

Research Article

Heat Transfer Characteristics of Filmwise and Dropwise Condensation on Vertical Tubes in the Presence of Air: An Experimental Study

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Abstract:

Condensers are essential to many industrial processes, such as desalination, refrigeration, and power generation. The condenser performance largely depends on the mode of condensation (filmwise or dropwise) which significantly affects heat transfer rates. In this paper, An experimental investigation has been made on filmwise condensation (FWC) & dropwise condensation (DWC) to determine the heat flux and heat transfer coefficient (HTC) for three steam temperatures of 100°C, 80°C and 60°C. The results are compared with predicted values. It is observed that HTC of DWC is more than FWC. Experimental calculations of the influence of non-condensable gas (air) on heat flux and HTC have also been done and it was proved that the effect of air (1% and 2%) has a very detrimental effect on both FWC and DWC. The experimental values of HTC in presence of air are compared with predicted values obtained by two latest empirical correlations and deviation is identified in FWC. A satisfactory relation is obtained between actual and predicted values.

Keywords: Filmwise, dropwise, air, heat flux, heat transfer coefficient

1. Introduction

The phase change in condensation affects the performance of many applications such as steam power plant, refrigeration, water harvesting and chemical refining. Process efficiency is increased in these applications by increasing heat transfer rates during condensation. There are two ways that condensation can happen: dropwise condensation which forms droplets and filmwise condensation which forms a continuous liquid film [1,2].

The condensate produces a continuous coating over the cooling surface, preventing heat transfer in FWC. Heat transfer is improved by DWC droplets shedding and migrating, exposing the vapor to a greater cooling surface area. In dropwise condensation, the absence of a continuous liquid coating reduces thermal resistance, increasing heat transmission. The reduced thermal resistance and increased surface area of dropwise condensation result in heat transfer coefficients that are 10 or more times higher than FWC [3-5]. The representation of filmwise and dropwise condensation is shown in figure 1.

Non-condensable gas (NCG) presence reduces the condensation process performance due to several reasons such as thermal resistance, do not allow for changing phase, cause for impeding vapour transport. So, NCG presence is a key factor for better design of condenser used in power plants, refrigerators, air conditioners and other thermal applications [6-12].

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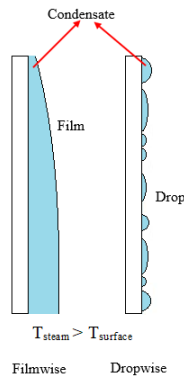


Fig. 1. Filmwise and dropwise condensation

Different authors studied about different NCGs such as air, CO₂ (carbon dioxide), H₂ (hydrogen), HC (hydrocarbon) and Ar (Argon), CH₄ (methane) and N₂ (nitrogen) on various factors such as NCG concentration, vapour velocity, condenser surface orientation & geometry and system pressure & temperature. NCG presence creates resistance layer, increases condenser pressure (initially partial pressure of system increases), helps to lead uneven heat transfer rates and temperature distribution which cause hot spots on surface and surface damage. From the observations in literature, Air, CO₂, Ar contributed for high thermal resistance, condensation interface and created high effect to reduce heat transfer rate. H₂ created lower and other gases created moderate effect on heat transfer rate. [13-18]

Initial empirical correlations by Uchida et al. [19], Tagami [20], Kataoka et al. [21], and Murase et al. [22] predominantly examined the influence of non-condensable gas (NCG) mass fraction, thereby simplifying the intricate condensation process by representing the heat transfer coefficient h exclusively as a function of air mass fraction W_a . Uchida et al. [19] formula became notable for its simplicity and ease of engineering application, making it especially valuable for preliminary containment analyses. Uchida et al. [19] study involved air mass fractions from 0.23 to 0.91, pressure between 0.1 and 1.8 MPa, subcooling from 10 to 140 K, and tube lengths from 0.3 to 0.9 m. With a constant length of 0.3 m, a wall temperature of 322 K, and an air mass fraction varying from 0.38 to 0.83, derived by Tagami [20]. Kataoka et al. [21] provided similar functional forms, differing slightly in exponents and constants. Murase et al. [22] extended the formula range by testing up to high W_a values up to 0.97; he also included only mass fraction in the correlation, though the experiments varied pressure, 0.1–0.35 MPa, temperature, 295–383 K, and tube length, 0.9–4.2 m.

Recognizing the limitations of mass-fraction models, researchers like Dehbi et al. [23], Lee et al. [24], and Liu et al. [25] developed multi-parameter correlations. Dehbi et al. [23] model includes air mass fraction, wall subcooling, total pressure, and condenser length, with ranges of 0.28–0.9 (mass fraction), 1.5–4.5 bar (pressure), and 10–50 K (subcooling), making it more physically representative of condensation processes under varying environmental conditions. Lee et al. [24] used dimensionless analysis involving Grashof and Jakob numbers, factoring in mass fraction (0–0.9), pressure (0.2–0.5 MPa), and subcooling (20–70 K). Liu et al. [25] work introduced steam quality as a key variable, with correlations for pressure from 253 to 456 kPa, subcooling between 4 and 25 K, and steam quality from 0.395 to 0.873. Kawakubo et al. [26] expression also involved pressure and subcooling, with air mole fractions from 0.5 to 1.0 and subcooling from 5 to 50 K. These correlations offer greater flexibility and realism, better capturing the interplay between multiple thermodynamic effects.

In more recent Su et al. [27, 28] models, Fan et al. [29] and Zhu et al. [30] refine predictive accuracy by tailoring the correlation structure to specific regimes. Su proposed separate correlations for high and low subcooling ranges: the model in 2013 for subcooling of 27–70 K and air mass fractions of 0.2–0.8, while the model in year 2014 covered 0.07–0.52 (mass fraction) and 13–25 K (subcooling). Both included pressure effects (0.2–0.6 MPa). Fan et al. [29] expression accounted for air mass fraction, pressure (0.2–0.5 MPa), and subcooling (10–70 K), with the novelty of making the subcooling exponent itself variable-dependent. Based on Rayleigh, Jakob, and Schmidt numbers, Zhu et al. [30] was applied a dimensionless correlation to helium-nitrogen mixtures with NCG mass percentage range between 0.001 and 0.05 and the range 0.1–0.9 used for pressure. These models are more accurate and robust for experimental interpretation and simulation but require comprehensive input data for proper application.

An experimental study of FWC and DWC has been conducted in this work to investigate heat transfer characteristics for different steam temperatures of 100°C, 80°C and 60°C under pure steam and air presence (1% and 2%). The experimental results of FWC are compared with predicted values. Rohsenow [31] Nusselt theory for pure steam and two latest Lee et al. [24] & Su et al. [27] correlations for air presence case are used for validation on HTC in FWC in this work.

2. Experimental Setup

The equipment shown in figure 2 is placed on a high quality glass reinforced plastic on which the components such as steam chamber, heating element, condensers, air extraction system, heater control, electronic temperature indicator, flow meters and pressure gauge are mounted. A vertical glass cylinder with thick walls and about 0.5 kg of water is contained in a brass lid that has been nickel-plated. The lower plate has an electric heating element that uses electrical power to convert water into steam at a specific pace. As the steam ascends to the dropwise and filmwise condensers, heat is transferred to the cooling water, causing the steam to condense. For further evaporation, the resultant condensate goes back to the vessel lower end.

To lessen the splashing caused by the vigorous boiling, three baffles are installed in the chamber. The element for heating the water has a helical structure and a power rating of three kilowatts. The element's upper coils come into thermal contact with a stainless steel pocket that contains a thermocouple. If overheating results from a water shortage, this thermocouple acts as a controller to isolate the heater. A combination of brass and copper makes up the condenser, which aids in filmwise condensation. To show the mean temperature of the condenser shell, three thermocouples that are evenly spaced and fastened to the shell are connected to the temperature indicator. To facilitate dropwise condensation, the exterior of the shell is gold-plated.

2.1 Air extraction

Condensers and other heat exchangers that operate at low pressure are equipped with an air extraction system to mitigate the adverse effects of a minute quantity of air on their performance. The device for film and dropwise has a water jet vacuum pump that can remove air from the steam room. An anticlockwise rotation of the valve hand wheel opens an air extraction valve between the chamber and the vacuum pump. Air exits the stem chamber and traverses an uninsulated copper conduit during the air extraction system's operation. Typically, the air extraction system is activated at the commencement of a test and then periodically for a few seconds at 5 to 15 minutes throughout the test duration.

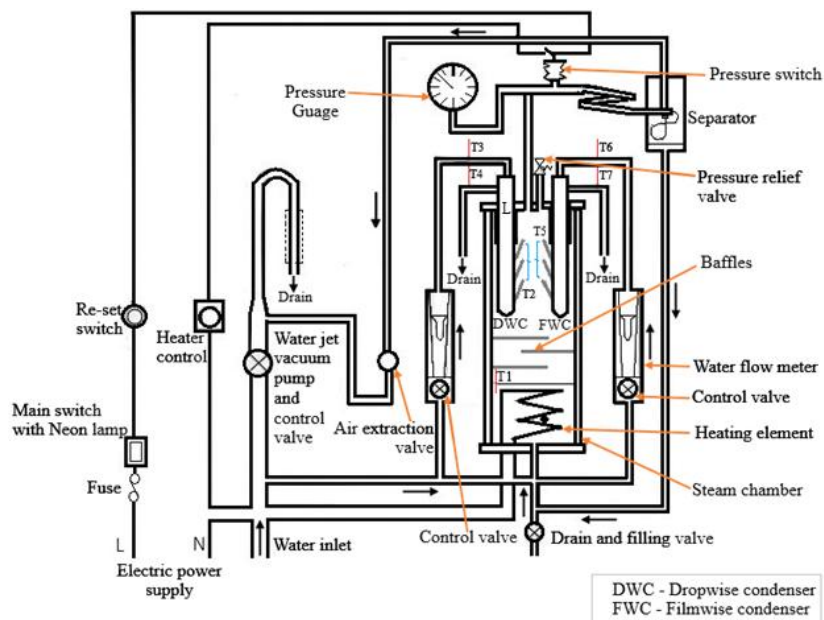


Fig. 2. Line diagram of the experimental setup.

Heater control regulates the heating element output from 0 to 3.0 kW. Press the black reset button (left of the control knob) to enable operation. The heater reset button must be pressed after the mains switch is turned off, if the pressure exceeds approximately 10 kN/m²(g), or after an overheating event caused by water accumulation in the chamber.

There are five function keys on the front of the digital temperature monitor. Flow meters with control valves were used in this project to show how fast the water was moving through the condenser. The pressure gauge compares the absolute pressure of the steam chamber and the ambient pressure.

2.2 Procedure

The water level in the chamber was carefully set, and air was removed from the system as part of the initial setup. The apparatus has been operated about 5 min at 100 °C where saturation occurs, employing a lower cooling water flow to assist heat all the parts and lessen glass condensation. The test was conducted at a consistent steam temperature. The heater was adjusted to maintain the steam temperature, and water was allowed to pass through the dropwise condenser gently. Temperatures of the steam (T_1), surface (T_2, T_5), cooling water inlet (T_3, T_6), outlet (T_4, T_7), and water flow rate (m) were recorded throughout the experiment. After altering the water flow rate, the heater was repositioned to restore the steam temperature to the desired level. The $T_1, T_2, T_3, T_4, T_5, T_6, T_7$ and m measurements were repeated. This process was carried out multiple times with varying water flow rates, up to the maximum. The same set of steps was then carried out using the filmwise condensation, with suitable water flow rates, and recording the corresponding temperatures and flow rate. Finally, both tests were repeated at additional constant steam temperatures of 80 °C and 60 °C.

Experimental analysis

In this experiment, the heat taken up by the cooling water was measured from its flow rate and the temperature rise between inlet and outlet (T_o and T_i). This heat is assumed to be equal to the heat released during steam condensation (q).

$$q = m C_p (T_o - T_i) \quad (1)$$

C_p is specific heat of water which is taken at the average of inlet and outlet temperatures.

Using this heat value, the average heat transfer coefficient is calculated with Newton's cooling law,

$$h = \frac{q}{A \Delta T} \quad (2)$$

Here A is condenser tube surface area and ΔT is temperature difference between the tube wall and the steam. Correction for temperature drop through condenser shell (ΔT_{shell}) is also included for the ΔT , so that the final heat transfer coefficient reaches accurate. From the Fourier equation,

$$\Delta T_{shell} = \frac{q z}{A k} = \frac{Q z}{k} \quad (3)$$

Here z is shell thickness and k is thermal conductivity.

The corrected steam to surface temperature difference ΔT considered for this work is

$$\Delta T = T_{steam} - T_{surface} - \Delta T_{shell} \quad (4)$$

Predicted analysis

Rohsenow's [31] improvement of Nusselt's theory improves the analysis of laminar FWC by taking temperature change inside the condensate film. It takes into consideration the liquid film's non-isothermal properties, in contrast to the classic Nusselt model. Hence it predicts the heat transfer coefficient with more precise mainly for vertical surfaces. Rohsenow [31] improved the Nusselt number by adjusting the constant and mathematical behavior of film

thickness and temperature distribution. This model gives better reliability for real engineering applications involving condensing pure steam.

Filmwise HTC for pure steam from Rohsenow [31]

$$h_{ps} = 0.943 \left[\frac{k^3 \rho (\rho - \rho_v) g (h_{fg} + 0.68 C_p \Delta T)}{L \mu \Delta T} \right]^{1/4} \quad (5)$$

In equation (5), L is condenser tube length.

Nusselt number for non-condensable gases (film wise) from Lee et al. [24]

$$Nu_{ncg} = 890 Gr^{0.125} (1 - W_a^{0.01})^{0.966} \left(\frac{C_p \Delta T}{h_{fg}} \right)^{-0.327} \quad (6)$$

Where $0 \leq W_a \leq 0.9$; $0.2 \leq P \leq 0.5 \text{ MPa}$; $20 \leq \Delta T \leq 70 \text{ K}$;

From the Nusselt number in equation (6), the HTC (h) can be estimated for filmwise by

$$Nu = \frac{h L}{k} \quad (7)$$

The HTC (h_{ncg}), for filmwise from Su et al. (2013) for non-condensable gases is given by

$$h_{ncg} = [10819 + 90416P - (4314 + 46537P) \log(100W_a)] \Delta T^{-0.6} \quad (8)$$

In equation (8), P is pressure in MPa.

The heat flux (Q) is given by

$$Q = h \Delta T \quad (9)$$

3. Results and Discussion

3.1 ΔT vs HTC

The observations in figures 3, 4 and 5 highlighted the critical importance of maintaining pure steam conditions in condensers to maximize heat transfer efficiency. Even small concentrations of air can severely impair performance, which must be considered in the design and maintenance of steam systems.

At steam temperatures of 100°C, 80°C and 60°C under pure steam, higher HTCs were obtained in dropwise condensation than filmwise. From observations of figures 3, 4 and 5, HTCs in dropwise were higher than filmwise. Because dropwise condensation exposes a greater surface area to the vapors and provides less thermal resistance during phase change. For similar ΔT at 100°C and 80°C steam temperature in figures 3 and 4, dropwise HTC reached nearly three times that of filmwise. For similar ΔT at 60°C steam temperature in figure 5, dropwise HTC reached nearly 2-2.5 times that of filmwise. Even with the same temperature differential, a decrease in steam temperature typically results in a reduced HTC in both dropwise and filmwise condensation. The reason is that heat transfer from the steam to the cooling surface might be impeded by the condensate film in filmwise condensation and the condensate droplets in dropwise condensation acting as thermal barriers. As ΔT increases, h decreases in both cases, but more steeply in filmwise. Dropwise maintains superior efficiency due to reduced thermal resistance from discrete droplets.

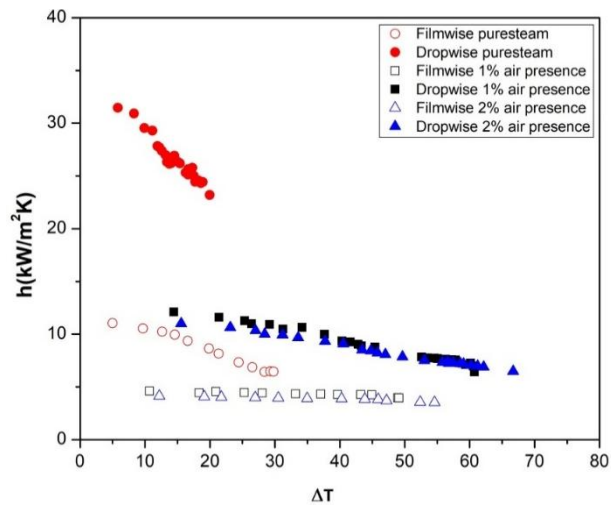


Fig. 3. Relation between HTC (h) and temperature difference ΔT at $T_{steam} = 100^\circ\text{C}$

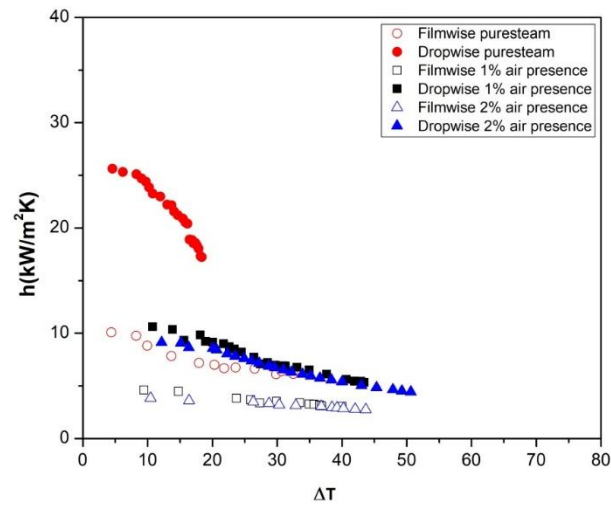


Fig. 4. Relation between HTC (h) and temperature difference ΔT at $T_{steam} = 80^\circ\text{C}$

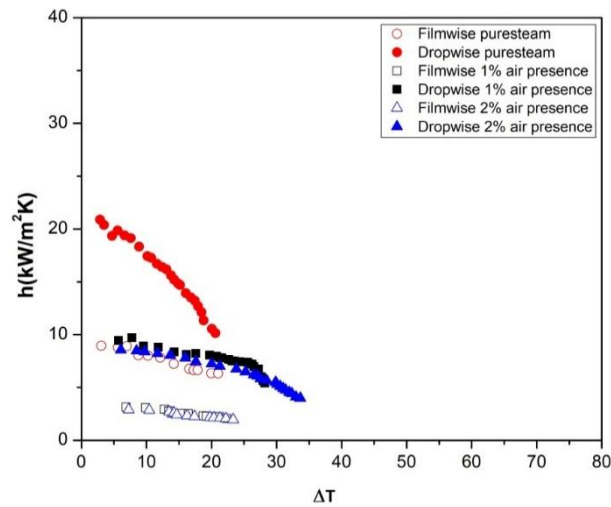


Fig. 5. Relation between HTC (h) and temperature difference ΔT at $T_{steam} = 60^\circ\text{C}$

The air presence created a diffusion barrier near the condensate interface, hindering the steam's ability to reach the surface and condense. This led to increase in thermal resistance and consequently a lower HTC (h). Furthermore, higher air concentrations exacerbated this barrier effect, caused reduction in HTC's progressively. A substantial reduction in HTC's was taken place with the presence of air. It corresponds to a reduction of approximately 55–60% in heat transfer performance even with only 1% air presence. The trend was continued with increasing ΔT ; however, the rate of decrease became more gradual. For example from observations at $\Delta T \approx 19^\circ\text{C}$ in filmwise condensation, the HTC fallen to around 4.461 kW/m²K at 1% air and 4.09 kW/m²K at 2% air, compared to 8.636 kW/m²K for pure steam. Similarly, at $\Delta T \approx 15^\circ\text{C}$ in dropwise condensation, the HTC fallen to around 9.322 kW/m²K at 1% air and 9.07 kW/m²K at 2% air, compared to 20.877 kW/m²K for pure steam. These results demonstrate that even minor quantities of air can significantly impede condensation heat transfer. For filmwise at 100°C, 1% air presence caused to 66.51% drop in HTC, while a 2% concentration lead to decrease of 68.81%. At 80°C, these values are 64.53% and 68.79% and similarly at 60°C, values are 51.18% and 60.45% for 1% and 2% air presence. For dropwise at 100°C in figure 3, the HTC dropped by 47.21% at 1%, and 52.79% at 2% air concentration. At 80°C in figure 4, the values are 51.76% and 57%, while at 60°C in figure 5, they are 66.1% and 68.47% for air presence of 1% and 2%, respectively.

3.2 ΔT vs Heat Flux

The dependance of Q on ΔT for FWC and DWC at different steam temperatures shown in figures 6, 7 and 8. Heat flux increased with increase in temperature difference (ΔT) in both filmwise and dropwise condensation. Because, the temperature gradient is directly proportional to heat flux. A greater temperature difference creates larger amount of thermal energy flowing each unit of time, which raises the condensation rate and increases heat flux.

For the same ΔT , dropwise condensation produced higher heat flux values and 2–3 times greater than filmwise. Because, heat transfer decreased by the liquid film created by filmwise condensation which increases thermal resistance. But dropwise condensation reduced this resistance due to better vapour surface contact.

The sudden drop in heat flux during condensation was taken place at constant ΔT in presence of air compared to pure steam. Because, air creates a diffusion barrier that prevents steam from reaching the surface and also decreases the partial pressure of steam while increasing thermal resistance. This reduced condensation rate and total heat transfer. Compared to pure steam, even minor quantities of air can greatly impair heat flux. From the readings at $\Delta T \approx 17^\circ\text{C}$ in dropwise condensation under $T_{\text{steam}} = 60^\circ\text{C}$, the heat flux fallen to around 144.60 kW/m² at 1% air and 130.99 kW/m² at 2% air, compared to 230.465 kW/m² for pure steam. Similarly, at $\Delta T \approx 20^\circ\text{C}$ in FWC, the heat flux dropped to around 45.189 kW/m² at 1% air and 43.57 kW/m² at 2% air, compared to 126.756 kW/m² for pure steam. The greater steam temperature results high heat flux even if the difference in temperature (ΔT) remains constant. Because, at constant ΔT , higher steam temperatures caused to increase the steam's saturation pressure and energy content which enhance condensation rate. This leads to a more vigorous generation and removal of droplets which enhanced surface contact and reduced thermal resistance. For example from readings, at $\Delta T \approx 26^\circ\text{C}$ under 1% of air presence in dropwise condensation, the heat flux were increased from 188.86 kW/m² at $T_{\text{steam}} = 60^\circ\text{C}$ to 203.35 kW/m² at $T_{\text{steam}} = 80^\circ\text{C}$ and 290.34 kW/m² at $T_{\text{steam}} = 100^\circ\text{C}$. Similarly, at $\Delta T \approx 10^\circ\text{C}$ under 1% of air presence in FWC, the hat flux values were obtained as 30.277 kW/m², 43.269 kW/m² and 49.369 kW/m² at T_{steam} of 60°C, 80°C and 100°C. It was observed at higher heat flux under T_{steam} of 60°C in figure 8, droplet removal from the surface became less efficient. The retained condensate droplets acted as an additional thermal resistance which caused a drop in overall heat flux for pure steam, 1% and 2% air presence cases.

3.3 Experimental vs Predicted

Figure 9 presents the experimental and theoretical (Nusselt) condensation HTC's in FWC for pure steam at various temperature differences (ΔT). It is observed that the heat transfer coefficient is more for experimental than predicted under pure steam case due to surface roughness, condensate subcooling, and increased turbulence in the condensate film. The experimental values fairly fitted with Nusselt predictions and average deviation reached to -14.21%. As expected, both experimental and theoretical values of the HTC (h) decreased with increasing ΔT , attributed to the thickening of the condensate coating thickening across its vertical face.

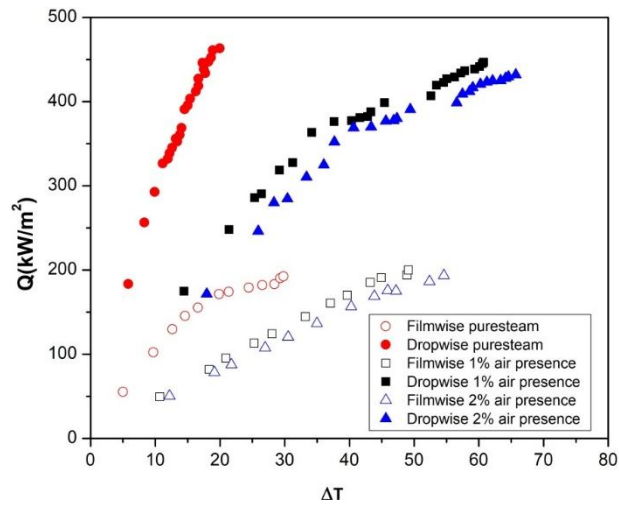


Fig. 6. The dependence of heat flux (Q) on temperature difference ΔT at $T_{steam} = 100^{\circ}\text{C}$

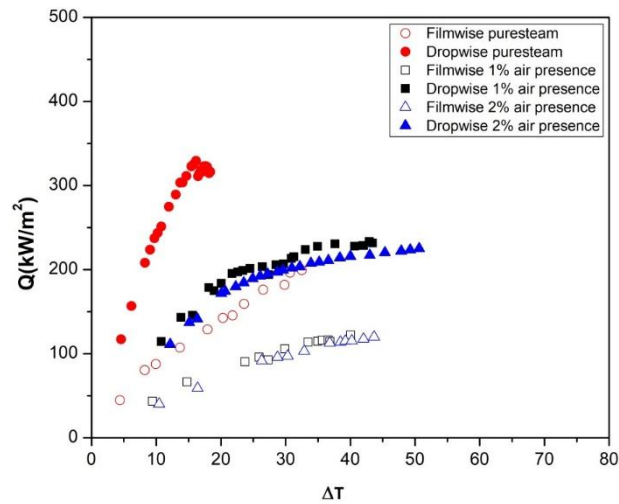


Fig. 7. The dependence of heat flux (Q) on temperature difference ΔT at $T_{steam} = 80^{\circ}\text{C}$

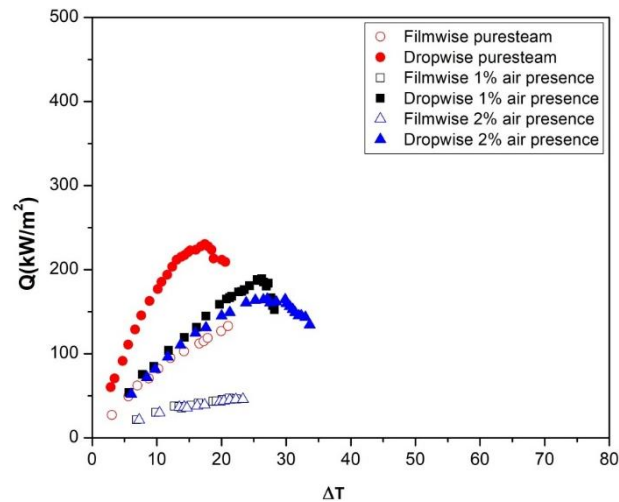


Fig. 8. The dependence of heat flux (Q) on temperature difference ΔT at $T_{steam} = 60^{\circ}\text{C}$

This study employed two empirical correlations established by Lee et al. [24] and Su et al. [27] for the assessment of Coefficient of heat transfer for the steam with air during Film-Wise Condensation. In order to assess how well these correlations worked, the measured data is compared with the theoretical values (figure 10). The percentage difference between the expected and actual heat transfer coefficients for the correlation put out by Lee et al. [24] varied from -15.42% to 17.49%. The range of error, however, was rather different in the correlation found by Su et al. [27], which varied from -6.29 % to 26.55 %. These differences emphasize the relative accuracy and limitations of each empirical model in capturing the complicated effects of the presence non condensable gases in the steam on condensation. Despite the discrepancies, the measured and theoretical findings (shown in figure 10) are in fairly good accord.

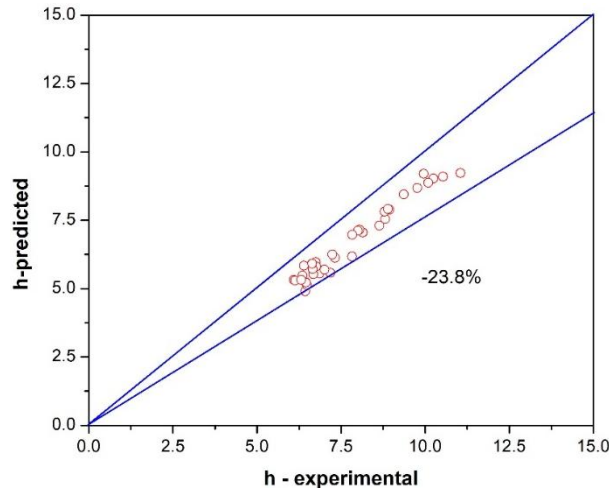


Fig. 9. Comparison of theoretical and measured HTC in FWC Under Pure steam

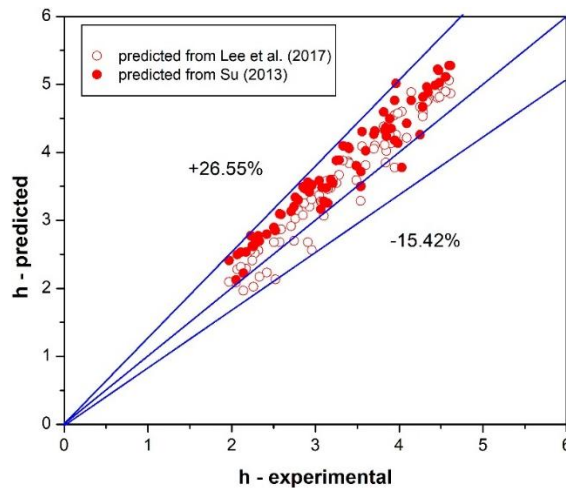


Fig. 10. Comparison of theoretical and measured HTC in FWC under NCG

4. Conclusion

- This work investigated the impact of air presence such as 0% (pure steam), 1% and 2% on heat transfer characteristics (HTC and heat flux) for dropwise and filmwise condensation at different steam temperatures (100°C, 80°C and 60°C). The HTC for DWC is 2 to 3 times greater than that for FWC.
- Sudden drop in HTC and heat flux was observed on air presence in both the FWC and DWC. HTC was decreased by an average of 55.02% and 59.92% during FWC and 60.74% and 66.01% during DWC at air concentrations of 1% and 2% compared to pure steam.

- Compared to pure steam, heat flux was reduced by an average of 28.67% and 30.54% during FWC at air concentrations of 1% and 2%, respectively, and by 14.28% and 18.54% during DWC.
- The average deviation observed in HTC when comparing the experimental results of FWC with predicted values is -14.21% under pure steam conditions and 6.69% using Lee et al. [24] model & 14.423% using Su et al. [27] model for the steam with the non-condensable gases.

Nomenclature

A	tube surface area (m^2)
C_p	Specific heat (kJ/kgK)
Gr	Grashof number
g	gravitational acceleration (m/s^2)
h	heat transfer coefficient ($\text{W/m}^2 \text{K}$)
h_{fg}	Latent heat kJ/kg
k	thermal conductivity (W/mK)
L	tube length (m)
m	mass flow rate (kg/s)
Nu	Nusselt number
P	pressure (Pa)
Q	heat flux (W/m^2)
q	heat transfer rate (W)
T	temperature ($^{\circ}\text{C}$)
W	mass fraction
z	shell thickness (mm)

Greek letters

ρ	density (kg/m^3)
μ	viscosity (kg/ms)

Subscript

a	air
ncg	non-condensable gas
ps	pure steam
steam	steam
surface	surface

Abbreviation

DPW	dropwise condensation
FPW	filmwise condensation
HTC	heat transfer coefficient
HC	hydrocarbon
NCG	Non-condensable gas

References

- [1] Rose JW. Film and dropwise condensation. In: Handbook of Thermal Science and Engineering. Cham: Springer; 2018.
- [2] David GW, Leroy AB, Stanley MR. Dropwise and filmwise condensation of water vapor on gold. AIChE Journal. 1973;19(1):119-123.
- [3] Kim DE, Ahn HS, Kwon TS. Experimental investigation of filmwise and dropwise condensation inside transparent circular tubes. Applied Thermal Engineering. 2016.
- [4] Bachir EF, Girish K, Srinivas G. A review of dropwise condensation: Theory, modeling, experiments, and applications. International Journal of Heat and Mass Transfer. 2020;160:120172.
- [5] Chung B-J, Kim S, Kim MC. Experimental comparison of film-wise and drop-wise condensations of steam on vertical flat plates with the presence of air. International Communications in Heat and Mass Transfer. 2004;31(8):1067-1074.

- [6] Colburn AP, Hougen OA. Design of cooler condensers for mixtures of vapors with noncondensing gases. *Industrial Engineering Chemical*. 1934;26:1178-1182.
- [7] Tanner DW, Pope D, Potter CJ, West D. Heat transfer in DWC at low steam pressure in the absence and presence of non-condensable gas. *International Journal of Heat and Mass Transfer*. 1968;11(2):181-190.
- [8] Tanner DW, Pope D, Potter CJ, West D. Heat transfer in DWC Part I. The effect of heat flux steam velocity and non-condensable gas concentration. *International Journal of Heat and Mass Transfer*. 1965;8:419-426.
- [9] Wang CY, Tu CJ. Effects of non-condensable gas on laminar film condensation in a vertical tube. *International Journal of Heat and Mass Transfer*. 1988;31(11):2339-2345.
- [10] Minkowycz WJ, Sparrow EM. Condensation heat transfer in the presence of non-condensables, interfacial resistance, super heating variable properties and diffusion. *International Journal of Heat and Mass Transfer*. 1966;9:1125-1144.
- [11] Ma X-H, Zhou X-D, Lan Z, Li Y-M, Zhang Y. Condensation heat transfer enhancement in the presence of non-condensable gas using the interfacial effect of dropwise condensation. *International Journal of Heat and Mass Transfer*. 2008;51:1728-1737.
- [12] Poredoš P, Petelin N, Žel T, Vidrih B, Gatarić P, Kitanovski A. Performance of the condensation process for water vapour in the presence of a non-condensable gas on vertical plates and horizontal tubes. *Energies*. 2021;14(8):2291.
- [13] Jakub K, Jan H, Tomaš D. Theoretical and experimental study of water vapour condensation with high content of non-condensable gas in a vertical tube. *Acta Polytechnica*. 2022;62(3):352-360.
- [14] Lu J, Cao H, Li JM. Condensation heat and mass transfer of steam with non-condensable gases outside a horizontal tube under free convection. *International Journal of Heat and Mass Transfer*. 2019;139:564-576.
- [15] Minghui G, Shixue W, Jun Z, Yulong Z, Liansheng L. Condensation of steam with high CO₂ concentration on a vertical plate. *Experimental Thermal and Fluid Science*. 2016;75:147-155.
- [16] Gu HF, Chen Q, Wang HJ, Zhang HQ. Condensation of a hydrocarbon in the presence of a non-condensable gas: Heat and mass transfer. *Applied Thermal Engineering*. 2015;91:938-945.
- [17] Albdour SA, Addad Y, Alyammahi N, Afgan I. Steam condensation heat transfer in the presence of noncondensable gases (NCGs) in nuclear power plants (NPPs): A comprehensive review of fundamentals, current status, and prospects for future research. *International Journal of Energy Research*. 2024:Article ID 2880812.
- [18] Ma X, Ma J, Tong H, Jia H. The investigation on heat transfer characteristics of steam condensation in presence of noncondensable gas under natural convection. *Science and Technology of Nuclear Installations*. 2021:Article ID 6689597.
- [19] Uchida H, Oyama A, Togo Y. Evaluation of post-incident cooling systems of light-water power reactors. Third International Conference on the Peaceful Uses of Atomic Energy; 1964 Aug 31-Sep 9; Geneva, Switzerland. New York: United Nations; 1965. p. 93-104.
- [20] Tagami T. Interim report on safety assessments and facilities establishment project. Japanese Atomic Energy Research Agency; 1965.
- [21] Kataoka Y, Fukui T, Hatamiya S, Nakao T, Naitoh M, Sumida I. Experiments on convection heat transfer along a vertical flat plate between pools with different temperatures. *Nuclear Technology*. 1992;99:386-396.
- [22] Murase M, Kataoka Y, Fujii T. Evaporation and condensation heat transfer with noncondensable gas present. *Nuclear Engineering and Design*. 1993;141:135-143.
- [23] Dehbi A, Golay M, Kazimi M. Condensation experiments in steam-air and steam-air-helium mixtures under turbulent natural convection. *National Heat Transfer Conference*; 1991.
- [24] Lee Y, Jang Y, Choi D. An experimental study of air-steam condensation on the exterior surface of a vertical tube under natural convection conditions. *International Journal of Heat and Mass Transfer*. 2017;104:1034-1047.
- [25] Liu H, Todreas NE, Driscoll MJ. An experimental investigation of a passive cooling unit for nuclear plant containment. *Nuclear Engineering and Design*. 2000;199:243-255.
- [26] Kawakubo K, Aritomi M, Kikura H, Komeno T. An experimental study on the cooling characteristics of passive containment cooling systems. *Journal of Nuclear Science and Technology*. 2009;46:339-345.
- [27] Su J, Su Z, Fan G, Ding M. Experimental study of the effect of non condensable gases on steam condensation over a vertical tube external surface. *Nuclear Engineering and Design*. 2013;5:201-208.
- [28] Su J, Su Z, Ding M, Fan G. Analysis of experiments for the effect of non condensable gases on steam condensation over a vertical tube external surface under low wall subcooling. *Nuclear Engineering and Design*. 2014;278:644-650.

- [29] Fan G, Tong P, Sun Z, Chen Y. Development of a new empirical correlation for steam condensation rates in the presence of air outside vertical smooth tube. *Annals of Nuclear Energy*. 2018;113:139-146.
- [30] Zhu SL, Li Y, Zhang RP. Experimental study on the condensation characteristics of nitrogen with non-condensable gas. *Cryogenics*. 2019;98:29-38.
- [31] Rohsenow WM. Heat transfer and temperature distribution in laminar film condensation. *Transactions of the ASME*. 1956;78:1645-1648.