EXPERIMENTAL INVESTIGATION ON FLOW CONDENSATION PRESSURE DROP OF STEAM IN A HORIZONTAL TUBE

by

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Original scientific paper
https://doi.org/10.2298/TSCI211212061G

The pressure drop characteristics of pure steam condensation are studied in a horizontal tube with an inside diameter of 38 mm and a length of 3.4 m. The mass flux ranges from 3-7.5 kg/(m²s). The saturation temperatures of the inlet steam are 50 °C, 60 °C, and 70 °C. The temperature difference between the inlet steam and the inlet cooling water is from 3-7 °C. The influences of mass flux, total heat transfer temperature difference and inlet saturation temperature total pressure drop are investigated. The new correlation is proposed to calculate the total pressure drop of pure steam flow in a horizontal tube. The Dukler correlations based on the homogeneous flow model are good predictors for calculating the frictional pressure drop in this work.

Key words: *pressure drop, steam condensation, horizontal tube*

Introduction

Low-temperature multi-effect desalination (LT-MED) is dominant in thermal desalination technology. For the steam condensation in horizontal tubes of the LT-MED system, gravity acts as the main driving force. The condensate gathers at the bottom of the tube when the steam flows and condenses in the tube. This phenomenon is the stratified flow which has a large impact on the characteristics of two-phase flow.

Many scholars have conducted various experimental studies [1-5] of the steam condensates in horizontal tubes. The range of the mass flux in these experiments is from 100-800 kg/m^{2}s. The results of these studies indicate that the total pressure drop decreases with the decrease of mass flux. The saturation temperature has less effect on pressure drops. To predict the total pressure drop, the formulas obtained by different scholars are limited for the different experimental conditions.

Ferguson and Spedding [6] conducted an experimental and comparative study on the pressure drop of air-water two-phase flow in a horizontal glass tube with an inner diameter of 9.35 mm and a length of 12.8 m. They found that the model proposed by Olujic [7] was the most accurate, especially in stratified flow states. They also recommend corresponding prediction models for different flow patterns and explain the causes of prediction errors for specific flow patterns.

Cavallini [8] conducted experimental research on the total pressure drop for refrigerants. The experimental result showed that the pressure drop reaches a maximum at the saturation temperature of 30 °C and decreases with the decrease of mass flux. Dalkilic [9] studied

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the condensation process of R600a in a smooth tube and experimented with the process of R134a flow in a vertical tube. After comparing the experimental results, the model of Chen [10] predicted the experimental results well. The experiments of R134a condensation in parallel micro-channel tubes were carried out by Goss [11]. The pressure drop increased with the decreasing saturation temperature and decreased with decreasing mass flux, but the heat flow rate did not play a role in the pressure drop. Müller-Tribbe [12] found the predictions of Müller-Steinhagen and Heck [13] were accurate. However, these empirical formulas were accurate for their experimental condition, but could not reach agreements in prediction results at different conditions. Therefore, there is a need to do more study to determine the applicability of the accuracy to specific conditions. Hwang [14] investigated the characteristics of pressure drop for various types of enhanced titanium tubes. By comparing the results of experiments and calculations, the Blasius equation had good prediction accuracy. Xu *et al.* [15] collected 3480 data points from the previous pressure drop experiments, calculated and compared the friction pressure drop model of 29 in-pipe two-phase flow six-bucket cabinet, and studied the influence of steam dryness, mass flux, fluid physical properties and pipe diameter on the friction pressure drop. The results showed that the models of Muller-Steinhagen and Heck [13] and Sun and Mishima [16] could accurately predict the pressure drop under different experimental conditions.

The current research focuses on the pressure drop of refrigerants condensation process at high mass flux, but there are few studies on pure steam condensation at low mass flux. Therefore, it is of great significance for experimental research under LT-MED conditions.

Experimental apparatus

An experimental apparatus is established to investigate the pressure drop of low mass flux steam flow in a horizontal tube. The system consists of four parts, which are the medium supply system, the experimental section, the medium recovery system and the data recording system, as shown in fig. 1. The characteristics of the used measuring instruments are shown in tab. 1.

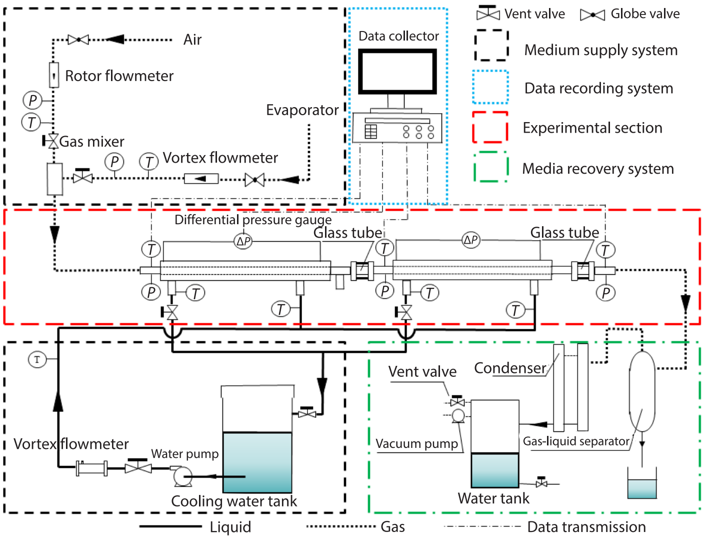

Figure 1. Schematic of experiment system

The steam is produced in an evaporator consisting of six 6 kW heating rods. The evaporation power is controlled by adjusting the power regulator, which controls the mass of

Table 1. Measurement instruments and uncertainties

Instruments	Parameters	Range	Uncertainties
Differential pressure transducer GE LPM1512-C1SNW-1	Differential pressure	–100 kPa to 100 kPa	0.04%
Absolute pressure GE 5073-TC-A3-CA-H0-PA	Absolute pressure	–1 kPa to 1 kPa	0.02%
Flowmeter Shanghai Weikuang LZB-25	Mass-flow rate	0.08-0.83 kg/s	0.4%
Vernier caliper	Length	0-150 mm	±0.02 mm
Data acquisition YOKOGAWA GM10+ GX90XA	Data	–	–

vapor produced in a different experimental condition. The bottom of the cooling water tank is equipped with two 9 kW heating rods, which are used to regulate the temperature of inlet cooling water. Before the cooling water entered the experimental tube section, the cooling water volume is adjusted by valves. The temperature of the cooling water is measured by *K*-type thermocouples.

There are two sections of the experimental equipment and each section is a segmented casing heat exchanger. The length of each section is 1.7 m. The inner tube is an aluminum brass tube with an inside diameter of 38 mm and a wall thickness of 1 mm. The roughness of the tube wall is 0.008 mm. The pressure sensors and temperature sensors are equipped at the inlet and outlet of the tube to measure the pressure and temperature. A quartz glass tube is also installed between the two tube sections to observe the changes in fluid-flow patterns during the experiment. Each experimental tube section is equipped with a differential pressure sensor to record the pressure changes of the condensation flow in the tube during the experiment.

After passing through the tube, the two-phase flow enters the recovery system. This system consisted of three parts, gas-liquid separator, condenser and vacuum pump. The gas-liquid separator is equipped with a level gauge and the mass of the condensate flow in the experimental tube section is obtained by measuring the height of the level gauge rise per second. After passing through the gas-liquid separator, the steam entered the condenser. The condenser is equipped with a level gauge and the mass of uncondensed steam at the outlet of the experimental tube section is obtained by measuring the height of the level gauge per second. The condenser is equipped with a vacuum pump at its tail to provide the vacuum required for the entire experimental tube section.

The data collector recorded real-time data values of 86 channels during the experiment, including pressure, differential pressure and temperature values at each position. Each set of experimental data is measured three times and averaged as the measured value of the experiment.

Data processing

There are two ways to calculate the total heat exchange of steam condensing in the experimental sections. One way is to calculate the temperature differences of the inlet and outlet cooling water in each experimental section:

$$Q = Q_1 + Q_2 \tag{1}$$

$$Q_\mathrm{i} = (T_\mathrm{c,out,i} - T_\mathrm{c,in,i})\dot{m}_{\mathrm{c},i} c_\mathrm{p} \tag{2}$$

where $i = 1$ or 2, Q – the total heat exchange in the experimental tube, Q_i – the heat exchange of condensation in the i^{th} experimental section, $T_{\text{c,in},i}$ and $T_{\text{c,out},i}$ are inlet and outlet cooling water temperature, respectively, $\dot{m}_{\text{c},i}$ – the mass-flow rate of the cooling water passing through the tube, which is obtained by a rotameter.

According to the structure of the experimental apparatus, it can be learned that the inlet of the first experimental section is pure steam, so we take the inlet vapor quality as 1. Similarly, according to the principle of energy conservation, it is deduced that the inlet and outlet vapor quality of the i^{th} experimental tube can be calculated:

$$x_{\text{out},1} = x_{\text{in},2} = 1 - \frac{Q_1}{\dot{m}r} \tag{3}$$

$$x_{\text{out},2} = 1 - \frac{Q_1 + Q_2}{\dot{m}r} \tag{4}$$

where x is the quality of steam at the inlet and outlet positions of the tube. Therefore, the average quality of steam in the i^{th} experimental section can be derived:

$$x_i = \frac{1}{2}(x_{\text{in},i} + x_{\text{out},i}) \tag{5}$$

where $i = 1$ or 2 and x – the average vapor quality in the i^{th} experimental section. In this study, vapor quality is regulated by mass-flow rate of cooling water and total heat transfer temperature difference of cooling water. The mass-flow rate of cooling water used in the experiment is between 0.08 and 0.83 kg/s. The temperature difference between inlet steam and inlet cooling water is 3 °C, 5 °C, and 7 °C.

The total temperature difference $\Delta T_{\text{s,c}}$ is the difference between the inlet saturated steam temperature and the inlet cooling water temperature, which can be derived:

$$\Delta T_{\text{s,c}} = T_{\text{s,in}} - T_{\text{c,in}} \tag{6}$$

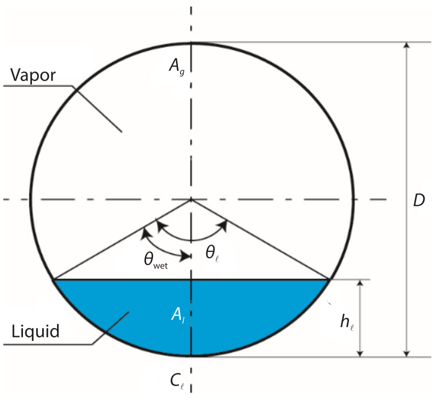

Figure 2. The distribution diagram of the vapor-liquid phase on the cross-section

Figure 2 shows the relationship between the vapor-liquid-flow cross-section and the area flowed by the vapor, which can be expressed:

$$\frac{A_{\text{g}}}{A_{\text{g}} + A_\ell} = 1 - \frac{\theta_\ell}{360} = 1 - \frac{\theta_{\text{wet}}}{180} \tag{7}$$

where A_{g} is the area occupied by the vapor on the cross-section, A_ℓ – the area occupied by the liquid on the cross-section, and θ_{wet} – the wetting wall angle, which can be calculated:

$$\theta_{\text{wet}} = \frac{C_\ell}{D}\frac{\pi}{180} \tag{8}$$

where C_ℓ is the wetted perimeter on the cross-section which is shown in fig. 2.

The void fraction, α_g, of the vapor phase is called the vapor quality or void fraction and it equals the ratio of the volume occupied by a vapor in the multi-phase flow to the total volume:

$$\alpha_g = \frac{V_g}{V} \tag{9}$$

Normally the subscript g in α_g could be omitted and a can be calculated:

$$\alpha = \frac{A_g}{A_t} = \frac{A_g}{A_g + A_\ell} \tag{10}$$

The total pressure drop, ΔP, consists of three parts, the static pressure drop ΔP_{static}, the frictional pressure drop ΔP_{fric} and the momentum pressure drop ΔP_{mom}:

$$\Delta P = \Delta P_{\text{static}} + \Delta P_{\text{fric}} + \Delta P_{\text{mom}} \tag{11}$$

The total pressure drop, ΔP, is obtained from the experiment. For the horizontal tube, the static pressure drop ΔP_{static} is zero. So in this study, the frictional pressure drop ΔP_{fric} could be calculated:

$$\Delta P_{\text{fric}} = \Delta P - \Delta P_{\text{mom}} \tag{12}$$

where the momentum pressure drop, ΔP_{mom}, is calculated by the void fraction model [17]:

$$\Delta P_{\text{mom}} = G^2 \left\{ \left[\frac{(1-x)^2}{\rho_\ell (1-\alpha)} + \frac{x^2}{\rho_g \alpha} \right]_{\text{out}} - \left[\frac{(1-x)^2}{\rho_\ell (1-\alpha)} + \frac{x^2}{\rho_g \alpha} \right]_{\text{in}} \right\} \tag{13}$$

where the void fraction a can be calculated by Chisholm [18] correlation.

The momentum pressure drop ΔP_{mom} which due to momentum change reflected the continuous kinetic energy change of the fluid. For condensing flow, the continuous fluid changes from the steam state at the inlet to the fluid state at the outlet. This process does not produce acceleration losses but rather a pressure recovery and shrinkage of volume.

Experimental results and analysis

Analysis of experimental results of the total pressure drop

The effect of mass flux of two-phase flow in the tube G and steam quality x on the total pressure gradient is shown in fig. 3, where the steam inlet saturation temperature $T_{s,\text{in}}$ is 60 °C. The increasing mass flux and vapor quality rise total pressure gradient. The total pressure gradient increased more rapidly at a higher mass flux. This is because, at a high mass flux, the steam velocity is higher, the wall surface has a greater friction resistance on the steam and the pressure gradient is larger. As shown in fig. 3, the total pressure gradient of steam flow decreased with the decrease of vapor quality.

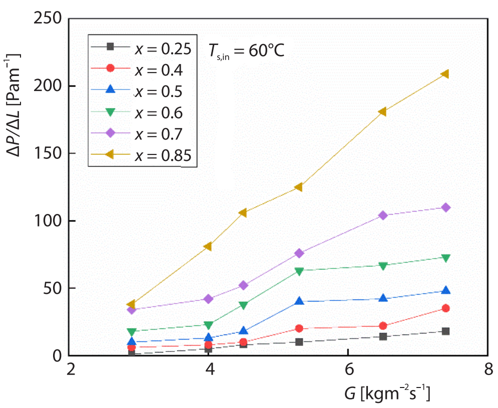

Figure 3. Effect of *G* and steam quality *x* total pressure gradient

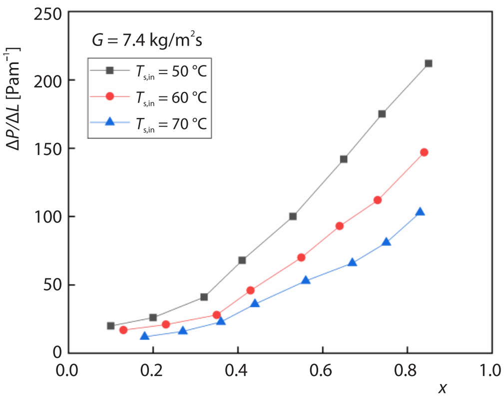

Figure 4. Effect of the inlet saturation temperature on the total pressure gradient

Figure 4 shows the relationship between total pressure gradient and vapor quality at inlet mass flux $G = 7.4$ kg/m^{2}s. The total pressure gradient decreases with the decrease of vapor quality. This is because, under the condition of high steam quality, the steam flow speed is fast, the momentum loss caused by friction with the wall surface becomes larger, which causes the total pressure gradient to rise. The total pressure gradient decreases with the increase of inlet saturation temperature. This is because the saturation temperature has a big impact on the physical characteristics of steam and condensate. As the steam density decreases with the decrease of saturation temperature, the steam velocity increases with the decrease of saturation temperature at the same mass flux. In the process of condensation, a liquid film formed on the upside of the wall, so the dynamic viscosity of vapor-liquid two-phase flow is an important factor. With the increase of saturation temperature from 50-60 °C and 70 °C, the dynamic viscosity of steam increases by 3% and 6%, respectively, while the dynamic viscosity of condensate decreases by 14.7% and 26.2%, respectively. The dynamic viscosity of condensate is about 36-50 times as steam, so the change of dynamic viscosity of condensate has a greater influence on flow. At high saturation temperature, lower steam velocity and smaller dynamic viscosity of condensate reduce the frictional pressure drop of flow significantly, therefore, the total pressure gradient decreases with the increase of saturation temperature.

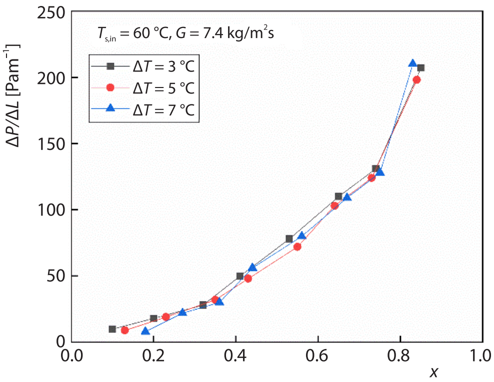

Figure 5. Effect of the total heat transfer temperature differences on the total pressure gradient

Figure 5 shows the variation rule between total pressure gradient and dryness under different total heat transfer temperature differences. At experiment condition of mass flux $G = 7.4$ kg/m^{2}s and the saturated temperature is 60 °C, the total temperature differences between the inlet saturated steam and cooling water range from 3-7 °C, the result of data almost overlap in these curves, so the total temperature difference has little influence on pressure gradient.

New correlation for a total pressure drop of steam inside the tube

In current studies, many scholars have studied refrigerant and steam at a high mass flux, but few have studied the formula of total pressure gradient at a low mass flux. In this study, the mass flux of the two-phase flow is low. The flow is considered homogeneous. Based on 154 data points obtained in this experiment, a new correlation formula of total pressure drop was established. The suitable condition is not more than 8 kg/m^{2}s for mass flux. The total heat transfer temperature difference range from 3-7 °C. In the gas-liquid two-phase flow theory, the homogeneous flow theory regarded the complex two-phase flow as an ideal single-phase mixture flow. In homogeneous flow models, the total pressure gradient can be expressed as a function of the mixture characteristics. The main factors affecting the total pressure gradient are mass flux, fluid dynamic viscosity, fluid density, dryness and other properties, which can be utilized work of Incropera [19]:

$$\frac{\Delta P}{\Delta L} = A\frac{2G^2}{\rho_{\mathrm{tp}}D}x^B\left(\frac{\mu_\ell}{\mu_{\mathrm{g}}}\right)^C \tag{14}$$

where A, B, and C are undetermined coefficients, x – the average vapor quality in the experimental section, and ρ_{tp} – the density of two-phase flow in the tube, that can be expressed:

$$\frac{1}{\rho_{\mathrm{tp}}} = \frac{x}{\rho_{\mathrm{g}}} + \frac{1-x}{\rho_\ell} \tag{15}$$

According to the results of this experiment, Levenberg-Marquardt iterative algorithm is used to calculate the aforementioned undetermined coefficients based on the least square criterion:

$$\frac{\Delta P}{\Delta L} = 0.073\frac{2G^2}{\rho_{\mathrm{tp}}D}x^{1.04}\left(\frac{\mu_\ell}{\mu_{\mathrm{g}}}\right)^{0.022} \tag{16}$$

The 274 data of experimental results and predicted results were compared, as shown in fig. 6. The data of predicted value error ranging from –25% to +25% is accounted for 80.0%.

Experimental data of the frictional pressure drop

As previously described in eqs. (12) and (13), the frictional pressure drop could be obtained from this study. The influence factor of steam quality x and mass flux G on the frictional pressure drop can be seen from fig. 7 at $T_{\mathrm{s,in}} = 60$ °C and $\Delta T_{\mathrm{s,c}} = 5$ °C. The friction pressure drop decreases with the decrease of mass flux. Because at the same inlet saturation temperature, the steam velocity is higher with a higher mass flux. Higher steam velocity means losing more energy because of more friction on the wall surface and leading to more frictional pressure drop. At the same inlet vapor mass flux G, the frictional pressure drop decreased with the decrease of steam quality. It is because, at the same inlet steam mass flux and same inlet saturation temperature, low vapor quality means more steam condenses in the tube and the decreased vapor velocity.

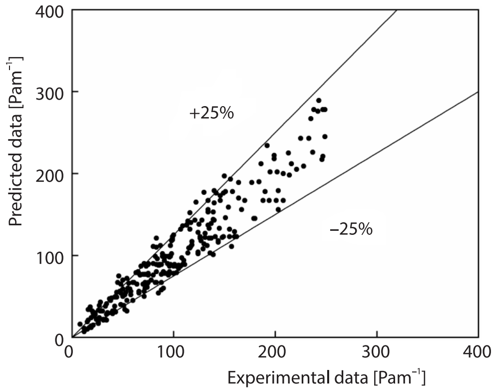

Figure 6. Comparison of the pressure drop of experimental data and predicted data

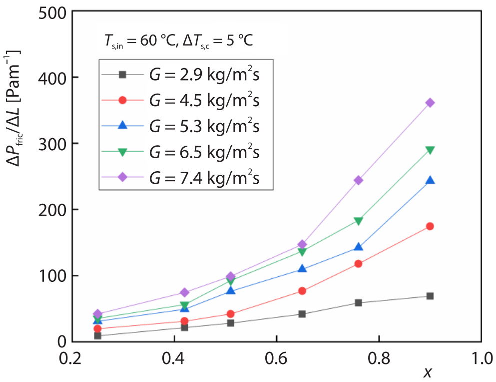

Figure 7. Effects of mass flux *G* on the frictional pressure drop

Comparison with existing models

For evaluating the forecast accuracy of numerous current frictional pressure drops, statistical functions are used commonly. Mean relative error (MRE) and mean absolute error (MAE) can be expressed:

$$\mathrm{MRE} = \frac{1}{N}\sum_{k=1}^{N}\left[\frac{\left(\dfrac{\Delta P}{\Delta L}\right)_{\mathrm{fric,pred,k}} - \left(\dfrac{\Delta P}{\Delta L}\right)_{\mathrm{fric,exp,k}}}{\left(\dfrac{\Delta P}{\Delta L}\right)_{\mathrm{fric,exp,k}}}\times 100\%\right] \tag{17}$$

$$\mathrm{MAE} = \frac{1}{N}\sum_{\mathrm{k}=1}^{N}\left[\frac{\left|\left(\dfrac{\Delta P}{\Delta L}\right)_{\mathrm{fric,pred,k}} - \left(\dfrac{\Delta P}{\Delta L}\right)_{\mathrm{fric,exp,k}}\right|}{\left(\dfrac{\Delta P}{\Delta L}\right)_{\mathrm{fric,exp,k}}}\times 100\%\right] \tag{18}$$

where the subscript character fric represents the frictional pressure drop, pred and exp are the prediction data and experiment data, k – the different data, and N – the number of experiments. The 58 sets of data are compared in the present work.

Correlation comparison of the friction pressure drop

In homogeneous flow model theory, the slip rate of two-phase flow is assumed to be 1. It represented the gas-liquid two-phase flowed at the same speed, therefore, the complex two-phase flow could be regarded as a single phase. Similar to single-phase flow, in the homogeneous flow model, the predicted value can be described [19]:

$$\left(\frac{\Delta P}{\Delta L}\right)_{\mathrm{fric,tp}} = 2 f_{\mathrm{tp}} \frac{G^2}{\rho_{\mathrm{tp}} D} \tag{19}$$

where ρ_{tp} is the density of two-phase flow in the tube, that can be calculated by eq. (15). The f_{tp} is the friction coefficient which is given by follows, the subscript tp means the value of two-phase flow:

$$f_{\mathrm{tp}} = 16\mathrm{Re}_{\mathrm{tp}}^{-1} \quad \text{for } \mathrm{Re}_{\mathrm{tp}} < 2000 \tag{20}$$

$$f_{\mathrm{tp}} = 0.079\mathrm{Re}_{\mathrm{tp}}^{-0.25} \quad \text{for } 2000 < \mathrm{Re}_{\mathrm{tp}} < 20000 \tag{21}$$

$$f_{\mathrm{tp}} = 0.046\mathrm{Re}_{\mathrm{tp}}^{-0.2} \quad \text{for } \mathrm{Re}_{\mathrm{tp}} > 20000 \tag{22}$$

$$\mathrm{Re}_{\mathrm{tp}} = \frac{GD}{\mu_{\mathrm{tp}}} \tag{23}$$

where Re is Reynolds number for two-phase mixture flow and μ_{tp} – the dynamic viscosity of the two-phase mixture flow. Many scholars have proposed different ways to calculate it and tab. 2 provides six of the widely used formulas:

The current experimental values are compared with the aforementioned six groups of frictional pressure drop empirical correlations for the homogeneous flow model, as is demonstrated in fig. 8. The Cicchitti correlation and Owens correlation deviated greatly from the current experimental values and the experimental values are lower than the predicted values, with MRE of 70.3% and 107.6%, respectively and MAE of 72.4% and 108%. The data calculated by the other empirical correlations are accurate. The data calculated by the McAdam correlation and the Dukler correlation are higher in the high pressure drop condition, while it is more accurate in other experimental conditions. The MRE of the McAdam correlation and the Dukler correlation is –20.7% and 4.9% and the MAE is 21.4% and 22.5%. The MRE and MAE of Dukler correlation are 4.9% and 22.5%, respectively. Considering MRE and MAE, the Dukler correlation is the most accurate of the six correlations.

Table 2. Six homogeneous flow models of frictional pressure drop

References	Correlations
[20]	$\dfrac{1}{\mu_{\text{tp}}} = \dfrac{x}{\mu_{\text{g}}} + \dfrac{1-x}{\mu_\ell}$
[21]	$\mu_{\text{tp}} = x\mu_{\text{g}} + (1-x)\mu_\ell$
[22]	Modified McAdams model [18] $\mu_{\text{tp}} = \mu_\ell$
[23]	$\mu_{\text{tp}} = \mu_{\text{g}}\beta + \mu_\ell(1-\beta),\ \beta = \dfrac{\rho_\ell x}{\rho_\ell x + \rho_{\text{g}}(1-x)}$
[24]	$\mu_{\text{tp}} = \dfrac{\mu_1 \mu_{\text{g}}}{\mu_{\text{g}} + x^{1.4}(\mu_\ell - \mu_{\text{g}})}$
[25]	$\mu_{\text{tp}} = \mu_\ell(1-\beta) + \mu_{\text{g}}\beta + 2\left[\beta(1-\beta)\mu_\ell\mu_{\text{g}}\right]^{0.5}$

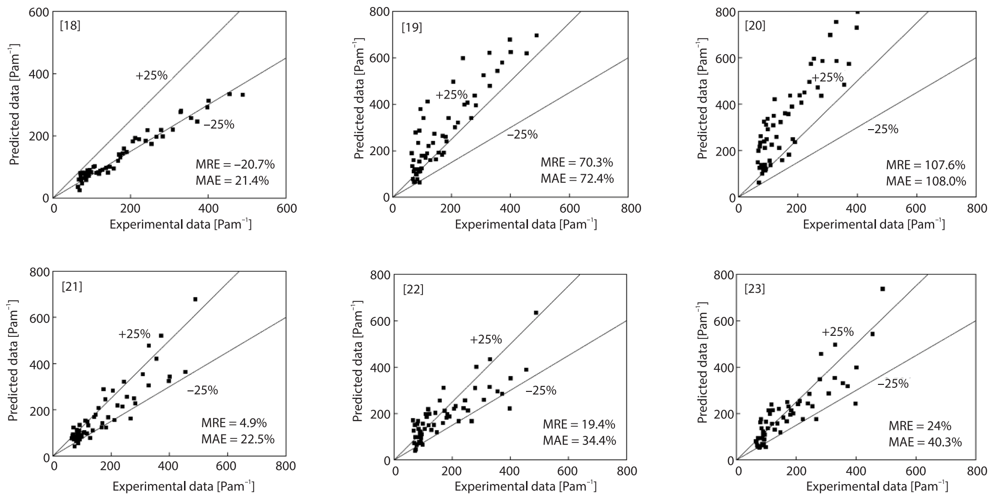

Figure 8. Comparisons between experimental data and predicted data based on homogeneous models of frictional pressure drop

Conclusions

The pressure drop of pure steam flow in the horizontal tube is investigated with the mass flux from 3-7.5 kg/m^{2}s and the inlet saturation temperatures 50 °C, 60 °C, and 70 °C. The following conclusions are obtained in the experiments, are as follows.

- The increase of the mass flux and inlet saturated temperature lead to an increase in the total pressure drop. The total pressure drop of steam flow decreases with the decrease of vapor quality. The total heat transfer temperature difference has little influence on the total pressure drop.

- A new correlation is proposed for calculating the pressure drop of steam condensation flow in a horizontal tube. The data with predicted value error ranging from –25% to +25% accounted for 80.0%.
- The Dukler correlations of the homogeneous frictional pressure drop model have good precision for calculating the frictional pressure drop.

Nomenclature

A – area, [m^2]
C – arc length, [m]
c_p – isobaric heat capacity, [Jkg^{-1}K^{-1}]
D – diameter, [m]
G – mass flux, [kgm^{-2}s^{-1}]
h – heigh, [m]
L – length, [m]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
P – pressure, [Pa]
Q – heat transfer, [J]
Re – Reynolds number, [–]
r – latent heat, [kJkg^{-1}]
T – temperature, [K]
V – volume, [m^3]
x – vapor quality, [–]

Greek symbols

α – void fraction, [–]
θ – angle, [°]
μ – dynamic viscosity, [Pa·s]
ρ – density, [kgm^{-3}]

Subscripts

c – cool water
exp – experiment value
g – gas
in – inlet of the tube
ℓ – liquid
out – outlet of the tube
pred – predicted value
s – saturation
t – tube
tp – two-phase
wet – wetted

Acknowledgment

The research is supported by the National Natural Science Foundation of China (No. 51936002) and Fundamental Research Funds for the Central Universities (DUT-21LAB128).

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Paper submitted: Decembar 12, 2021
Paper revised: April 4, 2022
Paper accepted: April 9, 2022