to additional liquid damping in lower outlet channel, Fig.3.16. The effect of damping by outlet channel length decreases along with reducing outlet channel length. Accordingly, in the junction of medium and short outlet channel, the magnitude velocity between upper and lower outlet channel devise in different significant value. This affects to which the part of gas phase in lower outlet channel is sucked back to upper outlet channel, intermittently. Thus, the flow pattern tends to transform to slug flow in lower outlet channel, not in upper outlet channel.

Fig. 3.19 Pressure loss or gain in junction to upper outlet channel of distributor at different flow patterns.
Obviously, in the junction the effect of oscillating and damping among phases exist in all flow patterns with different impact level. The highest impact level is in slug flow, followed by stratified flow and the lowest one is in bubble flow. Therefore, the modeling pressure drop of slug flow in the junction has the biggest gap with experimental results, while the bubble flow has the lowest gap with them, Fig. 3.19. The analytical pressure drop defined by correlation has slightly gap with experimental data. Because, the analytically pressure drop, the effect of oscillating and damping as well as the interface shear stress of phases are denied.

Fig. 3.20 Pressure loss or gain in junction to lower outlet channel of distributor at different flow patterns.
Similar to junction to upper outlet channel, Fig. 3.19, the junction to lower outlet channel is, as depicted on pressure drop distribution in Fig. 3.20. The pressure loss or gain of junction to lower outlet channel by the bubble flow tend to in similar level between modeling based and analytical based, but for experimental based exhibit higher than others. The effect of outlet channel length is not significant, influencing pressure loss, due to minimum damping effect resulted by combination low gas superficial velocity and more liquid superficial velocity. The higher pressure loss in experimental shall be affected by more irreversibility producing bigger two-phase pressure coefficient than formulated in equation (2.79). Hence, the two-phase flow coefficient for this bifurcation channel distributor shall be reconstructed.

### 3.4.3 Outlet channel Region of Distributor

Two section of outlet channel was investigated; first and second section. The first section begins from zero coordinate, outlet of junction, up to 16 mm distance of \( x \) coordinate and the second section starts from 16 mm up to 30 mm of \( x \) coordinate, Fig. 3.8c.

The pressure loss and gain of upper outlet channel is figured by Fig. 3.21. In the bubble flow, the pressure drop is increase by increasing upper outlet channel length and followed by increasing pressure gain due to upper outlet channel damping effects, proportional to upper outlet channel length too. Therefore, the pressure loss in short upper outlet channel is small, than increase in medium upper outlet channel first section followed by small pressure gain in second section and finally reduced pressure loss in the first and second section of long upper outlet channel of distributor.

In the slug flow, the pressure loss in short upper outlet channel is the highest due to limited liquid damper or minimum pressure gain. In medium, the effect of upper outlet channel damping is dominant in second section of medium upper outlet channel. Unfortunately, more additional upper outlet channel length, the damping effect by upper outlet channel length is weaker than increasing pressure loss by upper outlet channel length. This can be concluded that in slug flow, the damping effect by upper outlet channel length plays effectively in certain length of upper outlet channel.

In the stratified flow, though the pressure gain is the highest in the entrance region, due to impacting process, but in the junction the effect of damping is covered by the big
pressure loss. This condition is continued in the upper outlet channel, resulting minimum pressure gain and increasing pressure loss along with additional upper outlet channel length.

![Graphs showing pressure loss or gain in upper outlet channel of distributor at different flow patterns.](image)

Fig. 3.21 Pressure loss or gain in upper outlet channel of distributor at different flow patterns.

The performance of pressure loss or gain in lower outlet channel is shown in Fig. 3.22. In the bubble flow, the pressure drop is increase by increasing lower outlet channel length and followed by increasing pressure gain due to lower outlet channel damping
effects. Unfortunately, the increasing pressure gain is very limited, owing to limited bubble. Accordingly, the pressure loss in medium lower outlet channel is smaller than in short lower outlet channel and the pressure loss increase again in long lower outlet channel for both first and second section of lower outlet channel. Also, this can be concluded that in bubble flow, the damping effect by lower outlet channel length plays effectively in certain length of lower outlet channel.

Fig. 3.22 Pressure loss or gain in lower outlet channel of distributor at different flow patterns.
In the slug flow, in short lower outlet channel, instead of disappearing pressure loss is existing pressure gain. The pressure loss increases along with increasing lower outlet channel length, but pressure gain exhibits decrease in medium lower outlet channel and increase again in long lower outlet channel. This phenomenon is caused by fluctuating velocity as described above.

Now, the stratified flow behavior is different from others. In the experimental result shows high pressure gain for both first and second section of lower outlet channel. The gas phase is sucked back to upper outlet channel, due to significant different phase velocity between upper and lower outlet channel.

Similar as in upper outlet channel, the pressure loss and gain of lower outlet channel experimental based exhibit bigger than both modeling and analytical based. Except in bubble flow, wherein pressure loss modeling based is bigger than experimental results. These are caused by uncovered effect of oscillating and damping as well as interface shear stress of phases along outlet channel in both of modeling and analytical based. Perhaps, in slug and stratified flow their effect will be dominant within impacting process, but for bubble flow will be dominant in straight outlet channel.

Only in entrance region of distributor with long outlet channel, the slug flow and stratified flow have higher pressure gain than the bubble flow has. This indicated that the Beguin’s works is not valid for junction and outlet channel. Therefore, the Beguin’s correlation shall be revisited.

3.4.4 Summary

Numerical simulations of two-phase flow in a distributor were carried out. Although the simulation conditions are limited and further discussion is required, the experimental results give better support to obtain following conclusions;

1. The mal-uniformity of two phase flow distribution in bifurcation channel distributor is originated by two-phase flow pattern entering distributor, length of outlet channel and the inclination.

2. The bubble flow tends to give mal-uniform phase distribution and the bubble pattern does not transform along outlet channel in various length of outlet channel. All of two phase flow pattern become uniform in distribution, when the distributor is set vertically
3. The slug have better phase distribution and the flow pattern changes in both lower and upper outlet channel, while in stratified flow, it changes in lower outlet channel only.

4. The impacting two phase flow in the entrance region of bifurcation channel distributor promotes varying of pressure loss and pressure gain not only in entrance zone, but also in junctions as well as in the outlet channels.

5. The varying of pressure loss and pressure gain tend to what be caused by oscillating and damping process of air and water phase, promoting fluctuating velocity and pattern transformation in outlet channels.

6. The oscillating and dumping process among phases in entrance region has good agreement with Beguin’s works (2009), but junction and outlet channels region does not.
3.5 References


Chapter 4
Two-phase Flow Distribution in Plate Heat Exchanger

Introduce to heat exchanger including; process, classification and application. Further, describe more detail about Plate Heat Exchanger, PHE, particularly Brazed Plate Heat Exchanger

Although the improving design of plate heat exchanger has been done, the optimum design has not been achieved yet. This initial work was begun by comparing the performance of protrusion plate and corrugation ones, modeling based. The investigation had been undertaken for single phase laminar flow of water with constant surface temperature approach for entire plate. The boundary conditions are; water inlet velocity: 0.8 m/s on temperature 26°C with constant plate temperature 0°C. It is revealed that protrusion plate afford to increase heat flux 111% more than in corrugation plate and 24 % for dimple channel. The protruded channel contributes to reduce pressure drop per unit volume 49% less than in corrugation one, but the dimple channel is liable to increase it up to 204%. This is promoted by crash flow in between adjacent intersection of dimple compartments. It is recommended to improve the intersection of protrusions channel to reduce crash flow effect in dimple channel region or to combine dimple and protrusion in the same plate.

The other hand, the mal-uniformity phenomenon of plate heat exchanger has not disclosed clearly. The second work enhances the 3D modeling of corrugate channels of brazed plate heat exchanger completely, covering geometry in inlet and outlet distributor zone. The full acrylic corrugated channel is as test section used as validator, tested in two-phase flow mode in order to track the flow distribution clearly. It is revealed that the mal-uniformity is generated by geometry of inlet and outlet distributor zone for single phase flow and it tend to be promoted by flow pattern, buoyancy effect as well as geometry of inlet and outlet distributor zone for two-phase flow.

Meanwhile, the uniform flow distribution plays major role for enhancing plate heat exchanger performance. It is not easy to be achieved particularly for two phase flow. The third work investigated air-water two phase flow behavior through acrylic corrugated channels as mockup of plate heat exchanger channel. The effect of increasing quality on flow distribution and void fraction was investigated as well as channel orientation. The two-phase stratified flow is fed to the inlet tube and distributed to corrugated channel, set vertically upward and downward by quality variation from 0.1 to 0.6. The acrylic corrugate channel has chevron angle 30° and corrugation depth 1.5 mm. The void fraction is measured carefully by quick acting valve aid. The flow pattern and flow distribution through corrugated channels is observed using high speed video camera. It is disclosed that the increasing air quality of two phase flow promotes to increasing void fraction in certain trend line quite different from existing correlations, followed by increasing uniformity distribution along channels for both upward flow and downward flow. The upward flow tends to be more lack uniformity of phase distribution than downward flow.

Also, the void fraction is critical for determining pressure drop and heat transfer, and for the sizing of heat exchange equipment. Both have interrelationship each other and are not easy to be achieved. Neither, there has not been discovered such accurate meter, nor there has not been accomplished such accurate correlation for measuring or determining void fraction. This fourth work investigated void fraction and flow distribution in more detail by means CFD modeling and validated by experimental results. All boundary conditions are set similar to third work. The modeling shows that the increasing air quality of two phase flow promotes to increasing void fraction in certain trend line slightly different from experimental measuring and correlations. Meanwhile, the phase distributions tend to have good agreement between modeling and experiment. Upward flow tend to have worst uniformity than downward flow for all void fraction, due to different orientation of inertia force promoted by buoyancy effect on gas phase in entrance distributor zone. The gas phase tends to go directly to exit port in upward flow, vice versa for down flow.
4.1 **Introduction to Plate Heat Exchanger**

A **heat exchanger** is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. In heat exchangers, there are usually no external heat and work interactions. Typical applications involve heating or cooling of a fluid stream of concern and evaporation or condensation of single- or multicomponent fluid streams. In other applications, the objective may be to recover or reject heat, or sterilize, pasteurize, fractionate, distill, concentrate, crystallize, or control a process fluid. In a few heat exchangers, the fluids exchanging heat are in direct contact. In most heat exchangers, heat transfer between fluids takes place through a separating wall or into and out of a wall in a transient manner. In many heat exchangers, the fluids are separated by a heat transfer surface, and ideally they do not mix or leak.

### 4.1.1 Heat Exchanger Classification

According to construction, the heat exchanger is classified as following chart;

![Classification of heat exchangers](Shah, 1981)

Fig. 4.1 Classification of heat exchangers (Shah, 1981)

### 4.1.2 Plate-Type Heat Exchanger

Plate-type heat exchangers are usually built of thin plates (all prime surfaces). The plates are either smooth or have some form of corrugation, and they are either flat or wound in an exchanger. Generally, these exchangers cannot accommodate very high pressures, temperatures, or pressure and temperature differences. Plate heat exchangers (PHEs) can be classified as gasketed, welded (one or both fluid passages), or brazed, depending on the
leak tightness required. Other plate-type exchangers are spiral plate, lamella, and Plate coil exchangers.

4.1.2.1 Gasketed Plate Heat Exchangers

Fig. 4.2 Gasketed plate-and-frame heat exchanger.

*Basic Construction.* The plate-and-frame or gasketed plate heat exchanger (PHE) consists of a number of thin rectangular metal plates sealed around the edges by gaskets and held together in a frame as shown in Fig. 4.2. The frame usually has a fixed end cover (headpiece) fitted with connecting ports and a movable end cover (pressure plate, follower, or tailpiece). In the frame, the plates are suspended from an upper carrying bar and guided by a bottom carrying bar to ensure proper alignment. For this purpose, each plate is notched at the center of its top and bottom edges. The plate pack with fixed and movable end covers is clamped together by long bolts, thus compressing the gaskets and forming a seal. For later discussion, we designate the resulting length of the plate pack as $L_{\text{pack}}$. The carrying
bars are longer than the compressed stack, so that when the movable end cover is removed, plates may be slid along the support bars for inspection and cleaning.

Each plate is made by stamping or embossing a corrugated (or wavy) surface pattern on sheet metal. On one side of each plate, special grooves are provided along the periphery of the plate and around the ports for a gasket, as indicated by the dark lines in Fig. 4.3.

![Diagram of plate geometry and flow paths](image)

Fig. 4.3 Plates showing gaskets around the ports (Shah and Focke, 1988).

Typical plate geometries (corrugated patterns) are shown in Fig. 4.4, and over 60 different patterns have been developed worldwide. Alternate plates are assembled such that the corrugations on successive plates contact or cross each other to provide mechanical support to the plate pack through a large number of contact points. The resulting flow passages are narrow, highly interrupted, and tortuous, and enhance the heat transfer rate and decrease fouling resistance by increasing the shear stress, producing secondary flow, and increasing the level of turbulence. The corrugations also improve the rigidity of the plates and form the desired plate spacing. Plates are designated as hard or soft, depending on whether they generate a high or low intensity of turbulence.
Fig. 4.4 Plate patterns: (a) washboard; (b) zigzag; (c) chevron or herringbone; (d) protrusions and depressions; (e) washboard with secondary corrugations; (f) oblique washboard (Shah and Focke, 1988).

Sealing between the two fluids is accomplished by elastomeric molded gaskets [typically, 5 mm (0.2 in.) thick] that are fitted in peripheral grooves mentioned earlier (dark lines in Fig. 4.3). Gaskets are designed such that they compress about 25% of thickness in a bolted plate exchanger to provide a leaktight joint without distorting the thin plates. In the past, the gaskets were cemented in the grooves, but now, snap-on gaskets, which do not require cementing, are common. Some manufacturers offer special interlocking types to prevent gasket blowout at high pressure differences. Use of a double seal around the port sections, shown in Fig. 4.3, prevents fluid intermixing in the rare event of gasket failure. The interspace between the seals is also vented to the atmosphere to facilitate visual indication of leakage (Fig. 4.3).

Each plate has four corner ports. In pairs, they provide access to the flow passages on either side of the plate. When the plates are assembled, the corner ports line up to form distribution headers for the two fluids. Inlet and outlet nozzles for the fluids, provided in the end covers, line up with the ports in the plates (distribution headers) and are connected to external piping carrying the two fluids. A fluid enters at a corner of one end of the
compressed stack of plates through the inlet nozzle. It passes through alternate channels in either series or parallel passages. In one set of channels, the gasket does not surround the inlet port between two plates (see, e.g., Fig. 4.3a for the fluid 1 inlet port); fluid enters through that port, flows between plates, and exits through a port at the other end. On the same side of the plates, the other two ports are blocked by a gasket with a double seal, as shown in Fig. 4.3a, so that the other fluid (fluid 2 in Fig. 4.3a) cannot enter the plate on that side. In a 1 pass–1 pass two-fluid counterflow PHE, the next channel has gaskets covering the ports just opposite the preceding plate (see, e.g., Fig. 4.3b, in which now, fluid 2 can flow and fluid 1 cannot flow). Incidentally, each plate has gaskets on only one side, and they sit in grooves on the back side of the neighboring plate.

In Fig. 4.2, each fluid makes a single pass through the exchanger because of alternate gasketed and ungasketed ports in each corner opening. The most conventional flow arrangement is 1 pass–1 pass counterflow, with all inlet and outlet connections on the fixed end cover. By blocking flow through some ports with proper gasketing, either one or both fluids could have more than one pass. Also, more than one exchanger can be accommodated in a single frame. In cases with more than two simple 1-pass–1-pass heat exchangers, it is necessary to insert one or more intermediate headers or connector plates in the plate pack at appropriate places (see, e.g., Fig. 4.5). In milk pasteurization applications, there are as many as five exchangers or sections to heat, cool, and regenerate heat between raw milk and pasteurized milk.

![Three-fluid plate heat exchanger](image-url)
Flow Arrangements. A large number of flow arrangements are possible in a plate heat exchanger, Fig. 4.6, depending on the required heat transfer duty, available pressure drops, minimum and maximum velocities allowed, and the flow rate ratio of the two fluid streams. In each pass there can be an equal or unequal number of thermal plates. Whether the plate exchanger is a single- or multi-pass unit, whenever possible, the thermodynamically superior counterflow or overall counterflow arrangement

In the plate exchanger, the two outer plates serve as end plates and ideally do not participate in heat transfer between the fluids because of the large thermal resistance associated with thick end plates and air gaps between the end plates and the header/follower. The remaining plates, known as thermal plates, transfer heat between the fluids

One of the most common flow arrangements in a PHE is a 1-pass–1-pass U configuration, Fig. 4.6a. This is because this design allows all fluid ports to be located on the fixed end cover, permitting easy disassembly and cleaning/repair of a PHE without disconnecting any piping. In a multi-pass arrangement, the ports and fluid connections are located on both fixed and movable end covers. A multi-pass arrangement is generally used when the flow rates are considerably different or when one would like to use up the available pressure drop by multi-passing and hence getting a higher heat transfer coefficient
Fig. 4.6 Single-and multi-pass plate heat exchanger arrangements.

Looped or single-pass arrangement: (a) U arrangement, (b) Z arrangement
Multi-pass arrangement: (c) 2 pass-1pass, (d) 3pass-1pass, (e) 4pass-2pass and (f) series flow

4.1.2.2 Welded Plate Heat Exchangers

One of the limitations of the gasketed plate heat exchanger is the presence of gaskets, which restricts their use to compatible fluids (noncorrosive fluids) and which limits operating temperatures and pressures. To overcome this limitation, a number of welded plate heat exchanger designs have surfaced with welded pairs of plates on one or both fluid sides. To reduce the effective welding cost, the plate size for this exchanger is usually larger than that of the gasketed PHE. The disadvantage of such a design is the loss of disassembling flexibility on the fluid sides where the welding is done. Essentially, laser welding is done around the complete circumference, where the gasket is normally placed. Welding on both sides then results in higher limits on operating temperatures and pressures.
(350°C and 4.0MPa) and allows the use of corrosive fluids compatible with the plate material. Welded PHEs can accommodate multiple passes and more than two fluid streams. A Platular heat exchanger can accommodate four fluid streams. Figure 8.7 shows a pack of plates for a conventional plate-and-frame exchanger, but welded on one fluid side. Materials used for welded PHEs are stainless steel, Hastelloy, nickel-based alloys, and copper and titanium.

![Fig. 4.7 Section of welded plate heat exchanger](image)

4.1.2.3 Brazed plate heat exchanger

A vacuum brazed plate heat exchanger is a compact PHE for high-temperature and high-pressure duties, and it does not have gaskets, tightening bolts, frame, or carrying and guide bars. It consists simply of stainless steel plates and two end plates, all generally copper brazed, but nickel brazed for ammonia units. The plate size is generally limited to 0.3m². Such a unit can be mounted directly on piping without brackets and foundations. Since this exchanger cannot be opened, applications are limited to negligible fouling cases. The applications include water-cooled evaporators and condensers in the refrigeration industry, and process water heating and heat recovery.

The advantages of brazed plate heat exchanger are:

- Practically maintenance-free due to absence of gaskets
- Simple assembly with individual connection design
- Highest operational reliability thanks to comprehensive quality checks
- High resistance to pressure and temperature
- Compact design with low weight
4.2 References Study

The plate for plate heat exchanger has been enhanced several years. Now days, its application is more growing up, due to their compactness, enhanced heat transfer performance and flexibility in altering the size and obtaining different flow arrangements, Fernández-Seara et al. (2013). There are several researches regarding plate heat exchanger, grouped in following issues; flow configuration among plates, fluid substances effect on plates, plate heat exchanger in the systems, fouling factors, heat transfer and pressure drop.
The works about plate geometry of plate heat exchanger are limited, whereas several contours of plates were distributed widely.

The newly plate contours is the protrusion plate. The protrusion plate is the plate of plate heat exchanger made protrusion in certain configuration on its surface. By this way, the fluid passes through plate in surrounding of protrusion easier than through corrugate channels. Accordingly, the pressure drop entire protruded channel is reduced significantly. On the other hand the heat transfer rate over protruded plate can be enhanced. These results are claimed by a Danfoss® (2011) company.

Unfortunately the investigation of its performance has been carried out for protrusion channels only. Whereas the arrangement of protrusion plates in plate heat exchanger cannot form equal channel profile among plate layers. They have two different profiles channel; protrusion in odd layer and dimple in even layer. Their fluid flow behavior is different also. Accordingly the performances of both channels are unequal.

The initial study investigates the temperature, pressure and magnitude velocity distribution of protrusion channel and corrugation ones. For protrusion channel, both side of protrusion and dimple channel are observed. The performances of all are compared each other to recognize their excellences and weaknesses. Eventually the work is useful to direct the development of future plate contour for plate heat exchanger in order to attain better performance.

Meanwhile, the works studying the effect of protrusion surface to enhance heat transfer performance in different application have been done. These are; thermal performance of protruded rectangular fin, Leung et al. (1985), heat transfer with dimple/protrusion arrays in a rectangular duct, Hwang et al. (2008), split-dimple interrupted fin configuration for heat transfer augmentation, Elyyan (2009), periodically dimple-protrusion patterned walls for compact heat exchangers, Hwang et al. (2010), rotating channel with dimples and protrusions, Elyyan et al. (2010) and heat transfer in a cooling passage with protrusion-in-dimple surface, Kim et al. (2012). Although there are many works about protrusion effect for various implementations, the works particularly for plate heat exchanger are limited.
There are several works about plate heat exchanger including brazed type, experimentally and or modeling. Unfortunately, the mal-uniformity phenomenon has not disclosed clearly. The most of existing study was analyzed the macroscopic impacts of flow through plate heat exchanger channel via measuring pressure, temperature and flow at their inlet and outlet ports, Macin et al. (2013). Some works have investigated the flow distribution by using sight windows, but the sighting is limited on corrugate channel zone and is not in the inlet and outlet distributor zone, Nilpueng et al. (2010). Moreover the results are presented as 2D video file. Consequently, the detail information about flow distribution still has not adequate yet. The modeling study is an attractive option to reveal the mal-uniformity distribution phenomenon, and that had been done, Han et al. (2010). Again, the analysis tends to lack, due to complicated geometry in inlet and outlet distributor zone which has not been modeled properly.

The second work had enhanced the brazed plate heat exchanger model closed to test section made by acrylic, molded equal to real plate heat exchanger surface. The test section had been tested in two-phase flow mode in order to track the flow distribution clearly. The integration between complete modeling and experimental reveals cause of mal-uniformity distribution clearly.

The other hand, to enhance plate heat exchanger performance, the uniform flow distribution is absolutely required. For single phase flow, the uniformity flow is simpler to be investigated for both modeling and experimental works. Shaji et al. (2010) developed mathematical modeling for flow distribution, supplied by plate heat exchanger header to each channels, disregarding effect of each channel contour. Further, Han et al. (2010) detailed analysis to investigated flow distribution for each channel, CFD modeling and experimental based, but disregarding the inlet outlet distributor of each channel. Eventually, Gherasim et al. (2011) completed investigation on the flow and thermal fields CFD modelling and experimental based for single channel of plate heat exchanger including effect of inlet and outlet distributor.

For two phase flow, the flow distribution tends to be more complicated to investigate due to the complex in phase characteristics. The most studies emphasized on observing flow distribution by using diverse instruments such as; high speed video camera,
radiography, electrical impedance etc. Unfortunately, all of instruments still have had significant deviations, Winkler et al. (2014).

The third work expands to investigate two-phase flow distribution through corrugated channel of test section in various qualities at constant total mas flux 10 kg.s\(^{-1}\)m\(^{-2}\). The two-phase flow distributions are observed refer to each void fraction at certain air quality. The effects of increasing air quality on void fraction and phase distribution along corrugated channel as well as channel orientation are disclosed obviously.

The fourth works enhanced third work through investigating two-phase flow distribution and void fraction through corrugated channel of test section more detail through CFD Modeling. The boundary conditions are set to equal to forgoing works. The comparison between experimental and Modeling for phase distribution and void fraction were analyzed. The effects of increasing quality to void fraction and uniformity phase distribution along corrugated channel are disclosed clearly more than other method. Also, the pressure and local velocity distribution can be observed clearly.

### 4.3 Performance of Protrusion and Corrugation Plate for Plate Heat Exchanger

#### 4.3.1 Modeling of Plate Heat Exchanger Channel

As shown on Figure 4.9, the protrusion plate arrangement for plate heat exchanger form different channel passage; protruded channel in odd layer and dimple channel in even layer. The protrusion channel is the channel having flow constrains as double truncated cone attached each other on their apex, Figure 4.10(a). It consist 2mm protrusion elevation which equal to inter-plate distance \(b\), 6mm major protrusion diameter, 2 mm minor protrusion diameter as well as 45\(^{\circ}\) protrusion angle. Conversely, in the dimple channel, its double truncated cone is attached each other on their base, Figure 4.10(b). Exactly, the dimple size is same as protrusion one, because it take place on back side of protrusion.
Fig. 4.9 Protrusion plate heat exchanger.

Fig. 4.10 Protrusion channels in mm; (a) Protrusion channel, odd layer, (b) Dimple channel, even layer.
The corrugation plate arrangement of plate heat exchanger, Figure 4.11, is different from the protrusion plate arrangement. The plates always form equal channels passages for all of their channel layers. Figure 4.12 illustrates single channel model section of equal channels, having equal inter-plate distance \( b = 2 \text{mm} \) as in protrusion ones. The corrugation plate design is based on, Fernandes et al. (2000), in which has corrugation ratio \( \gamma = 0.474 \) and enlargement factor \( \phi = 1.17 \), formulated by following equation respectively.

\[
\gamma = \frac{2b}{P_x}
\]  \hspace{1cm} (4.1)

\[
\phi = \frac{1}{6} \left[ 1 + \frac{\pi}{2 \cos(\beta)} \right]^{0.5} + \left[ 1 + \frac{\pi}{2 \sqrt{2} \cos(\beta)} \right]^{0.5} \]  \hspace{1cm} (4.2)

Where \( b \) is inter-plate distance and \( P_x \) is the corrugation pitch in the main flow direction.

Fig. 4.11 Corrugation plate heat

Fig. 4.12 Corrugation channel.
For calculation efficiency, the investigation had not been carried out for complete plate heat exchanger, but it had been done for simple model section as depicted on Figure 4.10 for protrusion channels and Figure 4.12 for corrugation channel. The each model section of plate configurations forms heat transfer area per volume ($\sigma$) as brief in table 4.1;

<table>
<thead>
<tr>
<th>Channel</th>
<th>$A$ (m$^2$)</th>
<th>$V$ (m$^3$)</th>
<th>$\sigma$ (m$^2$.m$^3$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Protrusion, odd layer</td>
<td>0.000286</td>
<td>1.82x10$^{-7}$</td>
<td>1569</td>
</tr>
<tr>
<td>Dimple, even layer</td>
<td>0.000318</td>
<td>2.08x10$^{-7}$</td>
<td>1536</td>
</tr>
<tr>
<td>Corrugation</td>
<td>0.000639</td>
<td>4.94x10$^{-7}$</td>
<td>1294</td>
</tr>
</tbody>
</table>

The investigation of modeling had been done under certain boundary conditions. The working fluid is water, fed in inlet by constant velocity, $u_{wi}$: 0.8 m/s and constant temperature, $t_{wi}$: 26°C. The flow direction of inlet is towards $x$ negative of coordinate, as depicted on Figure 4.10 and Figure 4.12. The water flow out from model section on constant pressure outlet: 1000 Pa. The temperature of plates is $t_s$: 0°C constant for whole plate surfaces.

Although the heat transfer of odd layer and even layer of protrusion channel is dependent each other, the investigation was carried out independently. Because, the aim of this study is to compare the performance of plate shape of plate heat exchanger, not to exhibit the complete heat exchanger performance. The performances which investigated are temperature distribution and static pressure distribution as well as particular velocity vector to see detail fluid flow phenomenon. Those were emphasized on observing $y$ plane zone of center model section (projected plane of each couple plate).

### 4.3.2 Results and Discussion

The temperature distributions of water in center plane of channel between a pair of plates are depicted on Figure 4.13. The pair of protrusion plates derives outlet temperature of its channels, Figure 4.13(a), more uniform than that driven by the pair of corrugation plates, Figure 4.13(c).
Moreover, the protrusion plates reduces temperature more than the corrugation one. This indicates the protrusion plate has better heat transfer performance than the other one. In the

Fig.4.13 Temperature distribution of water in center plane of channels; (a) Odd layer protrusion, (b) Even layer protrusion, (c) Corrugation
dimple channel formed by another side of protrusion plate, even layer protrusion, Figure 4.13(b), apparently the performance of heat transfer is worse than in the corrugation one. But if the investigation is extended to the total surface, than the dimple channel still has heat transfer performance better than the corrugation channel has.

Table 4.2 show the total heat transfer released through plate of each model section. The total heat released through corrugation plate exhibit greater than others. But the total plate area of each protrusion and dimple is about half of corrugation one. Hence, the total heated released per total heat transfer plate area or heat flux of corrugation plate is worse than whose of both protrusion and dimple are. Eventually the protrusion channel increases heat flux more than two times of corrugation channel correspond to 111% and the dimple channel slightly increase heat flux, up to 24%.

Table 4.2 Total heat released over plates of models section

<table>
<thead>
<tr>
<th>Channel</th>
<th>Q (w)</th>
<th>q (kW.m⁻²)</th>
<th>Δq (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Protrusion, odd layer</td>
<td>62.2</td>
<td>217</td>
<td>111</td>
</tr>
<tr>
<td>Dimple, even layer</td>
<td>40.6</td>
<td>128</td>
<td>24</td>
</tr>
<tr>
<td>Corrugation</td>
<td>65.9</td>
<td>103</td>
<td></td>
</tr>
</tbody>
</table>

The evidences correspond to a resulted by Hwang et al. [3] in which the average Nusselt number of fluid flow over protruded plate is bigger than over dimpled plate. Also the Nusselt number of dimple plate is bigger than that of flat plate. The Nusselt number of fluid flow over corrugate plate would be lower than that, in between dimpled plate and flat plat. Consequently, the heat flux released from fluid flow to protrusion or dimple plates is much more than to corrugation plates.

Likewise, the pressure distributions of water in center plane of channel between pair of plates are depicted on Figure 4.14. The best pressure distribution is for protrusion channel followed by it is for dimple channel and the worst is for corrugation ones. Also, the pressure drop between inlet and inlet model section of protrusion channels less than others. While the dimpled channels exhibit the highest pressure drop. Hence, the total pressure drop of protruded plate heat exchanger is worse than corrugation ones.

The highest pressure drop is caused by what the water flow adjacent dimple compartments crash each other. The flow decelerates each other increasing significant
pressure gradient inter dimple compartment. This is promoted by what the flow vector adjacent compartment format obtuse angle, $\varphi$, Figure 4.15 (b). The problem crash flow can be resolved by modifying intersection inter protruded surfaces such that the flow vector adjacent dimple compartment format acute angle ($\varphi<90^\circ$).

Fig.4.14 Pressure distribution of water in center plane of channels;
(a) Odd layer protrusion, (b) Even layer protrusion, (c) Corrugation
These are supported by following facts. The total pressure drop per unit volume of model section computed as integral pressure distribution over total volume divided by volume of each model section and the results are as described on table 4.3.

Table 4.3 Pressure drop per unit volume of models section

<table>
<thead>
<tr>
<th>Channel</th>
<th>ΔP</th>
<th>ΔP/V</th>
<th>Δ(ΔP)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(Pa)</td>
<td>(Pa.m⁻³)</td>
<td>(%)</td>
</tr>
<tr>
<td>Protrusion, odd layer</td>
<td>3.56x10⁻⁴</td>
<td>1.95x10⁻⁴</td>
<td>-49</td>
</tr>
<tr>
<td>Dimple, even layer</td>
<td>2.40x10⁻³</td>
<td>1.16x10⁻⁴</td>
<td>204</td>
</tr>
<tr>
<td>Corrugation</td>
<td>1.88x10⁻³</td>
<td>3.81x10⁻³</td>
<td>0</td>
</tr>
</tbody>
</table>

As shown on table 4.3 that the protruded channel able to reduce pressure drop up to 49%. Unfortunately in the dimpled channel, back channel of protruded one, contribute to increase pressure drop significantly reach to 204%. So in the protrusion plate heat exchanger, the average increasing pressure drop per unit volume for both protrusion and dimple channel attain to 77%. Similarly, it increases average heat flux up to 67%. Ultimately the increasing heat flux in protrusion plate heat exchanger has to be compensated by increasing significant pressure drop. Therefore, the evaluation of intersection inter protruded surface is absolutely urgent.

The velocity distribution of water in the center plane of channels, Figure 4.15, for corrugated channels is the best ones, indicated by the best uniformity of flow whole channel, followed by dimple channel and protruded channel respectively. There is lowest velocity zone in protruded channel as well as dimpled channels, promoting decrease heat transfer performance. However, their heat transfer performances are exhibited better performance than in corrugation channel. Because, as disclosed by Hwang et al. (2008), the effect of existing dimple or protrusion for increasing Nusselt number is stronger than increasing Reynolds number.

Hence, the big problem of protrusion plate heat exchanger is whose the great pressure drop on dimple channel shall be resolved. The evaluation shall be taken by enhancement of protrusion or dimple configuration such that the flow vector adjacent dimple compartment format acute angle (φ<90°) or by the combination of protrusion and
dimple in the same plate. This combination has better heat transfer performance than only protrusion or dimple in the plate has, Elyyan et al. (2009). This combination not only shall enhance heat transfer performance significantly, but also shall form uniform channel for whole channel layer such as in corrugate channel.

Fig. 4.15 Velocity Vector distribution of water in center plane of channels; (a) Odd layer protrusion, (b) Even layer protrusion, (c) Corrugation
4.3.3 Summary

The forgoing discussion is drawn conclusions, they are;

- The temperature and pressure distribution in protruded channel is more uniform than in corrugated channel. But, in the dimple channel, in back of protrusion channel, tends to increase pressure drop significantly more than corrugated channel.
- The protrusion and dimple plate afford to increase heat flux more than in corrugation plate up to 111% and 24%, respectively.
- The protrusion channel contributes to reduce pressure drop per unit volume 49% less than in corrugation one. Unfortunately, the dimple channel is liable to increase pressure drop per unit volume up to 204% more than in corrugation channel.
- The increasing pressure drop is contributed by crash flow inter adjacent dimple compartment. Therefore, modifying intersection inter protruded surfaces such that the flow vector adjacent dimple compartment format acute angle ($\phi<90^\circ$) potentially reduce pressure drop per unit volume.
- The velocity distribution in the corrugation channel is more uniform followed by dimple channel and protrusion channel. The effect existing dimple and protrusion for increasing heat flux is stronger than increasing velocity. Accordingly, making the combination of protrusion and dimple in the same plate, forming uniform channel for whole channel layer such as in corrugate channel as well as potentially increase the heat transfer capacity and reduce pressure drop significantly.

4.4 Mal-uniformity Distribution of Fluid Flow through Corrugated Channel of Brazed Plate Heat Exchanger

4.4.1 Modeling and Experimental Setup

![Original plate of brazed plate heat exchanger](image)
Fig. 4.17 Molded acrylic test section and its detail drawing

Fig. 4.18 Full test section model

Fig. 4.19 Experimental setup
The original plate of brazed plate heat exchanger, Fig. 4.16, is derived as test section made from acrylic molded equally, Fig. 4.17. Also, it is modeled similarly, as Fig. 4.18. Fig. 4.19, air and water are fed to two-phase generator 5 to develop various flow patterns. Each flow rate is regulated by valve 3a and 3b as well as fixed by rotameter 4a and 4b. The generated flow pattern is fed to corrugated acrylic channel 6, as test section, observed flow distribution using high speed camera 10 and discharged to air water separator 7. The actual flow rate of each air and water is measured using bath flow meter 8 and 9 with maximum errors up to 0.393% and 0.106% respectively.

4.4.2 Result and discussion

The boundary conditions are; constant water inlet velocity: 0.2 m.sec\(^{-1}\), equivalent to total volume flow rate 2.265 l.min\(^{-1}\), at 26.85\(^\circ\)C and constant wall surface temperature 0\(^\circ\)C. The model was run on steady state mode resulting velocity distribution Fig. 4.20, temperature distribution Fig. 4.21 and pressure distribution Fig. 4.22. All figures are about center plane of corrugate channel of brazed plate heat exchanger model.

![Velocity distribution](image)

Fig. 4.20 Velocity distribution

Fig. 4.20, mal-uniformity is occurred, since the highest velocity is distributed along edge closed to inlet and outlet manifold, the moderate velocity is distributed along edge away from inlet and outlet manifold. The lowest velocity is distributed in center line of plate or in intersection of corrugated herringbone zone. The mal-uniformity tends to be promoted by geometry of inlet and outlet distributor zone, formed by attached of couple plates. Also, it is due to parallel inlet outlet configuration. Potentially, the cross inlet outlet
configuration as well as modifying their surface geometry shall enhance uniformity flow distribution.

Fig. 4.21 Temperature distribution

In single flow, though the velocity distribution ununiformed, the temperature distribution, Fig. 4.21 and pressure distribution, Fig. 4.22 are still closed to uniform. Meanwhile, the pressure drop tends to significantly high in inlet port, due to changing velocity vector orientation from horizontally inlet tube to vertically corrugated channel.

Fig. 4.22 Pressure distribution
The Fig. 4.23 is two-phase flow distribution through test section. The inlet tube boundary conditions are; the air superficial velocity $j_G$: 0.28 m.sec$^{-1}$ and the water superficial velocity $j_L$: 0.08 m.sec$^{-1}$. Equivalent with total air flow rate, $Q_G$: 0.789 l.min$^{-1}$ and total water. The flow rate, $Q_L$: 0.676. The flow direction is vertically downward and depicted as towards negative $x$ axis. It is revealed that the two phase flow distribution is corresponds with single phase velocity distribution, in which the air/bubble tend to distribute more unevenly along corrugate channel. The bubble along edge, away from inlet outlet port is more than it along edge, adjacent inlet outlet port. While, in the in center line, in the intersection of herringbone zone, the air/bubble is more limited. This mal-uniformity tends to be promoted by channel geometry, flow pattern and buoyancy effect

4.4.3 Summary

The geometry of inlet and outlet distributor zone of corrugated brazed plate heat exchanger primarily promotes mal-uniformity for single phase flow moreover for two-phase flow. More unevenly flow distribution of two phase flow is promoted by additional effect of flow pattern and buoyancy.

4.5 Air-Water Flow Behavior through Corrugated Channel of Plate Heat Exchanger

4.5.1 Experimental Setup

The test section of original plate heat exchanger, made from milled acrylic, is shown on Fig. 4.17. Fig. 4.24 is the experimental setup. Air and water are fed to two-phase generator 5 to develop various flow patterns. Each flow rate is regulated by valve 3a and 3b as well as fixed by rotameter 4a and 4b. The generated flow pattern is fed to corrugated
acrylic channel 6, as test section, observed flow distribution using high speed camera 10 and discharged to air water separator 7. The actual flow rate of each air and water is measured using bath flow meter 8 and 9 with maximum errors up to 0.393% and 0.106% respectively.

The two-phase stratified flow is fed to the inlet tube and distributed to corrugated channel in different volume flow rate correspond to total mass flux $10 \text{ kg.s}^{-1}\text{m}^{-2}$ with quality variation from 0.1 to 0.6. The experiment was carried out in two flowing modes; vertically upward and downward. The acrylic corrugate channel, test section, has chevron angle 30° and corrugation depth 1.5 mm. The void fraction is measured by digital scale 10 carefully, weighting channel occupied by stagnant air-water two phase flow trapped by quick acting valve in both of inlet and outlet. The experiment setup is also completed by recording flow pattern and flow distribution through corrugated channels using high speed video camera. The dynamic flow behavior can be observed clearly.

![Fig. 4.24 Experimental setup](image)

4.5.2 Results and Discussion

The flow distributions of air-water two-phase flow for specific air quality are depicted in Fig. 4.25 and Fig. 4.26. The uniformity gas phase distribution is increased by increasing gas quality for both downward and upward flow. However, the downward flow tend to more uniform in phase distribution than the upward flow. On the other hand,
increasing quality promotes increasing volume void fraction, as shown in Fig. 4.27, but the volume void fraction of downward flow tend to bigger than of upward flow. In same vapor quality, the bigger volume void fraction tends to attain the more uniformity. Consequently, the uniformity strongly depends on volume void fraction for all vapor quality.

The different in void fraction for upward from for downward flow, could be driven by different slip factor ($S$), promoted by different gap in superficial velocity between air/gas and water/liquid. In constant quality, upward flow tends to have bigger superficial velocity, supported by buoyancy effect than downward flow has, because dumped by buoyancy force of gas. Consequently, upward flow has bigger slip factor than downward flow has. Accordingly, the upward flow has smaller void fraction and less uniformity phase distribution. Here is proportional to the Fig. 4.27 that the void fraction of homogeneous correlation ($\alpha_H$), Eq. (4.1), Collier et al. (1994) is away from experimental data, due to disregarding slip factor assumption or both of liquid and gas have equal velocity.

Fig. 4.25 Two-phase distribution of downward flow at gas quality: (a) 0.1, (b) 0.2, (c) 0.4 and (d) 0.6.
Fig. 4.26 Two-phase distribution of upward flow at gas quality: (e) 0.1, (f) 0.2, (g) 0.4 and (h) 0.6.

Fig. 4.27 Experiment based and analytical correlation of void fraction.

\[
\alpha_H = \frac{1}{1 + \frac{(1-x) \rho_G}{x \rho_L}}
\]  

For separated flow model of void fraction; Zivi (1964), Smith (1969) and Chisolm (1973), the correlations, Eqs. (4.4)-(4.6), Collier et al. (1994) are slightly closed to experimental data, because, they concern slip factor.
\[
\alpha = \frac{1}{1 + \left(\frac{1 - x}{x}\right) \frac{\rho_G}{\rho_L} S}
\]  
(4.4)

\[
S = \frac{u_G}{u_L} \approx \left(\frac{\rho_L}{\rho_G}\right) \frac{\alpha}{\alpha + 1}
\]  
(4.5)

Zivi’s void fraction (1964);

\[
\alpha = \frac{1}{1 + \left(\frac{1 - x}{x}\right)^{2/3} \left(\frac{\rho_G}{\rho_L}\right)^{2/3}}
\]  
(4.6)

Smith’s void fraction (1969);

\[
\alpha = \frac{1}{1 + 0.79 \left(\frac{1 - x}{x}\right)^{0.78} \left(\frac{\rho_G}{\rho_L}\right)^{0.58}}
\]  
(4.7)

Chisolm’s slip factor (1973);

\[
S = \frac{u_G}{u_L} = \left[1 - x \left(1 - \frac{\rho_L}{\rho_G}\right)^{1/2}\right]
\]  
(4.8)

Although, their correlation is still away from experimental data, the trend line is quite similar. Therefore, the void fraction model for plate heat exchanger potentially could be derived from separated model by modification of slip factor correlation.

Fig. 4.27, different from others, the drift flux model of void fraction correlation \(\alpha_d\).

Eq. (4.7), Thome (2010);

\[
\alpha_d = \frac{x}{\rho_G} \left[ C_o \left(\frac{x}{\rho_G} + \frac{1 - x}{\rho_L}\right) + \frac{\bar{U}_{GUV}}{\dot{m}}\right]^{-1}
\]  
(4.9)

In which \(\bar{U}_{GUV}\), the drift velocity correlations are;

Zuber (1967) for bubbly flow

\[
\bar{U}_{GUV} = 1.41 \left[\frac{\sigma g (\rho_L - \rho_G)}{\rho_L^2}\right]^{1/4}
\]  
(4.10)

Wallis (1967) for bubbly flow
\[
U_{GU} = 1.53 \left( \frac{\sigma g (\rho_L - \rho_G)}{\rho_L^2} \right)^{1/4}
\]

(4.11)

Zuber (1967) for slug flow

\[
U_{GU} = 0.35 \left( \frac{g (\rho_L - \rho_G) d_t}{\rho_L} \right)^{1/2}
\]

(4.12)

Ishii (1976) for annular vertical flow

\[
U_{GU} = 23 \left( \frac{\mu_L U_L}{\rho_G d_t} \right) \left( \frac{\rho_L - \rho_G}{\rho_L} \right)
\]

(4.13)

Zuber (1967) and Wallis (1969), has the best trend line and proportionality with experimental data, excluding Ishii’s correlation. The flow pattern reference of each correlation is different. Though the fed flow pattern in inlet conduit is stratified regime, it tends to transforms to slug flow regime. Therefore, Zuber’s correlation using slug flow regime is the best void fraction correlation ones. However, his correlation shall be readjusted in order to match to the recent boundary condition. Hence, the drift flux model presented more realistic correlation.

Although the most of forgoing correlations are constructed based on two-phase flow in circular tube, the correlations show good agreement and reasonable. Consequently, for deriving void fraction correlation based on the drift flux model of Zuber (1967) correlation shall consider; flow pattern, channel geometry and its orientation.

Furthermore, the uniformity of two-phase flow distribution is promoted by channel geometry, especially in inlet and outlet channel of distributor zone, Mustaghfirin et al. (2013) and is also promoted by effective length variation along channel entire corrugated plate, in accordance with the initial study, Mustaghfirin et al. (2012). Similarly, mal-uniformity also is driven by damping effect of channel length, gas-phase impact and low velocity of liquid. They affect to transform flow pattern, Mustaghfirin et al. (2013). Though, all of previous study was carried out on merge pipe distributor, the two-phase flow distribution effect tend to similar, locally as well as tend to more complicated for entire corrugated channel of plate heat exchanger.
4.5.3 Summary

The characteristics of two-phase flow in plate heat exchanger are summarized as follows:

- The increasing air quality of two phase flow promotes to increasing its uniformity distribution. Also the bigger volume void fraction tends to attain the more uniformity.
- The upward flow tends to has more lack uniformity of phase distribution than downward flow, due to bigger slip factor as well as effect of damping of liquid and geometry.
- While the increasing gas quality drives increasing void fraction in certain trend line quite different from existing correlations.
- From both of separated model and drift flux model can be derived new correlation of void fraction for plate heat exchanger. Nevertheless, the drift flux model is the best one.

4.6 Two-phase Flow Distribution and Void Fraction in Corrugated Channel of Plate Exchanger

4.6.1 Modeling of Test Section

The two-phase stratified flow is fed to the inlet tube and distributed to corrugated channel in different volume flow rate correspond to total mass flux $10 \, \text{kg.s}^{-1} \, \text{m}^{-2}$ with quality variation from 0.1 to 0.6, Table 4.4. The experiment was carried out in two flowing modes; vertically upward and downward, as carried out in section 4.5. Unfortunately, the dynamic flow behavior can be observed only in 2d. Therefore the two-phase flow distribution is not observed clearly. The limitation of camera in capturing figure shall be enhanced by modeling based investigation, using CFD software. All detail information about velocity, pressure and phase distribution entire test section model can be presented in detail in 3d. Also, the void fraction can be calculated perfectly.
Fig. 4.28 shows, that the model is made closed to test section, to achieve better agreement. The two-phase flow is formed by mixing air and water on particular velocity in inlet tube. The air is fed to center of inlet, while the water is fed coaxially. The normal area of air inlet $A_G$ is $6.72 \times 10^{-5}$ m$^2$ and the normal coaxial area of water inlet $A_L$ is $2.02 \times 10^{-4}$ m$^2$. The boundary conditions of air velocity inlet $U_G$ and the water velocity inlet $U_L$ for both downward and upward flow, corresponding to a datum of experiment, is presented on Table 4.4. All of velocity boundary conditions result stratified flow in inlet tube at equal total mass flux $G=10.0$ kg.s$^{-1}$m$^{-2}$.

Statistic of mesh; number of nodes are 104,383 and number of elements are 449,239. Minimum size of element is $1.0979 \times 10^{-4}$ m and maximum one is Default $2.1957 \times 10^{-2}$ m.
Table 4.4 Velocity inlet boundary conditions

<table>
<thead>
<tr>
<th>Quality of air</th>
<th>Measured quantity $Q$ (m$^3$.sec$^{-1}$)</th>
<th>Velocity inlet B.C (m.sec$^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$Q_G$ (Air)</td>
<td>$U_G$ (Air)</td>
</tr>
<tr>
<td>0.1</td>
<td>8.599x10$^{-5}$</td>
<td>1.279</td>
</tr>
<tr>
<td>0.2</td>
<td>1.711x10$^{-4}$</td>
<td>2.545</td>
</tr>
<tr>
<td>0.4</td>
<td>3.435 x10$^{-4}$</td>
<td>5.110</td>
</tr>
<tr>
<td>0.6</td>
<td>5.184 x10$^{-4}$</td>
<td>7.712</td>
</tr>
</tbody>
</table>

4.6.2 Result and Discussion

4.6.2.1 Phase distribution

Fig. 4.29 Experimental and CFD modeling of two-phase distribution of downward flow at gas quality: (a) 0.1, (b) 0.4 and (c) 0.6.
The phases distribution entire corrugated channel of test section of plate heat exchanger has good agreement with experimental results for all quality, as depicted on Fig. 4.29. Obviously, by the CFD modeling, the two-phase flow distribution in plate heat exchanger can be observed more clearly than others, even by advance measuring as neutron radiography, by Asano et.al. (2005). Also, by CFD modeling, the two-phase flow distribution can be seen in 3D mode. The

In experimental, using high speed video camera, the effect of increasing quality to uniformity distribution is disappeared. While in the CFD modeling their effects is recognized clearly. By increasing gas quality, the two-phase flow distribution becomes more uniform. Because, when the quality is increased, the gas phase tends to have higher momentum. Therefore, if this impacts to the wall, the flow pattern transforms to bubble with fine bubble size. The more quality, the more momentum density, the finer bubble size, the more uniform distribution entire corrugated channels is.

Although, the gas phase distribution entire plate heat exchanger channel is different. Mostly, the gas-phase distribution still exhibits more uniform than that is in the upward flow, Fig. 4.26. This tends to be promoted by buoyancy and inertia force. The buoyancy force floats the gas away from entrance distributor channel and the body force of liquid closed liquid to entrance distributor channel. The inertia force tends to drive gas phase away from the wall, the gas tend to travel through channel distributor. Therefore, liquid tend to travel straight toward outlet manifold and the gas tend to travel with longer distance through perimeter of channel to outlet manifold.

4.6.2.2 Velocity distribution

Fig. 4.30 shows, the velocity distribution is proportional to phase distribution level. The more quality, the more velocity, the more uniform velocity distribution, the more uniform phase distribution in the corrugated channel of plate heat exchanger is. This evidence is strength to forgoing achievement in section 4.4, that the two-phase flow distribution could be track via single phase flow modeling. The gas phase tends to flow through the high velocity path line. This evidence approved that the inertia force strongly affect to uniformity of two-phase flow distribution. The higher momentum density on two-
phase flow capable to transport of gas phase event in the far distance entire corrugated channel.

![Velocity distribution](image)

Fig. 4.30 Velocity distribution for downward flow at gas quality: (a) 0.1, (b) 0.4 and (c) 0.6.

### 4.6.2.2 Pressure distribution

![Pressure distribution](image)

Fig. 4.31 Pressure distribution for downward flow at gas quality: (a) 0.1, (b) 0.4 and (c) 0.6.
All pressure, shown in Fig. 4.31 is static pressure (gauge) of mixture. The increasing quality affects to increase static pressure distribution, promoting higher pressure loss along corrugated channel of plate heat exchanger. The anomaly effect of two-phase flow distribution appear clearly, there is that increasing uniformity phase distribution by increasing quality is followed by increasing pressure loss.

4.6.2.3 Void fraction

![Void fraction graph](image)

Fig. 4.32 Experimental and analytical correlation as well as modeling of void fraction, downward flow

As shown in Fig. 4.32 the void fraction of modeling have similar trend line with experiment and analytical correlation, but void fraction modeling based have smaller void fraction than others for all gas quality. This indicates that the transport equation for modeling shall be reform in order to obtain better conservativeness and realistic model of two phase flow particularly related to interphase region. The CFD modeling for two-phase flow exhibited limited convergence level of continuity equation due to complexity of geometry and meshes as well as inadequate model in inter-phase zone. It is recommended
to enhance equation for continuity during separation and collation of phases as well as inter-phase equation based on oscillation and damping equation.

4.6.3 Summary

The characteristics of two-phase flow in corrugated channel of plate heat exchanger are summarized as follows;

- Uniformity two-phase flow distribution strongly proportional to increasing quality of gas phase followed by increasing void fraction.
- These are caused by buoyancy force and inertia force. Buoyancy force separate gas phase on downward flow away from entrance distributor. The inertia force drives the gas toward channel perimeter away from the wall. The high momentum density promotes to transport gas phase through higher velocity along their path lines.
- The more uniform flow distribution the more pressure drop is.
- The advanced model of two-phase flow in plate heat exchanger with fully developed inlet outlet tends to have good agreement with experimental results. The void fraction modeling based tend to be lower than experimental and correlation, due to lower convergence level of conservativeness.
4.7 References


Chapter 5
Conclusions and Recommendations

5.1 Conclusions

All foregoing investigation can be summarized in following general conclusions, these are,

*For bifurcation channel;*

- The curved reducer distributor has the best uniformity of phase distribution, conversely has the highest pressure drop. While the bifurcation channel distributor has the lack-uniformity of phase distribution due to stream line separation on sharp edge of main channel cap, but has lower pressure drop than curve reducer distributor. The straight reducer and T-reducer exhibit the optimum condition but both of them have to be enhanced their design. The straight distributor must be reduced stagnation area and optimized reducer slope. The T-reducer shall be narrowed the stagnation area while developed to 3d design.

- The mal-uniformity of two phase flow distribution in bifurcation channel distributor is originated by two-phase flow pattern entering distributor, length of outlet channel and the inclination.

- The bubble flow tends to give mal-uniform phase distribution and the bubble pattern does not transform along outlet channel in various length of outlet channel. All of two phase flow pattern become uniform in distribution, when the distributor is set vertically

- The slug have better phase distribution and the flow pattern changes in both lower and upper outlet channel, while in stratified flow, it changes in lower outlet channel only.

- The impacting two phase flow in the entrance region of bifurcation channel distributor promotes varying of pressure loss and pressure gain not only in entrance zone, but also in junctions as well as in the outlet channels.

- The varying of pressure loss and pressure gain tend to what be caused by oscillating and damping process of air and water phase, promoting fluctuating velocity and
pattern transformation in outlet channels.

- The oscillating and dumping process among phases in entrance region has good agreement with Beguin’s works (2009), but junction and outlet channels region does not.
- The phase impact in end wall of distributor junction promotes pressure gain in entrance region accumulate and or followed by fluctuating static pressure entire distributor
- Perhaps, the fluctuating pressure is promoted by successive oscillation and damping in interface zone and this will be reduced by increasing channel length, than vanish in fully developed region
- The Friedel (1979) correlation has better agreement to experimental results than others for all domains. While, the pressure gain/loss correlation by Hwang et al. (1988) is the best for bubble flow in junction only.

For plate heat exchanger:

- The temperature and pressure distribution in protruded channel is more uniform than in corrugated channel. But, in the dimple channel, in back of protrusion channel, tends to increase pressure drop significantly more than corrugated channel.
- The protrusion and dimple plate afford to increase heat flux more than in corrugation plate up to 111% and 24%, respectively.
- The protrusion channel contributes to reduce pressure drop per unit volume 49% less than in corrugation one. Unfortunately, the dimple channel is liable to increase pressure drop per unit volume up to 204% more than in corrugation channel.
- The increasing pressure drop is contributed by crash flow inter adjacent dimple compartment.
- The velocity distribution in the corrugation channel is more uniform followed by dimple channel and protrusion channel. The effect existing dimple and protrusion for increasing heat flux is stronger than increasing velocity.
- The geometry of inlet and outlet distributor zone of corrugated brazed plate heat exchanger primarily promotes mal-uniformity for single phase flow moreover for two-
phase flow. More unevenly flow distribution of two phase flow is promoted by additional effect of flow pattern and buoyancy.

- The increasing air quality of two phase flow promotes to increasing its uniformity distribution. Also the bigger volume void fraction tends to attain the more uniformity.
- The upward flow tends to has more lack uniformity of phase distribution than downward flow, due to bigger slip factor and effect of damping of liquid and geometry as well as buoyancy.
- While the increasing gas quality drives increasing void fraction in certain trend line quite different from existing correlations.
- The advanced model of two-phase flow in plate heat exchanger with fully developed inlet outlet tends to have good agreement with experimental results.

5.2 Recommendations

For bifurcation channel;

- The design development of distributor should refer to aerofoil contour by using aerodynamics or hydrodynamics theory. The CFD modelling is recommended to append viscous effect to detail turbulence phenomenon. The best modelling have to expand to 3d modelling analysis for perfection design.
- The transport equation for two-phase flow shall accommodate oscillating and damping effect particularly in impingement flow.
- Almost of all two-phase flow correlation should be reformed, while implemented on all distributors in order to reach better agreement and the future correlation shall be developed based on Friedel correlation (1979)

For plate heat exchanger;

- Modifying intersection inter protruded surfaces such that the flow vector adjacent dimple compartment format acute angle ($\varphi<90^\circ$) potentially reduces pressure drop per unit volume.
- Making the combination of protrusion and dimple in the same plate, forming uniform channel for whole channel layer such as in corrugate channel as well as potentially increase the heat transfer capacity and reduce pressure drop significantly.
• From both of separated model and drift flux model can be derived new correlation of void fraction for plate heat exchanger. Nevertheless, the drift flux model is the best one.
• Uniformity two-phase flow distribution strongly proportional to increasing quality of gas phase followed by increasing void fraction. These are caused by buoyancy force and inertia force.
• The buoyancy force separate gas phase on downward flow away from entrance distributor. The inertia force drives the gas toward channel perimeter away from the wall. The high momentum density promotes to transport gas phase through higher velocity along their path lines.
• The more uniform flow distribution the more pressure drop is.
• The advanced model of two-phase flow in plate heat exchanger with fully developed inlet outlet tends to have good agreement with experimental results. The void fraction modeling based tend to be lower than experimental and correlation, due to lower convergence level of conservativeness.

**General recommendation:**

The combination between experimental and modelling has been done. It is very good method to obtain some detail information about two-phase flow in flow field which cannot be achieved by experiment only. Some evidence has not been revealed yet and become a good challenge in future research. There are; oscillating and damping in two-phase flow and unrealistic model resulted by modeling in interphase region of two-phase flow. These two problems are apparently having relationship each other. Accordingly, the following works should be done in future research;

• Try to develop new two-phase flow model using ‘damping and vibration’ approach instead of existing model; homogeneous and separated model
• Start work with developing related equation and tested via modeling using CFD package by creating related equation in UDF (User Define Function). This shall enhance two-phase flow model particularly in complicated problem, interphase region
- The problem of oscillation and damping is not change basic transport equations, but only enhance source term equation to accommodate the oscillation and dumping effect.

- Together, the experimental work should be done for validation purpose, the merge pipe distributor is still relevant for this purpose.

If the forgoing basic fundamental work is well done, the work related to plate heat exchanger must be more realistic to explore such detail two-phase flow phenomenon in there i.e.: phase, velocity, pressure distribution as well as quality and void fraction. By enhancing source term of transport equation to vibration correlation, the calculation time shall increase significantly. This is special challenge for future study.
Appendix 1

Table 2.1 Air properties in atmospheric pressure, Thermopedia™

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<th>T °C</th>
<th>ρ kg.m⁻³</th>
<th>c_p kJ.(kg.K)⁻¹</th>
<th>k w.(m.K⁻¹)</th>
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<th>b x10⁻⁴ K⁻¹</th>
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Table 2.2 Water properties in atmospheric pressure, Thermopedia™

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<th>c_p kJ.(kg.K)⁻¹</th>
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Note:
T: temperature  b: expansion coefficient
ρ: density  Pa: vapor pressure
c_p: Specific heat  h: enthalpy
k: Thermal conductivity  Pr: Prandl’s number
ν: kinematic viscosity