

Pressure drop characteristics of steam condensation flow in triangular channels for multi-channel cylinder dryers

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ABSTRACT: This study investigates the condensation pressure drop characteristics of steam in the channels of multi-channel drying cylinders, which remain insufficiently understood. An experimental system was established using a horizontal mini-channel with a triangular cross-section and a hydraulic diameter of 4.09 mm. Flow visualization combined with a high-precision data acquisition system was employed to systematically examine the effects of steam mass flux (10–60 kg/(m²·s)), inlet temperature (110°C–140°C), and cooling water flow rate (100–500 kg/h) on the frictional pressure drop during condensation. The results indicate that the two-phase frictional pressure drop increases with rising steam mass flux but decreases with increasing saturation temperature, while the effect of cooling water flow rate becomes more pronounced under lower temperature conditions.

In addition, several existing two-phase flow pressure drop correlations were evaluated against the experimental data. Based on a comparative assessment of eight existing correlations, a new predictive model was developed through regression analysis, achieving a mean absolute percentage error (MAPE) of 19.56%. These findings elucidate the pressure drop mechanism of steam condensation in triangular channels and establish a high-precision predictive model, providing theoretical guidance and a critical tool for the engineering design and energy consumption optimization of multi-channel drying cylinders.

Application: This study clarifies the pressure drop characteristics of steam condensation within triangular channels and develops a predictive correlation. This model offers direct theoretical guidance and serves as a critical computational tool for the engineering design and energy efficiency optimization of multi-channel drying cylinders.

Accurate prediction of two-phase flow pressure drop is essential for heat exchanger design, energy consumption assessment, and the operational stability of thermal systems. Over the past decades, extensive experimental and theoretical investigations have been conducted to elucidate the condensation pressure drop characteristics within channels, leading to the development of two principal categories of predictive models: the homogeneous flow model and the separated flow model.

The homogeneous flow model represents a simplified approach to multiphase flow analysis in which the two-phase working fluid is treated as a single, uniform mixed medium. This model assumes that both phases move with the same velocity, thereby neglecting interphase slip and phase coupling effects. Consequently, all thermophysical parameters are evaluated using the average values of the gas–liquid mixture. Several studies have contributed to development and application of homogeneous flow models, including those by McAdams et al. [1], Cicchitti et al. [2], Dukler et al. [3], Beattie and Whalley [4], Lin et al. [5], and Awad and Muzychka et al. [6]. Despite its simplicity and computational convenience, the homogeneous flow model exhibits significant limitations in representing realistic two-phase flow behavior. By assuming complete mixing and identical phase velocities, it fails to cap-

ture the complex interactions between the liquid and vapor phases, including the influence of distinct flow regimes and interfacial shear stresses on flow resistance. These inherent simplifications restrict the model's capability to accurately predict key parameters, such as pressure drop and heat transfer performance, under many practical and complex operating conditions.

The separated flow model is more widely applied in the prediction of two-phase flow pressure drop. In this framework, the gas and liquid phases are treated as two distinct fluids, each regarded as a continuous medium flowing within the channel. The model assumes that the two phases possess different velocities and physical properties and move through the channel at their respective velocities while interacting through interfacial forces. Consequently, at the same spatial location and time, the gas and liquid phases may exhibit different velocities, temperatures, and densities. This modeling framework was originally proposed by Lockhart and Martinelli [7] and has since been further developed and refined by numerous researchers, including Chisholm [8], Kim [9], Mishima and Hibiki [10], Tran et al. [11], H.J. Lee and S.Y. Lee [12], Hwang and Kim [13], Quan et al. [14], and Cavallini et al. [15]. Because the separated flow model more realistically captures key phys-

ical mechanisms — such as interfacial velocity slip, interfacial shear stress, and flow pattern evolution — it generally provides higher prediction accuracy than the homogeneous flow model, which treats the two phases as a single uniform mixture. Consequently, the separated flow model has gradually become the predominant framework for predicting two-phase flow pressure drop.

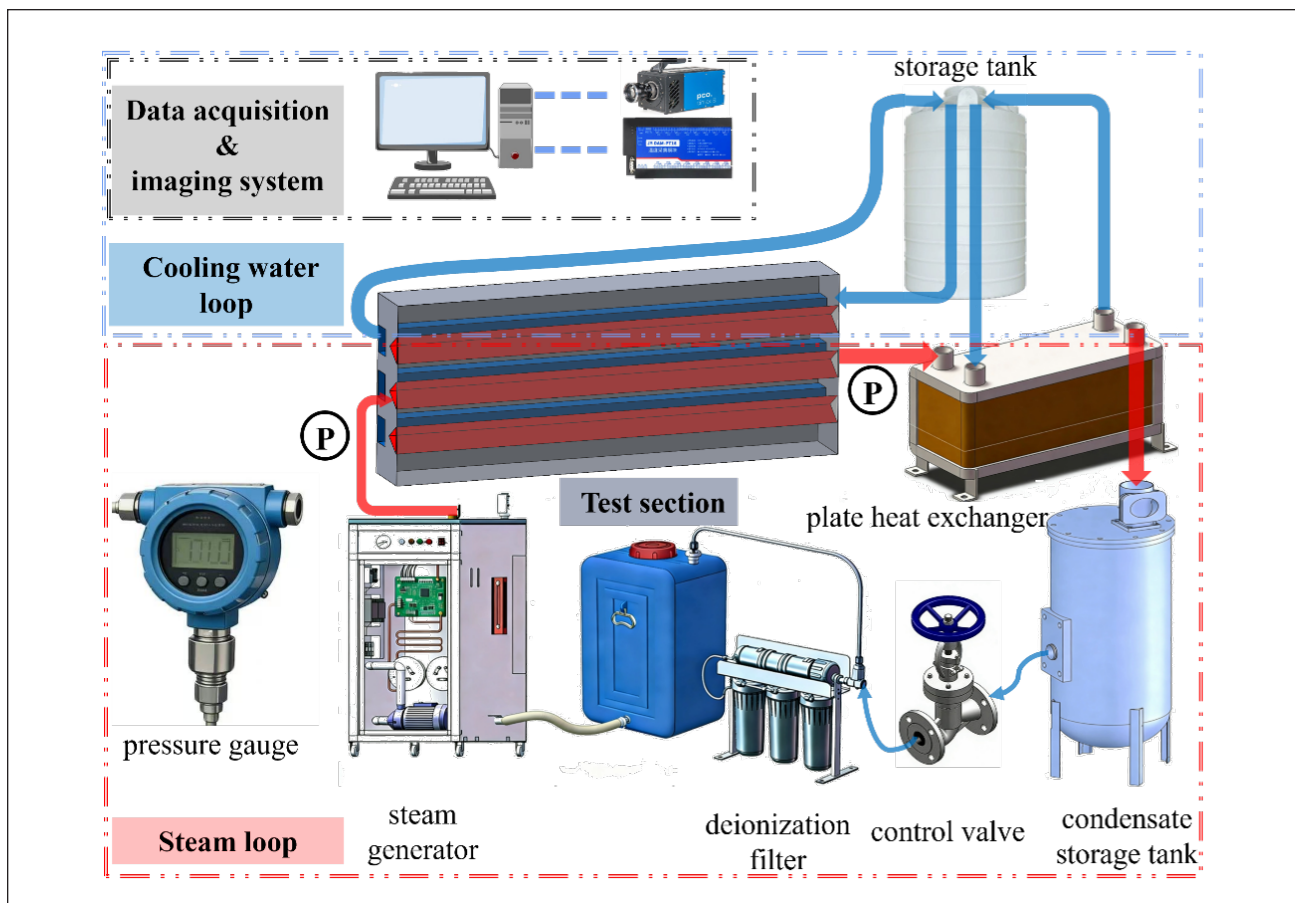
Currently, academic research on condensation two-phase flow pressure drop in channels primarily focuses on two major categories: conventional channels of traditional size and microchannels with characteristic dimensions ranging from micrometers to millimeters. These two research directions collectively constitute the main body of current knowledge on condensation pressure drop in channels. However, notable differences exist between them in terms of research methodologies, governing parameters, and the applicability of predictive models. The hydraulic diameter criterion of the present channel ($D_h=4.09$ mm) is clearly specified. Based on the channel classification criterion proposed by Kandlikar [16], the test channel adopted in this work is defined as a macrochannel. Further mechanistic analysis of Zhang's group [17] reveals that condensation heat transfer in macrochannels is dominated by forced convection and interfacial shear effects, while condensation occurring in mini/microchannels is governed by surface tension. Such essential differences in physical mechanisms well explain why the heat transfer and pressure drop characteristics obtained in this study are distinct from those reported in previous microchannel investigations.

In studies of two-phase flow and heat transfer in conventional horizontal tubes, the range of working fluids extends from water to various refrigerants. Early research demonstrated that total pressure drop decreases with a reduction in mass flux, whereas the influence of saturation temperature is comparatively weak [18]. For steam–water two-phase flow, investigations by Ferguson and Spedding indicated that the model proposed by Olujić [19] provides the highest prediction accuracy under stratified flow conditions [20]. For refrigerants, a similar trend — namely, a reduction in pressure drop with decreasing mass flow rate — is widely observed. Numerous empirical correlations, such as those proposed by Chen et al. [21], Tribbe and Müller-Steinhagen [22], and Müller-Steinhagen and Heck [23], exhibit satisfactory predictive performance under specific experimental conditions; however, their general applicability remains limited. Dalkilic et al. [24] experimentally investigated the condensation pressure drop characteristics of R600a and R134a and, based on annular flow conditions, identified the correlation that best agrees with their experimental data. For the specific case of pure steam condensation, Gu et al. [25] proposed a new correlation for calculating total pressure drop. Overall, most existing studies focus on high mass flux conditions, while pressure drop characteristics at low mass flux conditions — particularly for pure steam condensation — have received comparatively little

attention. With the rapid development of microscale technologies, condensation in microchannels has attracted considerable attention due to its superior heat transfer performance. It has been widely reported that surface tension effects become increasingly significant in microchannels, leading to pressure drop characteristics that differ markedly from those in conventional channels [26,27]. Existing studies generally highlight coupled effects of operating parameters and geometric factors: pressure drop typically increases with increasing mass flux and vapor quality, while it decreases with increasing saturation temperature and hydraulic diameter [28]. Specifically, the study by Goss et al. [29] on R134a in parallel microchannels found that heat flux has a negligible effect on pressure drop. Similar trends were confirmed by Liu et al. [30,31] for refrigerants such as R32, as well as by Agarwal and Garimella [32] for R134a in rectangular microchannels. Geometric effect is particularly pronounced. Garimella et al. [33], Agarwal and Garimella [34], and Coleman and Garimella [35] demonstrated a significant influence of channel aspect ratio, and Lee et al. [36] further reported that pressure drop increases with the aspect ratio of flat tubes. In addition, the effects of special structures (e.g., converging sections) [37,38], as well as vapor density and surface roughness [39–42], have also been confirmed as important influencing factors.

To predict pressure drop, extensive efforts have been devoted to the development and evaluation of frictional pressure drop correlations suitable for microchannels. Early improvements were largely based on classical models (e.g., the Lockhart-Martinelli model), in which the Chisholm parameter C was modified through the incorporation of dimensionless groups, such as the Bond number (Bo) and Reynolds number (Re), to extend model applicability across a wider range of conditions [43]. For example, Kim and Mudawar developed a generalized predictive model based on a large-scale database [44]. Fan et al. [45] proposed an enhanced Friedel-type model specifically tailored for steam condensation. More recently, data-driven approaches, including machine learning techniques, have been increasingly employed to establish more robust and potentially universal predictive models [46,47]. Despite these advancements, systematic evaluations of existing correlations frequently reveal inconsistent predictive performance and limited generalizability. Numerous assessment studies have shown that many correlations exhibit deviations when applied to new working fluids or operating conditions, with average prediction errors often exceeding 30%. For example, Pabon et al. [48] reported that only 65% of R1234yf data points could be predicted within a $\pm 30\%$ error band, while the comprehensive evaluation by Xu and Fang [49] of 29 correlations further highlighted their generally poor universality.

While preliminary studies, such as Qiao et al. [50] and Yan [51], have explored steam condensation in industrial channels (e.g., multi-port cylinders), several critical limitations remain. Existing empirical correlations generally ex-



1. Experimental system.

hibit poor universality, as they tend to perform well only under restricted operating conditions. More importantly, there is a notable lack of both high-quality experimental data and mechanistic understanding for pure steam condensation under low mass flux conditions.

To address these deficiencies and provide guidance for the optimal design of multi-port flat tubes, the present study conducts a systematic experimental investigation of steam condensation pressure drop under low mass flux conditions in specified channel geometries. The work aims to elucidate the dominant influencing factors and their underlying mechanisms, evaluate the performance of existing predictive correlations, and ultimately develop an improved, empirically grounded correlation with enhanced accuracy and reliability for engineering design purposes.

TEST FACILITY

As shown in **Fig. 1**, the experimental apparatus consists of four core parts: the steam loop, the cooling water loop, the test section, and the data acquisition and imaging system.

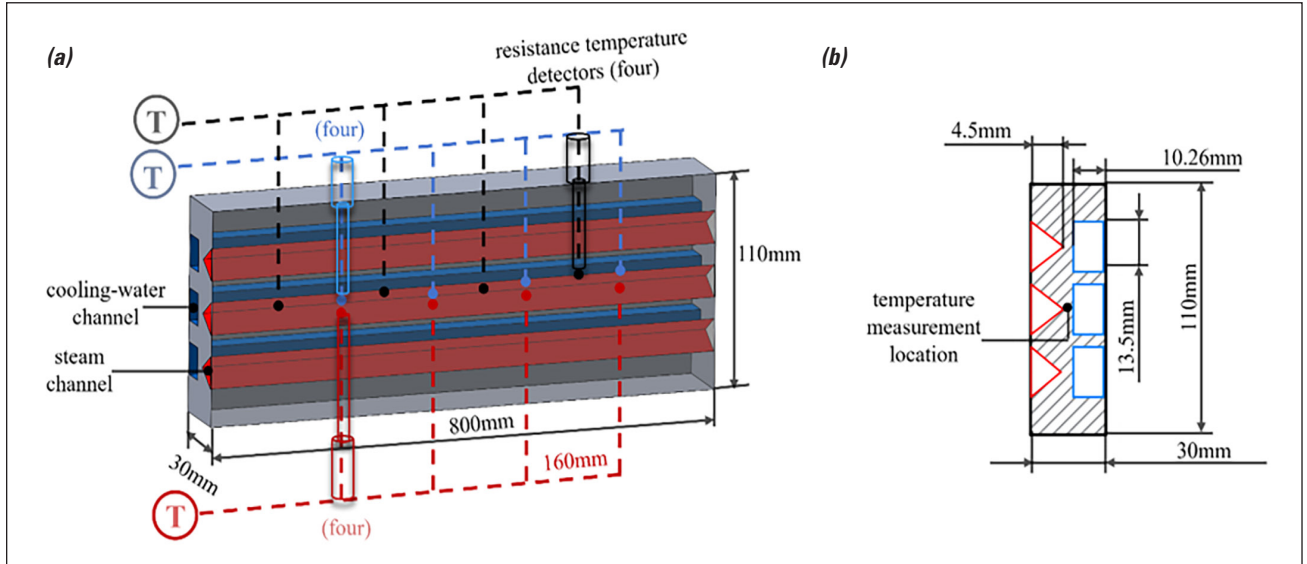
Working process

The experimental procedure commenced with the transport of deionized water from a storage tank to a steam generator, driven by a high-pressure vortex pump. Following

flow regulation by a needle valve, the generated saturated steam entered the test section to facilitate thermal exchange with a counter-flow cooling stream. Upon exiting the test section, any uncondensed steam and the resulting condensate were directed through a plate heat exchanger for cooling prior to being recirculated to the storage tank. To preserve stable experimental conditions, the cooling water inlet temperature was precisely maintained at 14°C. To assure the fidelity of the experimental results, the outlet steam quality was rigorously maintained above zero under all operating conditions, thereby ensuring the outlet subcooling approached zero. This rigorous control strategy confirmed that no identifiable subcooled region formed at the test section's outlet.

Test section: micro-structured plate heat exchanger

The test section consists of a dual-channel structure fabricated from rectangular aluminum plates, with overall dimensions of 800 mm × 110 mm × 30 mm, as shown in **Fig. 2a–2b**. During experiments, the test section was oriented horizontally. To facilitate visual observation, the steam-side channel was equipped with a high-temperature polycarbonate viewing window providing optical access, while the cooling water side was sealed with stainless steel



2. (a) Test section and (b) cross-sectional view of test section (red: steam data points; blue: condensate water data points; black: wall-surface data points).

plates. The entire assembly was wrapped with insulating cotton to restrict heat loss to below 0.5%.

Along the flow direction within both the steam and cooling water channels, PT100 resistance temperature detectors (RTDs) were installed at intervals of 160 mm. Additionally, four RTDs were uniformly embedded into the channel metal walls for accurate wall temperature measurement. It can be noted that wall temperature sensors were installed only in the middle channel of the three-channel structure, with data collected accordingly; no wall temperature sensors were arranged in the two side channels. System pressure was continuously monitored by pressure transducers, while the flow rates of steam and coolant were precisely measured by turbine flowmeters. Under steady-state conditions, the data acquisition system synchronously recorded temperature, pressure, and flow parameters at a sampling frequency of 10 ms. Flow pattern transition processes were captured by a pco.dimax S high-speed camera (Excelitas Technologies; Pittsburgh, PA, USA) with image data transmitted in real-time to a computer for subsequent analysis. Detailed specifications of the experimental apparatus are summarized in **Table I**.

DATA ANALYSIS

Data reduction

In the experiment, the thermodynamic and transport properties of fluids were evaluated using REFPROP software (National Institute of Standards and Technology; Gaithersburg, MD, USA). The steam quality was calculated using the following formula:

$$x_i = \frac{h_i - h'_i}{h''_i - h'_i} \quad (1)$$

where h_i represents the specific enthalpy of steam at the i -th segment (kJ/kg); h'_i denotes the saturated liquid specific enthalpy at the corresponding condition (kJ/kg); and h''_i is the saturated steam enthalpy at the corresponding condition (kJ/kg). The value of h_i is obtained from Eq. 2:

$$h_i = \frac{(h_{i,in} + h_{i,out})}{2} \quad (2)$$

where $h_{i,in}$ represents the inlet steam specific enthalpy at the i -th segment (kJ/kg), and $h_{i,out}$ denotes the outlet steam specific enthalpy at the same segment (kJ/kg). From the law of energy conservation:

$$h_{i,out} = h_{i,in} - Q_{c,i} \quad (3)$$

where $Q_{c,i}$ represents the heat absorption power by coolant in the i -th segment (W). Deionized water served as the coolant in this experiment. Resistance measurements provided spatially resolved temperature profiles along the entire cooling channel, enabling the calculation of segmental heat absorption according to Eq. 4:

$$Q_{c,i} = C_p m_c (T_{c,i+1} - T_{c,i}) \quad (4)$$

where m_c denotes the cooling water mass flow rate (kg/h); C_p is the isobaric specific heat capacity (kJ/(kg·K)); $T_{c,i+1}$ represents the coolant outlet temperature of the i -th segment (°C); and $T_{c,i}$ indicates the corresponding inlet temperature (°C).

Within the channel, the heat released by steam is partially absorbed by the coolant, with the remainder dissipating into the environment. Based on thermocouple arrangement, the entire channel is divided into five segments, each

Experimental Facility	Model	Range	Accuracy
Temperature transducers	WZP-02 PT100	-50°C–200°C	± 0.25%
Pressure sensors	PX409-1.5GV/2.5GT	0–150 psig	± 0.50%
Steam flowmeter	FTB-1411	50–500 L/min	± 1.00%
Cooling water flowmeter	FTB-1411	2.3–11.3 L/min	± 1.00%
Steam generator	NBS-AH9	9–36 kW	
Plate heat exchanger	MZ14-40D		
High-speed camera	Pco.dimax S		
Data acquisition module	JY-PT16, DAM-4AIAO		

1. Experimental facility details.

satisfying the law of energy conservation (Eq. 5):

$$Q_{s,i} = Q_{c,i} + Q_{e,i} \quad (5)$$

where $Q_{s,i}$ denotes the heat release power of steam in the i -th segment (W); $Q_{c,i}$ represents the heat absorption power by coolant in the same segment (W); and $Q_{e,i}$ signifies the thermal dissipation power from the test section to the environment in the i -th segment (W).

Given that the experimental apparatus is insulated with thermal insulation cotton, segmental heat losses remained below 0.5% of the total condensation heat transfer. Thus, the approximation $Q_{s,i} = Q_{c,i}$ is considered valid. The heat release rate in the steam channel within the i -th segment can be derived via Fourier's law:

$$Q_{s,i} = A_i h_i (T_{s,i} - T_{w,i}) \quad (6)$$

where h_i denotes the condensation heat transfer coefficient in the i -th segment of the steam channel (W/(m²·K)); A_i represents the corresponding heat transfer area (m²); $T_{w,i}$ is the measured wall temperature at the monitoring point (°C); and $T_{s,i}$ indicates the steam temperature within the i -th segment of the steam channel (°C).

The average condensation heat transfer coefficient of the whole steam channel (h_{avg}) can be determined from the local condensation coefficients of individual segments:

$$h_{avg} = \frac{\int h_i}{l} \approx \sum_{i=1}^n h_i \Delta l / l \quad (7)$$

The two-phase frictional pressure drop (ΔP) of the entire test section is calculated by subtracting the other pressure drop parts from the experimental overall pressure drop (ΔP_{tot}); i.e., acceleration pressure drop (ΔP_{acc}), single-phase frictional pressure drop (ΔP_{sp}), and the manifold

pressure drop at the inlet and outlet (ΔP_{man}) [52]:

$$\Delta P = \Delta P_{tot} - \Delta P_{acc} - \Delta P_{man} - \Delta P_{sp} \quad (8)$$

$$\Delta P_{acc} = -\frac{G^2(1-x_m)}{\rho_m} \quad (9)$$

$$\Delta P_{man} = 1.4 \frac{G^2}{2\rho} \quad (10)$$

The mean density of the working fluid (ρ_m) is calculated based on the homogeneous model (Eq. 11):

$$\rho_m = \left(\frac{x_m}{\rho_g} + \frac{1-x_m}{\rho_l} \right)^{-1} \quad (11)$$

The single-phase frictional pressure drop is the sum of single-phase frictional pressure drop in these regions. They are calculated by:

$$\Delta P_{sp} = \frac{f_{sp} G^2 L_{sp}}{2d_h \rho_{sp}} \quad (12)$$

where the single-phase friction factor (f_{sp}) is determined by the Martin correlation [53].

Uncertainty analysis

The analysis of experimental uncertainty is crucial during the experiment, and the calculation of uncertainty R is determined by Eq. 13 [54]. The main error parameters and uncertainty in the experiment are described below.

$$R = \left[\left(\frac{\partial R}{\partial x_1} \Delta x_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} \Delta x_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial x_n} \Delta x_n \right)^2 \right] \quad (13)$$

Derived Parameters	Uncertainty
Heat transfer rate (Q)	$\pm 6.58\%$
Steam quality (x)	$\pm 3.48\%$
Average heat transfer coefficient (h_{avg})	$\pm 7.90\%$
Two-phase frictional pressure drop (ΔP)	$\pm 5.87\%$

II. Uncertainty analysis results for derived parameters.

where x is the uncertainty of the variable, and different variables are represented by 1, 2, ..., n. The corresponding results are summarized in **Table II**.

EXPERIMENTAL RESULTS AND DISCUSSION

This section presents the experimental results obtained from steam condensation in a horizontal tube. The operating conditions for all experiments are summarized in **Table III**. For each test, every data point corresponds to the mean value measured over 10 min after steady state was achieved.

To assess the validity of the experimental data and the robustness of the evaluation procedure, the heat transfer coefficient of the steam condensation is compared with prediction values by established correlations. **Figure 3** presents a comparison between the experimental results obtained in this study and the corresponding values calculated using the correlations proposed by Marinheiro et al. [55] and Shah [56]. The discrepancy between the experimental and calculated results is quantified using the root mean square error (RMSE), as defined in Eq. 14.

Experimental Parameters	Values
Heat transfer fluid	Steam
Coolant	Deionized water
Steam inlet temperature (T_{in})	111.4°C–138.9°C
Cooling water inlet temperature (T_{ci})	14°C
Steam mass flux (G)	10–60 kg/(m ² ·s)
Cooling water mass flow rate (m_c)	100–500 kg/h

III. Experimental variables.

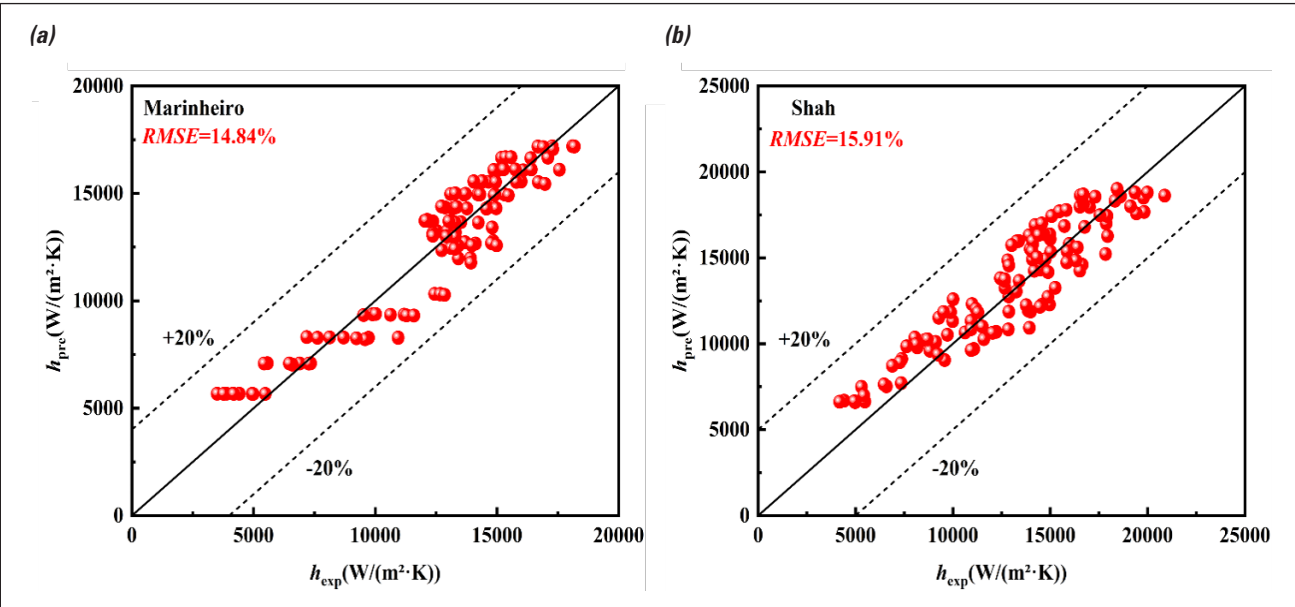
$$RMSE = \sqrt{\frac{\sum_N \left(\frac{h_{exp} - h_{pre}}{h_{exp}} \right)^2}{N}} \times 100\%$$

(14)

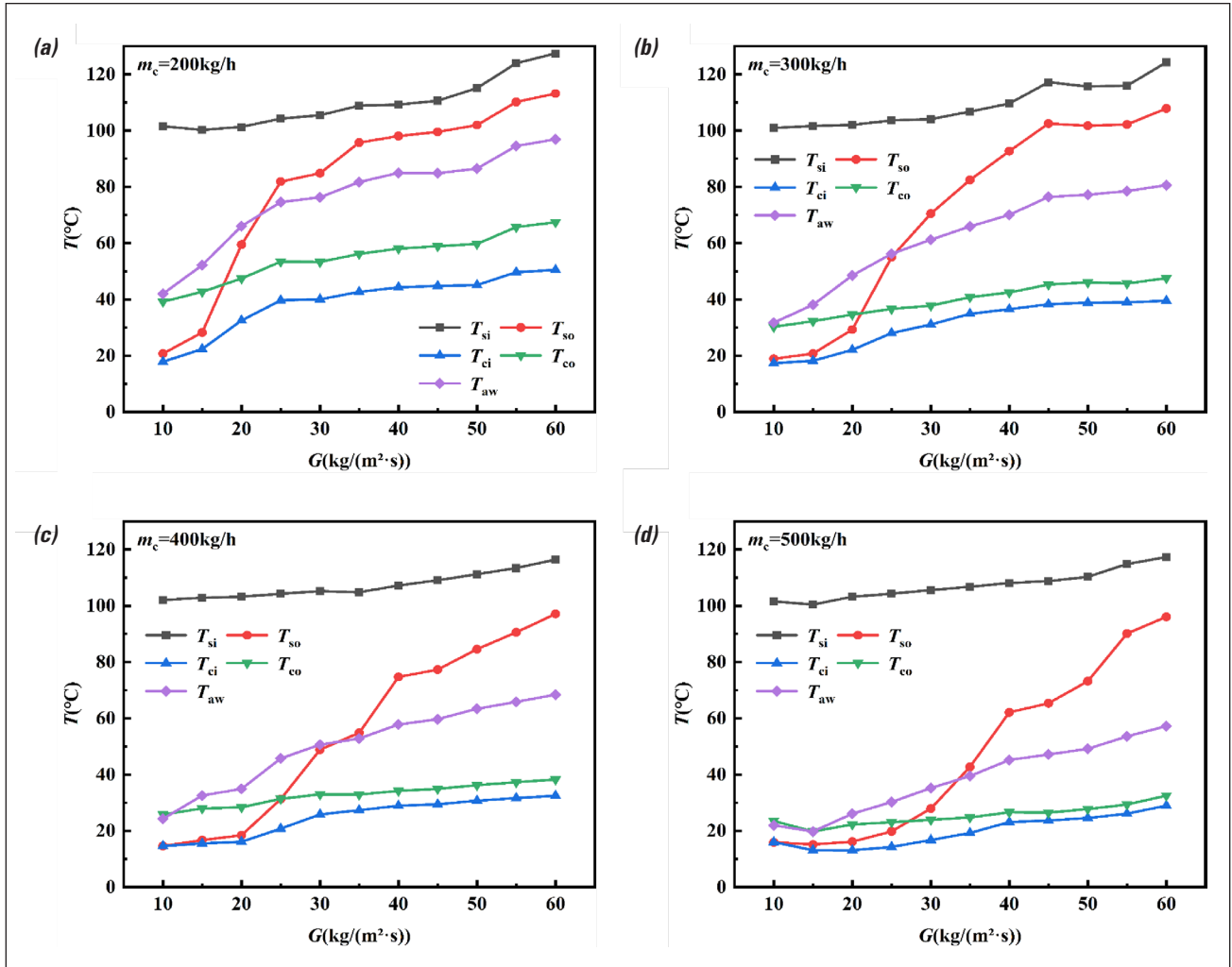
The RMSE values for the correlations of Marinheiro et al. and Shah are 14.84% and 15.91%, respectively, indicating good agreement between the present measurements and the correlation-based predictions. This agreement further confirms the validity of the experimental data and demonstrates the reliability of the evaluation procedure adopted in this work.

Heat transfer

Figure 4 illustrates the temperature distribution on both the steam side and the cooling water side under varying steam mass flux (G) and cooling water mass flow rate (m_c).



3. Comparison of heat transfer data from this work during steam condensation and predictions from the correlations of (a) Marinheiro et al. [55] and (b) Shah [56].



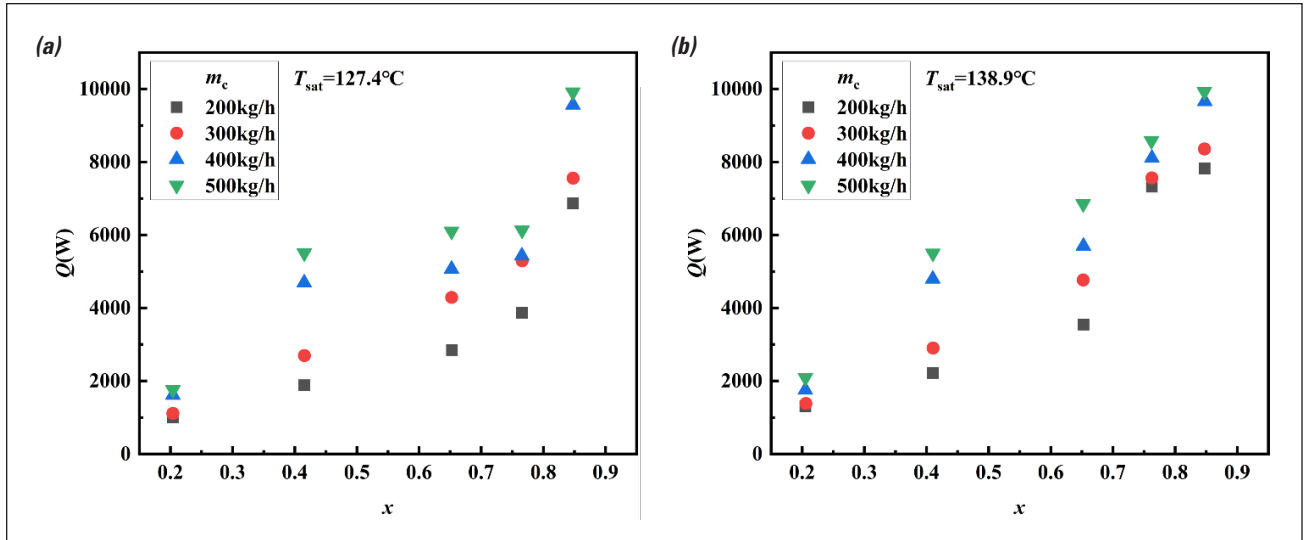
4. Temperature distributions with changes in steam mass flux, G , and cooling water mass flow rate, m_c : (a) m_c of 200 kg/h; (b) m_c of 300 kg/h; (c) m_c of 400 kg/h; and (d) m_c of 500 kg/h.

The figure presents, respectively, for cooling water mass flow rates of 200, 300, 400, and 500 kg/h, the variation curves of steam inlet temperature (T_{si}), steam outlet temperature (T_{so}), cooling water inlet temperature (T_{ci}), cooling water outlet temperature (T_{co}), and average wall temperature (T_{aw}) with steam mass flux. The results indicate that, under a constant cooling water mass flow rate, the temperatures at all measurement points on both the steam and cooling water sides show an increasing trend as the steam mass flux increases. This is primarily attributed to the increased heat carried by the steam.

Furthermore, a comparison of the subfigures corresponding to different cooling water flow rates reveals that as the cooling water flow rate increases, the rise in steam outlet temperature is significantly suppressed. In other words, a larger steam mass flux is required to achieve the same temperature rise level. Concurrently, the increasing trend of the cooling water temperature difference ($T_{co} - T_{ci}$) with steam mass flux becomes less pronounced, or its magnitude diminishes more notably. This indicates that, at the

same steam mass flux, the condensation heat transfer process is inhibited. The reason is that the increased cooling water flow rate reduces the effective temperature difference for heat transfer, leading to a decrease in the amount of heat effectively absorbed by the cooling water, thereby affecting the overall heat transfer performance.

Figure 5 systematically illustrates the variation of steam condensation heat transfer rate (Q) with steam quality (x), clearly revealing the significant influence of cooling water mass flow rate and steam saturation temperature (T_{sat}) on the heat transfer process. Based on a hydraulic diameter (d_h) of 4.09 mm, the channel in this study belongs to macrochannels, while for $d_h \leq 4$ mm, the channel belongs to the minichannel or the microchannel ($d_h \leq 1$ mm) category [57]. Unlike condensation in microchannels, which is predominantly governed by surface tension [58], forced convective condensation dominates in macrochannels, with its heat transfer mechanism primarily controlled by shear stress [59]. The figure clearly demonstrates the pronounced effects of cooling water mass flow rate and saturation tem-



5. Heat transfer rate (Q) vs. steam quality (x) at saturation temperatures (T_{sat}) of (a) 127.4°C and (b) 138.9°C (m_c : cooling water mass flow rate).

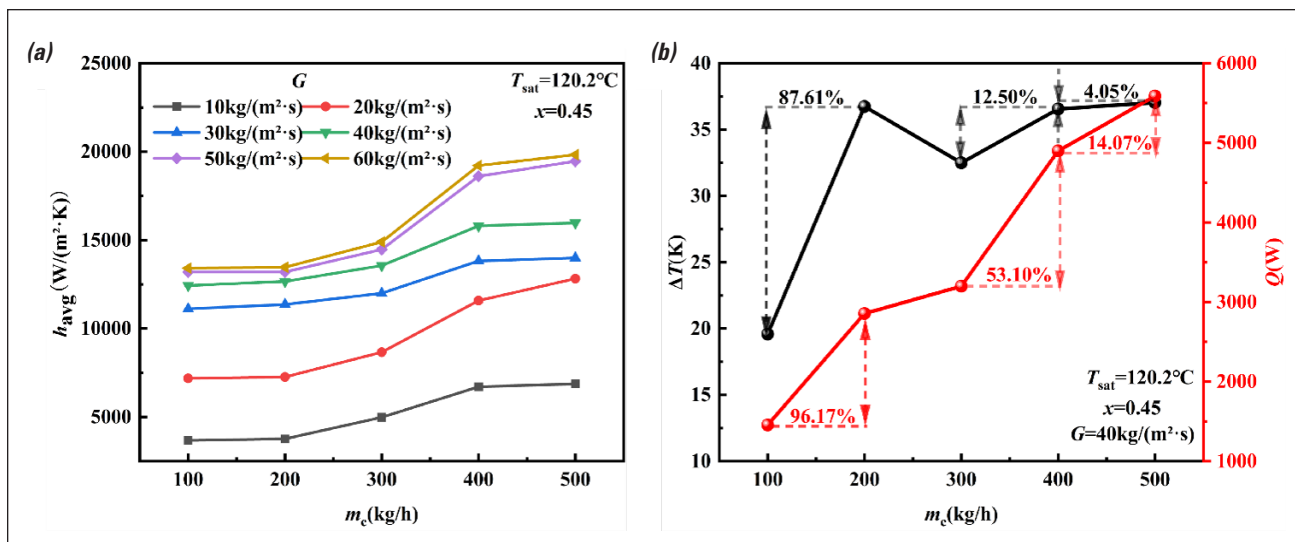
perature on the heat transfer process. Under the same saturation temperature (Fig. 5a: $T_{\text{sat}} = 127.4^\circ\text{C}$; Fig. 5b: $T_{\text{sat}} = 138.9^\circ\text{C}$), as the cooling water mass flow rate increases progressively from 200 kg/h to 500 kg/h, the heat transfer rate curves shift systematically upward, indicating a significant enhancement in heat transfer capacity with increasing cooling water flow. This is mainly attributed to the increase in heat transfer coefficient due to higher flow velocity and the increase in logarithmic mean temperature difference resulting from a lower cooling water outlet temperature. Notably, the degree of influence of the cooling water mass flow rate varies non-uniformly with steam quality. The influence is most pronounced within the steam quality range of 0.4–0.7. For instance, at $T_{\text{sat}} = 127.4^\circ\text{C}$ and $x \approx 0.41$, increasing the cooling water flow rate from 200 kg/h to 500 kg/h can elevate the heat transfer rate by approximately 191.7%. In contrast, the effect is markedly weaker at lower ($x < 0.4$) or higher ($x > 0.7$) steam qualities. For example, at $x \approx 0.8$, the same change in flow rate yields only a 44.3% increase. This phenomenon can be explained by flow regime transitions: in the low-quality region, the steam phase is transported as discrete bubbles; in the intermediate quality region, the flow may transition to intermittent slug flow or the initial stage of annular flow [60], where flow instability increases, making heat transfer more sensitive to changes in cooling water conditions; in the high-quality region, a stable annular liquid film flow develops, the flow becomes more stable, and the influence of cooling water flow rate variations is relatively diminished.

On the other hand, under fixed cooling water flow rate and saturation temperature conditions, the condensation heat transfer rate shows a monotonically increasing trend with increasing steam quality. Taking $m_c = 300$ kg/h and $T_{\text{sat}} = 127.4^\circ\text{C}$ as an example, when x increases from 0.2 to 0.8, Q rises from approximately 1100 W to nearly 7500 W.

This indicates that higher steam quality enhances the gas–liquid interface and heat transfer area on the condensation side, thereby intensifying the phase-change heat transfer. Furthermore, comparing cases with the same cooling water flow rate but different saturation temperatures reveals that increasing the saturation temperature can further improve the overall heat transfer performance. For example, at $m_c = 400$ kg/h, the heat transfer rates corresponding to various qualities under $T_{\text{sat}} = 138.9^\circ\text{C}$ are significantly higher than those under $T_{\text{sat}} = 127.4^\circ\text{C}$, especially in the high-quality region around $x \approx 0.8$, where the increase in heat transfer rate can exceed 50%. This is primarily because a higher saturation temperature increases the temperature difference between the steam and the cooling water, thereby strengthening the driving force for condensation heat transfer.

In summary, the heat transfer performance of the steam condensation process is synergistically influenced by three key parameters: cooling water mass flow rate, steam saturation temperature, and steam quality. Increasing the cooling water flow rate enhances convective heat transfer; raising the saturation temperature expands the heat transfer temperature difference; and increasing the steam quality strengthens the phase-change heat transfer intensity. By systematically regulating these parameters, a clear technical pathway can be provided for optimizing the condensation heat transfer process.

Figure 6a and Fig. 6b systematically illustrate the variations of the average condensation heat transfer coefficient (h_{avg}), heat transfer temperature difference (ΔT), and heat transfer rate with cooling water mass flow rate and steam mass flux under fixed saturation temperature ($T_{\text{sat}} = 120.2^\circ\text{C}$) and steam quality ($x = 0.45$). As shown in Fig. 6a, at a given steam mass flux, the average condensation heat transfer coefficient increases monotonically with rising cooling water



6. (a) Variation of condensation heat transfer coefficient (h_{avg}) and heat transfer temperature difference (ΔT) with cooling water mass flow rate (m_c), and (b) variation of heat transfer rate (Q) with m_c , under the conditions of saturation temperature (T_{sat}) of $120.2^\circ C$ and steam quality (x) of 0.45 (G : steam mass flux).

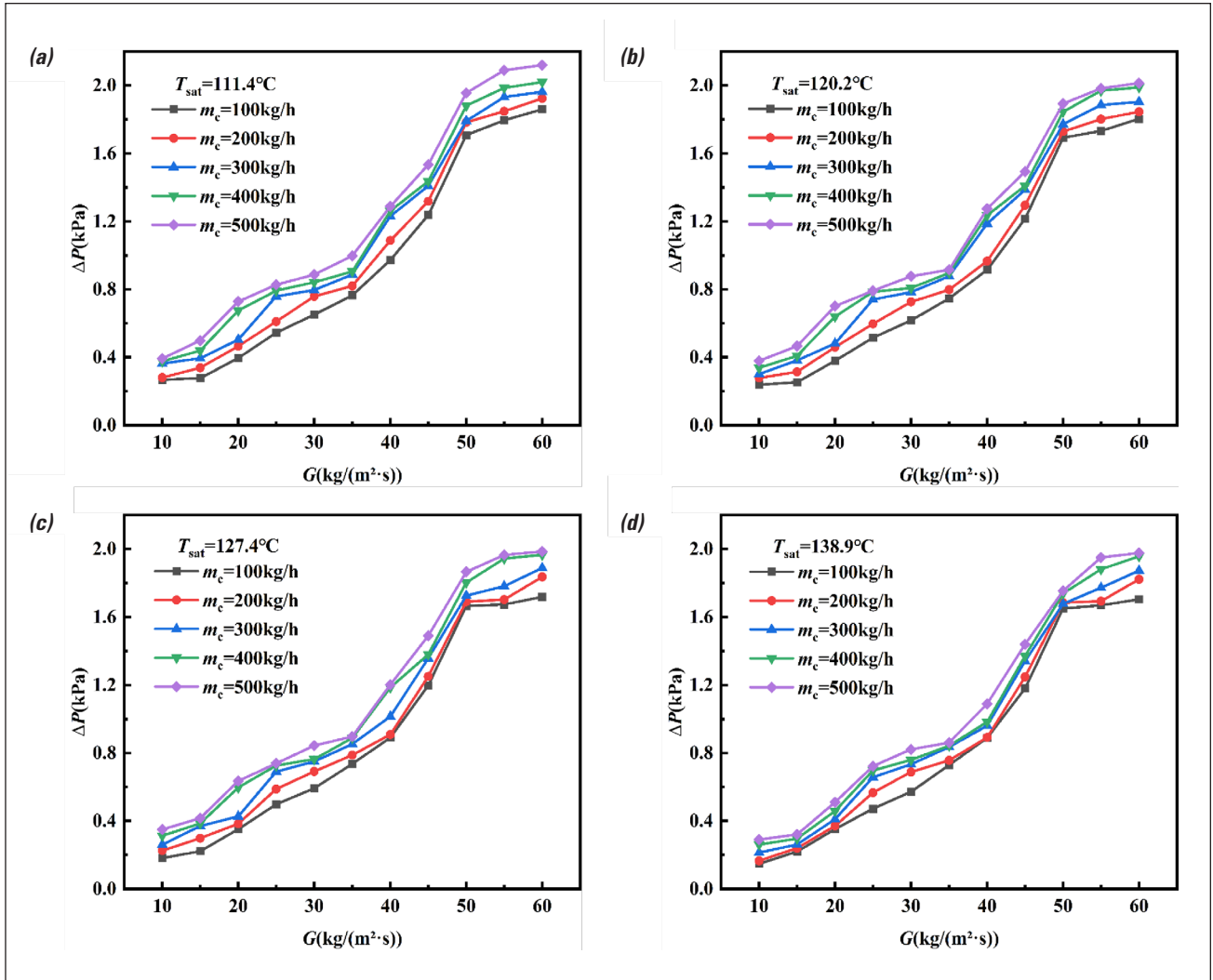
mass flow rate. For instance, at $G = 40 kg/(m^2 \cdot s)$, h_{avg} increases by approximately 28.46% as m_c rises from 100 kg/h to 500 kg/h . This indicates that increasing the cooling water flow rate significantly enhances condensation heat transfer. The underlying mechanism lies in the improved convective heat transfer on the cooling water side and the resulting increase in the temperature difference between steam and cooling water, thereby boosting overall heat transfer performance. Moreover, at a fixed m_c , h_{avg} also rises notably with increasing steam mass flux, particularly within the range of $G = 10$ to $40 kg/(m^2 \cdot s)$. This is attributed to the transition to annular flow induced by higher steam flux, which reduces liquid film thickness and thermal resistance, thereby strengthening condensation heat transfer. When G exceeds $40 kg/(m^2 \cdot s)$, the growth of h_{avg} gradually levels off, indicating that the liquid film stabilizes and heat transfer enhancement approaches saturation under high-flux conditions. Figure 6b further presents the variations of ΔT and Q with m_c at $G = 40 kg/(m^2 \cdot s)$. Notably, the growth rate of Q consistently exceeds the rate of increase in ΔT across the entire range of m_c variation. According to the defining equation $h_{avg} = Q/(\Delta T \cdot A)$, this directly leads to a continuous increase in the condensation heat transfer coefficient, corroborating the trend observed in Fig. 6a.

Combining Fig. 6a and Fig. 6b, it is evident that under the operating conditions of $T_{sat} = 120.2^\circ C$ and $x = 0.45$, simultaneously increasing the cooling water mass flow rate and steam mass flux synergistically enhances the condensation heat transfer coefficient. The underlying mechanisms include enhanced cooling-side convection and modified temperature difference driven by higher cooling water flow, combined with reduced thermal resistance through flow regime optimization and liquid film thinning induced by elevated steam flux. These findings provide important in-

sights for optimizing condenser performance and regulating operating parameters under variable conditions.

Two-phase frictional pressure drop

Figure 7 presents the relationship between the experimental two-phase frictional pressure drop (ΔP) and steam mass flux at different saturation temperatures and cooling water mass flow rates. The figure consists of four subplots corresponding to the following saturation temperatures: (a) $111.4^\circ C$, (b) $120.2^\circ C$, (c) $127.4^\circ C$, and (d) $138.9^\circ C$. Each subplot contains five curves representing the cooling water mass flow rates of 100, 200, 300, 400, and 500 kg/h , respectively. The two-phase frictional pressure drop was extracted by subtracting the acceleration pressure drop, single-phase frictional pressure drop, and manifold pressure drop from the total measured pressure drop, as given by Eq. 8. The data reveal two predominant trends. First, at a given saturation temperature and cooling water flow rate, ΔP increases consistently with rising steam mass flux. This trend is primarily due to enhanced momentum exchange and intensified interfacial friction at higher steam velocities, which is consistent with the findings reported by Yan et al. [61]. Second, at a constant mass flux, the pressure drop decreases as the saturation temperature increases. This behavior is closely linked to changes in steam thermophysical properties, such as reduced steam density and surface tension at elevated temperatures, which collectively lower flow resistance. Furthermore, the effect of cooling water flow rate is more pronounced at lower saturation temperatures, as clearly observed in subplot (a) for $T_{sat} = 111.4^\circ C$, where the curves corresponding to different cooling water flow rates are distinctly separated. This indicates that condensation intensity significantly influences steam quality and, consequently, the frictional pressure drop in the two-phase flow regime.



7. Variation of two-phase frictional pressure drop (ΔP) with steam mass flux (G) under different saturation temperatures (T_{sat}) of (a) 111.4; (b) 120.2°C; (c) 127.4°C, and (d) 138.9°C; and cooling water mass flow rate (m_c).

Models for frictional pressure drop

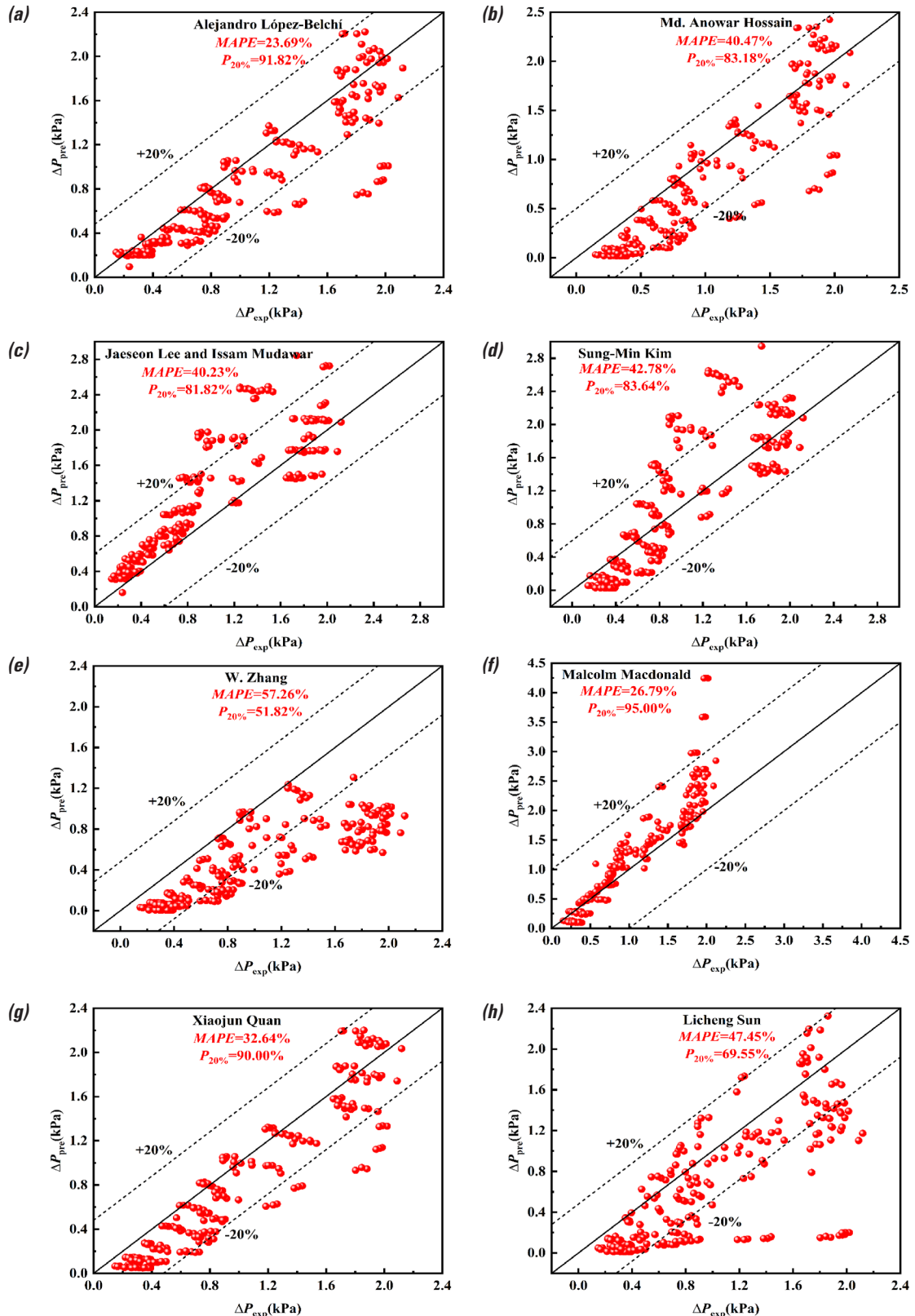
Regarding the two-phase frictional pressure drop, the experimental results were compared and analyzed with the estimated values from eight correlations derived based on classical correlations. **Figure 8** shows the comparison between the friction pressure drop calculated by the evaluated correlations and the experimental data, where the abscissa represents the measured pressure drop and the ordinate represents the predicted pressure drop. The error band of $\pm 20\%$ is also shown by the solid lines in these figures. The consistency between the correlations and the experimental data was quantified using the mean absolute percentage error (MAPE) and the fraction of data points falling within the relative error range of $\pm 20\%$, as detailed in **Table IV** [62–68]. The MAPE was calculated according

$$MAPE = \frac{1}{N} \sum_N \frac{|\Delta p_{exp} - \Delta p_{pre}|}{\Delta p_{exp}} \times 100\% \quad (15)$$

where the subscripts “exp” and “pre” denote the experimental value and predicted value, respectively.

According to the data in Table IV [62–68], the correlation proposed by Macdonald and Garimella [62] provided the best prediction, with a MAPE of 26.79% and as high as 95.00% of the data points lying within the $\pm 20\%$ error band. The correlations of López-Belchí et al. [63] and Quan et al. [14] also demonstrated good predictive capability, with MAPEs of 23.69% and 32.64%, respectively, and fractions of data points within the error band of 91.82% and 90.00%, respectively. In contrast, the correlations of Hossain et al. [64], Lee and Mudawar [65], and Kim and Mudawar [44] showed similar and moderate prediction accuracy, with MAPEs ranging from about 40.00% to 42.78% and fractions of data points within the error band between approximately 81.82% and 83.64%.

The correlations of Zhang et al. [66] and Sun and Mishima [67] exhibited relatively large prediction deviations



8. Comparison of two-phase frictional pressure drop (ΔP) with prediction results in Table IV for (a) López-Belchi et al. [63]; Hossain et al. [64]; (c) Lee and Mudawar [65]; (d) Kim and Mudawar [44]; (e) Zhang et al. [66]; (f) Macdonald and Garimella [62]; (g) Quan et al. [14]; and (h) Sun and Mishima [67].

Model	G, kg/m ² s	Working Fluid	Test Section	MAPE, %	Fraction of Data Points within the Relative Error of ±20%, %
López-Belchí et al. [63]	350–940	R1234yf, R134a, and R32	multiport minichannel tube $d_h = 1.16$ mm	23.69	91.82
Hossain et al. [64]	147–403	R1234ze(E), R32, R410A, dimethyl ether (DME), and R1234ze(E)/R32 mixtures (30/70 and 45/55 wt%)	horizontal smooth tube $d_h = 4.35$ mm	40.47	83.18
Lee and Mudawar [65]	127–654	R134a	231 μm * 713 μm	40.23	81.82
Kim and Mudawar [44]	4.0–8528	17 working fluids	$0.0695\text{ mm} \leq d_h$ $\leq 6.22\text{ mm}$	42.78	83.64
Zhang et al. [66]		21 working fluids	$0.007\text{ mm} \leq d_h$ $\leq 6.25\text{ mm}$	57.26	51.82
Macdonald and Garimella [62]	150–450	Propane, mixtures, R410A, R404A, pentane, and R245fa	$d_h = 7.75$ and 14.45 mm	26.79	95.00
Quan et al. [14]	187, 231 and 143	Steam	smooth trapezoidal silicon microchannels; $d_h = 109\text{ }\mu\text{m}$, $142\text{ }\mu\text{m}$, $151\text{ }\mu\text{m}$, and $259\text{ }\mu\text{m}$	32.64	90.00
Sun and Mishima [67]	50–1080	R123, R134a, R22, R236ea, R245fa, R404a, R407C, R410a, R507, CO ₂ , water and air	$0.506\text{ mm} \leq d_h$ $\leq 12\text{ mm}$	47.45	69.55

G: steam mass flux; MAPE: mean absolute percentage error; d_h : hydraulic diameter;

IV. Summary of selected condensation friction pressure drop correlations.

under the experimental conditions of this study. Specifically, the correlation of Zhang et al. showed a MAPE as high as 57.26%, with only 51.82% of the data points falling within the $\pm 20\%$ error range. The correlation of Sun and Mishima yielded a MAPE of 47.45%, with 69.55% of the data points within the error band. This is likely because these correlations were primarily developed based on experimental data for refrigerants; thus their applicability may be limited for other working fluids. In summary, under the working fluids and operating conditions considered in this study, the correlations of Macdonald and Garimella [62], López-Belchí et al. [63], and Quan et al. [14] offer better prediction accuracy and reliability.

New correlation for friction pressure drop

Within the investigated parameter range, existing correlations generally showed slight deviations (overestimation or underestimation) in predicting the two-phase frictional

pressure drop in conventional channels. To improve prediction accuracy, a new correlation is proposed herein.

This correlation, a modification of the López-Belchí et al. [63] model, introduces a newly developed expression for the parameter, C . This expression was obtained through nonlinear regression analysis of 220 experimental data points and is a function of several dimensionless parameters characterizing fluid properties and vapor-liquid differences, including the liquid-phase inertia-to-viscosity ratio and the Martinelli parameter.

The proposed correlation for the two-phase frictional pressure drop during in-tube condensation is provided in Eq. 16.

$$\Delta P = \phi_{h,q}^2 \left(\frac{dp}{dz} \right)_{liq}$$

(16)

with liquid flow multiplier defined by:

$$\phi_{liq}^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \quad (17)$$

where:

$$\left(\frac{dp}{dz}\right)_{liq} = \frac{G^2(1-x^2)}{2D\rho_{liq}} f_{liq} \quad (18)$$

$$\left(\frac{dp}{dz}\right)_{gas} = \frac{G^2 x^2}{2D\rho_{gas}} f_{gas} \quad (19)$$

and f_{liq} and f_{gas} are the single-phase frictional factors calculated by applying the liquid or gas phase properties and liquid or gas phase mass fluxes, respectively, to the following equations:

$$f = 64/Re (Re \leq 2000) \quad (20)$$

$$f = 0.25 \left[\log \left(\frac{150.39}{Re^{0.98865}} - \frac{152.66}{Re} \right) \right]^{-2} (Re \geq 3000) \quad (21)$$

$$f = (1.525 Re + 895) \cdot 10^{-5} (2000 < Re < 3000) \quad (22)$$

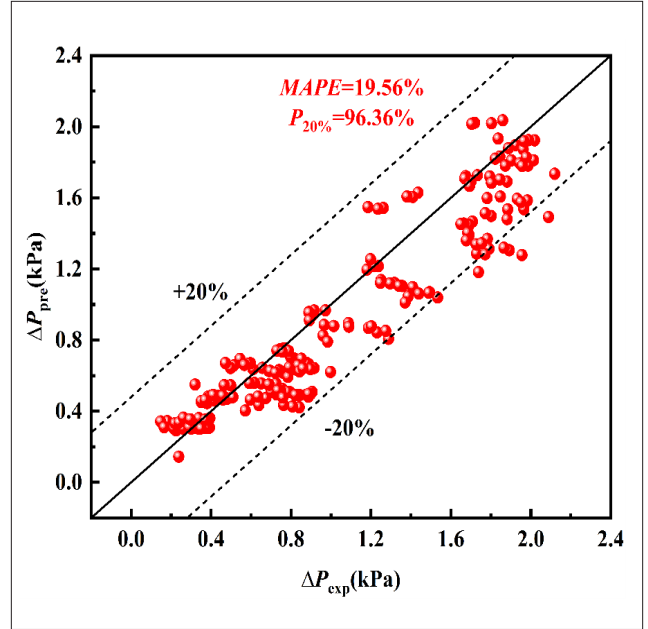
The Martinelli parameter, X , can be obtained as:

$$X = \sqrt{\left(\frac{dp}{dz}\right)_{liq} / \left(\frac{dp}{dz}\right)_{gas}} \quad (23)$$

Equations 16–23 were used for vapor and liquid phase to calculate the vapor phase pressure gradient and afterwards the Martinelli parameter. In addition, C was adjusted for best fitting experimental data.

$$C = 6.9521 \times 10^{-7} \left(\frac{P}{P_{crit}}\right)^{5.7124} Re_l^{0.4287} \left(\frac{\rho_l}{\rho_g}\right)^{5.3146} X^{-0.3853} \quad (24)$$

The proposed correlation was developed to minimize the deviation between experimental measurements and model predictions. The following dimensionless parameters are incorporated to capture the dominant physical mechanisms governing the two-phase frictional pressure drop during steam condensation in triangular channels: the reduced pressure, which accounts for the variation of fluid thermophysical properties with saturation conditions and its influence on condensation characteristics; the density ratio, which represents the significant disparity between the vapor and liquid phases and its



9. Comparison of experimental results with proposed new correlation (MAPE: mean absolute percentage error).

effect on interfacial momentum exchange; the liquid Reynolds number, which characterizes the relative importance of inertial and viscous forces in the liquid film flow and reflects the hydrodynamic contribution to pressure loss; and the Martinelli parameter, which describes the relative contributions of liquid- and vapor-phase frictional effects and reflects the phase distribution characteristics within the two-phase mixture. Therefore, these parameters collectively account for the coupled effects of fluid property variation, inertial forces, viscous resistance, interfacial interactions, and phase density differences, which are the primary mechanisms controlling the frictional pressure drop during condensation flow.

Figure 9 illustrates the comparison between the fitted results of the newly developed correlation and the 220 experimental data points. The two-phase frictional pressure drop values predicted by the new correlation show good consistency with the experimental measurements. The correlation exhibits a mean relative deviation of 19.56%, and 96.36% of the experimental data are predicted within a $\pm 20\%$ error band. Consequently, the proposed correlation demonstrates satisfactory predictive capability for the two-phase frictional pressure drop during steam condensation in triangular channels.

CONCLUSION

In this study, a comprehensive experimental and theoretical investigation is conducted to elucidate the heat transfer and flow resistance characteristics of steam condensation in triangular multi-port channels. The principal findings are summarized as follows:

(1) Effects of operating parameters and experimental validation. The influence of key parameters on condensation heat transfer and pressure drop is systematically analyzed. The reliability of the experimental data is validated through comparison with classical heat transfer correlations, showing prediction errors within approximately 15%. The results indicate that increasing the cooling water mass flux, steam saturation temperature, and vapor quality enhances the condensation heat transfer, although the magnitude of enhancement varies across different quality ranges. The two-phase frictional pressure drop increases with steam mass flux but decreases with rising saturation temperature. In addition, the effect of cooling water flow rate becomes more pronounced at lower temperature conditions.

(2) Evaluation of existing frictional pressure drop correlations. The applicability of eight classical predictive models is assessed against experimental data. The correlations proposed by Macdonald and Garimella. [62], López-Belchí et al. [63], and Quan et al. [14] demonstrate relatively good predictive performance, with mean absolute percentage errors ranging from 23.69% to 32.64%, and most data points falling within the $\pm 20\%$ error band. In contrast, correlations developed by Zhang et al. [66] and Sun and Mishima [67] — primarily based on refrigerant data — exhibit significant deviations when applied to steam condensation, underscoring their limited applicability across different working fluids.

(3) Development of a modified predictive correlation. To overcome the limitations of existing models, a modified correlation is proposed based on the framework of López-Belchí et al [63]. Using 220 experimental data points for regression, the proposed correlation shows strong agreement with experimental results, yielding a mean deviation of 19.56% with 96.36% of the data within the $\pm 20\%$ error band. This represents a notable improvement over most existing models and provides a more accurate and reliable method for predicting two-phase frictional pressure drop during steam condensation in triangular channels.

STATEMENTS AND DECLARATIONS

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.

DATA AVAILABILITY

Data will be made available on request.

AUTHOR CONTRIBUTIONS

All authors contributed to the study conception and design. Material preparation, data collection, and analysis were performed by Lijie Qiao, Haozeng Guo, and Kewei Guo. The first draft of the manuscript was written by Xingchen Liu and all authors commented on previous versions of the manuscript. All authors read and approved the final manuscript.

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ABOUT THE AUTHORS

Multi-channel cylinder dryers are increasingly used to raise drying efficiency and sheet quality in paper mills, but the pressure drop behavior of steam condensing inside their small triangular channels is still poorly understood, and this lack of both experimental data and reliable prediction tools directly limits the optimal design and energy-saving operation of such dryers.

This study extends our earlier work (Qiao et al.) and related studies by Yan on steam condensation in industrial multi-port channels, and it differs from most published studies — which focus on refrigerants, circular or rectangular microchannels, and relatively high mass fluxes — by supplying new experimental data for pure steam condensing in a horizontal triangular channel of 4.09 mm hydraulic diameter at very low mass fluxes of 10–60 kg/(m²·s), a region that has received little attention.

The most difficult aspect of this study was obtaining reliable data at very low steam mass fluxes and isolating the true two-phase frictional pressure drop. We solved this by subtracting the acceleration, single-phase frictional, and inlet/outlet manifold pressure drops from the measured total, keeping the cooling water inlet at 14°C and the outlet vapor quality above zero so no subcooled region formed, insulating the test section to keep heat loss below 0.5%, and validating the measurements against established condensation heat transfer correlations, which agreed within about 15% root mean square error (RMSE).

We discovered that the two-phase frictional pressure drop in these triangular channels increases with steam mass flux, decreases as saturation temperature rises, and becomes more sensitive to cooling water flow rate at lower saturation temperatures. We also confirmed that at a hydraulic diameter of 4.09 mm, the process behaves as a macrochannel process governed by forced convection and interfacial shear rather than by surface tension. A surprising finding was how unevenly the cooling water flow rate acts on heat transfer: raising it from 200 to 500 kg/h increased the heat transfer rate by about 191.7% near a steam quality of 0.41 but only about 44.3% near a quality of 0.8, because the middle quality range is dominated by unstable intermittent and developing annular flow. Equally striking is how poorly

refrigerant-based correlations perform here, with Zhang et al. and Sun and Mishima deviating by 57.26% and 47.45%.

Mills can use these results and the new correlation to predict steam-side pressure drop and condensate behavior in multi-channel drying cylinders; to size and arrange channels to avoid condensate accumulation and uneven drying; and to choose steam pressure, saturation temperature, and cooling conditions that reduce pressure loss and energy consumption while maintaining or increasing drying capacity.

The next step for this research is to extend the experiments to wider mass fluxes and saturation pressures; to other channel sizes, aspect ratios, and cross-sectional shapes; and to parallel multi-channel configurations. Then, we will examine the effect of non-condensable gases and validate the new correlation (MAPE 19.56%, 96.36% of data within ±20%) against operating data from full-scale dryer cylinders.

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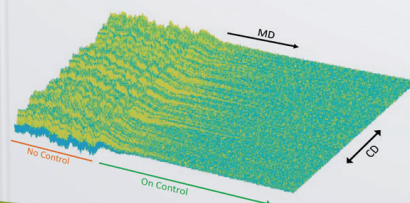
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