An improved heat transfer correlation for condensation inside inclined smooth tubes

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Abstract

To date, there has been no robust model that can satisfactorily predict the condensation heat transfer coefficients in smooth tubes when oriented at some angles other than horizontal and vertical. Therefore, it was the motivation of this investigation to develop a universally acceptable model capable of predicting the heat transfer coefficients during convective condensation inside inclined tubes subject to diabatic conditions. An extensive database of experimental results collected from our previous studies was used in the development of the proposed model. The database consisted of five hundred and fifty-nine data sets for tube orientation varying between - 90° and + 90°, mass velocities 100 kg/m^{2}s to 400 kg/m^{2}s, mean vapour qualities 10% to 90% and saturated condensing temperatures 30 °C to 50 °C. The proposed model showed a magnificient agreement with the experimental data within an global average and mean absolute deviations of −5.74% and 1.13% respectively. The performance of the new empirical model was validated with inclined flow data from three sources in the open literature and was found to predict them with high accuracy.

Keywords

Condensation
Correlation
Heat transfer coefficient
Inclined tube
Flow pattern map
Two-phase flow model

Nomenclature

ABS	absolute
AD	average deviation
c_p	specific heat, J kg^{-1}K^{-1}
C_T	constant
d	diameter, m
Eo	Eötvös number, dimensionless
g	gravitational acceleration, ms^{-2}
G	mass velocity, kgm^{-1}s^{-2}
h_{fg}	latent heat, J kg^{-1}
Ja	Jakob number, dimensionless
J_G	dimensionless gas velocity
J_G^T	transition dimensionless gas velocity
k	thermal conductivity, Wm^{-1}K^{-1}
MAD	mean absolute deviation
N	number of data points
Pr	Prandtl number, dimensionless
Re	Reynolds number, dimensionless
RMSE	root mean square error
T	temperature, K
We	Weber number, dimensionless
x	vapour quality
X_{tt}	Martinelli parameter

Greek symbols

α	heat transfer coefficient, Wm^{-2}K^{-1}
β	tube orientation or inclination, ^{o}
μ	dynamic viscosity, kg/m.s
ρ	density, kgm^{-3}

σ surface tension, Nm^{-1}

Subscripts

exp	experiment
l	liquid
m	mean
pred	predicted
s,sat	saturation
tp	two-phase
v	vapour
w	wall

1. Introduction

For many decades, the focus of many researchers was on the formulation of heat transfer models for condensation inside horizontal and vertical tubes. This was due to their relevance to the available configurations in the heating, ventilation and air-conditioning systems, process and chemical industries, thermal plants, etc. In this regard, various models were formulated for both horizontal [[1], [2], [3], [4], [5], [6]] and vertical flows [[7], [8], [9], [10]], with very limited studies on inclined flows [[11], [12], [13], [14]].

However, in recent times, condensation inside inclined tubes has gained awareness due to its use in large industrial A- and V-frame heat exchangers, and the need to develop heat exchangers for situations where there are space, size and environmental constraints. Its application can also be extended to the cases of refrigeration systems in inclined surfaces such as automotive movement uphill and downhill a slope and during take-off and landing in aeroplanes. For example, in space operations and applications, the understanding of the two-phase heat transfer characteristics in inclined tubes is imperative due to limitations of space and weight.

With respect to horizontal flows, Cavallini et al. [1] suggested a new heat transfer empirical correlation for smooth horizontal tubes of internal diameter larger than 3.0 mm. Their model was found to be suitable for different types of fluids and conditions, They validated their model by comparing it with 4,771 data points relative to CO_2, hydrocarbons, hydro-fluorocarbons, hydrochlorofluorocarbons, ammonia and water from many independent laboratories.

Han et al. [15] carried out experiments with R134a, R22 and R410A condensing inside a 7.92 mm internal diameter smooth copper tube. The test conditions were condensation temperatures from 30 to 40 °C, mass velocities between 95 and 410 kg/m^{2}s and vapour qualities from 0.15 to 0.85. The results of their experiments were compared with eight models developed for the annular flow regime. They proposed a heat transfer coefficient correlation for the annular flow regime in accordance with the heat-momentum relationship. It was found that the formulated correlation provided a very good prediction with its root mean square error of less than 9%.

Dobson and Chato [4] conducted an experimental investigation of flow regimes and heat transfer during the convective condensation of R11, R12, R134a and near azeotropic blends. They noted the flow distribution at the inlet and outlet of the test section and listed the flow regimes observed as stratified smooth, stratified wavy, wavy-annular, slug flows and annular mist. They posited that heat transfer behaviours were controlled by the prevailing flow regime and on that basis grouped the flow regimes into gravity-influenced and shear- influenced flows and concluded that the gravity- influenced regime was dependent on refrigerant temperature difference but independent of mass velocity. They then applied the two-phase multiplier approach to analysing annular flow which was influenced by vapour shear forces and developed a model which was able to give a good prediction of their experiments.

Tandon et al. [16] carried out condensation heat transfer experiments using R12 and R22 as their working fluids. They covered wavy, semi-annular and annular flows and studied the effects of refrigerant mass velocity, vapour quality, condensate film temperature drop and average vapour mass velocity on the average heat transfer coefficient. They noted that their set of data was best correlated by the model of Akers and Rosson [17]. Thereafter, they derived a correlation for the shear- influenced annular and semi-annular flow regimes.

Sapali and Patil [2] studied the convective condensation heat transfer when R404a and R134a were conveyed in both smooth and microfinned tubes in a horizontal orientation, at condensing temperatures

(55–65 °C). The significance of mass velocity and condensing temperature on the convective heat transfer were investigated and they concluded their investigation by proposing two heat transfer correlations for both types of tubes.

Jung et al. [3] performed experiments to determine the heat transfer coefficients during the condensation of pure refrigerants. Their test section was a smooth horizontal tube with an external diameter of 9.52 mm and length of 1.0 m. They examined the effects of vapour fraction and mass velocity on the heat transfer coefficient for various fluids. They concluded their investigation by developing a heat transfer model, which was a modification of the correlation of Dobson and Chato [4].

With respect to vertical flows, Dalkilic et al. [7,8] investigated the effect of mean vapour quality, mass velocity and saturated condensing temperature on the thermal performance during the downward flow of R134a transported in a smooth tube. It was found that the heat transfer coefficient was a notably varied with these parameters and that an increase in the mean vapour quality and mass velocity increased the heat transfer coefficient, but the reverse was the case for the saturated condensing temperature. It was also found that at high mass velocitIes, the interfacial shear effect was significant and they formulated a new correlation for practical applications.

Kim et al. [18] formulated a robust heat transfer model to predict turbulent gas-liquid transported inside vertical pipes using experimentally acquired data available in the literature. Their proposed model which was formulated from 255 data points accounted for the significance of both the gas and liquid phases.

Ghajar and Tang [10] developed a heat transfer empirical model for vertical pipes with respect to void fraction. The model predicted different two-phase flow mixtures with very good accuracy.

With respect to inclined flows, Ghajar and Kim [11] formulated a heat transfer coefficient model for horizontal and slightly inclined upward flow using a 408 data points. Varying the wall heat flux between 3,000 W/m^2 and 10,600 W/m^2 and tube orientation, an optimum tube orientation of 5^o was obtained for slug and bubbly flow, while 7^o was obtained for other flow regimes. The formulated heat transfer coefficient had a mean deviation of −4.22%, a standard deviation of 12.5% and a deviation range between −30.7% and 37.0%.

Chato [19] formulated analytical model to predict stratified-laminar condensation in horizontal and slightly oriented flows. The neglect of the liqiuid at the bottom of the tube and also of the change in void fraction with vapour quality resulted to large deviations at high mass velocities and low vapour fraction. He therefore employed the model of Chen [20] to formulate a correlation for low vapour fractions.

Akhavan-Behabadi et al. [12] studied the convective condensation of R134a in an inclined enhanced tube with an inner diameter of 8.92 mm at mass velocitIes of 54, 81 and 107 kg/m^{2}s, vapour quality of 0.2 to 0.8, tube orientation between −90^o and + 90^o and saturated condensing temperature of 32 °C. From their experimental data, an tube orientation of +30^o was found to give the maximum thermal performance. A heat transfer model that predicted their experimental data within ±10% accuracy, was developed.

Lips and Meyer [21] studied the convective heat transfer of R134a condensing in an inclined smooth tube for various conditions; for mass velocity of 200 to 600 kg/m^{2}s, mean vapour quality between 10% and 90%, tube orientation between -90^o and + 90^o at a saturated condensing temperature of 40 °C. Conclusively, they examined the impacts of some parameters on the heat transfer coefficient. However, they did not formulate a correlation capable of predicting their experimental data.

Other studies have been conducted on the effects of saturated condensing temperature, mass velocity, vapour quality and tube orientation on the heat transfer of R134a condensing in both horizonal and inclined tubes for low mass velocities [31,42] and for relatively high mass velocities [22]. Only Meyer and Ewim [42] formulated a correlation for horizontal tube orientation.

Shah [13,[23], [24], [25], [26], [27], [28]] proposed various empirical models by modifying earlier ones, the first of which was proposed in 1979 using a collection of experimental data from various researchers. The last one was aimed at incorporating the tube orientation so as to capture the effect of inclination on the heat transfer coefficients.

Nada and Hussein [29] Studied the film-wise condensation of saturated vapour on smooth tubes varying the orientation between vertical upward and horizontal flows. The results of the analytic solution only fairly agreed with experimental results for horizontal and vertical tube orientations. The error was observed to increase with tube orientation. Furthermore, a semi-empirical correlation was formulated from the experimental results to predict the range of tube orientations covered in their study.

Xing et al. [30] investigated the flow condensation of R245fa inside an inclined smooth tube at saturated condensing temperature of 55 °C. They considered mass velocitles within the range of 191 to 705 kg/m^{2}s and the vapour quality of 0.19 to 0.95. It was noted that an optimal tube orientation range of 15° and 30° existed and, they formulated an empirical model to predict heat transfer coefficient.

A review of previous works shows that there are just a few models in the literature on condensation inside inclined tubes, most of which were formulated for either slightly inclined, upward or specific flow orientations. Therefore, there is the need for more models that can predict the whole range of flow orientations. Furthermore, to optimise the efficiency and effectiveness of the condenseing units, hence their energy consumption, generalised methods of evaluating their thermal performance are required. Therefore, it was the motivation of this study to present a novel simple heat transfer coefficient correlation. The developed model is compared with the experimental data and 466 other data points obtained from the open literature, and some of the recent models available in the technical papers.

2. Text matrix

The experimental test rig shown in Fig. 1 was a vapour compression consisting of two high pressure lines (the test section and the bypass lines). On the test line were three condensing units (the pre-, test- and post- condensers) and one condensing unit (the bypass condenser) on the bypass line. The pre-condenser was meant to regulate the desired quality entering the test condenser while the post condenser was to ensure a sub-cooled liquid refrigerant passed on to the expansion valve. The bypass condenser controlled the amount of refrigerant supplied to the test line. The test condenser was a tube-in-tube condenser of 1.488 m long. The refrigerant was transported through the inner tube of inner and outer diameters of 8.38 mm and 9.52 mm respectively, while the cold water was pumped through the annulus in a counter flow arrangement. Connected to the test section at both end were high pressure flexible hoses. This enabled the test section to be tilted about a midpoint so that the angle could be measured by a digital inclinometer.

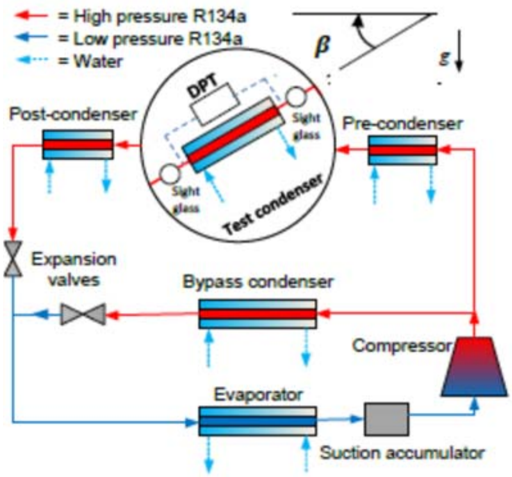

Fig. 1a. Schematic diagram of the test rig.

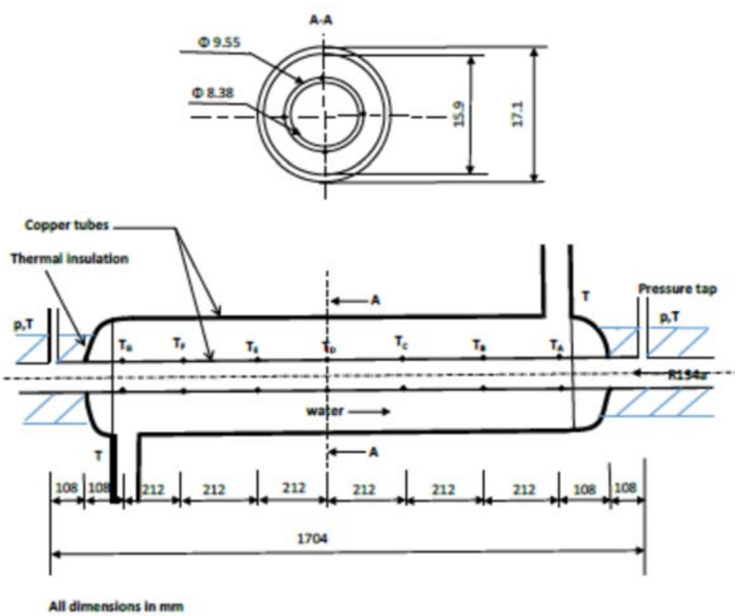

Fig. 1b. Schematic diagram of the test condenser.

Details of the experimental set-up and data deduction procedure have been extensively reported in the studies of Professor Josua Meyer and his co-researchers [21,22,[31], [32], [33], [34], [35], [36], [37], [38], [39], [40], [41], [42], [43], [44], [45], [46], [47], [48], [49]]. In this study, five hundred and fifty-nine (559) of the test data points for saturated condensing temperatures of 30 °C to 50 °C were used. The measured parameters and range of variables used for the experiment are presented in Table 1. Also, the database consisted of 43 data sets as listed in Table 2 regarding the mass velocity, *G,* and mean vapour quality, x_m. Each data point represents thirteen tube orientations considered (i.e. β = −90°, −60° - 30° - 15° -10°, −5°, 0°, 5°, 10°,15°, 30°, 60°, 90°).

Table 1. Parameters and range.

Parameter	**Range**	**Band**
T_{sat}[°C]	30–50	± 0.6
G [kg/m^{2}s]	100–400	± 5
x_m	0.1–0.9	± 0.01
β [°]	-90° – +90°	± 0.1
Q_{H2O} [W]	250	± 20
ΔP [kPa]	−2 - +12	±0.05

Table 2. Experimental data used for all tube orientations (i.e. -90^o ≤ β ≤ +90^o).

T_{sat} **[^{o}C]**	G **[kg/m^{2}s]**	x_m **[−]**
	100	0.5,
30	200	0.25, 0.5, 0.75
	300	0.5
	200	0.25, 0.5
35	300	0.5
	400	0.5
	100	0.25, 0.5, 0.62, 0.75
40	200	0.1, 0.25, 0.5, 0.62, 0.75, 0.9
	300	0.1, 0.25, 0.5, 0.62, 0.75, 0.9
	400	0.5, 0.62, 0.75, 0.9
	200	0.25, 0.5
45	300	0.25, 0.5
	400	0.5
50	200	0.25, 0.5
	300	0.1, 0.25, 0.5, 0.62, 0.75, 0.9
	400	0.5

3. Comparison of experimental results with other models

Five hundred and fifty-nine experimental data points of the convective condensation of R134a inside a smooth inclined tube were evaluated with six different established heat transfer models formulated for inclined tubes, that is, predictive models of Cao et al. [50], Xing et al. [30], Mohseni et al. [51], shah [13,26] and Adelaja et al. [52] using Eqs. (1), (2), as shown in Fig. 2. This was done to check the veracity of the data resulting from the experimental test rig.

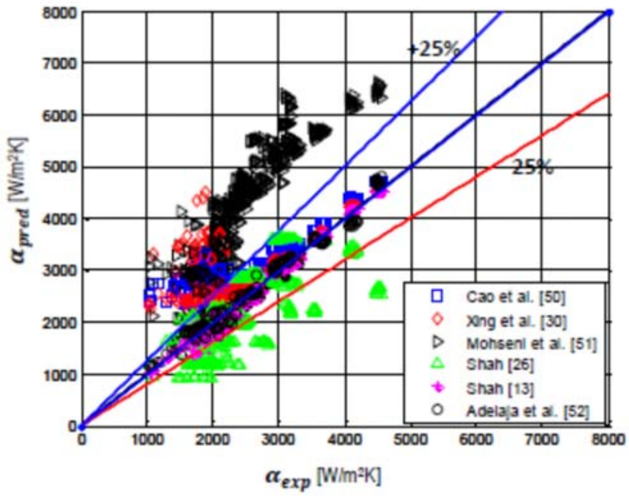

Fig. 2. Experimental data compared with seven inclined tube correlations.

Average deviation (AD):

$$AD = \frac{1}{N}\sum_{1}^{N}\left[\frac{(\alpha_{pred}-\alpha_{exp})\times 100\%}{\alpha_{exp}}\right] \tag{1}$$

Mean absolute deviation (MAD):

$$MAD = \frac{1}{N}\sum_{1}^{N} ABS\left[\frac{(\alpha_{pred}-\alpha_{exp})\times 100\%}{\alpha_{exp}}\right] \tag{2}$$

Fig. 2 shows the heat transfer coefficients of the experimental results of Meyer et al. [22], as predicted by the aforementioned models. All six models were employed to predict the inclined tube data. The statistical analysis; namely, AD and MAD, in Table 3, were used to evaluate the robustness of each of them. The models of Shah [13] and Adelaja et al. [52] performed better than the other four models because they had lower values for AD and MAD. The predictive model of Adelaja et al. [52] possibly showed a very good performance because the same data sets were used for its development. One of the purported reasons for the poor performance of Mohseni et al. [51] could be because it was developed for low mass velocitles where the heat transfer phenomena were quite different.

Table 3. Statistical analysis of deviations of current data by various models.

S/N	Researchers	AD (%)	MAD (%)
1	Cao et al. [50]	−23.05	8.03
2	Xing et al. [30]	−19.33	19.41
3	Mohseni et al. [51]	−72.63	72.63
4	Shah [26]	10.42	17.52
5	Shah [13]	0.61	3.70
6	Adelaja et al. [52]	0.78	5.91
7	Current model	−5.74	1.13

4. Heat transfer model development

As earlier mentioned, a few heat transfer coefficient models have been formulated to predict inclined tube flow. In the current study, the experimental data of [22,32] were employed in the correlation development. The model is based on the flow pattern map of Adelaja et al. [52], which is a modification of the model of Cavallini et al. [1].

The flow pattern map of Cavallini et al. [1] was formulated for horizontal tubes heat transfer data obtained at saturated condensing temperature of 40 °C. The model consists of two components depending on the flow categorisation noted during the experiment. The first classification is comprised of the annular, annular-wavy, intermittent and mist flow. These flow patterns are gravity independent. The second classification is comprised of stratified and stratified-wavy flows. They fall within the gravity-dependent regime. The flow transition criteria used by Cavallini et al. [1] were modified to develop the flow pattern map of Adelaja et al. [52]. The flow pattern was developed on the premise that the Cavallini et al. [1] model was unable to adequately predict the flow distribution for inclined flows as shown in Fig. 3. For every flow pattern map, flows around the transition lines are sometimes not correctly predicted because of the fine line between the transition. For the typical case of flow distribution for mass velocity of 300 kg/m^{2}s, quality of 50%, while Cavallini et al. [1] categorised all the flows in the gravity-dependent regime, the flow distribution in Fig. 3c shows that this is only true for the downward flows, that is, for tube orientations of -10° to -60°. Other flow patterns were observed for other orientations; annular-wavy for 0° to 30°, annular for -90°, 60° and 90°. Another reason why Cavallini et al. [1] could not predict inclined tube data was the assumption that for

, the flow was temperature difference independent. This may be applicable to horizontal tube orientation. However, it is insufficient for inclined tubes. Therefore, the authors agree with the designation of gravity dependence or independence of the two regimes but not with the temperature difference dependence or independence of the regions. This will be expatiated in the next paragraph.

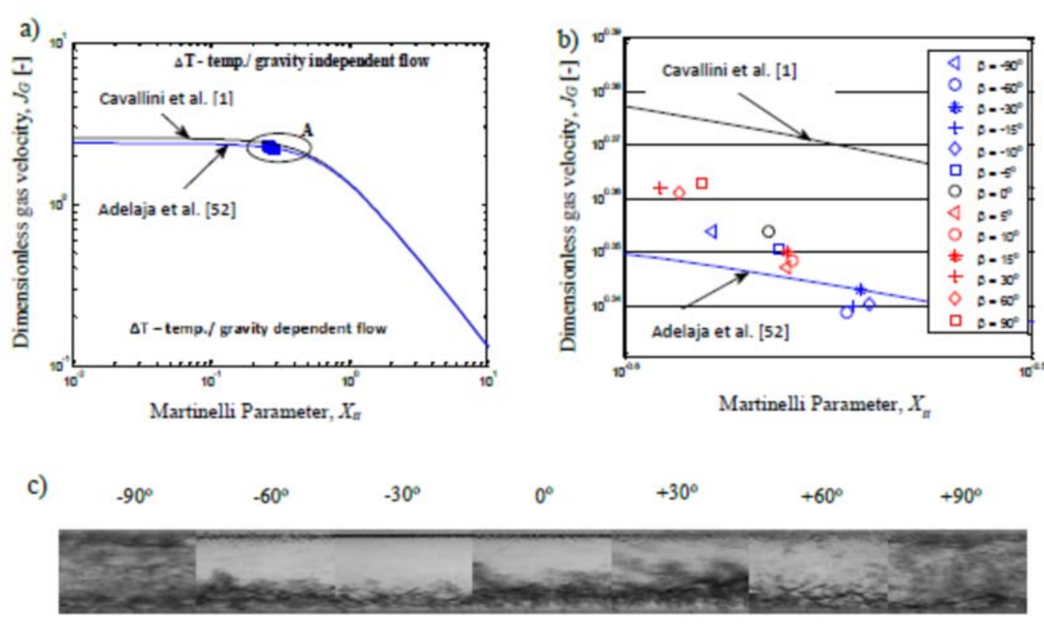

Fig. 3. a). Transition profile for gravity-independent and dependent flow regimes, b). Prediction of a typical experimental data set for mass velocity of 300 kg/m^{2}s, vapour quality of 50% and saturated condensing temperature of 40 °C at the transition curve using Cavallini et al. [1], the current model, c). the flow distribution for this case.

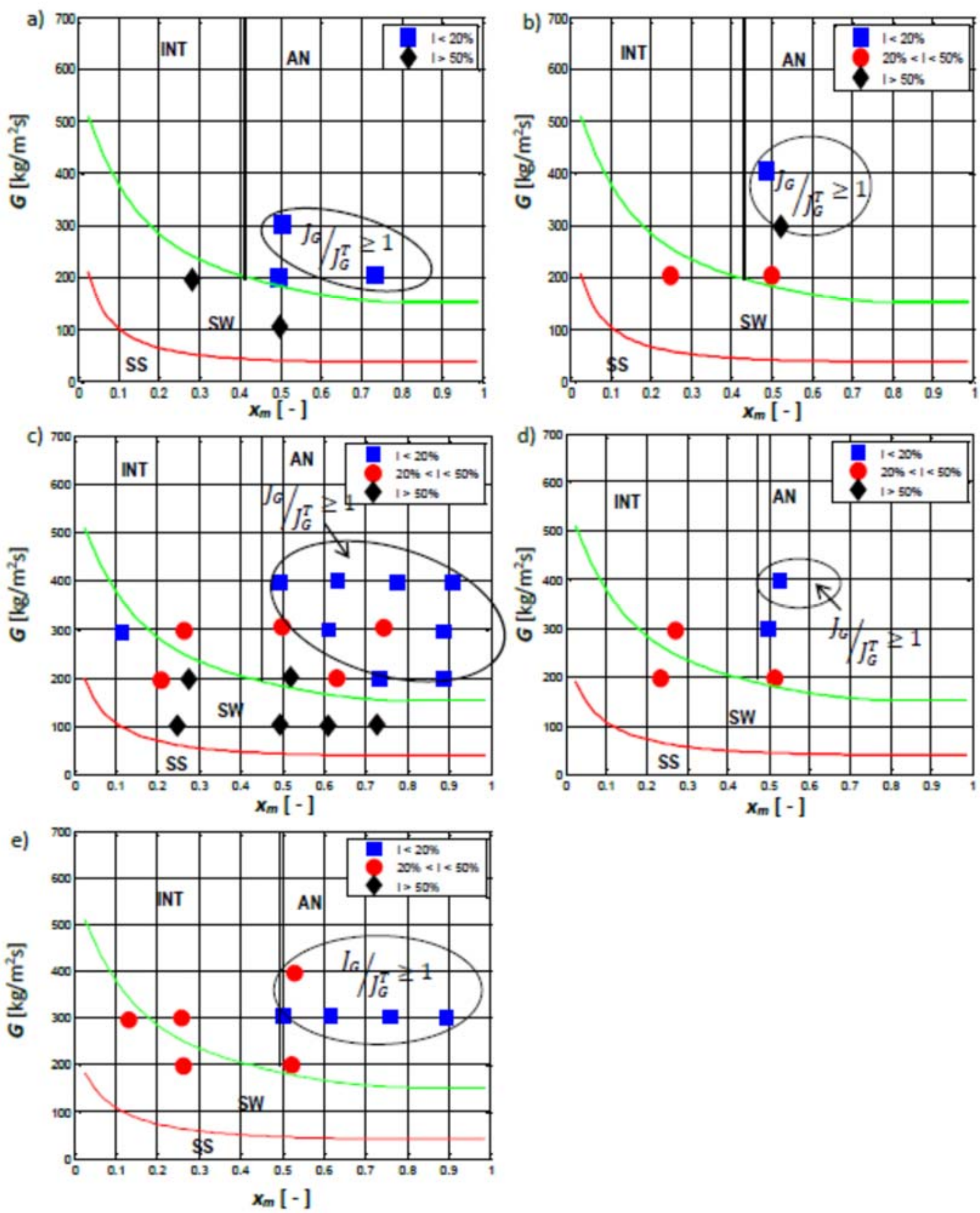

Fig. 4. The effect of tube orientation on the temperature difference on the El Hajal et al. [53] flow pattern map with the superposition of Adelaja et al. [52] transition criterion for saturated condensing temperature of a) 30 °C, b) 35 °C, c) 40 °C, d) 45 °C, e) 50 °C.

Fig. 4 shows the result of the effect of inclination as a function of temperature difference on the El Hajal et al. [53] flow pattern map as expressed in Eq. (3) on the mass velocity and quality for different saturation temperatures. The green line is the transition line between the stratified-wavy and smooth stratified flow patterns; the red line is the transition line between the stratified-wavy and the intermittent flows at the upstream and the annular flow at the downstream while the vertical black line is the transition line between the intermittent and annular flow patterns. SS, SW, INT and AN on the flow map represent the smooth stratified, stratified-wavy, intermittent and annular flows respectively. On the flow map is superimposed the transition criterion of Adelaja et al. [52], which determines whether the flow is gravity independent or not. The enclosed points represent the gravity-independent data, while other points outside the enclosure represent the gravity-dependent data. The figure shows that the effect of inclination can be significant for the gravity-independent region specified by the criterion $J_G/J_G^T \geq 1$. The region where this criterion exists is shown to have inclination effect as large as over 20% for saturated condensing temperature of 40 °C (Fig. 4c) and 50 °C (Fig. 4e) and over 50% for saturation temperature of 35 °C (Fig. 4b). For example, Fig. 4e shows that for the flow with mass velocity of 400 kg/m^{2}s, quality of 50% at saturated condensing temperature of 50 °C, the effect of inclination can be significant (i.e. between 20% and 50%). Also, the flow with mass velocity of 300 kg/m^{2}s, quality of 50% at a saturation temperature of 35 °C has an inclination effect greater than 50%. This inclination effect was captured by Adelaja et al. [52] by incorporating the Jakob number into their condensation model. If the so much-talked-about inclination effect has been catered for, what then is the purpose of the current study?

Fig. 10a shows that the earlier correlation by the authors is not robust enough to predict other inclined tube data, though it performed well with the authors' data sets. The inclined tube data of Xing et al. [30], Cao et al. [50] and Lips and Meyer [21] were evaluated with the model. While Lips and Meyer [21] used R134a, Xing et al. [30] used R245fa in tube-in-tube condensers; Cao et al. [50] used R245fa in a shell-and-tube heat exchanger. The full information regarding the fluid types, types of heat exchangers, saturation temperatures, range of mass velocitIes, qualities and tube orientations is presented in Table 4. Statistical analyses show that it was unable to predict the tested data with reasonable accuracy. This poor performance is due to the neglect of some phenomena characterised by the Weber number, Prandtl number ratio and the ratio of viscosity. Weber number, Eq. (11), accounts for the relative importance of the fluid inertia compared with the surface tension and is useful for the analysis of thin film flow and the formation of bubbles and droplets. The Prandtl number ratio accounts for the relative importance of the momentum diffusivity to thermal diffusivity of the vapour to liquid, while the viscosity ratio accounts for the relative viscosity of the liquid-vapour compared with the liquid viscosity.

Table 4. Operating conditions of validated data.

S/N	**Researchers**	**Types of heat exchanger**	**Tube Length m, Diameter mm**	**Refrigerant**	**Saturation temperature [°C]**	**Mass velocity kg/m^{2}s**	**Quality [−]**	**Tube orientation [°]**	**Boundary condition**
1	Cao et al. [50]	Shell-and-tube	1.6, 14.7	R245fa	63.1	199.0–699.2	0.2–0.64	−30, 0, +30	Diabatic
2	Xing et al. [30]	Tube-in-tube	1.2, 14.81	R245fa	55.48	199.0–699.2	0.1–0.69	−90, 0, +90	Diabatic
3	Lips and Meyer [21]	Tube-in-tube	1.488, 8.38	R134a	40	200–400	0.1–0.9	−90 - +90	Diabatic
4	Gu et al. [55]	Tube-in-tube	1.0, 4.57	R1234ze(E)	40	300, 600	0.55–0.95	−90 - +90	–
5	Wang and Du [56]	Tube-in-tube	1.0, 1.94/2.8	R718	100	10–100	0.18–0.88	0, 17, 34, 45	Diabatic

The proposed model comprised two parts; the first part predicts the gravity-independent flows, that is, the annular-wavy, annular, intermittent and misty flows as represented in Eq. (7) based on the flow transition criterion $J_G/J_G^T \geq 1$. The second part predicts the gravity-dependent regime, that is, stratified-wavy and stratified smooth, written in Eq. (12).

These equations are expressed in terms of dimensionless parameters given in Eq. (4) - Eq. (6), Eq. (8) - Eq. (10), Eq. (11) and Eq. (13).

The effect of inclination on the temperature difference is presented in new Eq. (3 a, b) as

$$I = f(\Delta T) \tag{3a}$$

$$I = \frac{\Delta T_{max} - \Delta T_{min}}{\Delta T_{\beta=0}} \tag{3b}$$

where, ΔTis the temperature difference between the saturation and wall temperatures, however, this value changes depending on the inclination angle. ΔT_{max} and ΔT_{min} are the maximum and minimum values respectively obtained as the inclination angle is changed from -90° to +90°. $\Delta T_{\beta=0}$ is the temperature difference between the saturation and wall temperatures for horizontal tube orientation. Employing the model of Cavallini et al. [1], the gas velocity, J_G, is used to designate where the flow lies. It is expressed as a function of mass velocity G, vapour quality x_m, tube inner diameter d, acceleration due to gravitation g, and the liquid and vapour phase densities, ρ_l and ρ_v:

$$J_G = \frac{x_m G}{[gd\, \rho_v(\rho_l - \rho_v)]^{0.5}} \tag{4}$$

The transition value J_G^T, is dependent on the fluid type and tube orientation. It is expressed in Eq. (5) and accounts for the turbulent nature of the flow:

$$J_G^T = \left\{ \left[7.5/\left(4.3X_{tt}^{1.111} + 1\right)\right]^{-3} + C_T^{-3} \right\}^{-\frac{1}{3}} \tag{5}$$

where the value of C_T was taken as 2.6 by Cavallini et al. [1]. This was modified by Adelaja et al. [52] as 2.4 to capture the effect of inclination and flow distribution. X_{tt} is the Martinelli parameter and is given as:

$$X_{tt} = ((1 - x_m)/x_m)\hat{}\ 0.9\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} \tag{6}$$

where μ_l and μ_v are the viscosities (dynamic) of the liquid and vapour respectively.

For the gravity-independent flows ($J_G \geq J_G^T$), Eq. (7) is employed to model the heat transfer coefficient, α_{tp}:

$$\alpha_{tp} = \alpha_l \left[a \left(\begin{array}{c} 1 + 177.0306 X_{tt}^{-0.2350} \left(\frac{Pr_v}{Pr_l}\right)^{2.85357} \left(\frac{\mu_l - \mu_v}{\mu_l}\right)^{18.07788} Ja^{-0.1405} \left(\frac{J_G}{J_G^t}\right)^{-0.1779} \\ +b\left(1 + 0.3119 x_m^{0.6683} X_{tt}^{-0.024} \left(\frac{Pr_v}{Pr_l}\right)^{-0.2117} We_l^{-0.0308}\right) \end{array} \right) \right] \tag{7}$$

here a = 0.6, b = 0.4, α_l is the liquid only heat transfer coefficient expressed by Dittus-Boelter Eq. [54] and presented in Eq. (8):

$$\alpha_l = 0.023\, Re_l^{0.8}\, Pr_l^{0.4}\, {}^{k_l}/_{d} \tag{8}$$

Where
$$Re_l = G(1-x_m)d/\mu_l \tag{9}$$

where Pr_v and Pr_l are the vapour and liquid Prandtl numbers respectively, Ja is Jacob number and We_l is the liquid Weber number and k_l is the liquid thermal conductivity.

The Jakob number, Ja, signifies the proportion of sensible to latent heat and accounts for the contributions of superheating and subcooling, heat capacity of the liquid, $c_{p,\,l}$, the difference in the saturation T_{sat} and wall temperatures T_w and the latent heat of vaporisation, h_{fg}.

$$Ja = \frac{c_{p,l}(T_{sat}-T_w)}{h_{fg}} \tag{10}$$

$$We_l = \frac{G^2(1-x_m)^2 d}{\rho_l \sigma} \tag{11}$$

The heat transfer coefficient for the gravity dependent flows ($J_G < J_G^T$) is presented in Eq. (12):

$$\alpha_{tp} = \alpha_l\left[1+1.1\left(X_{tt}^{-0.2350}\left(\frac{Pr_v}{Pr_l}\right)^{2.85357}\left(\frac{\mu_l-\mu_v}{\mu_l}\right)^{18.07788}Ja^{-0.1405}\left(\frac{J_G}{J_G^t}\right)^{-0.1779}\right)*\left(1+12404E\ddot{o}^{-0.8439}(cos(\beta))^{0.005479}\right)\right] \tag{12}$$

where β is the tube orientation and $E\ddot{o}$ is the Eötvös number (Eq. (13)):
$$E\ddot{o} = \frac{(\rho_l-\rho_v)gd^2}{\sigma} \tag{13}$$

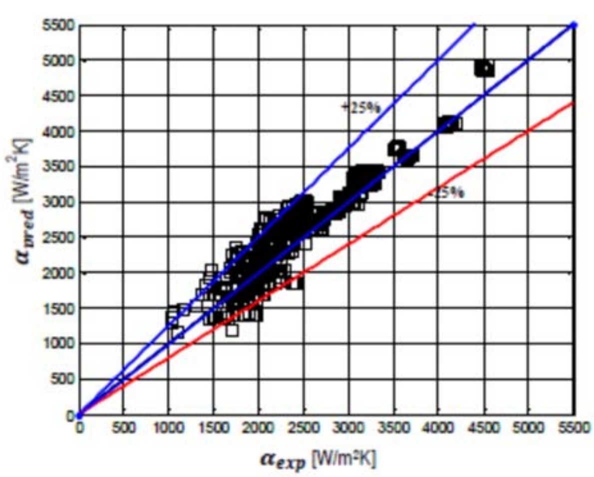

Fig. 5. Comparison between the current model and experimental data.

5. Model comparison with experimental results

Fig. 5 presents the application of the proposed model to the experimental data. The experimental data contain 43 data points each for the thirteen tube orientations considered between -90° and + 90°; five hundred and fifty-nine data points in all. AD and MAD are −5.74% and 1.13% respectively showing that the model is capable of predicting the experimental data accurately. Better accuracy was obtained for the high heat transfer coefficient region, which corresponded to the high mass velocitles, high quality flow rather than to the low heat transfer coefficient regime, that is, the low mass velocitles, low vapour qualities.

6. Evaluation of proposed model

In this section, the authors attempted to conduct a holistic analysis of all the variables that influence the thermal performance during flow condensation in inclined tubes. The effects of tube orientation, saturated condensing temperature, mass velocity and quality on the proposed flow condensation model were investigated.

Fig. 6 presents the variation of ratio of the heat transfer coefficient for the predicted to experimental data with tube orientation. Results reveal that there is a dense concentration of the data points around the 1.0 (100% accuracy) for all the tube orientations. The data for 30° inclination are the best predicted as all the data points fall within ±25% deviation, while the worst predicted are data for the -90° inclination.

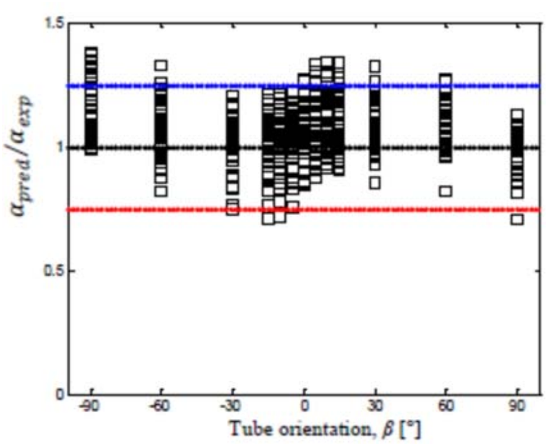

Fig. 6. Predicted to experimental heat transfer coefficient ratio versus tube orientation.

Fig. 7 presents the variation of saturated condensing temperature with the ratio of the predicted model to the experimental heat transfer coefficient within the ±25% deviation. The result shows a very good agreement for all data for the different saturation temperatures. The data points for the saturation temperature of 50 °C performed best as all of the data points are within the ±25% deviation. Considering the data points outside the deviation, 40 °C would be said to perform the worst. This may be attributed to the fact that the data points obtained for this saturation temperature account for about 47% of the total data analysed (see Table 2).

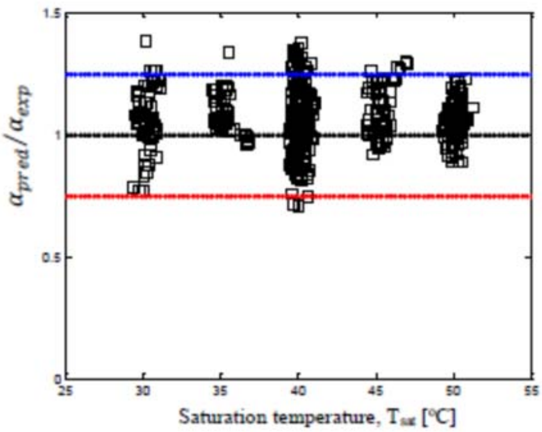

Fig. 7. Predicted to experimental heat transfer coefficient ratio versus saturation temperature.

Fig. 8 shows the ratio of the predicted heat transfer coefficient to the experimental heat transfer coefficient as it varies with mass velocity for the 559 data points. The results indicate that all the data for the mass velocity of 400 kg/m²s fall within ±25% deviation and are the best predicted, while the data points for 300 kg/m²s are the worst predicted. This may be attributed to the fact that the data for a mass velocity of 400 kg/m²s are the least represented, while that of mass velocity of 300 kg/m²s are the most represented (see Table 2).

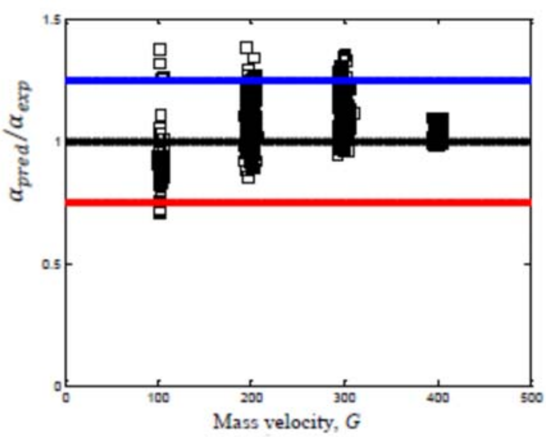

Fig. 8. Predicted to experimental heat transfer coefficient ratio versus mass velocity.

Fig. 9 shows the variation of the ratio of predicted to experimental heat transfer coefficient with vapour quality. The result shows that most of the data points agree within the ±25% deviation. The 90% quality is the most performing of the qualities, while the 25% quality has the worst performance. Also, it should be noted that the 90% quality is the least represented of all the data considered (see Table 2). Also, the flow patterns observed during the experiment show that data for 25% present a wide variety of flow distribution during the inclination; these range from intermittent (slug, plug and elongated bubbles), churn (bubbly/slug and annular/ bubbly/ slug) and stratified (smooth and wavy) flows.

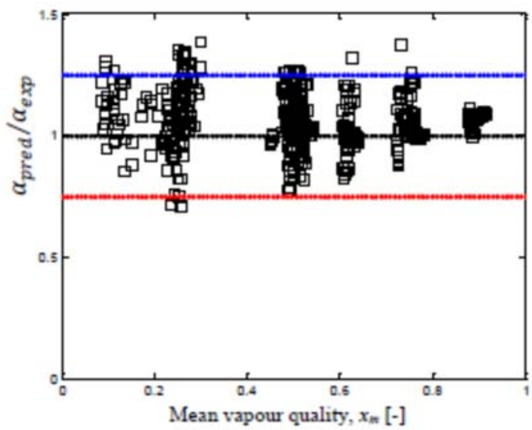

Fig. 9. Predicted to experimental heat transfer coefficient ratio versus mean vapour quality.

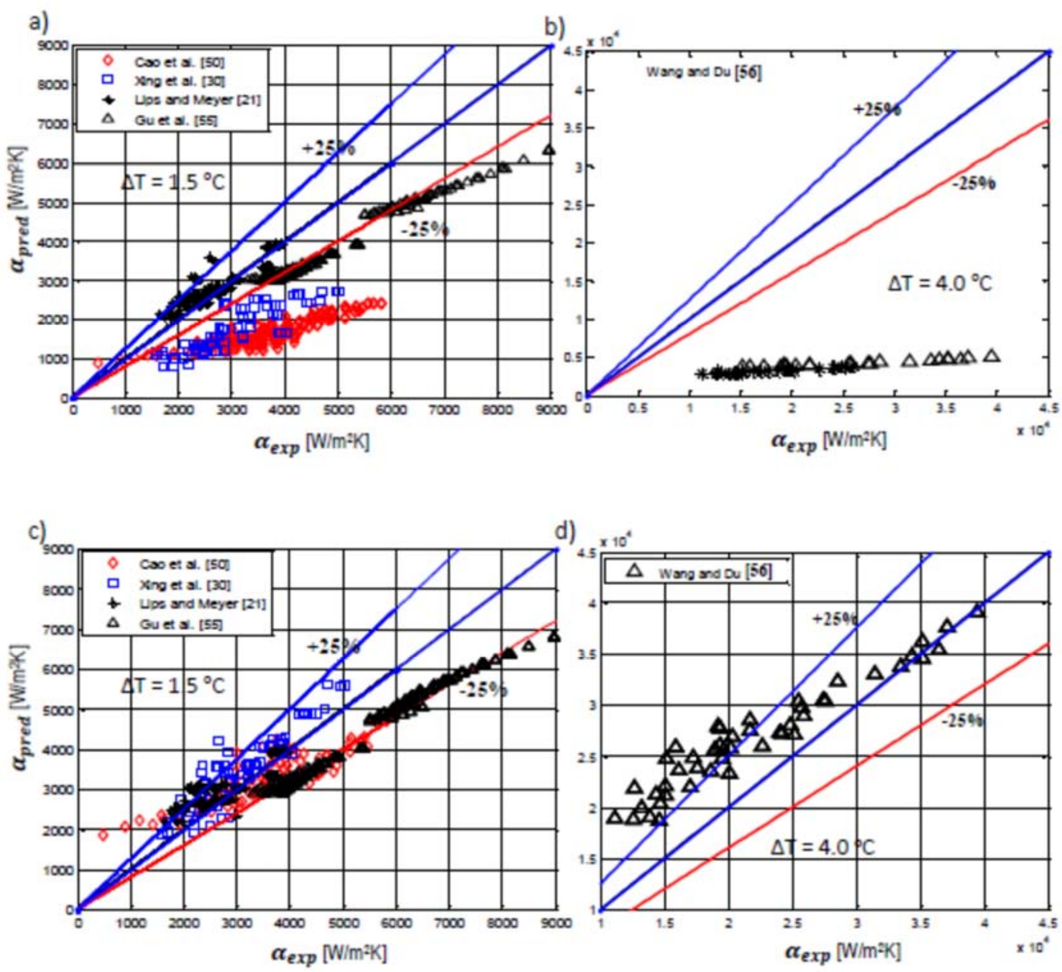

Fig. 10. Comparison of the condensation heat transfer coefficient models of a) & b) Adelaja et al. [52] and c & d) current study with the data of the R245fa in tube-in-tube condenser of Xing et al. [30]; R245fa in shell-and-tube of Cao et al. [50], R134a in tube-in-tube condenser of Lips and Meyer [21], R1234ze(E) in tube-in-tube condenser of Gu et al. [55] and, R718 in tube-in-tube condenser of Wang and Du [56].

Fig. 10 show the validation of Adelaja et al. [52] and the current model using the inclined tube flow condensation data of Cao et al. [50], Xing et al. [30], Lips and Meyer [21], Gu et al. [55] and Wang and Du [56]. The operating conditions, geometric parameters and the boundary conditions for the various experiments are listed in Table 4. Fig. 10a and b show that though the condensation model of Adelaja et al. [52] gave a very good prediction of the inclined tube data of Ref ([22,32]), as can be seen in Table 5, it is not robust enough to predict other inclined tube data. Using Ref ([52]) to predict the aforementioned data, the following statistical analyses gave a poor result. For example, the AD and MAD analyses gave 51.51% and 53.28% respectively for Cao et al. [50], Xing et al. [30] gave −42.73% and 42.73% respectively, Lips and Meyer [21] gave 8.94% and 4.81% respectively, Gu et al. [55] gave 22.08% and 22.08% respectively, while Wang and Du [56] gave 82.22% and 82.22% respectively. With the current model, better results are obtained in Fig. 10 c and d. For example, AD and MAD are 0.17% and 19.34% respectively for Cao et al. [50]; −13.05% and 19.25% respectively for Xing et al. [30]; −11.00% and 16.03% respectively for Lips and Meyer [21]; 20.01% and 20.01% respectively for Gu et al. [55], and 1.51% and 22.77% respectively for Wang and Du [56].

Table 5. Statistical analysis of other experimental data by the current model.

S/N	**Researchers**	**AD (%)**	**MAD (%)**	**AD (%)**	**MAD (%)**
		Current model		Adelaja et al. [52]	
1	Cao et al. [50]	0.17	19.34	51.15	53.28
2	Xing et al. [30]	−13.05	19.25	−42.73	42.73
3	Lips and Meyer [21]	−11.00	16.03	8.94	4.81
4	Gu et al. [55]	20.01	20.01	22.08	22.08
5	Wang and Du [56]	1.51	22.77	82.22	82.22

This is one of the few models that have attempted to predict the variety of complex flows that occur during flow condensation in inclined tubes.

7. Conclusions

A new heat transfer coefficient model was formulated for inclined tubes according to the flow pattern map of Adelaja et al. [52] model. The current study showed that the region said to be temperature difference independent may not be applicable to inclined flows, though, the claims of gravity independence are upheld. The model was formulated based on the 559 inclined tube data of Meyer et al. [22] and was found to give a very good predictive accuracy. AD and MAD were − 5.74% and 1.13% respectively.

For validation, the model was compared with the inclined tube flow condensation data of Cao et al. [50] for R245fa flowing in shell-and-tube heat exchanger, Xing et al. [30] for R245fa flowing in tube-in-tube condenser, Lips and Meyer [21] for R134a in tube-in-tube condenser, Gu et al. [55] for R1234ze(E) in tube-in-tube condenser and Wang and Du [56] for water/steam in tube-in-tube condenser. Statistical analyses showed that Cao et al. [50] gave AD and MAD values of 0.17% and 19.34% respectively; Xing et al. [30] -13.05% and 19.25% respectively; Lips and Meyer [21] -11.00% and 16.03% respectively, Gu et al. [55] 20.01% and 20.01% respectively and wang and Du [56] 1.51% and 22.77% respectively.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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