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Heat Transfer Measurements and Correlations Assessment for Upward Inclined Gas-Liquid Two Phase Flow

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ABSTRACT

Non-boiling heat transfer in gas-liquid two phase flow embraces some of the practical applications pertinent to chemical and oil and gas industry. In comparison to horizontal and vertical two phase flows, very little information is available about non-boiling heat transfer in upward pipe inclinations. To further understand this phenomenon, 679 experimental measurements of the local and space averaged two phase heat transfer coefficients are carried out in 0^o, $+5^o$, $+10^o$, $+20^o$, $+30^o$, $+45^o$, $+60^o$, $+75^o$ and $+90^o$ degrees of pipe inclinations using air-water fluid combination in a 12.5 mm I.D. stainless steel pipe. The measured non-boiling two phase heat transfer coefficients are found to be influenced by phase flow rates, pipe orientation and flow patterns. The two phase heat transfer coefficient is observed to increase with increase in pipe orientation from horizontal in the vicinity of $+45^o$ to $+60^o$ and then shows a decreasing trend as the pipe is inclined towards vertical position. The effect of change in flow pattern structure on the measured two heat transfer coefficient is noticeable for bubbly and slug flows compared to that in intermittent and annular flow patterns. The heat transfer coefficient in two phase flow is found up to 5-6 times greater than that in single phase flow. The experimental data (679 data points) measured in this work are also used to scrutinize the performance of some of the existing non-boiling two phase heat transfer correlations available in the literature. Based on a statistical performance, the top performing correlations for different flow patterns and pipe orientations are recommended.

KEY WORDS: Two phase flow, non-boiling heat transfer, upward inclination, flow patterns, heat transfer correlations

1. INTRODUCTION

Several practical applications occurring such as those seen in power plants, nuclear reactor cooling, heat exchangers (evaporators and condensers) in heating and cooling systems and simultaneous transport of oil and gas in petroleum industry involve gas-liquid two phase flow. The two component (non-boiling) two phase flow is of prime interest in chemical and petroleum industries. In particular, correct knowledge and proper understanding of this non-boiling flow of gas and liquid phase is essential to solve flow assurance related problems some of which involve gas hydrates formation and wax deposition on the inner wall of pipes carrying natural gas and hydrocarbons. These residues in the pipe block the flow causing heavy loss to the oil and natural gas industry which is shown in McClaflin and Whitfill [1]. To address these issues and have a proper understanding of the heat transfer phenomenon in non-boiling (without phase change) two phase flow, it is very crucial to know the interrelationships between the two phase heat transfer and two phase flow parameters such as flow patterns, void fraction, phase flow rates, fluid properties and pipe orientation.

In comparison to horizontal and vertical pipe orientations, not much is known about the two phase heat transfer phenomenon in upward inclined systems. Some of the existing studies focused on understanding the two phase heat transfer phenomenon in upward inclined systems are those of Hestroni et al. [2], Ghajar and Tang [3], Kalapatapu et al. [4] and Lips and Meyer [5]. Hestroni et al. [2] measured two phase heat transfer

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coefficient in a 25 and 49.2 mm I.D. near horizontally oriented pipes ($0^o \leq \theta \leq +8^o$) for annular flow regime and found that the circumferential heat transfer coefficient depended on the local film thickness and wetted pipe perimeter. Specifically for the annular flow at low gas and liquid flow rates, they found a substantial difference between the pipe upper and lower wall heat transfer coefficient that corresponded to the local dry out conditions. Ghajar and Tang [3] measured two phase heat transfer coefficient in a 27.9 mm I.D. stainless steel pipe in near horizontal upward inclinations ($0^o \leq \theta \leq +7^o$) using air-water fluid combination. Their work focussed on all major flow patterns and concluded that the increase in pipe orientation from horizontal increases the two phase heat transfer coefficient. They also found that the change in physical structure of the flow pattern was strongly coupled to the trends of two phase heat transfer coefficients. Similar conclusions were reported by a recent study of Kalapatapu et al. [4] for two phase flow of air-water in a 12.5 mm I.D. near horizontal upward inclined pipe carrying a mixture of air and water. The experimental work of Lips and Meyer [5] was carried out in upward inclined ($0^o \leq \theta \leq +90^o$) 8.38 mm I.D. copper pipe carrying two phase flow of liquid R-134a and its vapor. They noticed a little increase in two phase heat transfer coefficient with increase in the upward pipe inclination from horizontal. However, at low mass flux and low quality, two phase heat transfer coefficient was observed to decrease slightly in near horizontal upward pipe inclinations ($\theta \leq +20^o$).

To contribute to the current state of knowledge, the objective of this work is to experimentally measure the two phase heat transfer coefficient in upward inclined pipe orientations using two phase flow of air and water. The measured values of two phase heat transfer coefficient (h_{TP}) are analysed as a function of flow patterns and pipe orientation. Moreover, the heat transfer coefficient in two phase flow is also compared with that in single phase flow. Additionally, this work also attempts to validate the performance of existing non-boiling two phase heat transfer correlations by comparing them with the experimentally measured data in this study. Finally, based on a statistical comparison, top performing correlations applicable for a range of flow patterns and pipe orientations are identified and recommended for use.

2. EXPERIMENTAL SETUP

The experimental setup as shown in Fig. 1 (designed by Cook [6]) is used for the measurements of two phase heat transfer coefficient. The experimental setup consists of a 12.5 mm I.D. schedule 10 S stainless steel pipe having a surface roughness of 0.0152 mm. The test section also consists of a 12.7 mm I.D. transparent pipe made up of polycarbonate material that runs parallel to the heat transfer section. This section is used for void fraction measurements and flow visualization. The test section consisting of both pipes is mounted on a variable inclination frame that can be inclined to any orientation between vertical upward and downward ($-90^o \leq \theta \leq +90^o$) with the help of pulleys. Distilled water and compressed air are used as fluid combination to generate the two phase flow. The liquid phase i.e., water is stored in a 50 gallon storage tank and is circulated through the flow loop using a Bell and Gosset (series 1535, model number 3445 D10) centrifugal pump. The liquid phase is first filtered through an AquaPure AP12-T purifier before it is passed through a shell and tube heat exchanger (ITT model BCF 4063). Later, it flows through the Emerson Coriolis mass flow meter (Micro Motion Elite Series model number CMF100) and then allowed to mix with air in a static mixer. The gas phase that consists of compressed air supplied by (Ingersoll Rand T-30 Model 2545) compressor is first passed through filter and regulator circuit and then through the helical coiled heat exchanger immersed in water tank. Later on it is fetched to the Micro Motion Elite Series Model LMF 3M and CMF 025 Coriolis gas mass flow meters where the mass flow rate of air is controlled precisely using a Parker (24NS 82(A)-8LN-SS) needle valve. The compressed air is then allowed to enter the test section through a Koflo model 3/8-40C-4-3V-23/8 static mixer. The mixer is mounted right before the entrance to the test section. The two phase inlet and outlet temperature is measured using Omega TMQSS-06U-6 thermocouple probes. These thermocouple probes 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. Twenty eight thermocouples (Omega TT-T-30 T-type) with accuracy of $\pm 1^oC$ are cemented at pipe outer wall at seven different axial locations spaced 12.7 cm apart along the pipe length. At each axial location, four thermocouples are arranged circumferentially 90^o apart (top, bottom and each side) to measure the local and circumferentially averaged temperature and hence the two phase heat transfer coefficient. A thick Micro-Lok Fibre Glass insulator is used to provide insulation to ensure minimal heat loss from the setup to the surrounding. Uniform wall heat flux is provided to the setup via a Lincoln DC-600 welder having a maximum current supply of 750 amps. The local inside wall temperature, heat flux and convective heat transfer coefficient is calculated using a finite difference formulation based data reduction program developed by Ghajar and Kim [7]. The convective heat transfer coefficient is represented by the average of the measured local values of h_{TP} at each thermocouple station. In order to maintain a constant temperature difference of about 4ºC between the inlet and outlet of the test section, a heat flux in the range of 9000 to 60,000 W/m² is provided to the setup. The uncertainties of the experimental data for single phase heat transfer coefficient are calculated using Kline and McClintock [8] uncertainty analysis method.

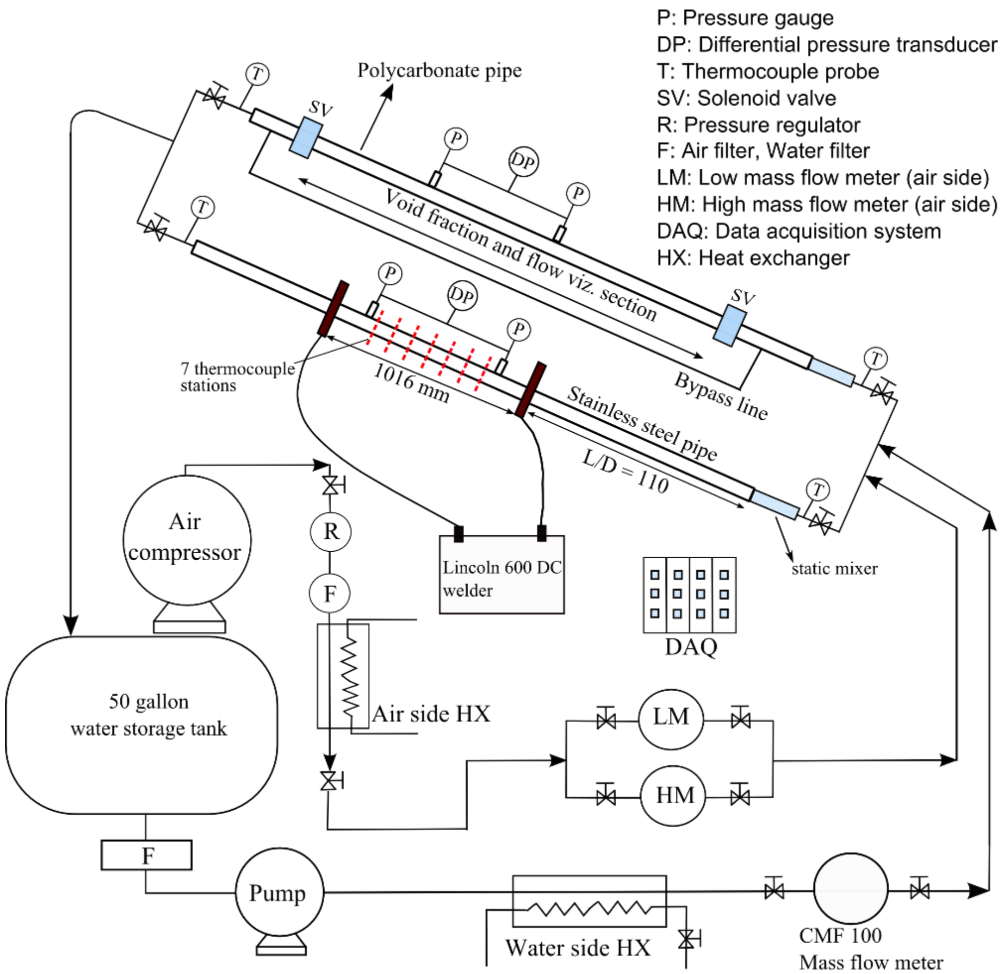

Fig. 1 Experimental setup used for non-boiling two phase heat transfer measurements (setup capable of moving in both upward and downward inclinations).

The uncertainty in measurement of two phase heat transfer coefficient and other associated two phase flow variables is shown in Table 1 for a sample run. The minimum and maximum uncertainty associated with each flow pattern is also reported in Table 2. The higher magnitude of uncertainty in stratified, intermittent and annular flows is due to inability to maintain a higher temperature difference across the pipe inlet and exit and the higher values of heat balance error associated with these flow patterns. As shown in Fig. 2, the accuracy of this experimental setup is also checked by comparing the measured single phase liquid heat transfer coefficient against the single phase heat transfer correlations of Dittus and Boelter [9], Ghajar and Tam [10], and Sieder and Tate [11].

Table 1: Uncertainty in measured values of two phase heat transfer coefficient (worst case scenario).

Variable	Value	± Uncertainty	± % Uncertainty
Inner pipe diameter (m)	0.01252	0.0000127	0.1
Outer pipe diameter (m)	0.0171	0.0000127	0.07
Heat transfer length (m)	0.889	0.003175	0.31
Thermal conductivity (W/m-K)	13.438	-	-
Current (I)	231.83	2.32	1.00
Voltage (V)	1.55	0.02	1.00
Inner wall temperature (^oC)	14.11	0.50	3.55
Heat transfer rate (W)	375.87	5.08	1.35
Heat transfer coefficient (W/m^2 K)	2998.31	907.12	30.25

Table 2: Minimum and maximum uncertainty in measured h_{TP} for different flow patterns.

Flow pattern	Minimum uncertainty (%)	Maximum uncertainty (%)	Average uncertainty (%)
Stratified	15.08	24.81	20.05
Slug	5.68	13.49	11.08
Intermittent	6.08	28.81	15.33
Bubbly	5.7	12.31	9.55
Annular	6.19	30.25	18.72

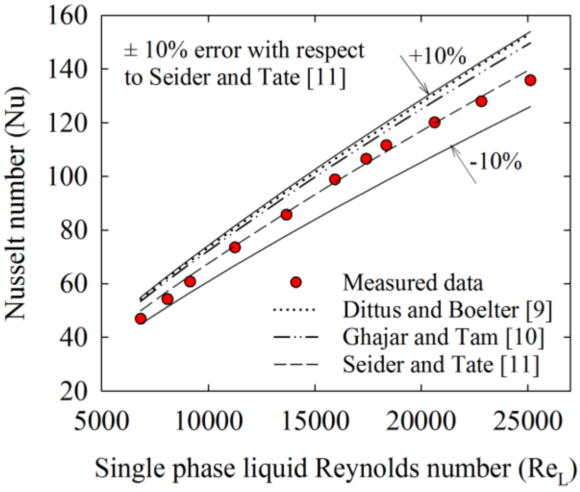

Fig. 2 Comparison between measured and predicted values of single phase heat transfer coefficients.

3. RESULTS AND DISCUSSION

Similar to any other two phase flow phenomenon, the heat transfer in gas-liquid two phase flow also depends to a great extent on the distribution of gas and liquid phase across the pipe cross section. This distribution of gas and liquid phase known as the flow pattern arises due to significant density difference between the two phases, compressibility nature of the gas phase and the interaction between body and surface forces. Before looking at the effect of flow pattern structure on two phase heat transfer coefficient, it is important to take a look at the key flow patterns observed in upward inclined gas-liquid two phase flows. In the present study, all major flow patterns i.e., bubbly, slug, intermittent and annular flow patterns are observed by varying the gas and liquid mass flow rates. A representative picture of these flow patterns is shown in Fig. 3. The two representative picture of the slug and intermittent flows are shown in Fig. 3 (a) and (b) to indicate variations in the physical appearance of the flow patterns during their onset and departure to other flow patterns.

The bubbly flow that appears at low gas and moderate to high liquid flow rates is characterized by the flow of small discrete gas bubbles dispersed in continuous liquid phase. The size, shape and pipe cross sectional distribution of these bubbles are a function of phase flow rates, pipe orientation and fluid properties. Due to asymmetry and buoyancy effects, these gas bubbles are always observed in the vicinity of the pipe upper wall. At low gas and low liquid flow rates, slug flow pattern is observed in form of elongated gas pockets of varying length (known as elongated bubbles) separated by liquid slugs while at moderate gas and liquid flow rates, these gas slugs are quite agitated with significant amount of aeration in liquid slugs. At fixed gas flow rate, the length and frequency of gas slugs is directly proportional to the liquid flow rate. Furthermore, the gas slug frequency is also proportional to the pipe orientation and hence the change in pipe orientation in slug flow regime also affects two phase flow parameters such as void fraction, pressure drop and heat transfer. Similar to bubbly flow, the gas slugs in horizontal and upward pipe inclinations are also oriented in the vicinity of the pipe upper wall. With increase in upward pipe inclination, the gas and liquid phase distribution across the pipe cross section becomes uniform and a symmetric two phase flow is observed in vertical upward pipe orientation. At moderate gas and liquid flow rates, the gas slug tends to disintegrate giving rise to the churning between the two phases and the two phase mixture appears frothy with increase in liquid flow rates. These flow patterns marked by vigorous mixing and intermittent flow tendency are sometimes identified as churn and froth flow in the two phase flow literature. However, considering the physical appearance and flow behavior of these flow patterns, in this study, we have combined these two flow patterns and have tagged it as intermittent flow. Finally, at low liquid and high gas flow rates, annular flow is observed. Annular flow is characterized by the flow of rough and wavy liquid film in contact with the pipe wall that surrounds the central moving gas core. The fast moving gas core shears the gas-liquid interface resulting into the entrainment of liquid droplets into the gas core. The liquid entrainment process decreases the liquid film thickness and hence affects the two phase heat transfer coefficient.

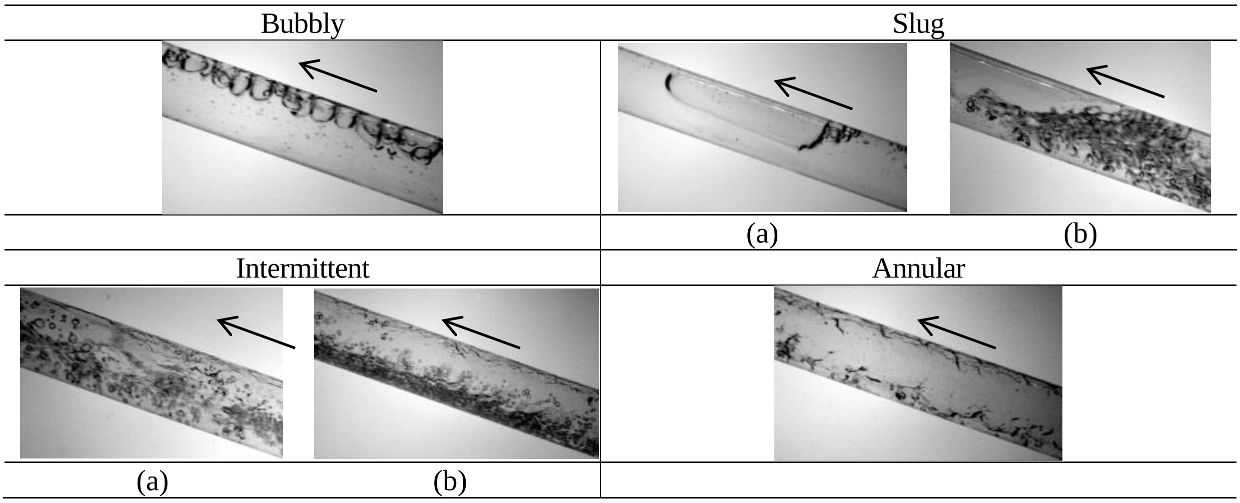

Fig. 3 Flow pattern observed in upward inclined two phase flow.

In the present study, two phase heat transfer coefficients are measured for all of these aforementioned major flow patterns in horizontal and upward pipe inclinations by varying gas and liquid mass flow rates (phase superficial Reynolds numbers) in a range of 0.001-0.2 kg/min ($270 \leq Re_{SG} \leq 19000$) and 1-10 kg/min ($2300 \leq Re_{SL} \leq 17000$), respectively. The superficial Reynolds numbers for gas and liquid flow rates are defined by Eqs. (1) and (2) where x is the two phase flow quality and G is the total two phase mixture mass flux. What follows is a brief description of the variation of h_{TP} as a function of flow patterns, phase flow rates and pipe orientations.

$$\mathrm{Re}_{SL} = \frac{\rho_L U_{SL} D}{\mu_L} = \frac{G(1-x)D}{\mu_L} \tag{1}$$

$$\mathrm{Re}_{SG} = \frac{\rho_G U_{SG} D}{\mu_G} = \frac{GxD}{\mu_G} \tag{2}$$

3.1 Effect of Flow Pattern and Phase Flow Rates

As mentioned earlier, the two phase heat transfer is bound to be influenced by the variation in physical structure of the flow patterns which in turn depends on the change in mass flow rates of individual phases. This variation of h_{TP} as a function of both flow patterns and gas and liquid flow rates expressed in terms of Re_{SG} and Re_{SL} is reported in Figs. 4 and 5. Although, these trends of h_{TP} are shown only for representative upward pipe inclinations of +30°, +60° and +90°, similar qualitative trends of h_{TP} vs. Re_{SL} and Re_{SG} are observed for all other pipe orientations. In general, it is found that at low liquid flow rates ($Re_{SL} \leq 6250$), for every pipe orientation, the two phase heat transfer coefficient is apparently insensitive to the increase in gas flow rates corresponding to $Re_{SG} \leq 2000$. A close observation shows that this region within the range of Re_{SL} and Re_{SG} values consists of slug flow pattern and transition region between slug and intermittent flow. In order to maintain the continuity, the slug flow in upward pipe inclinations is always accompanied by the partial flow of liquid film (surrounding the gas slug) in downward direction. Thus, the increase in two phase heat transfer coefficient with increase in the gas flow rates is possibly interfered by the downward falling liquid film which reduces the local (near pipe wall region) velocity of liquid phase and hence the heat transfer from pipe wall to the two phase mixture. It appears that a counter balance between these two phenomenon cause the two phase heat transfer coefficient to stay relatively independent of the increase in gas flow rates. Next, during the transition from slug to intermittent flow pattern (with increase in gas flow rate), the gas slug appears distorted (often wavy in nature) due to onset of its disintegration process and its length is increased considerably. Thus, for a fixed liquid flow rate, an increase in the gas slug length increases the effective length of the pipe occupied by the gas phase and hence inhibits a drastic increase in two phase heat transfer coefficient.

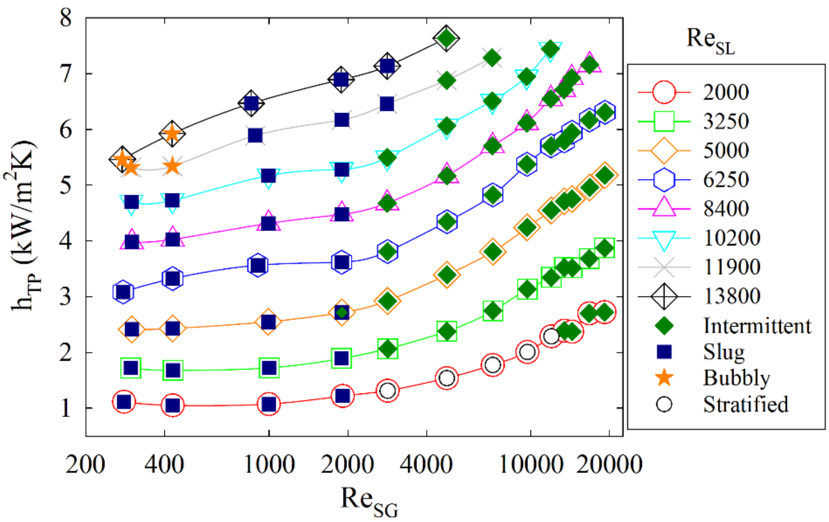

Fig. 4 Variation of h_{TP} with change in Re_{SL} and Re_{SG} in horizontal two phase flow.

For fixed liquid flow rates, as the gas flow rate is further increased such that $Re_{SG} \geq 5000$, the two phase heat transfer coefficient is found to increase sharply. In the present study, this region typically corresponds to intermittent and annular flow regimes. In particular, two different slopes of the increase in h_{TP} vs. Re_{SG} are observed during the onset of intermittent flow and during the departure from intermittent to annular flow pattern. After the onset of intermittent flow, increase in h_{TP} with respect to Re_{SG} is gradual while during the later transition, a rapid increase in h_{TP} is observed. These two different trends are essentially due to the difference in physical structure of the intermittent flow pattern at different gas flow rates. During the onset of intermittent flow (after the transition from slug flow at constant liquid flow rate), the flow pattern is wavy in nature with occasional elongated, aerated and distorted gas pockets flowing through the pipe. Whereas, during the end of intermittent flow regime or during its transition to annular flow regime, the intermittent flow structure is wavy in nature characterized by the formation of disturbance waves at the gas liquid interface. These disturbance waves create a sweeping action at the gas-liquid interface and also cause the two phases to mix and churn vigorously and consequently results into increased heat transfer coefficient. Note that in comparison to low liquid flow rate (Re_{SL} = 2000), the slope of increase of h_{TP} with increase in Re_{SG} is much steeper at higher liquid flow rates.

(a)

(b)

(c)

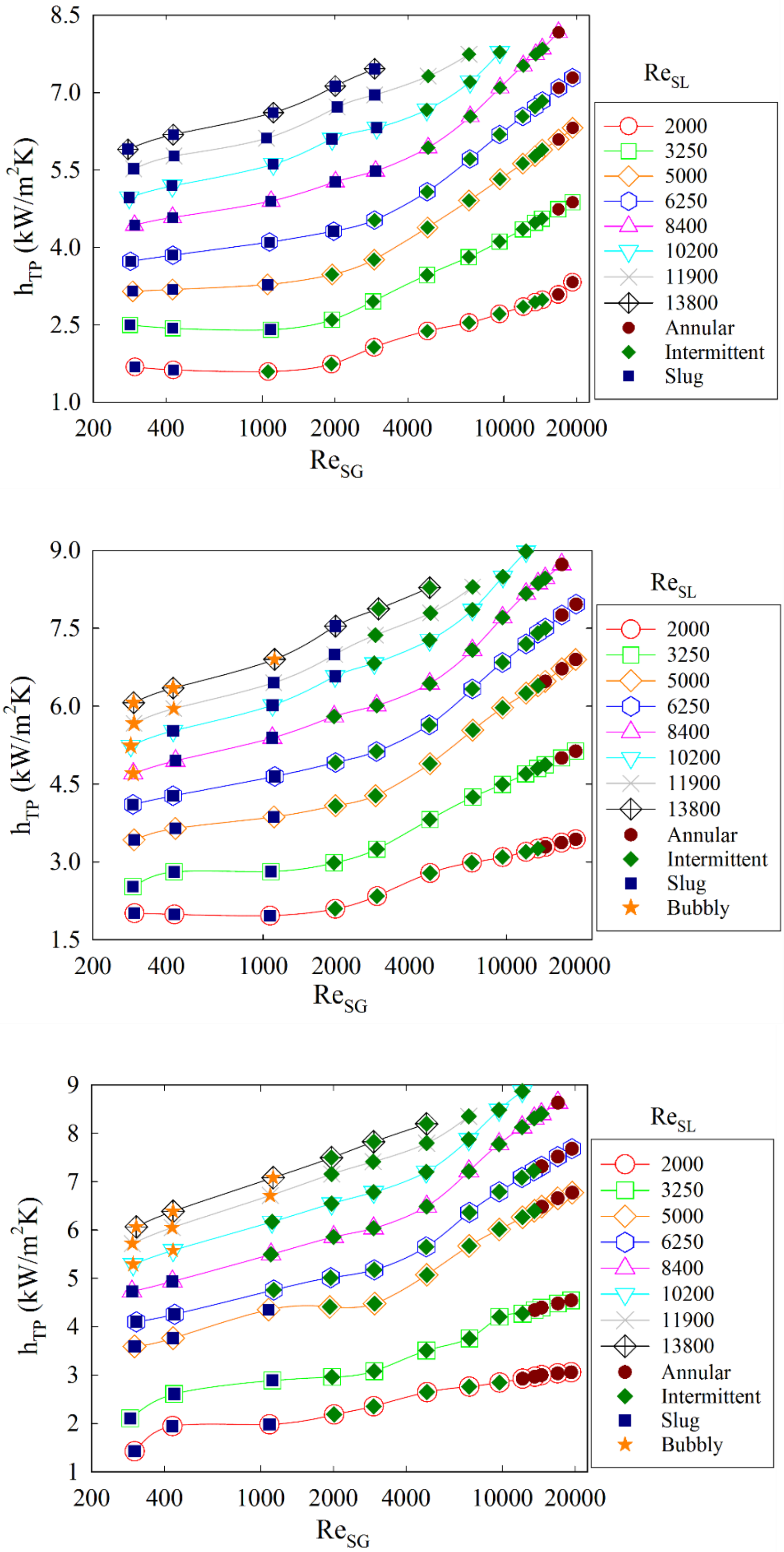

Fig. 5 Variation of two phase heat transfer coefficient with change in Re_{SL} and Re_{SG} in upward inclined two phase flow (a) θ = +30° (b) θ = +60° and (c) θ = +90°.

3.2 Effect of Pipe Orientation

The physical structure of two phase flow patterns is sensitive to the pipe orientation and hence the two phase heat transfer coefficient also depends upon the pipe orientation. The effect of pipe orientation on measured values of h_{TP} is depicted in Fig. 6. At low liquid and gas flow rates (or Re_{SL} and Re_{SG}), the two phase heat transfer coefficient is observed to increase with increase in the pipe orientation from horizontal. At low liquid flow rates and moderate to high gas flow rates, h_{TP} is first observed to increase from horizontal until up to $+45^o \leq \theta \leq +60^o$ and then reduce thereafter with further increase in the pipe inclination. This trend appears to be the consequence of the relative magnitude of buoyancy and gravitational forces that aids and opposes the motion of gas phase and hence the two phase mixture. For the slug and slug wavy (intermittent flow at low liquid and low gas flow rates) flows, this relative magnitude between these two forces is propositional to the pipe inclination. As suggested by Bendiksen [12], this proportionality is of the form $0.54 \cos(\theta) + 0.35 \sin(\theta)$ where θ is the pipe orientation measured from horizontal in upward direction. For fixed gas and liquid flow rates, this proportionality represents the translational velocity of the two phase mixture as a function of pipe orientation which attains a maximum in the vicinity of $+45^o$.

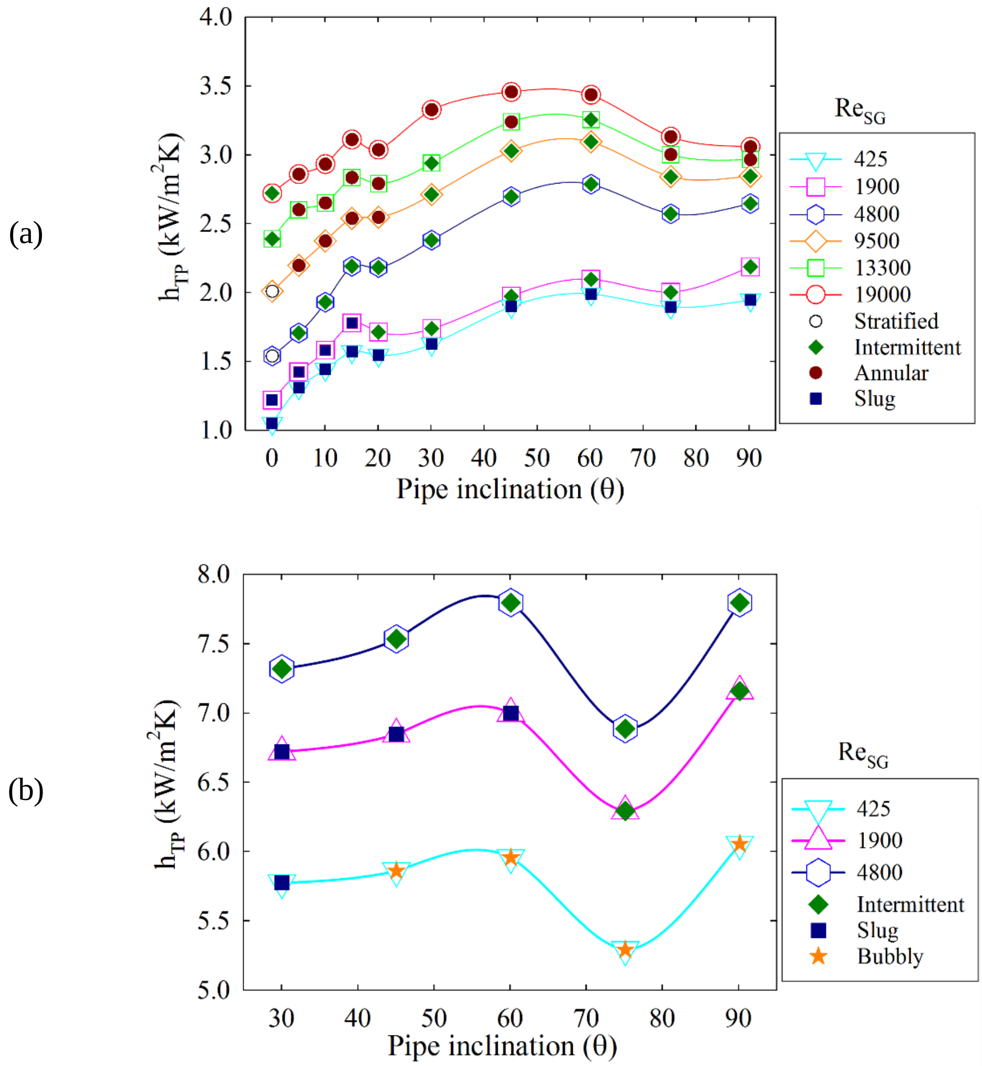

Fig. 6 Effect of pipe orientation on two phase heat transfer coefficient (a) Re_{SL} = 2000 (b) Re_{SL} = 12000.

This very similar increasing and decreasing tendency of two phase heat transfer coefficient is also observed in annular wavy (intermittent flow at low liquid and moderate to high gas flow rates) and annular flow regimes. At steeper pipe orientations (near vertical), the layer of liquid phase in contact with the pipe wall undergoes flow reversal under the influence of gravity. This situation is somewhat similar to that mentioned for a slug flow, however, it is an outcome of the prevalent hydrostatic forces compared to the shear forces at the gas liquid interface. Two phase flow literature reports that this type of a situation arises during the churn (intermittent flow in this study) flow and its transition to the annular flow. Experimentally, this flow reversal condition that inhibits the two phase heat transfer could be experimentally verified in terms of the decreasing trends of two phase pressure drop. Our past measurements of two phase pressure drop with this setup have shown the existence of flow reversal at $Re_{SL} \leq 4000$ and $Re_{SG} \geq 9000$. Considering the two phase flow physics, it appears that the decreasing tendency of the two phase heat transfer coefficient due to flow reversal may depend on the pipe geometry as well as the density of the liquid phase. More experimental data in this region for different pipe diameters, fluid combinations and the steeper pipe orientations is required to further verify the effect of flow reversal phenomenon on the two phase heat transfer coefficient in intermittent and during the intermittent-annular flow transition. At higher liquid flow rates, no significant effect of pipe orientation (with the exception of h_{TP} at $\theta = +75^{o}$) is observed on the two phase heat transfer coefficient. No visual evidence in terms of the physical structure of the flow pattern was observed at this pipe orientation that may support the anomalous behavior of h_{TP} at $+75^{o}$. Repeated experiments and measurements at these flow rates resulted in consistent h_{TP} values without any change in the overall trends. Conclusively more experimental work focused on near vertical pipe orientations is required to further validate these trends.

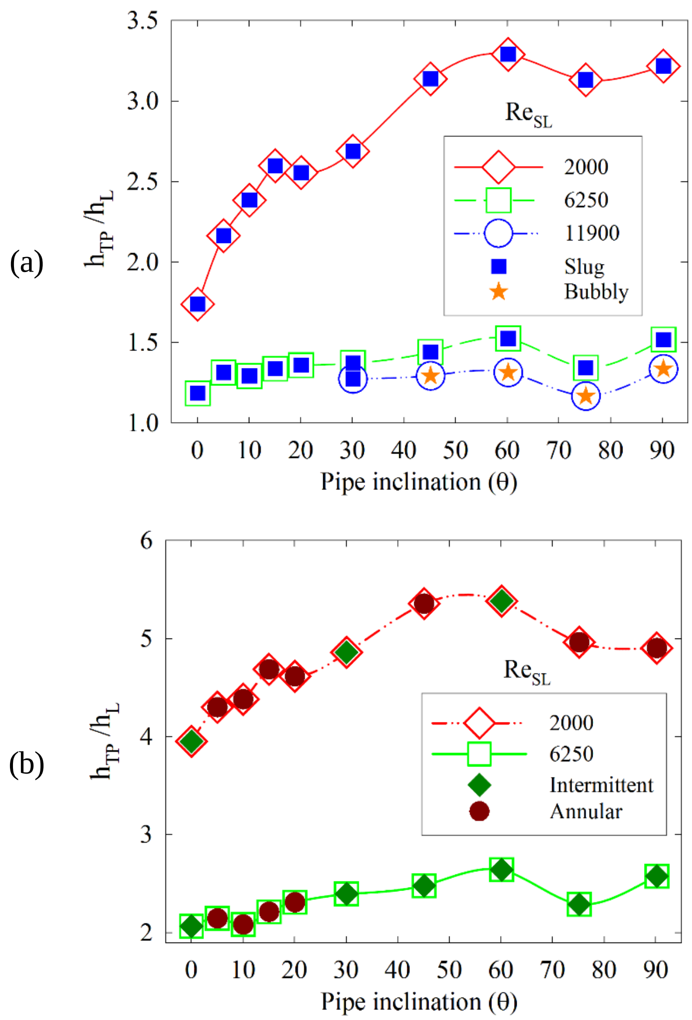

Fig. 7 Ratio of two phase to single phase heat transfer coefficient in upward pipe inclinations (a) $Re_{SG} = 420$ (b) $Re_{SG} = 13300$.

It is also of interest to check the magnitude of heat transfer coefficient in two phase flow with respect to that in single phase flow measured at equivalent liquid flow rates. The single phase heat transfer coefficient h_L is calculated assuming only G(1-x) amount of liquid mass flows through the pipe occupying the entire pipe cross section. The single phase superficial liquid Reynolds number using this mass flux is then used to calculate h_L using Sieder and Tate [11] correlation. As shown in Fig 7 (a) and (b), for a fixed gas flow rate, the ratio of h_{TP}/h_L is highest for lowest liquid gas flow rates. For all flow patterns and gas and liquid flow rates considered in this study the ratio of h_{TP}/h_L is always greater than unity with maximum amount of two phase heat transfer ($h_{TP}/h_L \approx 5$) obtained in annular flow with reference to that in single phase liquid flow. However, this gain in heat transfer is always accompanied by a substantial increase in the two phase frictional pressure drop compared to that in single phase flow.

4. PERFORMANCE ANALYSIS OF HEAT TRANSFER CORRELATIONS

As mentioned earlier, the two phase literature provides several correlations to predict non-boiling two phase heat transfer coefficient. However, these correlations are mostly developed and validated against data in horizontal and vertical pipe orientations and their accuracy for upward inclined two phase flow in not known. Thus, it is of interest to check the accuracy of these correlations against the experimental data for upward inclined two phase flow collected in this study. Based on the recommendations of Kim et al. [13], we have verified the performance of two phase heat transfer correlations [14-20] listed in Table 3.

Table 3 Two phase heat transfer correlations used in this study for performance analysis.

Sieder and Tate [11] (for single phase)	$h_L = 1.86\left(\mathrm{Re}_{SL}\,\mathrm{Pr}_L\,D/L\right)^{0.33}(k_L/D)\left(\mu_b/\mu_w\right)^{0.14}$ $\{\mathrm{Re}_{SL} \le 2000$ (Laminar) $h_L = 0.027\,\mathrm{Re}_{SL}^{0.8}\,\mathrm{Pr}_L^{0.33}(k_L/D)\left(\mu_b/\mu_w\right)^{0.14}$ $\{\mathrm{Re}_{SL} > 2000$ (Turbulent)
Aggour [14]	$h_{TP} = h_L(1-\alpha)^{-0.33}$ (Laminar) and $h_{TP} = h_L(1-\alpha)^{-0.83}$ (Turbulent)
Shah [15][1]	$h_{TP} = h_L \times \left(1 + U_{SG}/U_{SL}\right)^{0.25}$
Knott et al. [16]	$h_{TP} = h_L \times \left(1 + U_{SG}/U_{SL}\right)^{0.33}$
Kim and Ghajar [17][2]	$h_{TP} = h_L \times F_p\left(1 + 0.7\left[\left(\frac{x}{1-x}\right)^{0.08}\left(\frac{1-F_p}{F_p}\right)^{0.06}\left(\frac{\mathrm{Pr}_G}{\mathrm{Pr}_L}\right)^{0.03}\left(\frac{\mu_G}{\mu_L}\right)^{-0.14}\right]\right)$ $F_p = (1-\alpha) + \alpha\left[\frac{2}{\pi}\left(\tan^{-1}\sqrt{\frac{\rho_G(U_G-U_L)^2}{gD(\rho_L-\rho_G)}}\right)^2\right]$; $U_G = \frac{U_{SG}}{\alpha}$ and $U_L = \frac{U_{SL}}{1-\alpha}$
Tang and Ghajar [18][3]	$h_{TP} = h_L \times F_p\left(1 + 0.55\left[\left(\frac{x}{1-x}\right)^{0.1}\left(\frac{1-F_p}{F_p}\right)^{0.4}\left(\frac{\mathrm{Pr}_G}{\mathrm{Pr}_L}\right)^{0.25}\left(\frac{\mu_L}{\mu_w}\right)^{0.25} I^{0.25}\right]\right)$ $I = 1 + \frac{(\rho_L-\rho_G)gD^2\lvert\sin\theta\rvert}{\sigma}$
Chu and Jones [19]	$h_{TP} = (k_L/D)0.43\left(\mathrm{Re}_{SL}/(1-\alpha)\right)^{0.55}\mathrm{Pr}_L^{0.33}\left(\mu_b/\mu_w\right)^{0.14}\left(P_{atm}/P_{sys}\right)^{0.17}$
Martin and Sims [20]	$h_{TP} = h_L \times \left(1 + 0.64\sqrt{U_{SG}/U_{SL}}\right)$

[1] For Shah [15] consider turbulent flow if $Re_{SL} > 170$, [2] Use Chisholm [21] correlation to calculate void fraction required in F_p, [3] Use Woldesemayat and Ghajar [22] correlation to calculate void fraction, h_L is based on Sieder and Tate [11] with Re_{SL} replaced by $Re_L = Re_{SL}/(1-\alpha)^{0.5}$.

Note that the correlation of Aggour [14] uses two different correlations for laminar ($Re_{SL} \leq 2000$) and turbulent flows ($Re_{SL} > 2000$) and is based on the void fraction in given two phase flow conditions. The correlation of Shah [15] is a simple correlation based on the gas volumetric flow fraction (ratio of superficial gas velocity to the mixture velocity) and differentiates between laminar and turbulent flow using a threshold value of $Re_{SL} < 170$. The non-boiling to phase heat transfer correlation of Tang and Ghajar [18] uses in-situ Reynolds ($Re_L = Re_{SL}/(1-\alpha)^{0.5}$) number for calculation of liquid phase heat transfer coefficient. Also note that, Tang and Ghajar [18] specifically recommends use of Woldesemayat and Ghajar [22] correlation for the calculation of the void fraction required in calculation of Re_L.
The assessment of non-boiling two phase heat transfer correlations is carried out using statistical tools such as the mean absolute error, standard deviation and percentage of data points predicted within ±20% and ±30% error bands. The statistical parameters of average percentage error and standard deviation of percentage error are defined by Eqs. (3) and (4). The performance of any given correlation is called satisfactory if it can predict more than 80% and 90% of data points within ±20% and ±30% error bands, respectively. This is a custom definition of satisfactory performance based upon the accuracy of the correlations listed in Table 3. It is shown in the previous section that the two phase heat transfer coefficient is sensitive to the change in pipe orientation especially at low gas and liquid flow rates (or Re_{SG} and Re_{SL}). Thus, it is of interest to check the performance of the existing correlations and scrutinize their accuracy in prediction of two phase heat transfer correlation for a narrow range of two phase flow classified in terms of phase flow rates (or alternatively the flow patterns) and pipe orientations. For the purpose of the assessment of heat transfer correlations, the entire experimental data collected in this study is divided into three ranges of pipe orientations namely near horizontal (0°, +5°, +10° and +20°), intermediate (+30°, +45° and +60°) and near vertical (+75° and +90°). Each of these ranges of pipe orientations consists of all key flow patterns observed in this study.

$$\text{Average error} = \frac{1}{N}\sum_{j=1}^{N}\varepsilon_j \quad \text{where} \quad \varepsilon_j = \frac{h_{TP,cal} - h_{TP,\exp}}{h_{TP,\exp}} \times 100 \tag{3}$$

$$\%\,\text{Std. deviation} = \sqrt{\frac{1}{N-1}\sum_{j=1}^{N}\left(\varepsilon_j - \bar{\varepsilon}\right)^2} \times 100 \tag{4}$$

Preliminary analysis of the correlations listed in Table 3 showed that the correlations of Chu and Jones [19], Martin and Sims [20] did not perform satisfactorily (based on our criteria of satisfactory performance) for any of the flow patterns at any given pipe orientation and hence are not included in statistical analysis shown in Tables 4 to 7. Table 4 shows the performance of heat transfer correlations for individual and all flow patterns combined together in near horizontal pipe inclinations. Except for the intermittent flow pattern, Tang and Ghajar [18] is found to be the best performing correlations. Note that their correlation is developed based on the near horizontal and vertical upward pipe inclinations. The empirical constants used in their correlation are based on a majority of the data for bubbly, slug and annular flow patterns and hence probably does not give good accuracy when used against intermittent flow pattern. Nevertheless, their correlation still predicts 70.5% of data points within ±30% error bands.

Table 4 Performance of non-boiling two phase heat transfer correlations for $0^o \leq \theta < +30^o$.

Flow Pattern	All flow patterns				Bubbly		Slug		Intermittent		Annular	
No. of data points	250				7		107		77		59	
Correlations	(1)	(2)	(3)	(4)	(1)	(2)	(1)	(2)	(1)	(2)	(1)	(2)
Aggour [14]	24.6	38.5	-83.8	80.2	57.1	85.7	68.2	81.3	0	3.8	0	0
Shah [15]	68.3	80.8	-13.2	20.5	100	100	74.7	81.3	62.3	80.5	61	78.1
Knott et al. [16]	35.5	48.4	-41.3	36.4	100	100	57.1	67.2	24.6	37.6	3.3	22.3
Kim and Ghajar [17]	64.4	64.6	6.8	19	82.3	88.3	80.3	93.4	38.2	86.6	67.7	93.2
Tang and Ghajar [18]	64.9	90.9	-3.7	18.5	100	100	79.4	100	33.8	70.5	94.9	100

(1) % of data points predicted within ±20% error bands, (2) % of data points predicted within ±30% error bands, (3) average percentage error and (4) standard deviation of percentage error.

For the data in near horizontal upward pipe inclinations, Tang and Ghajar [18] has smallest average percentage error of -3.7% and the standard deviation of 18.5%. In intermittent flow regime, correlation of Shah [15] gives best accuracy by predicting 62.3% and 80.5% of data points within ±20% and ±30% error bands, respectively. For the intermediate range of pipe orientations reported in Table 5, Tang and Ghajar [18] appears to be the best performing correlation for all but intermittent flow pattern. In comparison to near horizontal pipe inclinations, the accuracy of Tang and Ghajar [18] for intermittent flow pattern is observed to increase for steeper pipe orientations. For all flow patterns considered together in $+30^o \leq \theta \leq +60^o$, both Knott et al. [16] and Tang and Ghajar [18] predict comparable number of data points within ±20% error bands essentially due to uneven number of data points in each flow pattern. However, the accuracy of Tang and Ghajar [18] in annular flow regime is much higher than Knott et al. [16] correlation. The correlation of Aggour [14] appears to work only for bubbly and slug flow patterns. Note that their correlations for laminar and turbulent flow of single phase liquid is based on the void fraction. Our preliminary analysis showed that this correlation is not very sensitive to the change in void fraction and any acceptable uncertainty/error (typically up to ±10%) involved in its measurement/calculation does not influence the predicted values of two phase heat transfer coefficient.

Table 5 Performance of non-boiling two phase heat transfer correlations for $+30^o \leq \theta \leq +60^o$.

Flow Pattern	All flow patterns				Bubbly		Slug		Intermittent		Annular	
No. of data points	257				18		75		131		33	
Correlations	(1)	(2)	(3)	(4)	(1)	(2)	(1)	(2)	(1)	(2)	(1)	(2)
Aggour [14]	35.1	38.1	-54.2	51.1	94.4	100	97.3	100	0	3.8	0	0
Shah [15]	88.7	100	10.2	8.6	100	100	86.6	100	90.1	100	81.8	100
Knott et al. [16]	70.8	86.7	-19.3	18.5	100	100	97.3	100	68.7	81	57.6	72.2
Kim and Ghajar [17]	41.6	60.3	24.5	15.3	94.4	100	73.3	100	3.8	22.1	90.9	100
Tang and Ghajar [18]	71.5	97.6	-3.8	15.4	100	100	97.3	100	46.5	95.4	96.9	100

(1) % of data points predicted within ±20% error bands, (2) % of data points predicted within ±30% error bands, (3) average percentage error and (4) standard deviation of percentage error.

Table 6 Performance of non-boiling two phase heat transfer correlations for $+60^o < \theta \leq +90^o$.

Flow Pattern	All flow patterns				Bubbly		Slug		Intermittent		Annular	
No. of data points	172				17		28		97		30	
Correlations	(1)	(2)	(3)	(4)	(1)	(2)	(1)	(2)	(1)	(2)	(1)	(2)
Aggour [14]	20.9	29.1	-31.1	51.7	47.1	88.2	78.5	85.7	6.1	11.4	0	0
Shah [15]	72.1	100	12.6	8.9	82.3	100	67.8	100	70.1	100	76.6	100
Knott et al. [16]	82.5	91.2	-5.6	17.9	100	100	92.8	100	86.6	92.7	50	73.3
Kim and Ghajar [17]	39.3	54.1	27.2	16.3	85.7	100	78.7	89.3	4.2	21.6	90	100
Tang and Ghajar [18]	69.1	94.7	-9.1	15.2	100	100	85.7	100	51.5	90.7	93.3	100

(1) % of data points predicted within ±20% error bands, (2) % of data points predicted within ±30% error bands, (3) average percentage error and (4) standard deviation of percentage error.

Table 6 shows the performance of the heat transfer correlations for individual flow patterns in near vertical pipe orientations. Similar to the other two ranges of pipe inclinations, Tang and Ghajar [18] is the top performing correlation for all flow patterns except for the intermittent flow. The overall accuracy of Knott et al. [16] correlation is observed to increase with increase in pipe orientation from horizontal. Although the Knott et al. [16] correlation appears to be the top performing correlation for near vertical pipe orientations, its accuracy in annular flow regime is significantly less than that of Kim and Ghajar [17] and Tang and Ghajar [18] correlations. Finally, for the entire data shown in Table 7, Tang and Ghajar [18] and Shah [15] are the top performing correlations that predict 94.3% and 92.9% of data points, respectively within ±30% error bands. These correlations are recommended to predict non-boiling two phase heat transfer coefficient in horizontal and upward inclined pipe orientations. For bubbly flow pattern, any of the Shah [15], Knott et al. [16] and Tang and Ghajar [18] correlations can be used to predict two phase heat transfer coefficient. For slug and annular flow patterns, Tang and Ghajar [18] correlation gives the best accuracy and hence is recommended to be used for horizontal and the entire range of upward pipe inclinations. For intermittent flow, correlation of Shah [15] gives highest accuracy and may be used to predict heat transfer coefficient for the entire range of upward pipe inclinations.

Table 7 Performance of non-boiling two phase heat transfer correlations for $0^o \leq \theta \leq +90^o$.

Flow Pattern	All flow patterns				Bubbly		Slug		Intermittent		Annular	
No. of data points	679				42		210		305		122	
Correlations	(1)	(2)	(3)	(4)	(1)	(2)	(1)	(2)	(1)	(2)	(1)	(2)
Aggour [14]	29.7	34.6	-59.1	66.8	71.4	90.4	79.5	87.6	1.6	4.3	0	0
Shah [15]	77.1	92.9	2.19	18.5	95.2	100	78.1	90.4	76.7	95.1	70.4	89.3
Knott et al. [16]	63.7	73.8	-20.1	31.1	100	100	76.1	83.3	61.9	72.7	34.6	51.3
Kim and Ghajar [17]	47.6	65.8	18.7	19.2	88.1	92.8	77.6	95.2	13.7	35.4	66.7	82.1
Tang and Ghajar [18]	75.6	94.3	-6.5	16.7	100	100	86.2	100	48.1	87.2	94.2	100

(1) % of data points predicted within ±20% error bands, (2) % of data points predicted within ±30% error bands, (3) average percentage error and (4) standard deviation of percentage error.

Note that although the correlation of Shah [15] works for the entire range of pipe orientation, its physical form does not contain the pipe orientation term to account for the effect of pipe orientation on two phase heat transfer coefficient and hence this correlation does not necessarily adheres to the two phase flow physics. In comparison to Shah [15], the correlation of Tang and Ghajar [18] contains a pipe inclination term (I) that considers the variation of h_{TP} as a function of pipe orientation. Their correlation also accounts for the change in physical structure of the two phase flow patterns through an empirical parameter in terms of flow pattern factor (F_p). The correlation of Tang and Ghajar [18] contains several adjustable empirical parameters that may be tweaked to improve its accuracy in the intermittent flow regime. Our preliminary analysis shows that by changing the leading multiplying factor (see Table 3) of 0.55 to 0.7 in their correlation, a better accuracy is achieved in intermittent flow regime. With this change in multiplying factor, for the entire range of upward pipe inclinations, Tang and Ghajar [18] predicts 87.2% and 99.3% of data points within ±20% and ±30% error bands, respectively. Thus, if the existence of intermittent (slug wavy/ wavy/ annular wavy/ churn/ froth) is known to the user, then a multiplying factor of 0.7 instead of 0.55 can be used to accurately calculate the two phase heat transfer coefficient using Tang and Ghajar [18] correlation.

5. CONCLUSIONS

New data on non-boiling two phase heat transfer coefficient for the entire range of upward inclined two phase flow is presented in this study. The measured values of two phase heat transfer coefficient are found to be influenced by the change in physical structure of the flow patterns (due to change in gas and liquid flow rates) and pipe orientations. In particular, the two phase heat transfer coefficient is sensitive to the change in pipe orientation at low gas and liquid flow rates. The trend of h_{TP} vs. pipe orientation showed an increasing and decreasing trend with maxima observed in the range of $+45^o \leq \theta \leq +60^o$. With increase in liquid flow rates, the dependency of h_{TP} on pipe orientation gradually decreased. The heat transfer coefficient in two phase flow is also compared with that in single phase flow calculated at equivalent mass flow rates. The maximum ratio of h_{TP}/h_L is obtained for low liquid flow rates and particularly in annular flow regime. The non-boiling two phase heat transfer correlations available in literature are compared with the measured values of h_{TP}. The comparisons are made for three different ranges of the pipe orientations and different flow patterns. The correlation of Tang and Ghajar [18] is found to predict the two phase heat transfer coefficient satisfactorily for all flow patterns except for the intermittent flow. For the entire data set, the correlations of Tang and Ghajar [18] and Shah [15] are found to be the top performing correlations. More experimental data is required to further validate the accuracy of these top performing correlations for more diverse two phase flow conditions.

NOMENCLATURE

D	pipe diameter	(m)
F_p	flow pattern factor	(-)
g	acceleration due to gravity	(m/s^2)
G	mass flux	(kg/m^2s)
h	heat transfer coefficient	(W/m^2K)
I	inclination factor	(-)
k	thermal conductivity	(W/mK)
L	Pipe length	(m)
N	no. of data points	
Nu	Nusselt number	(-)
P	pressure	(Pa)
Pr	Prandtl number	(-)
Re	Reynolds number	(-)
U	phase velocity	(m/s)
x	two phase flow quality	(-)

Greek symbols

α	void fraction	(-)
ε	error	(-)
μ	phase dynamic viscosity	(Pa-s)
ρ	phase density	(kg/m^3)
σ	surface tension	(N/m)
θ	pipe orientation	(deg.)

Subscripts

atm	atmospheric
b	bulk
cal	calculated
exp	experimental
G	gas
L	Liquid phase
sys	system
S	Superficial
TP	two phase
w	pipe wall

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