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Correlation of heat transfer coefficient for two-component two-phase slug flow in a vertical pipe

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ABSTRACT

The prediction of heat transfer coefficient for gas–liquid two-phase flows in vertical pipes is important for designing many industrial and engineering apparatuses such as nuclear power plants, solar collectors and petroleum pipelines. The mechanism of two-component two-phase heat transfer is more complicated than that of single-phase heat transfer. This study aims at developing a robust correlation of heat transfer coefficient for two-component two-phase slug flows in vertical pipes. The theoretical framework for the correlation development is inspired by considering classic Reynolds and Chilton–Colburn analogies. The framework demonstrates that the functional dependence of the two-component two-phase heat transfer coefficient on a void fraction or a two-phase multiplier is expressed by the Reynolds and Chilton–Colburn analogies. The correlation for the heat transfer coefficient of two-component two-phase slug flows in vertical pipes has been developed and validated by more than 200 existing data. The newly developed correlation predicts 95.1% of the collected two-phase heat transfer data within ± 30% error and the mean absolute relative deviation of the correlation is estimated to be 14.2%. The extended applicability of the newly developed correlation to other flow regimes such as bubbly, churn and annular flow regimes has been verified by the existing experimental data. The possible application of the newly developed correlation to large pipes is also discussed. In summary, it is expected that the newly developed correlation can predict the heat transfer coefficient of two-component two-phase flows under various conditions such as vertical upward and downward flows, developing and fully developed flows, laminar and turbulent flows and all two-phase flow regimes. The application of the correlation to vertical large pipes is also considered to be reasonable.

1. Introduction

Gas–liquid two-phase flows in vertical pipes are often encountered in many industrial and engineering applications such as nuclear power plants, solar collectors, and petroleum pipelines. Various flow regimes such as bubbly, slug, churn and annular flows appear depending on gas and liquid flow rates (Ishii and Hibiki, 2010). Among them, slug flow dynamics are complicated due to an intermittent flow of Taylor bubbles and liquid slugs. The heat transfer enhancement technique using adiabatic (or non-boiling) two-phase slug flow, especially, in micro-channels is receiving more attention in recent years (Kim et al., 2012; Hetsroni et al., 2009). Continuous efforts to understand and predict the basic flow and heat transfer characteristics of the slug flow have been made and several empirical correlations for the heat transfer coefficient of two-component two-phase slug flows in vertical pipes have been proposed. The heat transfer enhancement mechanism in two-component two-phase slug flows in vertical pipes has been studied for many years and several hypotheses have been proposed to figure out the appropriate mechanism to correlate the two-phase heat transfer coefficient.

Elamvaluthi and Srinivas (1984) and Chu and Jones (1980) hypothesized that the heat transfer mechanism of a two-phase flow was the same as that of a single-phase flow and the two-phase heat transfer enhancement was attributed to the increase of the mixture velocity. Based on this hypothesis, some two-phase heat transfer correlations for vertical two-phase flows were developed by modifying single-phase heat transfer correlations. An acceleration model for a liquid phase considered that the liquid phase was accelerated by a gas phase in a gas–liquid two-phase flow. Since the thermal conductivity of the liquid phase is much higher than that of the gas phase, the two-phase heat transfer coefficient was assumed to be the same as the heat transfer coefficient of the liquid phase flowing with the actual liquid velocity. Based on the acceleration model, Rezkallah and Sims (1989) estimated an actual liquid velocity with a superficial liquid velocity di-

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Nomenclature

a, A	empirical constant [-]
b	empirical constant [-]
C	Chisholm constant [-]
C_f,	friction coefficient [-]
C_0	distribution parameter [-]
D	diameter m
D_H	hydraulic equivalent diameter m
D_{SM}	bubble Sauter mean diameter m
f	fanning friction factor [-]
Fr	Froude number [-]
G	mass flux kg/(m$^2 \cdot$ s)
h	heat transfer coefficient W/(m$^2 \cdot$ K)
j	superficial velocity m/s
k	exponent [-]
L	length m
m	exponent [-]
m_d	mean error [-]
m_{rel}	mean relative deviation [-]
$m_{rel,ab}$	mean absolute relative deviation [-]
n	exponent [-]
N	number of samples [-]
Nu	Nusselt number [-]
p	pressure Pa
Pr	Prandtl number [-]
Q	flow rate m^3/s
Re	Reynolds number [-]
R_f	liquid holdup [-]
s_d	standard deviation [-]
T	temperature K
v	velocity m/s
V_{gj}	drift velocity m/s
x	mass fraction [-]
X	Lockhart–Martinelli parameter [-]
y	direction perpendicular to a wall [-]
z	axial direction [-]
Greek symbols	
α	void fraction [-]
μ	dynamic viscosity Pa $\cdot$ s
ρ	density kg/m^3
$\Delta\rho$	density difference kg/m^3
σ	surface tension N/m
τ	shear stress Pa
Φ_f^2	two-phase multiplier [-]
ν	kinematic viscosity m^2/s
Subscripts	
B	bulk [-]
cal	calculated [-]
exp	experimental [-]
f, 1ϕ	liquid [-]
F	friction [-]
g	gas [-]
m	gas–liquid mixture [-]
2ϕ	gas–liquid two-phase [-]
W, w	wall [-]
Superscripts	
*	non-dimensional [-]

vided by a liquid hold-up and developed a heat transfer correlation for two-phase flows in vertical pipes. Muzychka et al. (2011) and Talimi et al. (2012) hypothesized that the heat transfer enhancement of two-phase slug flows was attributed to the internal circulations inside the moving slugs.

Although several empirical heat transfer correlations for two-component two-phase slug flows in vertical pipes have been proposed, most of the empirical correlations were developed based on limited experimental data and the applicable range of the correlations was limited. In view of these, it is necessary to develop an accurate and robust heat transfer correlation of two-component two-phase slug flows in vertical pipes with a wide range of applicability, and the correlation to be developed should be supported by a physical two-phase heat transfer enhancement mechanism to be verified by experimental data.

This study is aiming at developing an accurate and robust heat transfer correlation for two-component two-phase slug flows in vertical pipes with a wide range of applicability. In order to develop the correlation successfully, an extensive literature survey is first conducted to collect more than 200 experimental data from 13 different sources with a variety of fluid systems, pipe diameters, and flow conditions. The literature survey also collects 22 existing two-component two-phase heat transfer correlations including 14 correlations for vertical pipes and 8 correlations for horizontal pipes. This study demonstrates that none of the existing correlations can predict the two-component two-phase heat transfer coefficient in the range of the whole database with a sufficient prediction accuracy. The development of the two-component two-phase heat transfer coefficient correlation in vertical pipes is performed based on the concept of classic Reynolds and Chilton–Colburn analogies. The newly developed correlation is verified by the existing data taken in a wide range of test conditions. The extended applicability of the correlation to other flow regimes such as bubbly, churn and annular flows and to large pipes is discussed.

2. Review of existing data and correlations and assessment of applicability of existing correlations

2.1. Existing data on heat transfer coefficient for two-component two-phase slug flows in vertical pipes

This section identifies more than 200 existing experimental data of two-component two-phase slug flows in vertical pipes from 13 different sources. The fluid systems, geometrical information of test section and flow conditions are detailed in Table 1 and the physical properties of the fluid systems are given in Table 2. The two-phase fluid systems in the collected database include six different systems such as air-water, air–oil, air–glycerine, helium–water, Freon–water and air–ethylene glycol mixtures. The pipe diameter, D, ranges from 1.17 to 7.00 cm. The threshold value between medium and large pipes is given by $D^* = 40$, where D^* is the non-dimensional pipe diameter defined by Eq. (1) (Ishii and Hibiki, 2010).

$$D^* \equiv \frac{D}{\sqrt{\frac{\sigma}{g\Delta\rho}}} \tag{1}$$

where σ, g, and $\Delta\rho$ are the surface tension, gravitational acceleration and density difference between two phases, respectively. The denominator on the right-hand side of Eq. (1) is Laplace length scale, which characterizes the bubble length scale. As given in Table 2, a preliminary analysis calculating the non-dimensional pipe diameter suggests that all collected data are categorized as the data taken in medium pipes where slug bubbles are formed. When the non-dimensional pipe diameter exceeds 40, slug bubbles cannot exist due to their surface instability and are disintegrated into small cap bubbles (Hibiki and Ishii, 2003). The two-phase flow characteristics in large diameter pipes are different from these in medium pipes. It should be noted here that the collected database include upward two-phase flows as well as downward two-phase

Table 1
Summary of heat transfer coefficient database for two-component two-phase slug flows in vertical pipes.

Sources	D/U	Fluids	Diameter [cm]	L/D [-]	j_g [m/s]	j_f [m/s]	Re_g [-]	Re_f [-]
Kudirka et al. (1965)	U	Air–water	1.59	17.5	1.05–21.7	0.300–1.37	–	–
Dorresteijn (1970)	U	Air–oil	7.00	15.7	0.280–8.13	1	–	–
Aggour (1978)	U	Freon–water	1.17	52.1	0.399–1.58	0.314–1.04	1.89×10^3–7.83×10^3	4.19×10^3–1.35×10^4
Aggour (1978)	U	Air–water	1.17	52.1	0.558–2.50	0.314–2.12	4.54×10^2–2.10×10^3	4.16×10^3–3.68×10^4
Aggour (1978)	U	Helium-water	1.17	52.1	1.01–6.11	0.314–1.04	1.03×10^2–6.56×10^2	4.01×10^3–1.43×10^4
Vijay (1978)	U	Air–glycerin	1.17	52.2	0.0700–2.28	0.0900–1.16	6.30×10^1–2.90×10^3	1.85–2.13×10^1
Vijay (1978)	U	Air–water	1.17	52.2	0.150–2.88	0.0200–0.314	1.18×10^2–2.173×10^3	2.65×10^2–3.32×10^4
Chu (1980)	U/D	Air–water	2.67	34.0	0.420–3.24	0.420–1.91	–	–
Rezkallah and Sims (1989)	U	Air–water	1.17	26.0	0.530–3.00	0.0560–0.314	–	–
Rezkallah and Sims (1989)	U	Air–glycerin	1.17	26.0	0.210–3.07	0.0220–0.997	–	–
Wang (2006)	U	Air–oil	2.60	181	2.50–9.30	0.390–1.88	–	–
Wang (2006)	U	Air–water	2.60	181	1.40–13.3	0.190–1.70	–	–
Bhagwat and Ghajar (2017)	D	Air–water	1.25	81.0	–	–	2.80×10^2–1.40×10^3	4.70×10^3–1.20×10^4

Notes: D and U represents downward upward two-phase flow, respectively.

Table 2
Physical properties of fluid systems in collected database.

Sources	σ_f [mN/m]	ρ_g [kg/m^3]	ρ_f [kg/m^3]	μ_g [Pa$\cdot$s $\times 10^6$]	μ_f [Pa$\cdot$s $\times 10^6$]	D^* [-]	N_μ [-]
Kudirka et al. (1965)	72.0	1.19	997	18.3	903	5.85	0.00463
Dorresteijn (1970)	–	1.19	750	18.8	52.5	–	–
Aggour (1978)	72.0	5.40	997	–	903	4.30	0.00204
Aggour (1978)	72.0	1.31	997	18.4	903	4.31	0.00204
Aggour (1978)	72.0	0.180	997	18.9	903	4.31	0.00205
Vijay (1978)	63.3	1.19	1263	18.3	60,000	5.17	0.141
Vijay (1978)	72.0	1.19	997	18.3	894	4.31	0.00203
Chu (1980)	72.0	1.19	997	18.3	894	9.83	0.00203
Rezkallah and Sims (1989)	72.0	1.19	997	18.3	894	4.31	0.00203
Rezkallah and Sims (1989)	67.0	1.19	1136	18.3	5100	4.77	0.0118
Wang (2006)	22.2	1.03	867	20.4	9760	16.1	0.0553
Wang (2006)	66.2	1.03	983	20.4	550	9.91	0.00133
Bhagwat and Ghajar (2017)	72.8	1.19	998	18.3	1005	4.58	0.00226

N_μ is a viscous number defined by $N_\mu \equiv \mu_f / (\rho_f \sigma \sqrt{\sigma/(\Delta\rho g)})$.

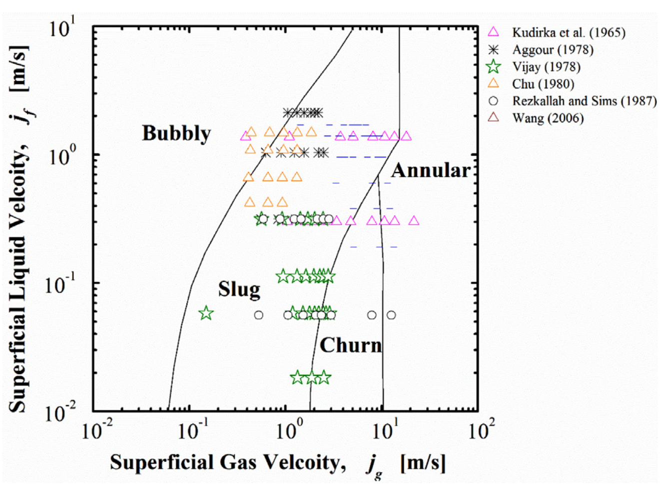


Fig. 1. Flow conditions of collected upward air-water flow database in flow regime map.

flow. The test section length, L, ranges from 17.5 to 181D, which indicates that the existing data are collected in both developing and fully developed flows.

The ranges of the superficial gas velocity, j_g, and the superficial liquid velocity, j_f, are from 0.0700 to 21.7 m/s and from 0.0200 to 2.12 m/s, respectively, which indicates that the existing data are collected in both laminar and turbulent flows of gas and liquid phases. Fig. 1 shows the flow conditions of the collected upward air-water flow database in a plane of j_g vs. j_f. Black solid lines are flow regime transition boundaries predicted by the flow regime transition criteria for an upward air-water flow in a vertical medium pipe with the inner diameter of 25.4 mm under atmospheric pressure condition (Mishima and Ishii, 1984). Symbols show the database taken by different researchers, and the figure indicates that most of the collected data are taken in a slug flow regime with a wide range of the superficial gas and liquid velocities. The rest of the collected data (air-oil, Freon–water, helium–water, air–glycerin and downward air–water flows) are taken in slug flow regime. This is stated by the researches who took each data set.

In summary, the collected database include vertical upward and downward flow data, developing and fully developed flow data and laminar and turbulent flow data. It is also emphasized here that this study is focused on developing a correlation of heat transfer coefficient for two-component two-phase slug flows in "medium" vertical pipes.

2.2. Existing correlations of heat transfer coefficient for two-component two-phase slug flow in pipes

This section performs an extensive literature review on existing correlations of a heat transfer coefficient for two-component two-phase slug flows in vertical as well as horizontal pipes. Tables 3 and 4 summarize the collected existing correlations for vertical and horizontal pipes, respectively. In total, 22 correlations of heat transfer coefficients are identified, including 14 correlations for vertical pipes and 8 correlations for horizontal pipes. Tables 3 and 4 indicate that the ratio of two-phase flow heat transfer coefficient (or two-phase Nusselt number), $h_{2\phi}$ (or $Nu_{2\phi}$), to single-phase heat transfer coefficient (or single-phase Nusselt number), $h_{1\phi}$ (or $Nu_{1\phi}$), is correlated by some dominant parameters speculated by researchers such as two-phase multiplier, Φ_f^2, void fraction, α, Reynolds number, Re, Prandtl number, Pr, superficial gas–

Table 3
Summary of existing heat transfer coefficient correlations for two-component two-phase slug flows in vertical pipe.

Sources	Two-phase correlations	Single-phase correlations
Groothuis and Hendal (1959)	$Nu_{2\phi} = 0.029Re_{tp}^{0.87}Pr_f^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$(A/W) $Nu_{2\phi} = 2.6Re_{tp}^{0.39}Pr_f^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$(A/G)	
Knott et al. (1959)	$\frac{h_{2\phi}}{h_{1\phi}} = (1+\frac{j_g}{j_f})^{1/3}$	$Nu_{1\phi} = 1.86(Re_f Pr_f \frac{D}{L})^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$ (L) $Nu_{1\phi} = 0.027Re_f^{0.8}Pr_f^{0.33}(\frac{\mu_B}{\mu_W})^{0.14}$ (T)
Kudirka et al. (1965)	$Nu_{2\phi} = 125(\frac{j_g}{j_f})^{0.125}(\frac{\mu_g}{\mu_f})^{0.6}Re_f^{0.25}Pr_f^{1/3}$ $\times(\frac{\mu_B}{\mu_W})^{0.14}$	
Ueda and Hanaoka (1967)	$Nu_{2\phi} = 0.075Re_m^{0.60}\frac{Pr}{1+0.35(Pr-1)}$ $Re_m = \frac{\rho j_m D}{\mu_f}$	
Dorresteijn (1970)	$\frac{h_{2\phi}}{h_{1\phi}} = (1-\alpha)^{-1/3}$ (L) $\frac{h_{2\phi}}{h_{1\phi}} = (1-\alpha)^{-0.8}$ (T)	$Nu_{1\phi} = 0.0123Re_f^{0.9}Pr_f^{0.33}(\frac{\mu_B}{\mu_W})^{0.14}$
Fedotkin and Zarudnev (1970)	$Nu_{2\phi} = 0.0182Re_m^{0.882}Pr_f^{0.43}(\frac{Pr_{f,B}}{Pr_{f,W}})^{0.25}Re_m = \frac{\rho_f j_f D}{\mu_f} + \frac{\rho_g j_g D}{\mu_g}$	
Aggour (1978)	$\frac{h_{2\phi}}{h_{1\phi}} = (1-\alpha)^{-1/3}$ (L), $\frac{h_{2\phi}}{h_{1\phi}} = (1-\alpha)^{-0.83}$ (T),	$Nu_{1\phi} = 1.615(Re_f Pr_f \frac{D}{L})^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$ (L) $Nu_{1\phi} = 0.0155Re_f^{0.83}Pr_f^{0.5}(\frac{\mu_B}{\mu_W})^{0.33}$ (T)
Chu and Jones (1980)	$Nu_{2\phi} = cRe_{tp}^{0.55}Pr_f^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}(\frac{Pa}{P})^{0.17}$, $c = 0.43$ (Upward), $c = 0.47$ (Downward)	
Vijay et al. (1982)	$\frac{h_{2\phi}}{h_{1\phi}} = \phi_f^{0.489}$,	$Nu_{1\phi} = 0.0155Re_f^{0.83}Pr_f^{0.5}(\frac{\mu_B}{\mu_W})^{0.33}$ (L) $Nu_{1\phi} = 1.615(Re_f Pr_f \frac{D}{L})^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$ (T)
Elamvaluthi and Srinvivas (1984)	$Nu_{2\phi} = 0.5(\frac{\mu_g}{\mu_f})^{0.25}Re_m^{0.7}Pr_f^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$ $Re_m = \frac{\rho_f j_f D}{\mu_f} + \frac{\rho_g j_g D}{\mu_g}$	
Rezkallah and Sims (1989)	$\frac{h_{2\phi}}{h_{1\phi}} = (1-\alpha)^{-0.8}$,	$Nu_{1\phi} = 0.023Re_f^{0.8}Pr_f^{0.33}(\frac{\mu_B}{\mu_W})^{0.14}$
Ghajar and Tang (2010)	$\frac{h_{2\phi}}{h_{1\phi}} = F_p[1+0.55(\frac{x}{1-x})^{0.1}(\frac{1-F_p}{F_p})^{0.4}(\frac{Pr_G}{Pr_L})^{0.25}(\frac{\mu_L}{\mu_G})^{0.25}(I^*)^{0.25}]$	
Bhagwat and Ghajar (2017)	$\frac{h_{2\phi}}{h_{1\phi}} = \phi_f^{0.55}$,	$Nu_{1\phi} = 0.027Re_f^{0.8}Pr_f^{1/3}(\frac{\mu_B}{\mu_W})^{0.13}$

Notes: 1. L and T represent laminar flow ($Re_f \le 2300$) and turbulent flow ($Re_f > 2300$), respectively.
2. A/W represents air and water mixture and A/G represents air and gas–oil mixture.

Table 4
Summary of existing heat transfer coefficient correlations for two-component two-phase slug flows in horizontal pipes.

Sources	Two-phase correlations	Single-phase correlations
King (1952)	$\frac{h_{2\phi}}{h_{1\phi}} = \frac{(1-\alpha)^{-0.52}}{1+0.025Re_f}(\phi_f^2)^{0.32}$,	$Nu_{1\phi} = 0.023Re_f^{0.8}Pr_f^{0.4}$
Oliver and Wright (1964)	$Nu_{2\phi} = 1.615(\frac{\mu_B}{\mu_W})^{0.14}(Re_{act}Pr_f\frac{D}{L})^{1/3}$ $\times(\frac{1.2}{R_f^{0.36}} - \frac{0.2}{R_f})$,$Re_{act} = \frac{(Q_f+Q_g)\rho_f D}{A\mu_f}$	
Martin and Sims (1971)	$\frac{h_{2\phi}}{h_{1\phi}} = 1+0.64\sqrt{\frac{j_g}{j_f}}$,	$Nu_{1\phi} = 0.027Re_f^{0.8}Pr_f^{0.33}(\frac{\mu_B}{\mu_W})^{0.14}$
Shah (1981)	$\frac{h_{2\phi}}{h_{1\phi}} = (1+\frac{j_g}{j_f})^{1/4}$,	$Nu_{1\phi} = 1.86(Re_f Pr_f \frac{D}{L})^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$ (L) $Nu_{1\phi} = 0.023Re_f^{0.8}Pr_f^{0.4}(\frac{\mu_B}{\mu_W})^{0.14}$ (T)
Kago et al. (1986)	$Nu_{2\phi} = (0.021Re_m^{0.8}+4.5)Pr_f^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$ $\times\{1+0.3\exp[-0.5(Fr_f-2)^2]\}$	
Kim (2000)	$\frac{h_{2\phi}}{h_{1\phi}} = (1-\alpha)[1+2.86(\frac{x}{1-x})^{0.42}(\frac{\alpha}{1-\alpha})^{0.35}$ $\times(\frac{Pr_g}{Pr_f})^{0.66}(\frac{\mu_g}{\mu_f})^{-0.72}]$,	$Nu_{1\phi} = 0.012(Re_f^{0.87} - 280)Pr_f^{0.4}$
Kalapatapu (2014)	$\frac{h_{2\phi}}{h_{1\phi}} = (\phi_f^2)^{0.39}$,	$Nu_{1\phi} = 1.86(Re_f Pr_f \frac{D}{L})^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$ (L) $Nu_{1\phi} = 0.027Re_f^{0.83}Pr_f^{0.5}(\frac{\mu_B}{\mu_W})^{0.33}$ (T)
Nada (2017)	$\frac{Nu_{2\phi}}{Nu_{1\phi}} = 1 - 0.2314Re_f^{-0.77}(10^{-5}Re_g^2 - Re_g)$,	$Nu_{1\phi} = \frac{(f/8)(Re_f-1000)Pr_f}{1+12.7\sqrt{f/8}(Pr_f^{2/3}-1)}[1+(\frac{D}{L})^{2/3}]$

Notes: L and T represent laminar flow ($Re_f \le 2300$) and turbulent flow ($Re_f > 2300$), respectively.

to-liquid velocity ratio, j_g/j_f, physical property ratio and so on. In what follows, a brief review of each correlation is given.

Groothuis and Hendal (1959) developed two sets of two-phase Nusselt number correlation for vertical pipes depending on a liquid phase (water or oil). The correlation was developed based on an extended form of Sieder and Tate correlation (Sieder and Tate, 1936). The applicable ranges of the correlations for air-water and air-oil systems in terms of the liquid Reynolds number were from 1400 to 3500 and above 5000, respectively.

Knott et al. (1959) measured a two-phase heat transfer coefficient for nitrogen–oil two-phase flows in vertical pipes. The two-phase heat transfer coefficient was non-dimensionalized by the single-phase heat transfer coefficient calculated by Sieder and Tate correlation (Sieder and Tate, 1936), and was correlated by j_g/j_f. The comparison between measured and calculated two-phase heat transfer coefficients indicated that the developed correlation tended to overestimate the data.

Kudirka et al. (1965) hypothesized that the heat transfer enhancement of a two-phase flow was attributed to the increase of a mixture velocity, and correlated two-phase Nusselt number with five non-dimensional parameters including superficial gas-to-liquid velocity ratio, viscosity ratio, liquid Reynolds number, liquid Prandtl number and liquid viscosity ratio between a wall and bulk fluid. Six empirical constants were determined by the data. The comparison between measured and calculated two-phase Nusselt numbers showed a good agreement with ±20% error.

Ueda and Hanaoka (1967) considered the effect of void fraction, liquid flow rates and Prandtl number on the two-phase heat transfer coefficient, and presented a complicated analytical solution to predict the heat transfer coefficient for two-phase slug flows in vertical pipes. It was assumed in their analysis that the heat was transferred and carried away by a liquid phase only. Ueda and Hanaoka showed a good agreement between measured and calculated two-phase Nusselt numbers.

Dorresteijn (1970) measured a forced heat transfer coefficient of adiabatic two-phase flows in a vertical pipe and developed a correlation for the ratio of the two-phase heat transfer coefficient to the single-phase heat transfer coefficient based on a liquid-acceleration model. The resulting correlation had a simple form such as a function of a liquid hold-up. Different exponents of the liquid hold-up were identified for laminar and turbulent flow regimes with the transition liquid Reynolds number of 2000.

Fedotkin and Zarudnev (1970) developed a correlation of a two-phase Nusselt number based on an extended single-phase Nusselt number correlation with modified empirical coefficients. A mixture Reynolds number in their correlation was defined as the sum of gas and liquid Reynolds numbers.

Aggour (1978) inferred that a gas phase introduced into a liquid phase contributed only to accelerating the liquid phase and the heat was mainly transferred and carried away by the liquid phase. Similar to the correlation developed by Dorresteijn (1970), Aggour developed a simple correlation for the ratio of the two-phase heat transfer coefficient to the single-phase heat transfer coefficient correlation with a function of the liquid hold-up. Correlations developed by Anon (1968) and Kays (1966) were used to calculate the single-phase heat transfer coefficient for laminar and turbulent flows, respectively. Aggour showed that 91% of the experimental data agreed with the correlation within ± 50%.

Chu and Jones (1980) developed a correlation of a two-phase Nusselt number for adiabatic upward and downward two-phase flows based on an extension of Sieder and Tate correlation (Sieder and Tate, 1936). A two-phase mixture Reynolds number in Chu and Jones correlation was defined as a liquid Reynolds number divided by a liquid hold-up. Chu and Jones demonstrated that the correlation could predict measured two-phase Nusselt number within ± 15% error.

Vijay et al. (1982) examined the relationship between a two-phase pressure drop and a two-phase heat transfer coefficient in vertical pipes, and correlated the ratio of the two-phase heat transfer coefficient to the single-phase heat transfer coefficient with a two-phase multiplier, Φ_f^2. The applicable range of the developed correlation in terms of the liquid Reynolds number was from 1.8 to 130,000. Vijay showed a good agreement between calculated and measured data with the root-mean-square of 17.7%.

Elamvaluthi and Srinivas (1984) proposed a correlation of a two-phase Nusselt number for vertical two-component two-phase flows based on an extension of a single-phase heat transfer coefficient correlation. In Elamvaluthi and Srnivas correlation, a mixture Reynolds number was defined as the sum of gas and liquid Reynolds numbers. The comparison between measured and calculated two-phase heat transfer coefficients indicated that 90% of the data could be predicted within ± 25% error.

Rezkallah and Sims (1989) studied heat transfer and hydrodynamics of two-component two-phase flows in a vertical pipe and proposed a correlation of a two-phase heat transfer coefficient based on the liquid-acceleration model. The correlation of the two-phase heat transfer coefficient was the same as Dorresteign correlation (Dorresteijn, 1970) for turbulent flow, but the component correlation to predict the single-phase heat transfer coefficient was different between them. Rezkallah and Sims reported that the overall algebraic and root-mean-square deviations of the correlation were −8.80% and 20.4%, respectively. However, they pointed out that the correlation tended to overestimate the two-phase heat transfer coefficient slightly.

Ghajar and Tang (2010) conducted experimental study on non-boiling two-phase flow heat transfer in horizontal and inclined pipes and developed a heat transfer correlation for two-phase flow in horizontal, vertical and inclined pipes based on a large amount of experimental data. In Ghajar and Tang correlation (Ghajar and Tang, 2010), six empirical exponents needed to be identified based on the experimental results. The comparison between experimental and calculated results indicated that the correlation could predict 90% of the data within ± 25% agreement with the calculated results. The root-mean-square was 18.4%. However, the correlation might not be applicable to laminar flow in vertical pipes because no laminar flow experimental data with $Re_f \le 2300$ in vertical pipes were considered in the development of Ghajar and Tang correlation (Ghajar and Tang, 2010).

Bhagwat and Ghajar (2017) measured non-boiling gas–liquid two-phase heat transfer coefficient in vertical downward pipes and correlated the ratio of a two-phase heat transfer coefficient to a single-phase heat transfer coefficient with a two-phase multiplier. The applicable range of the correlation in terms of gas and liquid Reynolds numbers were from 280 to 1400 and from 4700 to 12,000, respectively. Bhagwat and Ghajar adopted Sieder and Tate correlation (Sieder and Tate, 1936) to estimate the single-phase heat transfer coefficient. The comparison between measured and calculated two-phase heat transfer coefficients indicated that most of the data could be predicted within ± 30% error.

Table 4 summarizes existing correlations of a two-phase heat transfer coefficient in horizontal pipes. The modeling strategy for the two-phase heat transfer coefficient in horizontal pipes seems to be similar to that in vertical pipes. In other words, the two-phase heat transfer coefficient in horizontal pipes is modeled by an extension of a single-phase heat transfer coefficient correlation or correlated by a two-phase multiplier or a simple non-dimensional parameter such as j_g/j_f.

2.3. Performance evaluation of existing correlations with collected database

This section conducts a performance evaluation of the existing correlations with the collected database. In the following performance evaluation, the ratio of a two-phase heat transfer coefficient to a single-phase heat transfer coefficient is used as an objective parameter, and in what follows the ratio is referred to as two-phase heat transfer enhancement ratio. Fig. 2 shows the performance of the existing 14 correlations against the collected database. The abscissa and ordinate in the figure are the experimental and calculated two-phase heat transfer enhancement ratios, respectively, and solid and dotted lines indicate 0% and ± 30% error, respectively. Fig. 2 demonstrates that none of the existing 14 correlations can predict the two-phase heat transfer enhancement ratio for the collected database of two-component two-phase slug flows in vertical pipes satisfactorily. Fig. 2(e), (f), (g), (m) and (n) show that Dorresteign correlation (Dorresteijn, 1970), Fedotkin and Zarudnev correlation (Fedotkin and Zarudnev, 1970), Aggour correlation (Aggour, 1978), Ghajar and Tang correlation (Ghajar and Tang, 2010) and Bhagwat and Ghajar correlation (Bhagwat and Ghajar, 2017) tend to underestimate the experimental values when the two-phase heat transfer enhancement ratio is high. To the contrary, Fig. 2(a), (c), (d), and (k) show that Groothuis and Hendal correlation (Groothuis and Hendal, 1959), Kudirka et al. correlation (Kudirka et al., 1965), Ueda and Hanaoka correlation (Ueda and Hanaoka, 1967), Elamvaluthi and Srinvivas correlation (Elamvaluthi and Srinivas, 1984) tend to overestimate the experimental values at some test conditions. Fig. 2(b), (h), (i), (j) and (l) indicate that Knott et al. correlation (Knott et al., 1959), Chu and Jones correlation (Chu and Jones, 1980) (Upward). Chu and Jones correlation (Chu and Jones, 1980) (Downward), Vijay correlation (Vijay et al., 1982), and Rezkallah and Sims correlation (Rezkallah and Sims, 1989) predict the two-phase heat transfer en-

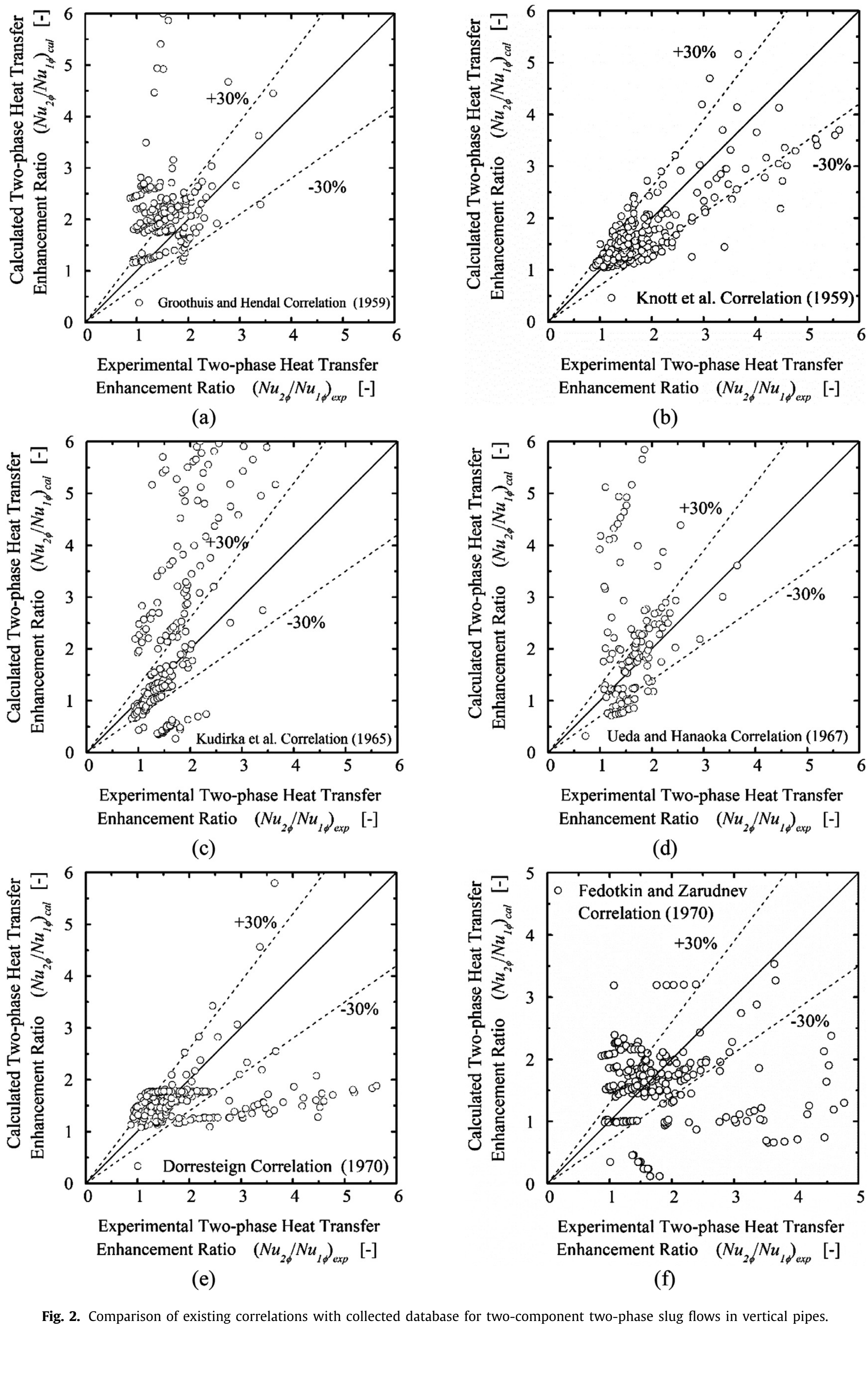


Fig. 2. Comparison of existing correlations with collected database for two-component two-phase slug flows in vertical pipes.

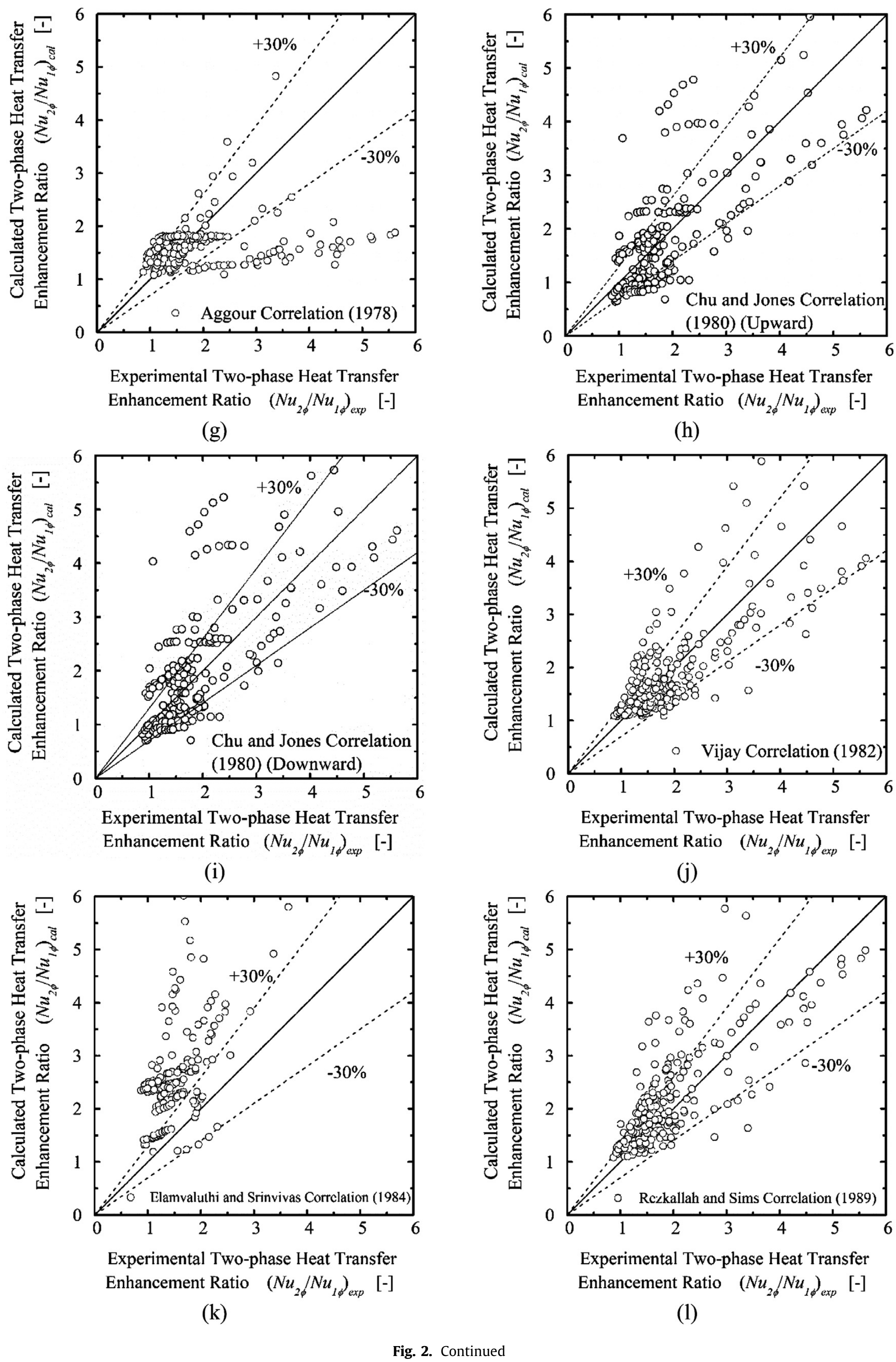
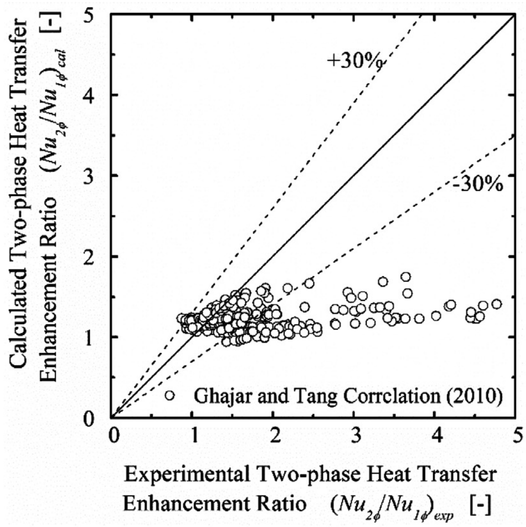
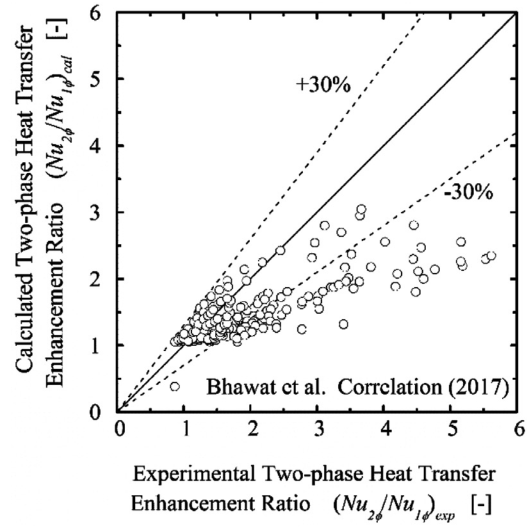


Fig. 2. Continued



(m)



(n)

Fig. 2. Continued

hancement ratio evenly around the experimental values but the degree of scattering is rather high, particularly, at high two-phase heat transfer enhancement ratio.

In order to evaluate the prediction performance of the 14 existing correlations for vertical pipes as well as the 8 existing correlations for horizontal pipes quantitatively, the following four statistical parameters are used. They are the mean error, m_d, defined by Eq. (2), standard deviation, s_d, defined by Eq. (3), mean relative deviation, m_{rel}, defined by Eq. (4) and mean absolute relative deviation, $m_{rel,\,ab}$, defined by Eq. (5).

$$m_d = \frac{1}{N}\sum_{i=1}^{N}\left[\left(\frac{h_{2\phi}(i)}{h_{1\phi}(i)}\right)_{cal} - \left(\frac{h_{2\phi}(i)}{h_{1\phi}(i)}\right)_{exp}\right] \tag{2}$$

$$s_d = \sqrt{\frac{1}{N-1}\sum_{i=1}^{N}\left[\left(\frac{h_{2\phi}(i)}{h_{1\phi}(i)}\right)_{cal} - \left(\frac{h_{2\phi}(i)}{h_{1\phi}(i)}\right)_{exp} - m_d\right]^2} \tag{3}$$

$$m_{rel} = \frac{1}{N}\sum_{i=1}^{N}\frac{\left(\frac{h_{2\phi}(i)}{h_{1\phi}(i)}\right)_{cal} - \left(\frac{h_{2\phi}(i)}{h_{1\phi}(i)}\right)_{exp}}{\left(\frac{h_{2\phi}(i)}{h_{1\phi}(i)}\right)_{exp}} \times 100 \tag{4}$$

$$m_{rel,ab} = \frac{1}{N}\sum_{i=1}^{N}\frac{\left|\left(\frac{h_{2\phi}(i)}{h_{1\phi}(i)}\right)_{cal} - \left(\frac{h_{2\phi}(i)}{h_{1\phi}(i)}\right)_{exp}\right|}{\left(\frac{h_{2\phi}(i)}{h_{1\phi}(i)}\right)_{exp}} \times 100 \tag{5}$$

where N and $h_{2\phi}(i)/h_{1\phi}(i)$ are the number of samples and two-phase heat transfer enhancement ratio for i-th data, respectively. Subscripts of cal and exp indicate the calculated and measured values, respectively.

Table 5 summarizes the results of the prediction performance for the two-phase heat transfer enhancement ratio. The mean absolute relative deviations for the 12 existing correlations for vertical pipes range from 21.1 to 97.4%, whereas the mean absolute relative deviations for the 8 existing correlations for horizontal pipes range from 21.9 to 86.6%. The mean absolute relative deviation for Rezkallah and Sims correlation (Rezkallah and Sims, 1989) is identified to be the best (21.1%), but as can be seen in Fig. 2(l), the correlation overestimates some experimental data significantly. This leads to the conclusion such that Rezkallah and Sims correlation

Table 5
Performance evaluation of exiting correlations for collected two-component two-phase slug flow database in vertical pipes.

Statistical parameters	m_d [-]	s_d [-]	m_{rel} [%]	$m_{rel,\,ab}$ [%]
Developed in vertical pipes				
Groothuis and Hendal (1959)	0.718	1.11	24.2	30.8
Knott et al. (1959)	−0.177	0.548	−10.4	21.7
Kudirka et al. (1965)	1.63	3.79	−4.07	55.1
Ueda and Hanaoka (1967)	0.582	1.28	4.32	35.2
Dorresteign (1970)	−0.264	0.940	−18.6	37.1
Fedotkin and Zarudnev (1970)	−0.330	1.13	−74.1	97.4
Aggour (1978)	−0.238	0.957	−17.3	37.5
Chu and Jones (1980) (Downward)	0.230	1.21	−2.78	26.8
Chu and Jones (1980) (Upward)	0.0494	1.10	−12.3	31.0
Vijay et al. (1982)	−0.0251	0.607	−3.80	21.5
Elamvaluthi and Srinvivas (1984)	1.19	0.993	38.7	41.4
Rezkallah and Sims (1989)	0.293	0.808	9.91	21.1
Ghajar and Tang (2010)	−35.5	0.453	−30.1	34.9
Bhagwat and Ghajar (2017)	−0.465	0.674	−28.6	32.8
Developed in horizontal pipes				
King (1952)	0.534	0.955	17.9	23.4
Oliver and Wright (1964)	−0.802	1.29	−52.2	65.3
Martin and Sims (1971)	0.432	0.682	16.8	21.9
Shah (1981)	−0.415	0.637	−24.7	28.9
Kago et al. (1986)	−0.258	1.54	−45.5	59.1
Kim (2000)	−0.871	0.905	−86.4	86.6
Kalapatapu (2014)	−0.197	0.574	−11.5	22.3
Nada (2017)	0.0280	1.07	−14.1	29.4

(Rezkallah and Sims, 1989) is not recommended for predicting the two-phase heat transfer enhancement ratio in a wide range of test conditions. Although Fig. 2(b) suggests that Knott et al. correlation (Knott et al., 1959) has the best overall performance, the mean absolute relative deviation of the correlation is still 21.7%. In addition, Knott et al. correlation (Knott et al., 1959) is developed by correlating the two-phase heat transfer enhancement ratio with j_g/j_f empirically. The physics behind the proposed functional dependence of the two-phase heat transfer enhancement ratio is not clear.

The above brief performance evaluation reveals the problems in the existing correlations of the two-phase heat transfer enhancement ratio as:

- The best mean absolute relative deviation for the existing correlation is 21.1% but the correlation, Rezkallah and Sims correlation (Rezkallah and Sims, 1989), overestimates some experimental data significantly. This leads to considerable uncertainty in applying Rezkallah and Sims correlation for conditions beyond the conditions validated for the correlation.

- For an overall point of view, Knott et al. correlation (Knott et al., 1959) can predict the two-phase heat transfer enhancement ratio fairly well. However, the correlation was developed based on a very simple formula which has not been theoretically justified.

In view of the above, it is necessary to develop an accurate, robust and theoretically-supported correlation for the two-phase heat transfer enhancement ratio which can be applicable to a wide range of conditions as well as conditions even beyond the conditions validated for the correlation. In the following section, a new correlation will be developed with the aid of the concept of classic Reynolds and Chilton–Colburn analogies.

3. Development of two-phase heat transfer coefficient correlation for two-component two-phase slug flows in vertical pipes

3.1. Reynolds and Chilton–Colburn analogies

In this section, two-dimensional laminar flow with negligible gravitational force and viscous dissipation is considered to simplify the problem. The boundary layer equations for a flow non-dimensionalized by proper characteristic length, velocity and temperature scales are formulated as (Incropera et al., 2007; Dong and Hibiki, 2018):

$$v_z^* \frac{\partial v_z^*}{\partial z^*} + v_y^* \frac{\partial v_z^*}{\partial y^*} = -\frac{dp^*}{dz^*} + \frac{1}{\mathrm{Re}} \frac{\partial^2 v_z^*}{\partial y^{*2}} \tag{6}$$

$$v_z^* \frac{\partial T^*}{\partial z^*} + v_y^* \frac{\partial T^*}{\partial y^*} = \frac{1}{RePr} \frac{\partial^2 T^*}{\partial y^{*2}} \tag{7}$$

where v_z, v_y, p, T, Re and Pr are the axial velocity (or z-directional velocity), velocity perpendicular to a wall, pressure, temperature, Reynolds number and Prandtl number, respectively.

The non-dimensionalized velocity and temperature profiles at a wall surface are related to the product of the friction coefficient, C_f, and Reynolds number and Nusselt number, respectively, as:

$$\frac{\partial v_z^*}{\partial y^*}\Big|_{y^*=0} = \frac{C_f Re}{2} \tag{8}$$

$$\frac{\partial T^*}{\partial y^*}\Big|_{y^*=0} = Nu \tag{9}$$

The definition of the friction coefficient is given by:

$$C_f \equiv \frac{\tau_w}{\frac{1}{2}\rho v_z^2} \tag{10}$$

where τ_w is the surface shear stress.

For the case of $dp^*/dz^* = 0$ and $Pr = 1$, the momentum and energy equations have the same form which indicates that the non-dimensionalized temperature profile at the wall surface should be the same as the non-dimensionalized velocity profile at the wall resulting in Reynolds analogy:

$$\frac{C_f Re}{2} = Nu \tag{11}$$

Chilton–Colburn extended the applicable range of Reynolds analogy to a wide range of Pr as:

$$\frac{C_f Re}{2} = Nu Pr^{-1/3} \quad (0.6 < Pr < 60) \tag{12}$$

It is important to be noted that in laminar flow Eq. (11) is only appropriate when $dp^*/dz^* = 0$ but in turbulent flow Eq. (11) is approximately valid due to a less sensitive effect of pressure gradients on Chilton–Colburn analogy (Incropera et al., 2007).

3.2. Framework of model development with the concept of Reynolds and Chilton–Colburn analogies

In this section, the model to predict the two-phase heat transfer enhancement ratio for two-component two-phase slug flows in vertical pipes is developed based on the concept of Reynolds and Chilton–Colburn analogies. Notations of 1ϕ and 2ϕ are used to identify the single-phase and two-phase flow, respectively, as needed basis.

The wall shear stress is expressed in terms of the frictional pressure drop, $(dp/dz)_F$, as:

$$\tau_w = \frac{D}{4}\left(\frac{dp}{dz}\right)_F = \frac{D}{4} \times 4f\frac{1}{D}\frac{1}{2}\rho v^2 = f\frac{1}{2}\rho v^2 \tag{13}$$

where f is the Fanning friction factor. The friction coefficient is represented by substituting Eqs. (13) into Eq. (10) as:

$$C_f = f \tag{14}$$

The Chilton–Colburn analogy is expressed by the Fanning friction factor as:

$$\frac{fRe}{2} = Nu Pr^{-1/3} \tag{15}$$

or

$$\frac{f_{1\phi} Re_{1\phi}}{2} = Nu_{1\phi} Pr^{-1/3} \tag{16}$$

Here, it is assumed that a two-phase flow is a homogenous mixture and the Chilton–Colburn analogy can be applied for the two-phase flow. Thus,

$$\frac{f_{1\phi}\Phi_f^2 Re_{2\phi}}{2} = Nu_{2\phi} Pr^{-1/3} \tag{17}$$

where Φ_f^2 is the two-phase multiplier. The two-phase heat transfer enhancement ratio is obtained by dividing Eq. (17) by Eq. (16) as:

$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = \Phi_f^2 \frac{Re_{2\phi}}{Re_{1\phi}} \tag{18}$$

In Eq. (18), the single-phase and two-phase Reynolds numbers are, respectively, defined by:

$$Re_{1\phi} = \frac{GD}{\mu_f} \text{ and } Re_{2\phi} = \frac{GD}{\mu_m} \tag{19}$$

where G and μ_m are the mass flux and mixture viscosity, respectively. The mixture viscosity is given by:

$$\frac{\mu_m}{\mu_f} = (1-\alpha)^{-n} \tag{20}$$

The exponent, n, depends on a flow regime. For a gas bubble in a liquid flow and a liquid drop in a gas flow, $n = 1$ and 2.5, respectively (Ishii and Hibiki, 2010).

From Eqs. (18) to (20), the two-phase heat transfer enhancement ratio is formulated in terms of a liquid hold-up and a two-phase multiplier as:

$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = (1-\alpha)^n \Phi_f^2 \tag{21}$$

The validity of the assumptions such as an approximated homogeneous mixture flow and an applicable Chilton–Colburn analogy for the two-phase flow is examined by comparing Eq. (21) with collected data. The void fraction is calculated by a drift-flux type correlation as (Ishii and Hibiki, 2010):

$$\alpha = \frac{j_g}{C_0 j + V_{gj}} \tag{22}$$

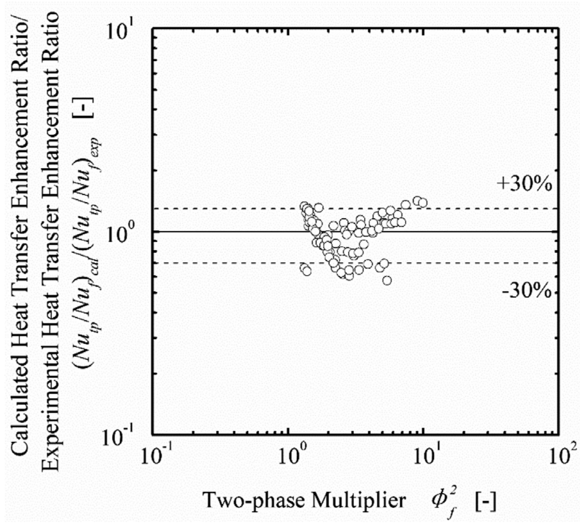


Fig. 3. Comparison of experimental two-phase heat transfer coefficients with theoretical based equation.

where j is the mixture volumetric flux given by the sum of superficial gas and liquid velocities. The distribution parameter, C_0, and drift velocity, V_{gj}, for a upward slug flow in a vertical medium pipe are given by (Ishii, 1977):

$$C_0 = 1.2 - 0.2\sqrt{\frac{\rho_g}{\rho_f}} \tag{23}$$

$$V_{gj} = 0.35\sqrt{\frac{\Delta\rho g D}{\rho_f}} \tag{24}$$

The distribution parameter, C_0, and drift velocity, V_{gj}, for a downward flow in a vertical pipe are given by (Goda et al., 2003):

$$\begin{aligned} &C_0 = (-0.0214\langle j^*\rangle + 0.772) + (0.0214\langle j^*\rangle + 0.228)\sqrt{\frac{\rho_g}{\rho_f}} \\ &\text{for } -20 \le \langle j^*\rangle < 0, \\ &C_0 = \left(0.2e^{0.00848(\langle j^*\rangle+20)} + 1\right) - 0.2e^{0.00848(\langle j^*\rangle+20)}\sqrt{\frac{\rho_g}{\rho_f}}, \\ &\text{for } \langle j^*\rangle < -20. \end{aligned} \tag{25}$$

$$\langle j^*\rangle = j/V_{gj} \tag{26}$$

$$V_{gj} = \sqrt{2}\left(\frac{\Delta\rho g\sigma}{\rho_f^2}\right)^{1/4} \tag{27}$$

where j^* is non-dimensional mixture velocity.

The two-phase multiplier is estimated by Chisholm's correlation (Chisholm, 1967) as:

$$\Phi_f^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{28}$$

where Martinelli parameter, X, is defined by:

$$X \equiv \sqrt{\frac{\left(\frac{dp}{dz}\right)_{F,f}}{\left(\frac{dp}{dz}\right)_{F,g}}} \tag{29}$$

The Chisholm's coefficient, C, is dependent on whether a flow is laminar or turbulent. When both gas and liquid flows are turbulent flows ($Re_g > 2300$ and $Re_f > 2300$), the value of the Chisholm's coefficient is 21. The exponent, n, is tentatively set at unity which is the value for a bubbly flow.

Fig. 3 shows the comparison between measured and calculated two-phase heat transfer enhancement ratio for gas and liquid turbulent flows. The abscissa and ordinate in the figure are the ratio of calculated $Nu_{2\phi}/Nu_{1\phi}$ to experimental $Nu_{2\phi}/Nu_{1\phi}$ and the two-phase multiplier, respectively, and solid and dotted lines indicate 0% and ±30% error, respectively. A reasonable agreement between calculated and experimental two-phase heat transfer enhancement ratio, $Nu_{2\phi}/Nu_{1\phi}$, without introducing new empirical adjustable coefficients into Eq. (21) demonstrates the validity of the assumptions such as an approximated homogeneous mixture flow and a applicable Chilton–Colburn analogy for the two-phase flow or Eq. (21).

3.3. Further discussion on the similarity of developed framework to some existing correlations

In order to demonstrate the similarity of the developed framework, Eq. (21) to some existing correlations, Eq. (21) is further modified as follows. Although the two-phase multiplier is given by Chisholm's correlation, Eq. (28), it is also given by a function of a void fraction as (Chisholm, 1967):

$$\Phi_f^2 = \frac{1}{(1-\alpha)^m} \tag{30}$$

The exponent, m, is typically in the range between 1.75 and 2. From Eqs. (21) and (30), the two-phase heat transfer enhancement ratio is formulated by a unique function of a void fraction or two-phase multiplier as:

$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = \frac{1}{(1-\alpha)^{m-n}} \tag{31}$$

or

$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = \Phi_f^{2(1-\frac{n}{m})} \tag{32}$$

The functional form of Eq. (31) is exactly the same as Dorresteign correlation (Dorresteijn, 1970), Aggour correlation (Aggour, 1978), and Rezkallah and Sims correlation (Rezkallah and Sims, 1989), whereas the functional form of Eq. (32) is exactly the same as Vijay correlation (Vijay et al., 1982) and Bhagwat and Ghajar correlation (Bhagwat and Ghajar, 2017). The above simple analysis demonstrates that Dorresteign correlation (Dorresteijn, 1970), Aggour correlation (Aggour, 1978), Vijay correlation (Vijay et al., 1982), Rezkallah and Sims correlation (Rezkallah and Sims, 1989), and Bhagwat and Ghajar correlation (Bhagwat and Ghajar, 2017) are the correlations based on the transformation of Eq. (21).

3.4. Development of two-phase heat transfer coefficient correlation for two-component two-phase slug flows in vertical pipes

In order to develop the correlation of the two-phase heat transfer coefficient for two-component two-phase slug flows in vertical pipes, an appropriate functional form is deduced from Eq. (21) as follows. Substituting Eq. (28) into Eq. (21) yields:

$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = (1-\alpha)^n\left(1 + \frac{C}{X} + \frac{1}{X^2}\right) \tag{33}$$

In order to ensure a sufficient flexibility to develop an accurate correlation, Eq. (33) is further recast as (Muzychka and Awad, 2010):

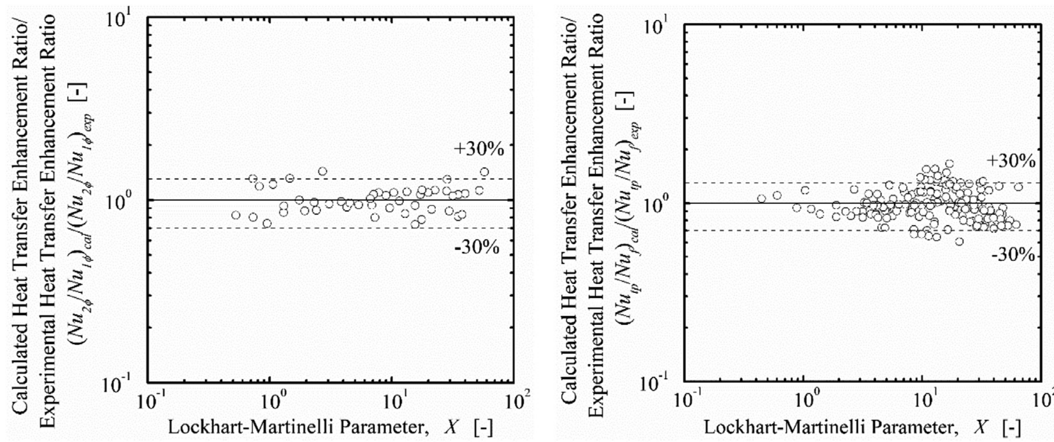
$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = (1-\alpha)^a\left(1 + \frac{A}{X^b}\right) \tag{34}$$

where a, b and A are the empirical constants. For a liquid single-phase flow ($\alpha \to 0$ and $X \to \infty$), Eq. (34) is reduced to:

$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = 1 \tag{35}$$

Table 6
Summary of newly-developed heat transfer coefficient correlations with different heat transfer coefficient correlations.

Correlations		$Nu_{2\phi}/Nu_{1\phi}$	$Nu_{1\phi}$
Correlation I	$Re_f > 2300$	$\frac{Nu_{2\phi}}{Nu_{1\phi}} = (1-\alpha)^{0.208}(1+4.12X^{-0.514})$	$Nu_{1\phi} = 0.023Re_f^{0.8}Pr_f^n$, $n=0.4$ (heating) or 0.3 (cooling)
	$Re_f \leq 2300$	$\frac{Nu_{2\phi}}{Nu_{1\phi}} = (1-\alpha)^{0.339}(1+4.65X^{-0.409})$	$Nu_{1\phi} = 1.86(Re_f Pr_f \frac{D}{L})^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$
Correlation II	$Re_f > 2300$	$\frac{Nu_{2\phi}}{Nu_{1\phi}} = (1-\alpha)^{-0.0200}(1+2.56X^{-0.508})$	$Nu_{1\phi} = \frac{(f/8)(Re_f-1000)Pr_f}{1+12.7\sqrt{f/8}(Pr_f^{2/3}-1)}[1+(\frac{D}{L})^{2/3}]$
	$Re_f \leq 2300$	$\frac{Nu_{2\phi}}{Nu_{1\phi}} = (1-\alpha)^{0.339}(1+4.65X^{-0.409})$	$Nu_{1\phi} = 1.86(Re_f Pr_f \frac{D}{L})^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$
Correlation III	$Re_f > 2300$	$\frac{Nu_{2\phi}}{Nu_{1\phi}} = (1-\alpha)^{0.0766}(1+2.90X^{-0.485})$	$Nu_{1\phi} = 0.027Re_f^{0.8}Pr_f^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$
	$Re_f \leq 2300$	$\frac{Nu_{2\phi}}{Nu_{1\phi}} = (1-\alpha)^{0.339}(1+4.65X^{-0.409})$	$Nu_{1\phi} = 1.86(Re_f Pr_f \frac{D}{L})^{1/3}(\frac{\mu_B}{\mu_W})^{0.14}$



(a) Laminar Flow ($Re_f \leq 2300$) (b) Turbulent Flow ($Re_f > 2300$)

Fig. 4. Predictive capability of correlation I for two-component two-phase slug flow heat transfer coefficient in vertical pipes.

This demonstrates that Eq. (34) has a built-in characteristic to match the limiting case of the two-phase heat transfer enhancement ratio at α=0.

As shown in Tables 3 and 4, several single-phase heat transfer correlations were recommended in the development of the two-phase heat transfer enhancement ratio. In this study, three single-phase heat transfer correlations are examined in the development of the two-phase heat transfer enhancement ratio. They are Dittus–Boelter correlation (Petukhov, 1970) ($Re_f > 2300$), Gnielinski correlation (Gnielinski, 1976) ($Re_f > 2300$) and Sieder–Tate correlation (Sieder and Tate, 1936). The parameters in Eq. (34) are determined based on the collected database using a non-linear regression analysis. Table 6 summarizes the finalized correlation of the two-phase heat transfer enhancement ratio for two-component two-phase slugs flow in vertical pipes based on the above three single-phase flow correlations.

4. Results and discussion

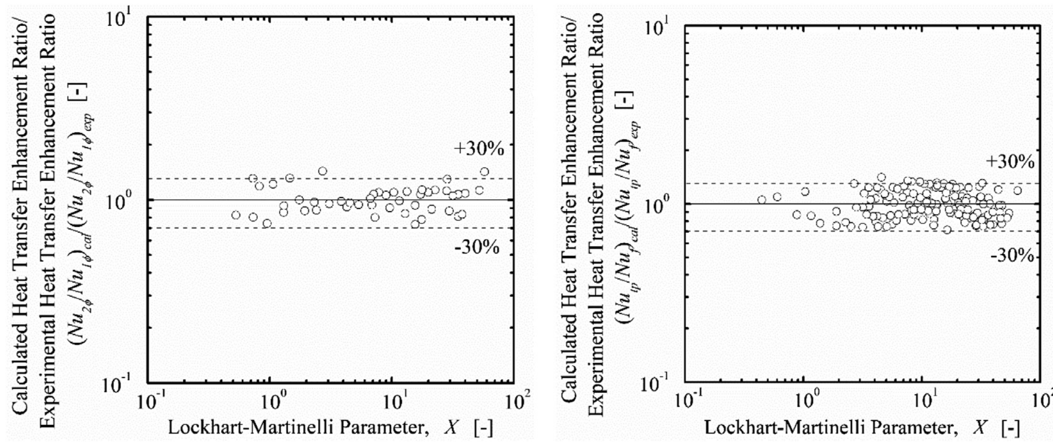
4.1. Performance analysis of newly developed two-phase heat transfer correlation for two-component two-phase slug flows in vertical pipes

Figs. 4–6 show the comparison between measured and calculated two-phase heat transfer enhancement ratios for the correlations I, II and III, respectively. The abscissa and ordinate in the figure are the ratio of calculated $Nu_{2\phi}/Nu_{1\phi}$to experimental $Nu_{2\phi}/Nu_{1\phi}$and the two-phase multiplier, respectively, and solid and dotted lines indicate 0% and $\pm$30% error, respectively. Left and right figures in Figs. 4–6 indicate the flow conditions being laminar and turbulent flows, respectively. As shown in Figs. 4–6, excellent agreement between the collected database and all correlations I, II and III is obtained.

Table 7
Performance evaluation of newly developed heat transfer coefficient correlations for two-component two-phase slug flows in vertical pipes.

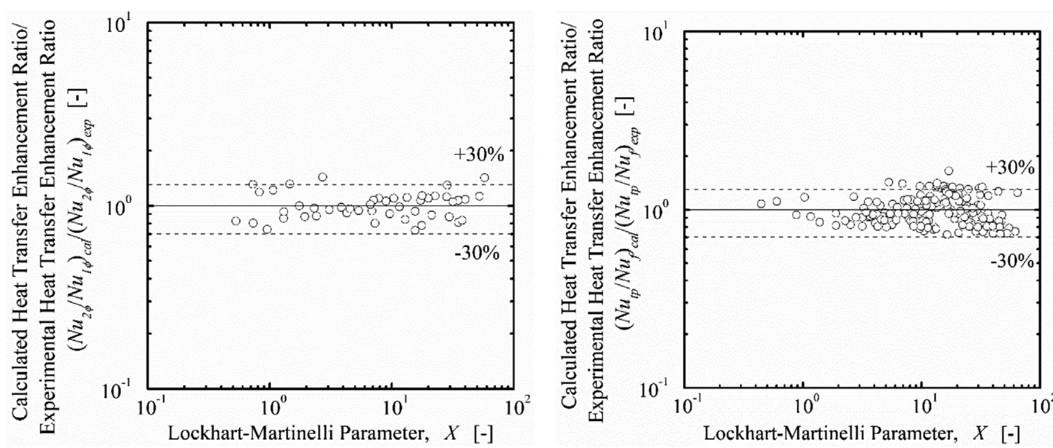
Statistical parameters		m_d [-]	s_d [-]	m_{rel} [%]	$m_{rel,\ ab}$ [%]
Correlation I	$Re_f > 2300$	−0.00202	0.376	−1.98	16.3
	$Re_f \leq 2300$	−0.0145	0.479	−0.880	13.7
	Total	0.00876	0.426	0.0314	15.7
Correlation II	$Re_f > 2300$	−0.00158	0.336	0.763	14.4
	$Re_f \leq 2300$	−0.0145	0.479	−0.880	13.7
	Total	0.00612	0.382	0.0221	14.2
Correlation III	$Re_f > 2300$	0.0279	0.358	1.395	16.0
	$Re_f \leq 2300$	−0.0145	0.479	−0.880	13.7
	Total	0.00713	0.397	0.116	14.9

Table 7 provides the quantitative predictive performance of the two-phase heat transfer enhancement ratio for the correlations I, II and III. For the correlation I with Dittus–Boelter correlation (Petukhov, 1970), 88.1% of the two-phase heat transfer enhancement ratio are predicted within $\pm$30% deviation from the measured values and the mean absolute relative deviation is estimated to be 15.7%. For the correlation II with Gnielinski correlation (Gnielinski, 1976), 95.1% of the two-phase heat transfer enhancement ratio is predicted within $\pm$30% deviation from the measured values and the mean absolute relative deviation is estimated to be 14.2%. For the correlation III with Sieder–Tate correlation (Sieder and Tate, 1936), 89.9% of the two-phase heat transfer enhancement ratio is predicted within $\pm$30% deviation from the measured values and the mean absolute relative deviation is estimated to be 14.9%. All correlations I, II and III show the excellent predictive performance. The mean absolute relative deviation is reduced from 21.1% (the value based on the best correlation in Table 5) to 14.2%, and the newly developed correlations do not show any systematic deviations within the tested conditions. Based on the



(a) Laminar Flow ($Re_f \leq 2300$) (b) Turbulent Flow ($Re_f > 2300$)

Fig. 5. Predictive capability of correlation II for two-component two-phase slug flow heat transfer coefficient in vertical pipes.



(a) Laminar Flow ($Re_f \leq 2300$) (b) Turbulent Flow ($Re_f > 2300$)

Fig. 6. Predictive capability of correlation III for two-component two-phase slug flow heat transfer coefficient in vertical pipes.

above quantitative predictive performance evaluation, the correlation II is recommended as:

Laminar flow ($Re_f \leq 2300$):

$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = (1-\alpha)^{0.339}\left(1+\frac{4.65}{X^{0.409}}\right) \quad (36)$$

$$Nu_{1\phi} = 1.86\left(Re_f Pr_f \frac{D}{L}\right)^{1/3}\left(\frac{\mu_B}{\mu_W}\right)^{0.14} \quad (37)$$

where μ_B and μ_W are the liquid viscosity estimated based on the temperature in bulk fluid and at wall surface, respectively.

Turbulent flow ($Re_f > 2300$):

$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = (1-\alpha)^{-0.0200}\left(1+\frac{2.56}{X^{0.508}}\right) \quad (38)$$

$$Nu_{1\phi} = \frac{(f/8)(Re_f - 1000)Pr_f}{1+12.7\sqrt{f/8}\left(Pr_f^{2/3}-1\right)}\left[1+\left(\frac{D}{L}\right)^{2/3}\right] \quad (39)$$

It should be noted here that Eqs. (36) and (38) are not continuous at the critical Reynolds number (=2300). This requires a simple interpolation scheme near the critical Reynolds number.

To examine the effect of the flow direction on the predictive performance of the newly developed correlation, the calculated-to-measured two-phase heat transfer enhancement ratios for upward and downward flows are plotted in Fig. 7(a) and 7(b), respectively.

Table 8
Predictive performance evaluation of the newly-developed two-phase heat transfer correlation for different two-phase flow regimes.

Statistical parameters	m_d [-]	s_d [-]	m_{rel} [%]	$m_{rel,\,ab}$ [%]
Upward flow	−0.0265	0.358	−1.93	15.0
Downward flow	0.0702	0.255	5.22	14.0

Table 8 provides the quantitative predictive performance of the two-phase heat transfer enhancement ratio for upward and downward flows. For upward flow, 93.2% of the two-phase heat transfer enhancement ratio is predicted within ± 30% deviation from the measured values and the mean absolute relative deviation is estimated to be 15.0%. For downward flow, 91.4% of the two-phase heat transfer enhancement ratio is predicted within ± 30% deviation from the measured values and the mean absolute relative deviation is estimated to be 14.0%. As demonstrated above, the newly developed correlation coupled with an appropriate drift-flux correlation can predict the two-phase heat transfer enhancement ratio well.

4.2. Applicability of newly developed two-phase heat transfer correlation to other flow regimes

In the previous section, an excellent prediction performance of the newly developed correlation of the heat transfer coeffi-

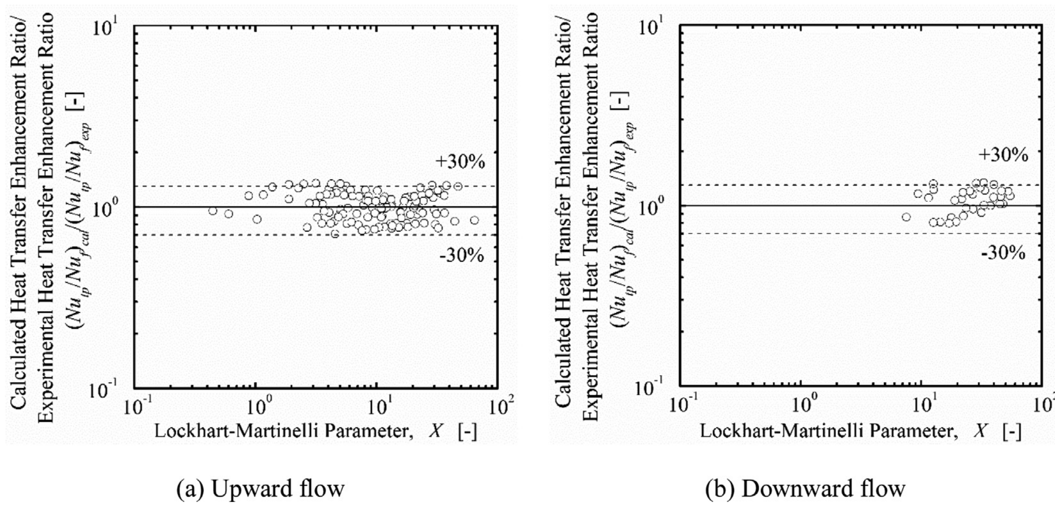


Fig. 7. Comparison of calculated-to-experimental two-phase heat transfer enhancement ratios for upward and downward flows.

cient for two-component two-phase slug flows in vertical medium pipes, Eqs. (36)–(39), has been demonstrated. This section examines the applicability of the correlation, Eqs. (36)–(39), to other flow regimes. In order to close the mathematical system of the correlation, a void fraction and Martinelli parameter should be calculated by operating parameters such as superficial gas and liquid velocities. As indicated in Eq. (29), the calculation procedure of Martinelli parameter only requires the calculation of gas and liquid single-phase frictional pressure drops and the calculation procedure is unchanged in applying the correlation, Eq. (36)–(39) for any two-phase flow regimes. The calculation of void fraction relies on a drift-flux type correlation which is commonly given on a flow regime to flow regime basis. The correlations of the distribution parameter and drift velocity for a slug flow regime are given by Eqs. (23) and (24) for upward flow and Eqs. (25) and (27) for downward flow, respectively. The correlations of the distribution parameter and drift velocity for other flow regimes for upward two-phase flows in vertical medium pipes are given by:

Bubbly flow (Hibiki and Ishii, 2002a):

$$C_0 = \left(1.2 - 0.2\sqrt{\frac{\rho_g}{\rho_f}}\right)\left\{1 - \exp\left(-22\frac{D_{Sm}}{D}\right)\right\} \tag{40}$$

$$V_{gj} = \sqrt{2}\left(\frac{\Delta\rho g\sigma}{\rho_f^2}\right)^{1/4}(1-\alpha)^{1.75} \tag{41}$$

where D_{Sm} is the bubble Sauter mean diameter, which can be calculated by Hibiki and Ishii correlation (Hibiki and Ishii, 2002b).

Churn flow (37):

$$C_0 = \left(1.2 - 0.2\sqrt{\frac{\rho_g}{\rho_f}}\right) \tag{42}$$

$$V_{gj} = \sqrt{2}\left(\frac{\Delta\rho g\sigma}{\rho_f^2}\right)^{1/4} \tag{43}$$

Annular flow (37):

$$\bar{V}_{gj}\left(\equiv (C_0-1)j + V_{gj}\right) = \frac{1-\alpha}{\alpha + \left(\frac{1+75(1-\alpha)}{\sqrt{\alpha}}\frac{\rho_g}{\rho_f}\right)^{1/2}} \times \left(j + \sqrt{\frac{\Delta\rho g D(1-\alpha)}{0.015\rho_f}}\right) \tag{44}$$

where $\bar{V}_{gj}$ is the mean drift velocity. The drift-flux type correlation for downward two-phase flows in vertical pipes, Eq. (25), is given by Goda et al. (Goda et al., 2003).

Table 9
Performance evaluation of the newly-developed two-phase heat transfer correlation for different two-phase flow regimes.

Statistical parameters	m_d [-]	s_d [-]	m_{rel} [%]	$m_{rel,\,ab}$ [%]
Bubbly flow	0.104	0.152	8.79	12.8
Churn flow	0.371	0.235	19.7	19.7
Annular flow	−0.595	1.02	−12.3	19.0
Total	−0.0745	0.776	4.40	17.5

The void fraction can be predicted accurately by the drift-flux type correlation. However, in order to select an appropriate drift-flux type correlation, a flow regime should be first identified by a flow regime transition criteriion as shown in Fig. 1 (Mishima and Ishii, 1984). This "two-step" method may be cumbersome for a purpose of quick design evaluation. For this case, Eqs. (23) and ((24) may be approximately utilized for all upward flow regimes in view of a weak dependence of void fraction on the two-phase heat transfer enhancement ratio such as $(1-\alpha)^{-0.0200}$ and $(1-\alpha)^{0.341}$as shown in Eqs. (36) and (38). In addition, for a relatively high mixture volumetric flux, the drift-flux type correlation is approximated by:

$$\alpha \simeq \frac{j_g}{C_0 j} \tag{45}$$

The difference in the distribution parameter among the flow regime is estimated to be less than 20%. The prediction uncertainty using Eqs. (23) and (24) for all flow regimes may not cause significant uncertainty in the predicted two-phase heat transfer enhancement ratio.

Experimental data of a two-component two-phase heat transfer coefficient for bubbly, churn and annular flows are collected from the data sources of Aggour (1978), Rezkallah and Sims (1989), and Bhagwat and Ghajar (2017). Fig. 8 shows the comparison between measured and calculated two-phase heat transfer enhancement ratios using the correlation, Eqs. (36)–(39). The abscissa and ordinate in the figure are the ratio of calculated $Nu_{2\phi}/Nu_{1\phi}$to experimental $Nu_{2\phi}/Nu_{1\phi}$ and the two-phase multiplier, respectively, and solid and dotted lines indicate 0% and ± 30% error, respectively. Fig. 8(a), (b) and (c) are the comparison for the data taken in bubbly, churn and annular flows, respectively, and show a reasonable agreement between the collected database and the newly developed correlation. Table 9 provides the quantitative predictive performance of the correlation for bubbly, churn and annular flows. The mean ab-

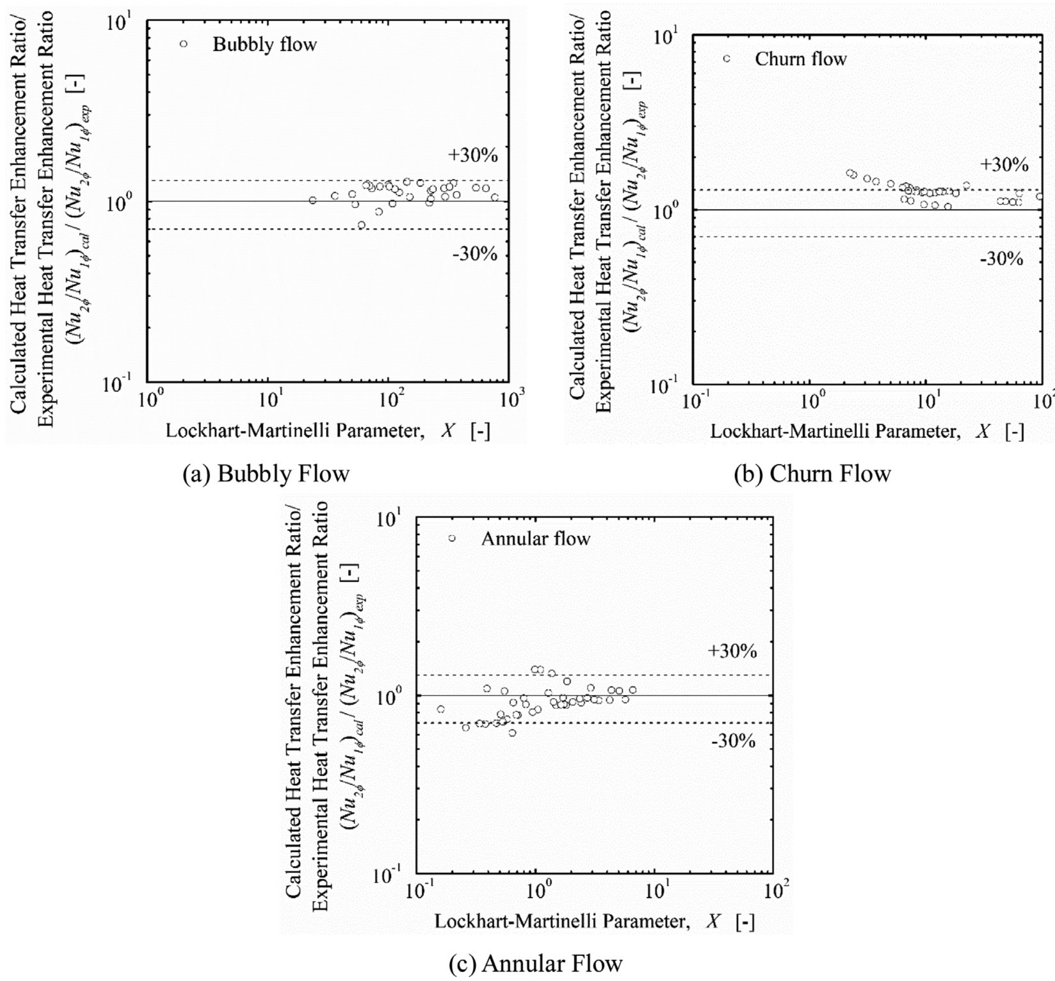


Fig. 8. Comparison of experimental and calculated two-phase heat transfer coefficients for other vertical two-phase flow regimes: (a) Bubbly flow, (b) Churn flow and (c) Annular flow.

solute relative deviations for bubbly, churn and annular flows are estimated to be 12.8%, 19.7%, 19.0%, respectively.

The above performance evaluation demonstrates that the newly developed correlation, Eqs. (36)–(39), is applicable to all flow regimes in vertical medium pipes.

4.3. Applicability of newly developed two-phase heat transfer correlation to large pipes

In the previous section, the applicability of the newly developed correlation of the heat transfer coefficient for two-component two-phase flows in vertical medium pipes, Eqs. (36)–(39), to all flow regimes has been demonstrated. This section discusses the applicability of the correlation, Eqs. (36)–(39), to vertical large pipes. As discussed in the previous section, in order to close the mathematical system of the correlation, a void fraction and Martinelli parameter should be given. The calculation procedure of Martinelli parameter only requires the calculation of gas and liquid single-phase frictional pressure drops and the calculation procedure is unchanged in applying the correlation, Eq. (36)–(39) for vertical large pipes. In the void fraction calculation, the constitutive correlations of the distribution parameter and drift velocity in vertical large pipes are different from these in vertical medium pipes. Hibiki and Ishii (2003) developed a set of the drift-flux type correlation for vertical large pipes, and Schlegel et al. (2010) summarizes a comprehensive set of the drift-flux constitutive equations for vertical pipes of various hydraulic diameters.

Similarity and difference of two-phase flow characteristics between vertical medium and large pipes are summarized as follows (Hibiki and Ishii, 2003):

- Two-phase bubbly flow characteristics in vertical large pipes are affected by a presence of cap bubbles and the formation of a secondary flow at a low liquid flow rate resulting in increased distribution parameter and drift velocity in vertical large pipes.
- In vertical large pipes, cap bubbly and cap turbulent regimes appear in the region of the void fraction where slug flow regime appears in vertical medium pipes. This is due to the disintegration of large bubbles into small cap bubbles caused by the surface instability in the large bubbles.
- The constitutive correlations beyond the bubbly flow regime are similar between vertical medium and large pipes.

Based on the similarity in two-phase flow characteristics of churn and annular flows between vertical medium and large pipes, the application of the newly developed correlation of the heat transfer coefficient for two-component two-phase flows in vertical medium pipes, Eqs. (36)–(39), to churn and annular flow regimes in vertical large pipes may be reasonable. The two-phase flow characteristics between vertical medium and pipes are different in the region of the void fraction where slug flow regime appears in vertical medium pipes. In vertical large pipes, cap-bubbly flow and cap-churn flow regimes appear instead of slug flow regime. In the previous section, it has been demonstrated that the newly developed correlation of the heat transfer coefficient for two-component

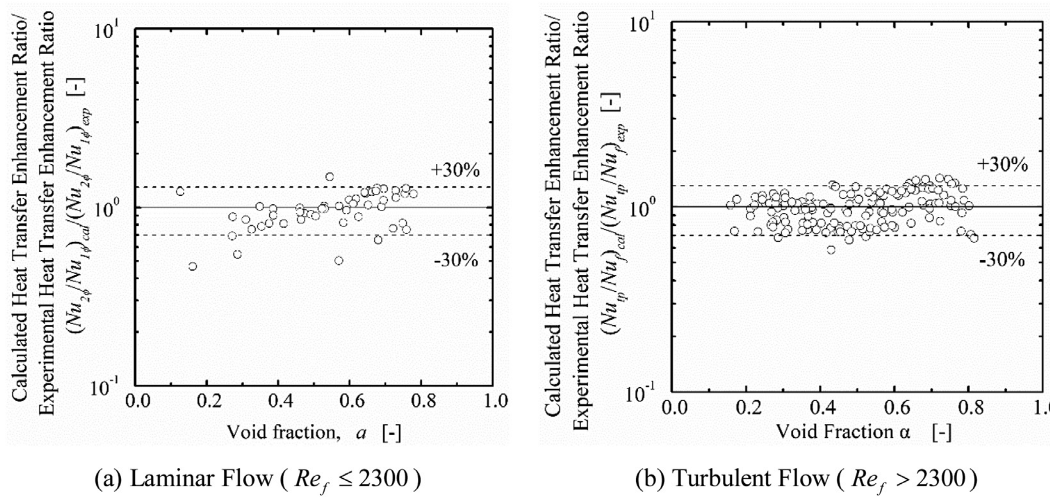


Fig. A1. Predictive capability of simplified heat transfer coefficient correlation for two-component two-phase slug flow heat transfer coefficient in vertical pipes.

Table A1
Performance evaluation of simplified heat transfer coefficient correlations for two-component two-phase slug flows in vertical pipes.

Statistical parameters		m_d [-]	s_d [-]	m_{rel} [%]	$m_{rel,\,ab}$ [%]
Simplified correlation	$Re_f > 2300$	−0.0500	0.422	−4.28	17.4
	$Re_f \leq 2300$	−0.119	0.633	−8.86	20.8
	Total	−0.0668	0.452	−5.40	18.2

two-phase flows in vertical medium pipes can be applicable to bubbly and churn flow regimes, which should be similar to two-phase flow characteristics in cap-bubbly and cap-turbulent flow regimes in vertical large pipes. Based on the similarity in the two-phase flow characteristics, the application of the newly developed correlation of the heat transfer coefficient for two-component two-phase flows in vertical medium pipes, Eqs. (36)–(39), to cap-bubbly and cap-churn flow regimes may be reasonable. Based on a weak function of the newly developed correlation against the void fraction, the application of the newly developed correlation of the heat transfer coefficient for two-component two-phase flows in vertical medium pipes, Eqs. (36)–(39), to the bubbly flow regime may be reasonable. However, due to more agitated bubbly flow in vertical large pipes at low liquid flow conditions, the newly developed correlation may underestimate the two-component two-phase heat transfer enhancement ratio of the bubbly flow in vertical large pipes. The applicability of the newly developed correlation of the heat transfer coefficient for two-component two-phase flows in vertical medium pipes, Eqs. (36)–(39), to vertical large pipes should be reasonable but examined by new data to be taken in a future study.

5. Conclusions

In this study, the heat transfer coefficient for two-component two-phase flows in vertical medium pipes has been investigated. The important achievements are summarized as follows:

- An extensive literature survey identified more than 200 experimental data of the heat transfer coefficient of two-component two-phase slug flows measured in vertical medium pipes. Some experimental data were also collected in bubbly, churn and annular flow regimes. The literature survey showed that a majority of the two-component two-phase flow data was taken for the slug flow regime in vertical medium pipes.
- An extensive literature survey identified 14 and 8 existing correlations for the heat transfer coefficient of two-component two-phase slug flows in vertical and horizontal pipes, respectively. The two-phase heat transfer coefficient has been correlated by some dominant parameters speculated by researchers such as a two-phase multiplier, Φ_f^2, void fraction, α, Reynolds number, Re, Prandtl number, Pr, superficial gas-to-liquid velocity ratio, j_g/j_f, physical property ratio and so on.
- The comparison between the collected existing data and correlations showed that none of the existing correlations could predict the two-component two-phase heat transfer coefficient in the range of the whole database with a sufficient prediction accuracy.
- The functional dependence of the existing correlations on a void fraction and a two-phase multiplier was theoretically explained with the aid of Reynolds and Chilton–Colburn analogies.
- The newly developed correlation for the heat transfer coefficient of two-component two-phase slug flows in vertical pipes has been developed with the aid of Reynolds and Chilton–Colburn analogies. The newly developed correlation could predict 95.1% of the two-phase heat transfer enhancement ratio data within ± 30% error and the mean absolute relative deviation of the correlation was estimated to be 14.2%.
- It was demonstrated that the newly developed correlation was applicable to other flow regimes such as bubbly, churn and annular flow regimes.
- In summary, it is expected that the newly developed correlation can predict the heat transfer coefficient of two-component two-phase flows under various conditions such as vertical upward and downward flows, developing and fully developed flows, laminar and turbulent flows and all two-phase flow regimes. The application of the correlation to vertical large pipes is also considered to be reasonable.

In relation to the implementation of the newly developed correlation into a computational code, an additional discussion on a simplification of the newly developed model is given in Appendix.

Acknowledgments

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Appendix

Several key heat transfer correlations have been modeled by Martinelli parameter or a quality. An example is Chen's correlation (Chen, 1963) for boiling heat transfer. When such a correlation is implemented in a computational simulation code such as USNRC TRACE (Trace V5.0, 2007), the modeling based on quality may induce some numerical instability around the onset of nucleate boiling. Based on this problem, USNRC prefers a correlation based on a void fraction rather than a quality. In view of this, a simplified correlation is developed as follows: Laminar flow ($Re_f \leq 2300$):

$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = (1-\alpha)^{-0.940} \tag{A1}$$

$$Nu_{1\phi} = 1.86\left(Re_f Pr_f \frac{D}{L}\right)^{1/3}\left(\frac{\mu_B}{\mu_W}\right)^{0.14} \tag{A2}$$

Turbulent flow ($Re_f > 2300$):

$$\frac{Nu_{2\phi}}{Nu_{1\phi}}\left(=\frac{h_{2\phi}}{h_{1\phi}}\right) = (1-\alpha)^{-0.756} \tag{A3}$$

$$Nu_{1\phi} = \frac{(f/8)(Re_f - 1000)Pr_f}{1 + 12.7\sqrt{f/8}\left(Pr_f^{2/3} - 1\right)}\left[1 + \left(\frac{D}{L}\right)^{2/3}\right] \tag{A4}$$

This simplification deteriorates the mean absolute relative deviation from 14.2% for the newly developed correlation, Eqs. (36)–(39) to 18.2% for the simplified correlation, Eqs. (A1)–(A4), see Table A1. However, as can be seen in Fig. A1, the simplified correlation predicts the heat transfer coefficient of two-component two-phase flows in vertical pipes reasonably well.

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