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EXPERIMENTAL INVESTIGATION AND EMPIRICAL ANALYSIS OF NON-BOILING GAS-LIQUID TWO PHASE HEAT TRANSFER IN VERTICAL DOWNWARD PIPE ORIENTATION

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ABSTRACT

The non-boiling gas-liquid two phase flow is pertinent to industrial applications like the reduction of paraffin wax depositions in petroleum transport lines, air lift systems and the chemical processes such as ethanol-water fractionation seeking enhanced heat and mass transfer. The non-boiling two phase heat transfer mechanism in horizontal and vertical orientations has been investigated by many researchers. However, till date very little experimental work and investigation has been performed for vertical downward flow. In order to contribute more to this research and have a better understanding of the non-boiling two phase heat transfer phenomenon for this pipe orientation, experimental investigation is undertaken for a vertical downward oriented 0.01252 m I.D. schedule 10 S stainless steel pipe using air-water as fluid combination. The influence of different flow patterns on the two phase convective heat transfer coefficient is studied using experimental measurements of 165 data points for bubbly, slug, froth, falling film and annular flow patterns spanned over the entire range of the void fraction. In general the two phase heat transfer coefficients are found to be consistently higher than that of the single phase flow. This tendency is observed to increase with increase in the gas flow rate as the flow regime migrates from bubbly to the annular flow. The concept of Reynolds analogy as implemented by Tang and Ghajar [1] for horizontal and vertical upward flow is analyzed against the vertical downward flow data collected in the present study. Due to lack of correlations available for predicting the two phase heat transfer coefficient in vertical downward orientation it was decided to perform the quantitative analysis of the seventeen two phase heat transfer correlations available for vertical upward flow. This analysis is concluded by the recommendation of the top performing correlations in the literature for each flow pattern. Based on the pressure drop data and using Reynolds analogy, a simple equation is proposed to correlate the two phase heat transfer coefficient with the single phase heat transfer coefficient.

INTRODUCTION

The heat transfer in gas-liquid two phase flow is of importance in several industrial operations and design considerations. So far, most of the research in this field has been dedicated to the boiling two phase flows in refrigeration systems and that used in the technology of nuclear reactor cooling. The non-boiling two phase flow i.e. the two component two phase flow finds its application in chemical processes, design of artificial gas lift systems. The research in the field of non-boiling two phase flow done so far is limited to upward and horizontal pipe orientations with seldom work done for the vertical downward orientation with the exception of Oshinowo et al. [2] and Chu and Jones [3]. For vertical downward flow, the flow patterns and hence the pressure drop and heat transfer trend is different than that in the vertical upward system. The purpose of this work is to present new data on two phase convective heat transfer coefficient for all flow patterns occurring in vertical downward flow and occupying the entire range of void fraction and compare it against different heat transfer coefficient correlations available in the literature. The concept of Reynolds analogy to determine heat transfer coefficient based on the pressure drop data reported by Tang and Ghajar [1] is also studied here. Exclusively for vertical downward flow, there are only two correlations available in the literature given by Oshinowo et al. [2] and Chu and Jones [3]. In addition to these correlations, we have presented the quantitative analysis of the performance of 17 other correlations developed for vertical upward flow, against our experimental data.

NOMENCLATURE

D	pipe diameter (m)
f	friction factor
L	pipe length (m)
h	convective heat transfer coefficient ($W/m^2\,^oC$)
k	thermal conductivity ($W/m\,^oC$)
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt number
Re	phase Reynolds number
T	Temperature (^oC)
U	phase velocity (m/s)
P	pressure (Pa)
Pr	Prandtl number

Greek symbols

α	void fraction
μ	phase dynamic viscosity (Pa-s)
ρ	phase density (kg/m^3)
ϕ	two phase friction multiplier in Equation (2)

Subscripts

b	bulk
f	friction
g	gas
i	pipe inlet
l	liquid
o	pipe outlet
s	superficial
TP	two phase
w	wall

EXPERIMENTAL SETUP

The experimental set up used for measuring two phase convective heat transfer coefficient as shown in Figure 1 consists of 0.01252 m I.D. schedule 10 S steel pipe of roughness 0.0000152 m. The fluid combination used for generating two phase flow is air-distilled water. The air is supplied through an Ingersoll Rand T-30 Model 2545, passed through a regulator and filter-lubricator circuit before it is fetched to the water submerged coil heat exchanger. Next, the air is passed through Coriolis mass flow meter and controlled by the Parker needle valve (Model 6A-NLL-NE-SS-V) before it is mixed with water in the static mixer. The liquid phase, i.e. distilled water is stored in a 55 gallon tank and is circulated in the system using a Bell and Gosset (series 1535, model number 3445 D10) centrifugal pump and passed through an Aqua-Pure AP12-T purifier followed by the flow through a ITT model BCF 4063 shell and tube heat exchanger. The water is then directed to flow meter through Emerson (Micro Motion Elite Series model number CMF 100) Coriolis mass flow meter and then allowed to mix with air in a static mixer. The water mass flow rate is controlled by a gate valve placed after the water mass flow meter. The CO1-T type thermocouples with an accuracy of $\pm 1^oC$ are used to measure wall temperatures at seven different stations spaced 0.127 m apart along the pipe length.

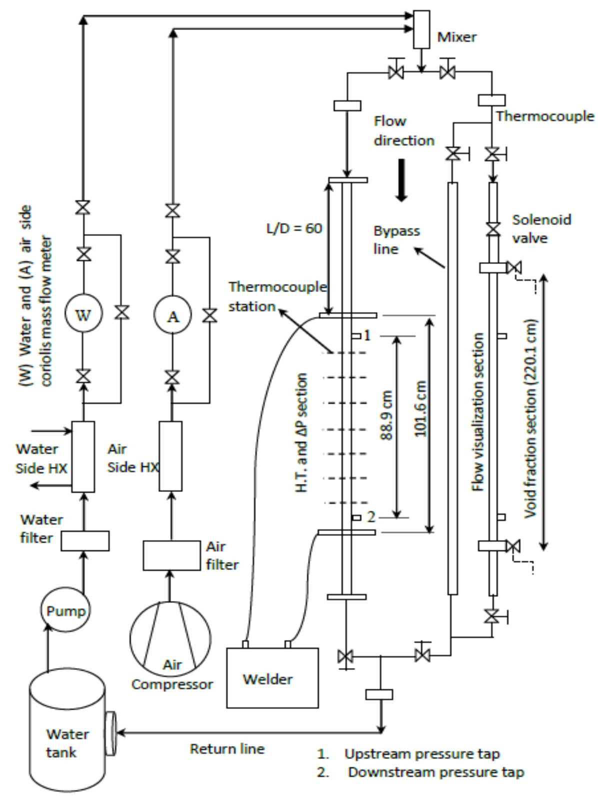

Figure 1 Schematic of experimental setup used for two phase heat transfer measurements in the present study.

The thermocouple probes (TMQSS-06U-6) used to measure temperature at the pipe inlet ($T_{i,b}$) and outlet ($T_{o,b}$) are inserted inside through pipe wall till it almost touched the other end of the pipe wall in order to ensure that the probes are always in contact with the two phase mixture.

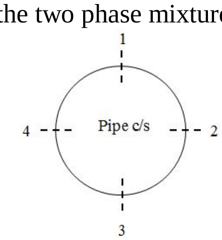

Figure 2 Radially arranged thermocouples along pipe circumference.

The thermocouples at each station (four thermocouples at each station) are arranged radially along the pipe circumference as shown in Figure 2. The uniform heat flux in a range of 7500 W/m^2 to 57000 W/m^2 is supplied by Lincoln DC-600 welder having a maximum current supply of 750 Amp. In order to ensure no heat transfer takes place from the system to surrounding, a 0.076 m (3 in.) thick Micro-Lok Fiber Glass insulation with thermal conductivity of 0.042 W/m^{o}C is used. The local inside wall temperature, wall heat flux and convective heat transfer coefficient is calculated using a data reduction program based on the finite difference formulation reported by Ghajar and Kim [4]. The convective heat transfer coefficient is represented by the average of the measured local values at each station.

The uncertainty of the experimental data for single phase heat transfer coefficient is calculated using Kline and McClintock [5] uncertainty analysis. The validity of the single phase heat transfer data is also confirmed by comparing it against the correlations of Gnielinski [6], Ghajar and Tam [7] and Seider and Tate [8]. As shown in Figure 3 the measured single phase heat transfer coefficient is found to be within ±10% of the predicted values by Seider and Tate [8]. The average and maximum deviation in measurement of single phase heat transfer coefficient with respect to the correlations of Gnielinski [6] and Ghajar and Tam [7] was found to be 7.2%, 14.8% and -6.1%, -10.7%, respectively. The average maximum uncertainty of measured single phase heat transfer coefficients was found to be 4.19%. Oshinowo et al. [2] did not report the uncertainty for two phase flow, but found their single phase data to be within ±15% of that predicted by Seider and Tate [8]. In comparison to this uncertainty, the data collected in the present study appears to have reasonable accuracy.

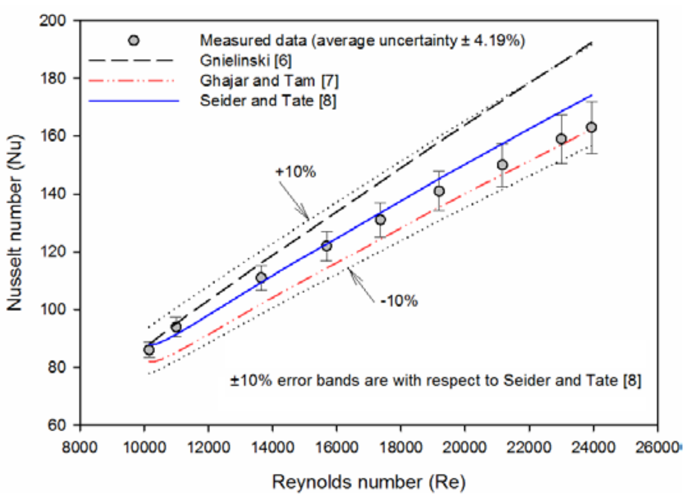

Figure 3 Uncertainty in calculation of single phase Nusselt number.

As seen from Figure 3 the absolute uncertainty associated in calculation of Nusselt number (Nu) increases with increasing Reynolds number. However, the percentage uncertainty is found to decrease due to increasing values of Nusselt number. It was observed in the present study that this higher absolute uncertainty is primarily due to the decreasing temperature difference between the pipe inlet and outlet temperature ($T_{o,b} - T_{i,b}$). At high flow rates as in case of annular flow, it was difficult to maintain the large temperature difference across the pipe inlet and outlet due to heat source limitation resulting into high uncertainty. The other possible reason for high uncertainty in annular and falling film flow is the high heat balance error. For these two flow patterns, the heat balance error was typically in the range of 5 to10% whereas for the flow patterns like bubbly, slug and froth the heat balance error was observed to be in the range of 1 to 6%. The detailed uncertainty of all the measured and calculated variables used for calculating two phase heat transfer coefficient is shown in Table 1 while the percentage uncertainty associated with each flow pattern is reported in Table 2.

Table 1 Uncertainty in measured variables and two phase heat transfer coefficient.

Variable	Value	±Uncertainty	± % Uncertainty
Inner pipe diameter (m)	0.0125	1.27E-05	0.1
Outer pipe diameter (m)	0.0171	1.27E-05	0.07
Heat transfer length (m)	1.016	3.175E-03	0.31
Thermal conductivity (W/m^{o}C)	13.438	NA	NA
Current (I)	577	5.77	1
Voltage (V)	3.91	0.0391	1
Inlet temperature (^{o}C)	16.9	0.5	2.95
Outlet temperature (^{o}C)	20.7	0.5	2.4
Inner wall temperature (^{o}C)	34.54	0.5	1.44
Heat transfer rate (W)	2256	31.91	1.41
Two phase heat transfer coefficient (h_{TP}) (W/m^{2}C)	7435	576.8	7.75

Table 2 Percentage uncertainty in calculation of two phase heat transfer coefficients for each flow pattern.

Flow pattern	Min. uncertainty (%)	Max. uncertainty (%)	Avg. uncertainty (%)
Bubbly	3.04	7.76	6.28
Slug	3.73	6.55	4.8
Froth	6.1	8.4	7.4
Falling Film	7.4	12.7	10.0
Annular	10.3	13.3	12.2

FLOW PATTERN MAP

The variation of flow patterns in vertical downward flow with respect to the phase flow rates can be mapped with superficial liquid velocity as ordinate and superficial gas velocity as abscissa. As shown in Figure 4, at the low liquid and moderate to high gas flow rates, the flow pattern shifts from falling film to annular while at moderate to high flow rates and with increasing gas flow rates, the bubbly, slug, froth and annular flow patterns are observed. As seen in the bubbly flow, the bubbles are oriented towards the pipe center (away from pipe wall) resulting into anomalous behavior of the two phase heat transfer coefficient. A detailed discussion on the variation of two phase heat transfer coefficient with respect to flow patterns and hence the phase flow rates is presented next.

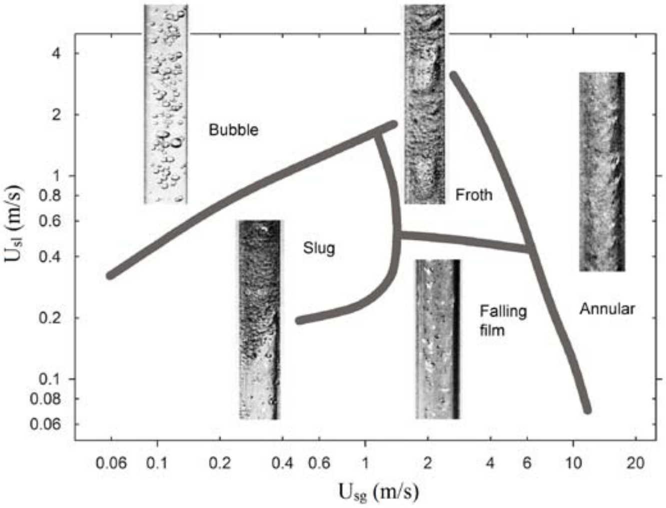

Figure 4 Flow pattern map for vertical downward flow.

RESULTS AND DISCUSSION

In all 165 data points were collected in the present study for two phase convective heat transfer coefficient for different flow patterns. The behavior of this data against increasing gas and liquid flow rates is analyzed in the next section. To add more insight to the effect of the injection of gas phase on two phase heat transfer coefficient, it is compared against the heat transfer coefficient in single phase flow calculated using Seider and Tate [8] correlation. In this section, we first discuss the overall trend of the two phase heat transfer data followed by the Reynolds analogy and finally propose a correlation to predict the two phase convective heat transfer coefficient based on the two phase frictional pressure drop data.

BUBBLY FLOW

The two phase bubbly flow often referred to as coring bubbly flow is featured by the flow of discrete bubbles in the continuous liquid media dispersed in the near pipe axis region. The heat transfer coefficient in bubbly flow regime is observed to increase gradually with increasing superficial gas Reynolds number measured at constant superficial liquid Reynolds number. It is seen from Figure 5 that, the two phase convective heat transfer coefficient is also a function of liquid flow rate. A careful observation indicates that the h_{TP} values stay virtually constant till Re_{sg} = 300 and thereafter increases gradually with increasing Re_{sg}. However at Re_{sg} > 300, the increase in two phase convective heat transfer coefficient at low liquid flow rate, i.e. Re_{sl} = 14200 is rapid and this tendency gets damped with increasing superficial liquid Reynolds number.

The comparison of the two phase convective heat transfer coefficient with its single phase counterpart shows that for bubbly flow, the value of h_{TP} is less than its single phase heat transfer coefficient. This difference between the two flow conditions decrease with increasing superficial gas Reynolds number and finally later is observed to be greater than former. The threshold value at which this trend flips also shifts to the higher Re_{sg} with increasing liquid flow rate. A plausible justification for this phenomenon is that for vertical downward bubbly flow, due to interaction between buoyancy and liquid inertia forces acting in opposite directions the bubbles are pushed towards the core (pipe axis) region by the repulsive forces. Thus the void fraction profile in vertical downward flow has peak in the near axis region and minima near the pipe wall. The repulsive force results in the increase of the viscous sub-layer thickness and hence more resistance is offered to the heat transfer. In other words, in the near pipe axis region due to presence of air bubbles (moving in a direction opposite to that of the mean flow) the average mixture velocity is less than that at the near wall region. This translates to a dynamic pressure difference between core region and near pipe wall region resulting into increase of the viscous sub-layer thickness and more resistance to heat transfer. However, with increasing gas and liquid flow rates, the bubbles are forced to move in the near wall region thus depleting the viscous sub-layer thickness and hence finally increasing the two phase convective heat transfer coefficient greater than its single phase counterpart.

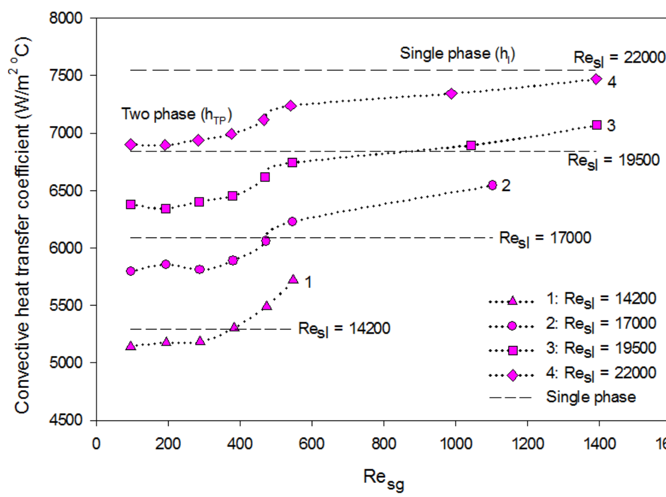

Figure 5 Variation of two phase convective heat transfer with increasing Re_{sg} in bubbly flow regime.

The spread of bubbles with increasing gas flow rate is documented by Bhagwat and Ghajar [9]. Oshinowo et al. [2] and Chu and Jones [3] did not perform comparison between two phase and single phase heat transfer coefficient, however, they reported that, in bubbly flow regime due to difference between void fraction profiles in upward and downward orientations, the two phase convective heat transfer coefficient in downward flow is always less than that in vertical upward flow. In view of the void profiles in vertical upward and downward orientations, the aforementioned justification appears logical.

SLUG FLOW

The slug flow is characterized by the alternate flow of air and liquid slugs creating fluctuations in the pipe wall temperature. A general tendency of the heat transfer data shows that the two phase convective heat transfer coefficient increases continuously with increasing gas flow rate at a given liquid flow rate. However, the slope of this trend of variation of h_{TP} with respect to gas superficial Reynolds number changes at approximately Re_{sg} = 600 making it less steeper further. As reported by Oliver and Wright [10], the two phase heat transfer in slug flow regime is a function of gas slug length. The shorter the gas slug length higher is the rate of heat transfer. Our results are in agreement with Oliver and Wright [10] observations. It is seen from Figure 6 that, the two phase convective heat transfer coefficient increases with increasing liquid flow rate where the slug length reduces with increasing Re_{sl} as reported Bhagwat and Ghajar [9]. As per his observations the increase in gas flow rate at constant liquid flow rate contributes only to enhanced mixture agitation and serves to maintain short length slugs. It should be also noted that the two phase heat transfer coefficient is greater than its single phase counterpart for slug flow regime. This advocates the influence of increasing gas flow rate on relative magnitudes of two phase heat transfer in context to corresponding single phase heat transfer coefficient. Further for froth, falling film and annular flow regimes, this deviation of two phase heat transfer coefficient from single phase parameter is observed to increase owing to the turbulent mixing caused by injection of gas phase.

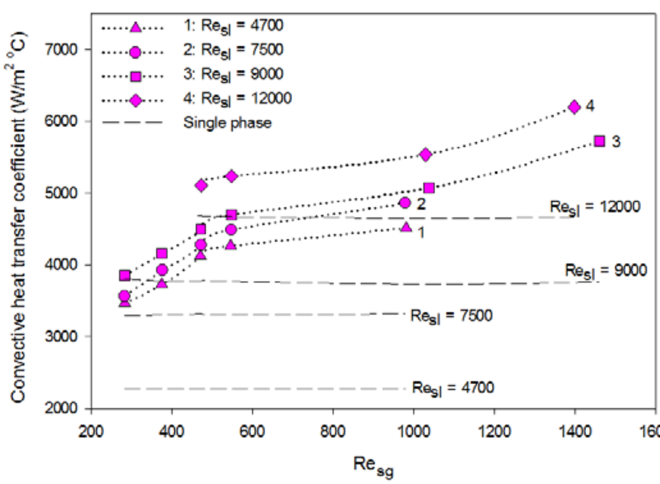

Figure 6 Variation of two phase convective heat transfer coefficient with increasing Re_{sg} in slug flow regime.

FROTH FLOW

The froth flow regime is marked by a significant qualitative turbulent flow due to vigorous mixing of the two phases. As a result the two phase convective heat transfer coefficient is significantly higher than its corresponding single phase value and contributes for the highest increase in the h_{TP} values among all other flow regimes. However, this flow pattern is observed to be less sensitive to the gas flow rate in comparison to the bubbly and slug flow regimes. The overall behavior of two phase heat transfer in froth flow regime with respect to increasing superficial gas Reynolds number is showed in Figure 7. It is also observed that with increasing gas flow rate, the deviation of the two phase heat transfer coefficient from its single phase equivalent increases.

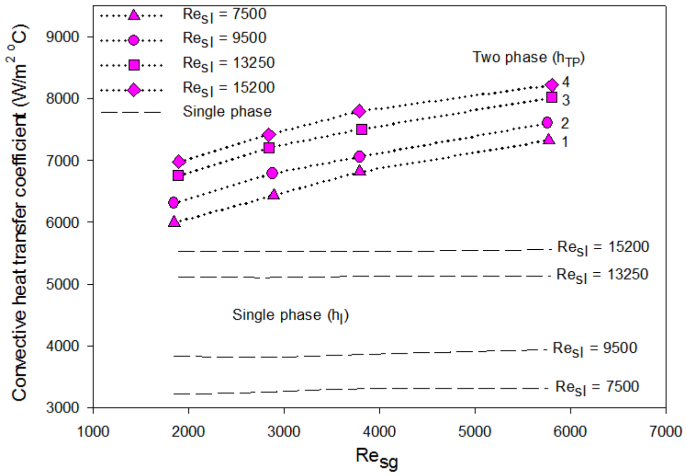

Figure 7 Variation of two phase heat transfer coefficient with increasing Re_{sg} in froth flow regime.

FALLING FILM FLOW

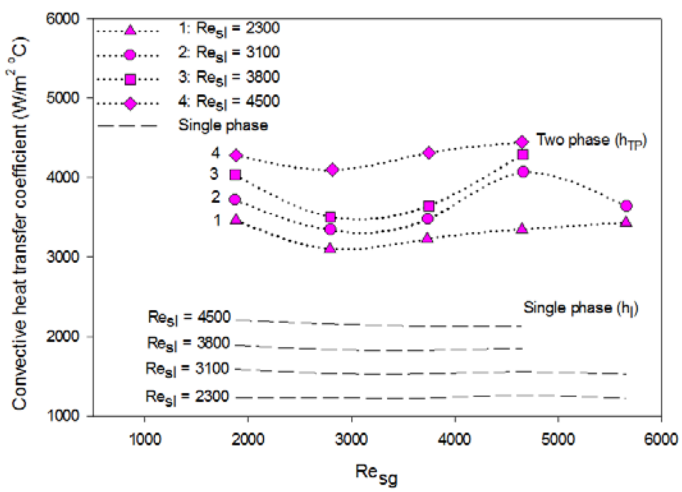

Figure 8 Variation of two phase convective heat transfer coefficient with varying Re_{sg} in falling film flow regime.

Falling film flow is characterized by the flow of liquid film in contact with the pipe wall and the gas phase flowing through the pipe core at moderate to high flow rates. It was observed by Bhagwat and Ghajar [9] that in falling film flow, at low liquid flow rates, the liquid film loses its contact with the pipe wall frequently. Due to heating of the pipe surface it is anticipated that at high pipe wall temperatures, this phenomenon of dry spots is aggravated and thus the heat transfer coefficient drops down due to absence of liquid phase in contact with the pipe wall. However, with increasing gas flow rate, the liquid phase is forced to reestablish its contact with the pipe wall and hence the two phase convective heat transfer coefficient is observed to increase. Bhagwat and Ghajar [9] observed that this tendency of occurrence of dry spots on the pipe surface decreases with increase in both gas and liquid flow rates. Thus, the tendency of drop in h_{TP} values is observed to decrease at higher liquid flow rates (Re_{sl} = 2300) as shown in Figure 8.

ANNULAR FLOW

As seen in the annular flow regime illustrated in Figure 9, the two phase convective heat transfer coefficient increases monotonically with increasing superficial gas Reynolds number with similar tendency indicated at all liquid flow rates. A similar observation is reported by Chu and Jones [3] and Oshinowo et al. [2] for vertical downward flow.

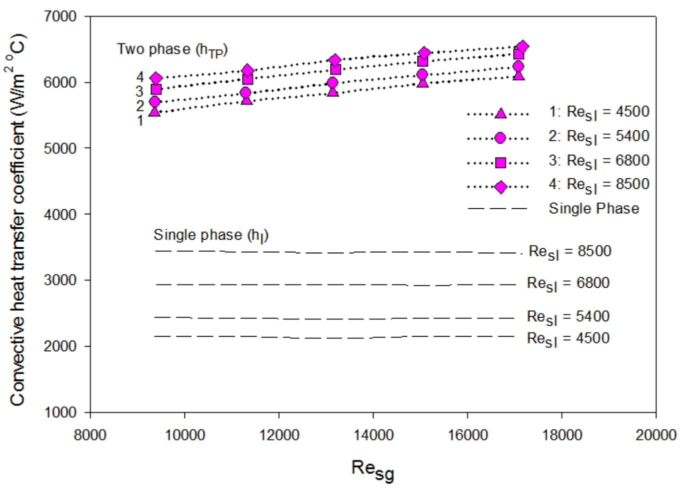

Figure 9 Variation of two phase heat transfer coefficient with increasing Re_{sg} in annular flow regime.

In addition to this it is also seen that similar to froth flow regime, the difference between the two phase and single phase heat transfer coefficients increases with increasing gas flow rate. Overall observation of froth and annular flow patterns show that the two phase heat transfer could be correlated to its single phase version by employing some two phase parameter that shows a similar tendency to increase with the increase in gas flow rate. This parameter could be either void fraction or two phase pressure drop. The variation of two phase convective heat transfer coefficient for all flow patterns measured at increasing gas flow rate and at constant liquid flow rate is shown in Figure 10. It is evident that the two phase convective heat transfer coefficient increases rapidly in the bubbly and slug flow regimes with increasing superficial gas Reynolds number. As the flow pattern shifts from slug to froth flow, this tendency of sharp increase in h_{TP} value decreases and thereafter a gradual rise in two phase heat transfer coefficient is observed. A careful observation of Figure 10 reveals that this trend of increase of two phase convective heat transfer coefficient with increasing gas flow rate is also influenced by the liquid flow rate. Particularly for the froth and annular flow regimes, this trend is more steep for increasing Re_{sl} in comparison to that observed to low Re_{sl} = 4500. However, the sharp increase in h_{TP} values with increasing gas flow rate in bubbly and slug flow regimes is not affected by the liquid flow rates. The overall trend of h_{TP} with respect to gas flow rate is very similar to the trend of void fraction against increasing gas flow rate reported by Bhagwat and Ghajar [9] and that of the parameter (ϕ) given by Equation (2) against increasing gas flow rate. Thus this trend indicates the possibility of using either void fraction or the ϕ parameter to establish a nexus between two phase and single phase convective heat transfer coefficients.

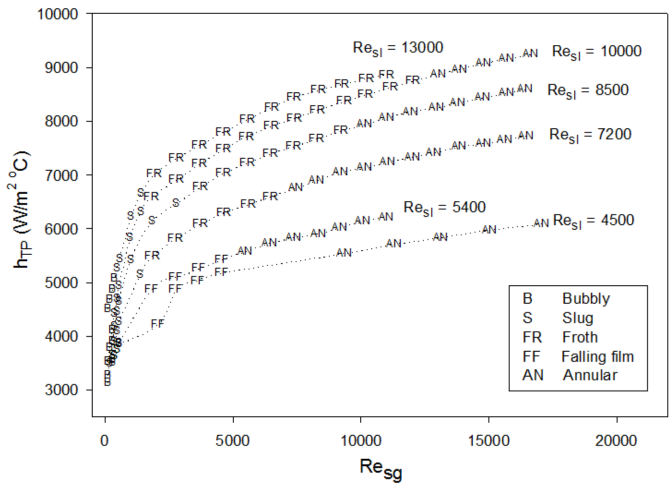

Figure 10 Variation of two phase convective heat transfer coefficient with increasing Re_{sg} for different flow patterns

REYNOLDS ANALOGY

It is a well-known fact that for any Newtonian fluid flow, the momentum transfer and heat transfer can be interrelated with each other through the concept of Reynolds analogy. Literature reports the theory of Reynolds analogy and correlations based on this concept applicable to two phase flow for horizontal and vertical upward flow. However, there are no reports of application of this method to vertical downward two phase flow. The study of Chu and Jones [3] and Oshinowo et al. [2] did not verify Reynolds analogy for vertical downward flow. Here we have showed the use of Reynolds analogy applicable to two phase flow to predict the two phase heat transfer coefficient as reported by Tang and Ghajar [1]. The connection between two phase heat transfer and two phase pressure drop is as shown in Equation (1). More details about this equation can be obtained from Tang and Ghajar [1].

$$\frac{h_{TP}}{h_l} = F_p^{\,m}\left(\frac{\dot{m}_l}{\dot{m}}\right)\left(\frac{\rho_{TP}}{\rho_l}\right)^{n} \phi_l^{\,p} \qquad (1)$$

The exponents m, n and p have the values of 0.5, -0.5 and 0.2, respectively. The F_p is a flow pattern factor and ϕ is the two phase friction multiplier as defined by Lockhart and Martinelli [11] as in Equation (2),

$$\phi = \sqrt{\frac{(dP/dL)_{f,TP}}{(dP/dL)_{f,l}}} \tag{2}$$

The details about calculation of the flow pattern factor (F_p) can be obtained from Tang and Ghajar [1]. The single phase fanning friction factor (f) to be used to calculate single phase frictional pressure drop is calculated using Equations (3) and (4)

For $Re_{sl} < 2000$,

$$f = \frac{16}{Re_{sl}} \tag{3}$$

For $Re_{sl} > 2000$,

$$f = \frac{0.079}{Re_{sl}^{0.25}} \tag{4}$$

The two phase density is calculated based on void fraction and is expressed by Equation (5),

$$\rho_{TP} = \alpha\rho_g + (1-\alpha)\rho_l \tag{5}$$

The single phase heat transfer correlation used in both Equations (1) and (7) is based on the superficial liquid Reynolds number and is calculated using Seider and Tate [8] correlation expressed as,

$$h_l = 0.027\left(\frac{k_l}{D}\right)Re_{sl}^{(4/5)}Pr_l^{(1/3)}\left(\frac{\mu_b}{\mu_w}\right)_l^{0.14} \tag{6}$$

The Equation (1) reported by Tang and Ghajar [1] is essentially developed for pipes inclined from horizontal to vertical upward orientation. The problem with using Reynolds analogy is that it requires the frictional pressure drop data or a robust and accurate correlation to predict frictional pressure drop in different flow patterns. The isothermal frictional pressure drop data used to analyze Tang and Ghajar [1] correlation was obtained from the pressure drop measurements done by Bhagwat et al. [12]. The Reynolds analogy given by Tang and Ghajar [1] is observed to predict the two phase convective heat transfer coefficient satisfactorily as shown in Figure 11. The satisfactory performance criterion is user defined and for the present work is outlined as the ability of the correlation to predict more than 75% of data points within ±20% error band and more than 85% of data points within ±30% error bands. The two phase convective heat transfer coefficient is predicted well within ±30% error bands by Tang and Ghajar [1] for all bubbly and annular flow regimes. The accuracy of this correlation drops down in slug, falling film and froth flow regimes. The other correlation based on Reynolds analogy and developed for vertical upward flow is that given by Vijay et al. [13]. Their correlation gave excellent results in bubbly flow pattern, but failed to give satisfactory performance in other flow regimes. The quantitative prediction of both Tang and Ghajar [1] and Vijay et al. [13] is documented in Table 3. Based on the Reynolds analogy it was attempted in the present study to correlate the two phase and single phase convective heat transfer coefficient through the non-dimensional frictional pressure drop defined by Equation (2). On the basis of the deviation of the two phase pressure drop from its single phase counterpart as shown in Figures 5-9, it is clear that the parameter that increases with increasing gas flow rate might correlate the two phase and single phase heat transfer coefficients. Bhagwat et al. [12] found the non-dimensional frictional pressure drop defined by Equation (2) to increase with increasing superficial gas Reynolds number (Re_{sg}) and with the trend similar to that observed for two phase heat transfer.

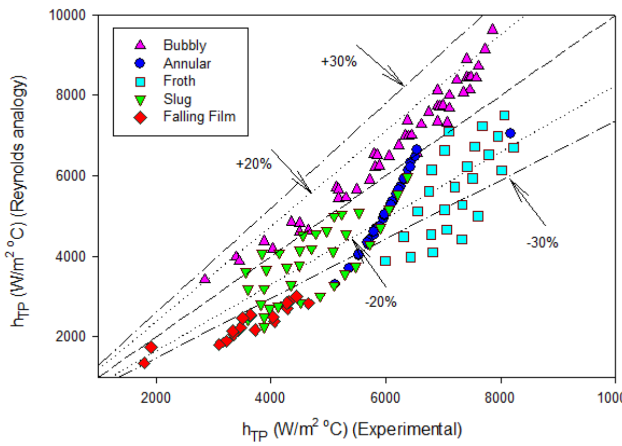

Figure 11 Performance of Tang and Ghajar [1] correlation based on Reynolds analogy against data in the present study.

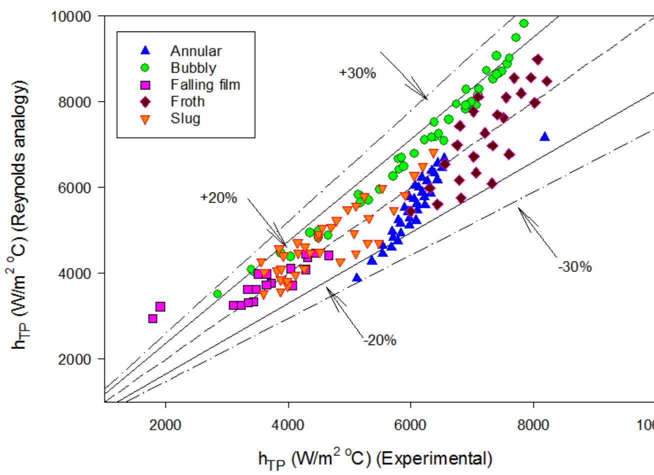

Figure 12 Performance of the proposed correlation Equation (7) against the data measured in the present study.

The major advantage of using this form of the correlation based on the Reynolds analogy is that it accounts for change in the flow condition such as the pipe diameter and fluid combination through the (ϕ) parameter. It was observed by Bhagwat et al. [12] that the two phase pressure drop is significantly influenced by the pipe diameter and the fluid thermo physical properties specially in the annular flow regime. It is anticipated that by using the (ϕ) parameter to correlate the two phase and single phase convective heat transfer coefficient, the influence of fluid combination and pipe diameter on the two phase heat transfer coefficient can be overcome. More experimental data for different pipe diameters and fluid combinations is required to verify this relationship. However, for the present experimental conditions, i.e. for D = 0.01252 m and air-fluid combination, the relationship given by Equation (7) was found to be apt. The (ϕ) parameter calculated from Equation (2) with exponent 0.55 gave the best fit to the experimental data as expressed in Equation (7).

$$\frac{h_{TP}}{h_l} = \phi^{0.55} \tag{7}$$

The performance of the proposed correlation against the data measured in the present study is presented graphically in Figure 12. It should be noted that the use of Equation (7) would require the experimental pressure drop and void fraction values or accurate models to predict these two phase flow parameters.

PERFORMANCE ANALYSIS OF TWO PHASE HEAT TRANSFER CORRELATIONS

As mentioned earlier, very little work is done till now dedicated to vertical downward flow and the only correlations available in the literature for predicting two phase convective heat transfer coefficient in vertical downward flow are by Chu and Jones [3] and Oshinowo et al. [2]. Hence it was decided to analyze the correlations developed for vertical upward flow in addition to aforementioned correlations. The preliminary results of the performance of some of these correlations against our vertical downward heat transfer data gave good results and thus supported our idea of analyzing them against vertical downward data. Due to space limitations the physical structure of these correlations are not reported here. The reader is suggested to refer to the original paper listed in the reference section. The performance of 17 convective heat transfer correlations as presented in Table 3 is documented in terms of the percentage of data points predicted within ±20% and ±30% error bands, respectively. In addition to this the accuracy of these correlations is also reported in terms of absolute percentage mean error and the standard deviation. Standard deviation as a statistical tool is useful to get an idea of any possibility of modifying the correlation to make it work for vertical downward flow. The absolute mean error and the standard deviation is calculated using Equations (8) and (9) where N and ε are the number of points and percentage error, respectively.

$$\text{Absolute mean error (\%)} = \frac{1}{N}\sum_{i=1}^{N}\left|\varepsilon_i\right| \tag{8}$$

Where, $\left|\varepsilon_i\right| = \left|\frac{pred_i - meas_i}{meas_i}\right| \times 100\%$

$$\text{Std. deviation (\%)} = \sqrt{\frac{1}{N}\sum_{i=1}^{N}\left(\varepsilon_i - \bar{\varepsilon}\right)^2} \tag{9}$$

The performance of the correlations are judged satisfactory if they were able to predict more than 75% of data points within ±20% error bands and more than 85% of data points within ±30% error bands, respectively for each flow pattern. The definition of satisfactory performance criteria is based on the overall results of the correlations used in the present study.

Starting with the bubbly flow regime where the interaction of buoyancy and liquid inertia forces influence two phase flow parameters, the correlation proposed in the present study successfully predicted 85% and 100% of the data points within ±20% and ±30% error bands, respectively. The correlations of Vijay et al. [13], Knott et al. [14], Kim et al. [15] and Shah [16] were identified as best performers, with Vijay et al. [13] and Shah [16] predicting 100% of data points within ±20% error bands The mean absolute error associated with Shah [16] was 9.7% with a standard deviation of 4.5. Although the correlation of Ravipudi and Godbold [17] did not meet the satisfactory performance criteria, it displayed some potential to be able to predict the two phase heat transfer coefficient in vertical downward flow by predicting 90.4% of data within ±40% error bands.

The prediction of two phase heat transfer coefficient in slug flow was thought to be little difficult as a consequence of its pulsating nature. In addition to the proposed work, the only correlations that could predict the slug flow heat transfer data well above the satisfactory criterion were by Kim et al. [15] and Ravipudi and Godbold [17]. The proposed correlation given by Equation (7), predicted 92% and 94.7% of data points within ±20% and ±30% error bands. The correlations of Kim et al. [15] and Ravipudi and Godbold [17] predicted 97.4% of data points within ±30% error bands with mean absolute error of 4.7% and 4.3%, respectively. The correlation of Knott et al. [14] did not meet the satisfactory performance criterion for ±20% error bands but predicted 89.5% of data points within the relaxed criterion of ±30% error bands. In addition to these the correlations of Tang and Ghajar [18] and Shah [16] showed their potential to predict the two phase heat transfer data in vertical downward flow. Both of these correlations predicted more than 75% and 90% of the data points for relaxed performance criteria of ±30% and ±40%, respectively.

The two phase heat transfer coefficient in froth flow regime could be predicted satisfactorily by the proposed correlation, Chu and Jones [3] and Knott et al. [14]. All three of these correlations were able to predict 100% of the data points within ±30% error bands. In fact, the proposed correlation predicted 100% of data points within ±20% error bands while the other two predicted 96% and 100% of the data points within ±20% tolerance bands. As can be seen from Table 3, the correlation of Ravipudi and Godbold [17] and Shah [16] predicted 100% of data points within ±30% error bands but failed to meet satisfactory performance criterion restricted within ±20%. In case of falling film flow, the proposed correlation predicted 89.5% of data points within the set criterion. Apart from the proposed correlation, the correlation of Knott et al. [14] could predict 89.5% and 94.5% of the data points within the set criteria. The correlation of Shah [16] predicted 100% of data points within ±30% error bands but could predict only 63.2% of data points within restricted criteria of ±20% error.

For annular flow regime, the proposed correlation gave the best performance by predicting 94.4% and 100% of the data points within ±20% and ±30% error bands, respectively. The next best performance was by Shah [16] followed by Tang and Ghajar [18]. These correlations predicted 86.1% and 77.8% of data points within ±20% error bands and 100% and 94.4% of data points within ±30% error bands, respectively. The correlation of Knott et al. [14] predicted 100% of data points within ±30% error bands but failed to perform satisfactorily for more restricted criteria. A careful observation of Table 3 shows that the performance of Tang and Ghajar [18] improves as the flow pattern shifts from bubbly to annular flow regime or in other words as void fraction increases from low to high values. The correlation of Tang and Ghajar [18] incorporates the void fraction predicted by Woldesemayat and Ghajar [19] in its structure and is conjectured to be sensitive to accurate estimation of void fraction. For vertical downward flow, Bhagwat and Ghajar [9] have shown that the accuracy of Woldesemayat and Ghajar [19] is not good in bubbly flow regime but improves significantly as the flow pattern shifts from bubbly to annular flow regime or alternatively as the void fraction increases from low to high values. Thus the inaccurate prediction of the void fraction in bubbly and slug flow regimes can be a probable reason of inaccuracy of Tang and Ghajar [18] correlation in these flow regimes. The correlations of Knott et al. [14] and Shah [16] which predicted two phase convective heat transfer data satisfactorily in majority of the flow patterns are observed to share a similar correlation structure with the only difference in the empirical exponents. The inaccuracy of later correlation for froth and falling film flow is probably due to the smaller exponent used in this correlation as compared to the former. The performance of Chu and Jones [3] was observed to increase as flow pattern shifted from bubbly to froth flow regime, however, its prediction of two phase heat transfer coefficient deteriorated in the falling film and annular flow regimes. The correlation of Oshinowo et al. [2] derived for vertical downward flow failed to predict the convective heat transfer data measured in the present study. Their correlation was based on the experimental data collected in a 0.0254 m I.D. pipe and thus indicates its sensitivity to the experimental conditions. Based on the quantitative analysis documented in Table 3, top three performing correlations in each flow pattern are recommended in Table 4. It is observed that the proposed correlation by Equation (7) is robust and gives comparable performance to top performing correlations for each flow pattern. It should be noted that the sequence of correlations mentioned in Table 3 is not based upon its performance, but gives alternative equations that could be used for a particular flow pattern. Conclusively the proposed correlation by Equation (7) can be used independent of flow patterns in vertical downward flow in a 0.01252 m diameter pipe using air-water fluid combination.

CONCLUSIONS

The undertaken study presented a new set of data on two phase convective heat transfer coefficient in vertical downward flow for all flow patterns. The trend of heat transfer data for bubbly, froth and annular flow regimes show a monotonous tendency to increase the two phase convective heat transfer coefficient with increasing superficial gas Reynolds number measured at different constant superficial liquid Reynolds numbers. The two phase convective heat transfer coefficient in bubbly flow regime is observed to be less than its single phase equivalent and this phenomenon is justified based on the void fraction profile in vertical downward bubbly flow. The trend of decreasing and increasing two phase heat transfer coefficient with increasing Re_{sg} in falling film flow is justified based on the observations of Bhagwat and Ghajar [9]. However, no confirm inference can be drawn due to limited data in the falling film flow regime. Overall enhancement in two phase heat transfer coefficient is observed due to introduction of gas phase.

In all, seventeen correlations available in the literature are compared against the experimental data measured in the present study. All the correlations used here except the correlations of Oshinowo et al. [2] and Chu and Jones [3] were devised for vertical upward flow. However, some of these correlations satisfactorily predicted the two phase heat transfer coefficient and showed a potential to be used for vertical downward flow. The concept of Reynolds analogy is also used to predict the two phase convective heat transfer coefficient. The proposed correlation, Equation (7), correlates the two phase and single phase convective heat transfer through the two phase frictional pressure drop. The proposed correlation is observed to perform satisfactorily in all flow regimes and overall predicts the two phase heat transfer coefficient with an absolute mean error of 11.4% and standard deviation of 17.2. Due to scarcity of the experimental data for vertical downward two phase flow, the success of the proposed correlation to predict the two phase heat transfer coefficient for different pipe diameters and fluid combinations could not be verified. In view of the general trend of bubbly and falling film flow, research focused on these two flow patterns is highly recommended.

Table 3 Performance analysis of various heat transfer coefficient correlations against the data collected in present study for various flow patterns.

Flow pattern	Bubbly (47 data points)				Slug (38 data point)				Froth (25 data points)				Falling Film (19 data points)				Annular (36 data points)			
Correlations	1*	2	3	4	1*	2	3	4	1*	2	3	4	1*	2	3	4	1*	2	3	4
Present study	85	100	14.5	4.5	92	94.7	2.1	24.9	100	100	0.7	9.1	89.5	89.5	8.6	21.1	94.4	100	8.7	7.1
Aggour [20]	0	0	39.9	6.1	18.4	36.8	29.4	16.8	0	0	51.7	9.5	0	0	68.8	20.4	0	2.8	69.9	19.5
Chu and Jones [3]	0	23.4	40.3	15.4	50	71.1	20	16.4	96	100	4.7	6.8	21.1	36.8	39.8	23.9	2.8	22.2	35.5	6.1
Dorresteijn [21]	0	0	47.4	7.7	26.3	47.4	28.1	19.2	0	0	56.6	13.0	0	0	57.7	18.3	2.8	5.6	66.1	21.5
Drucker et al. [22]	0	59.6	32.2	12.1	13.2	31.6	36	11.8	92.2	100	2.2	11.0	63.2	89.5	16.5	32.6	8.3	38.9	33.6	9.4
Katasuhara and Kazama [23]	0	0	85.3	33.1	0	0	96.4	20.2	0	0	48.5	7.6	0	0	138	56.6	0	0	60.9	7.1
Khoze et al. [24]	0	0	216.7	36.9	0	0	185.2	47	0	0	238.7	37.1	0	0	227.9	41.1	0	0	255.1	38.5
Knott et al. [14]	93.6	100	13.7	3.9	68.4	89.5	12.1	14.6	100	100	1.5	8.8	89.5	94.7	8.0	13.2	61.1	100	17.1	7.3
Kudiraka et al. [25]	8.5	19.1	59.7	36.9	0	0	113.2	19.5	0	0	69.3	6.7	0	0	215.4	74	0	0	118.7	10.3
Kim et al. [15]	78.7	100	16.1	3.9	92.1	97.4	4.7	12.6	60	96	18.4	7.2	47.4	68.4	27.1	16.0	38.9	69.4	21.9	13.9
Oshinowo et al. [2]	0	0	136	20	0	0	119.1	26.7	0	0	107.4	18.6	0	0	119.7	31.2	0	0	94.9	16.2
Ravipudi and Godbold [17]	55.3	74.5	19.7	15.7	97.4	97.4	4.3	8.9	72	100	17.1	5.5	0	0	80.5	29.8	0	0	70.2	5.0
Rezkallah and Sims [26]	27.7	72.3	24.9	6.4	28.9	71.1	23.2	15.3	0	0	47.1	10.3	0	0	81.9	23.0	0	0	79.7	19.5
Shah [16]	100	100	9.5	4.0	57.9	71.1	19.1	14.7	68	100	12.9	9.8	63.2	100	15.8	9.7	86.1	100	11.2	7.7
Tang and Ghajar [18]	0	8.5	35.9	4.9	52.6	78.9	6.8	22.4	64	84	9.3	18.6	5.3	5.3	38.5	6.7	77.8	94.4	10.3	11.4
Tang and Ghajar [1][a]	95.7	100	10.2	5.4	55.3	68.8	148.8	13.9	44	72	21.6	11.5	5.3	21.1	34	7.9	63.9	92.7	11.7	18.4
Ueda and Kanaoka [27]	0	0	671	238.8	0	0	776.8	134	0	0	49.5	3.4	0	0	126.8	40.6	0	0	102.9	10.0
Vijay et al. [13][a]	100	100	2.7	5.1	36.8	57.9	27.3	14.1	20	48	31.4	11.8	0	5.3	48.7	7.6	0	0	52.2	7.0

[a] Correlations based on Reynolds analogy.

* Note: In the above table, numbers 1, 2, 3 and 4 refer to:

1= % of data points within ± 20 % error bands

2= % of data points within ±30 % error bands

3= Mean absolute error (%)

4= Standard deviation

Table 4 Recommended correlations for different flow patterns in vertical downward flow.

Bubbly	Slug	Froth	Falling film	Annular
Present study	Present study	Present study	Present study	Present study
Shah [16][a]	Kim et al. [15]	Chu and Jones [3]	Knott et al. [14]	Shah [16]
Tang and Ghajar [1]	Ravipudi and Godbold [17]	Knott et al. [14]	Shah [16]	Tang and Ghajar [18]

[a] Shah [16] and Vijay et al. [13] give similar performance in bubbly flow regime.

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