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DECARBONISATION THROUGH INNOVATION

Experimental study on the Heat transfer performance of Enhanced tubes for falling film evaporation

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Abstract

In response to global carbon neutrality goals, improving energy efficiency in the industrial sector has become increasingly important, leading to the expansion of industrial heat pump applications utilizing waste heat. Development and application of low global warming potential (GWP) refrigerants to replace HFCs are accelerating. R1233zd(E) is suitable for industrial heat pump systems utilizing low-temperature waste heat due to its non-flammability, low toxicity, and relatively higher latent heat compared to other low-GWP refrigerants. Falling film evaporators spray refrigerant directly onto tube surfaces, offering the advantage of lower refrigerant charge compared to flooded evaporators. Falling film evaporators have primarily been used in small-capacity systems, and because refrigerant is directly sprayed to enhanced tube surfaces, accurate evaluation of the external heat transfer performance of enhanced tubes is necessary. This study experimentally investigates the overall heat transfer coefficient of enhanced evaporation tubes with different fins per inch (FPI). The Outer and overall heat transfer coefficients are measured over a falling film Reynolds number range of 50–300, with a refrigerant temperature of 36.5 °C. Overall heat transfer coefficient is 5 $kW/(m^2 \cdot K)$ for 45 FPI tube, 5.5 $kW/(m^2 \cdot K)$ for 48FPI tube, and 6 $kW/(m^2 \cdot K)$ for 52FPI tube at 0.9 m/s water velocity. At 2.0 m/s water velocity, the overall heat transfer coefficient for all enhanced tubes increases by more than 10% compared to the overall heat transfer coefficient at 0.9 m/s water velocity. The Outer heat transfer coefficient is 9 $kW/(m^2 \cdot K)$ for 45 FPI tube, 10 $kW/(m^2 \cdot K)$ for the 48 FPI tube, and 12 $kW/(m^2 \cdot K)$ for 52 FPI tube at 2.0 m/s water velocity.

Keywords: Enhanced tube; Falling film evaporator; Industrial heat pump; R1233zd(E)

1. Introduction

In recent years, global efforts toward carbon neutrality and the increasing emphasis on ESG (Environmental, Social, and Governance) principles have led to the tightening of energy-related regulations across industrial sectors. Industrial heat pumps are gaining attention for their ability to recover process waste heat with minimal input energy, making them suitable for decarbonizing medium to low-temperature heat demand in manufacturing processes.[1]

Falling film evaporators are particularly advantageous in reducing the total refrigerant charge, [2] as they distribute the refrigerant directly over the outer surface of the heat transfer tubes in the form of a thin liquid film. However, falling film evaporators require additional control mechanisms to ensure stable operation. [3] Moreover, since heat transfer occurs primarily on the outer surface of the tube, it is essential to accurately evaluate the heat transfer performance based on the external surface structure of the enhanced tube. [4]

This study is an extension of previous research [5], which evaluated the overall and outer heat transfer coefficients of enhanced tubes at a water velocity of 0.9 m/s. While the previous work established a fundamental performance baseline, industrial heat pumps require higher velocities to mitigate the risk of performance degradation caused by fouling. The present study focuses on evaluating the overall and outer heat transfer performance under industrial operating conditions, specifically at a water velocity of 2.0 m/s.

2. Methodology

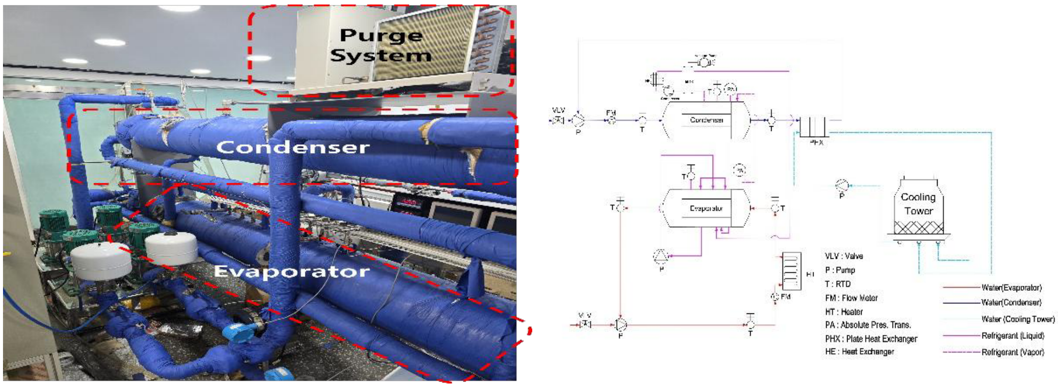

Fig. 1. Photo and schematic diagram of falling film evaporator and condenser set up

Due to the difficulty of accurately measuring the surface temperature of enhanced tubes in a shell and tube heat exchanger, a separate single-tube apparatus is installed to enable more precise measurement of the heat transfer coefficients. The inner Nusselt number correlation obtained from the single-tube apparatus is used to calculate the external and overall heat transfer coefficients.

Fig. 1 is set up of heat exchanger applied this study. Heated water flows through the inner side of tube and refrigerant are delivered to distributor and tray via a magnetic gear pump and sprayed over the outer surface in the form of falling film. Evaporated refrigerant is liquefied in the condenser and reused in the evaporator. Water from the condenser flows through a plate exchanger and exchanges heat with a cooling tower, enabling continuous condensation of the refrigerant.

The inner heat transfer coefficient is calculated based on Newton's law of cooling using the measured surface temperature of the tube and inlet and outlet temperatures of the working fluid. The heat flux of tube is determined by Eq. (1). The average value is determined by calculating the heat transfer coefficients at the inlet and outlet sides as shown in Eq. (2) to (4). The correlation for the inner heat transfer coefficient is expressed as a function of the Reynolds number (Re_i) and Prantl number (Pr_i) through linear regression as Eq. (5)

$$q" = \frac{\dot{Q}_a}{A_i} = \frac{\dot{m}_a \cdot C_p \cdot (T_{i,outlet} - T_{i,inlet})}{\pi \cdot d_i \cdot L} \tag{1}$$

$$h_{i,inlet} = \frac{q"}{(T_{i,s} - T_{i,inlet})} \tag{2}$$

$$h_{i,outlet} = \frac{q"}{(T_{i,s} - T_{i,outlet})} \tag{3}$$

$$h_i = \frac{h_{i,inlet} + h_{i,outlet}}{2} \tag{4}$$

$$Nu_i = \frac{h_i \cdot d_i}{k_a} = 0.0557 \cdot Re_i^{0.8} \cdot Pr_i^{0.3} \tag{5}$$

The energy balance between the water flowing inside the enhanced tube and refrigerant flowing on the outer surface can be defined as Eq. (5). Using overall heat transfer coefficient and inner heat transfer coefficient, the outer heat transfer coefficient is derived using Eq. (8).

$$\dot{Q_{ref}} = C_{P_w} \cdot \dot{m_w} \cdot \Delta T_w = U \cdot A \cdot \Delta T_{LMTD} \tag{6}$$

$$R_{ov} = R_o + R_w + R_i \tag{7}$$

$$\frac{1}{UA} = \frac{1}{h_o A_o} + R_w + \frac{1}{h_i A_i} \tag{8}$$

The falling film Reynolds number is defined as Eq. (9).

$$Re_f = \frac{4 \cdot \Gamma}{\mu_{liq}} = \frac{2 \cdot \dot{m_{ref}}}{\mu_{liq} \cdot L} \tag{9}$$

3. Results and discussion

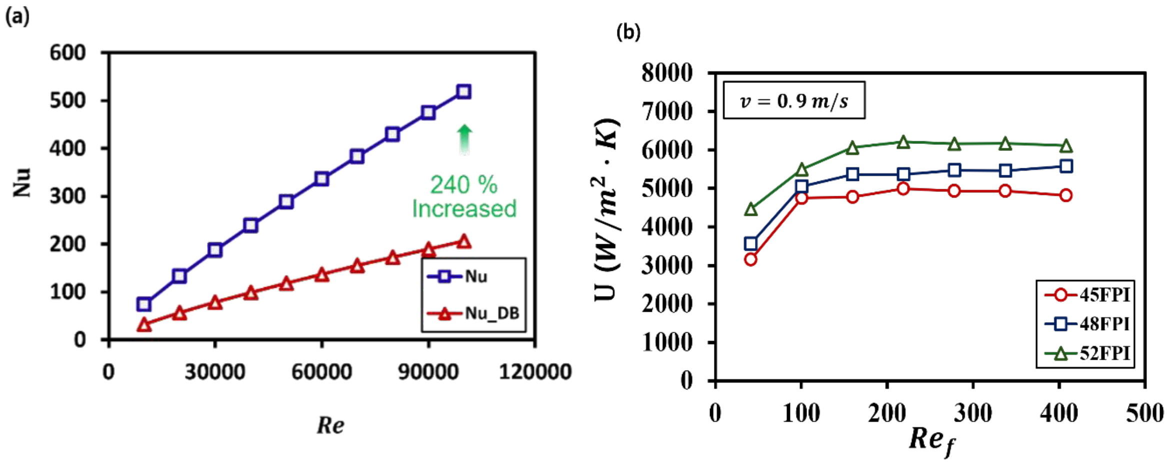

Fig. 2 Inner Nusselt number (a), overall heat transfer coefficient with respect to falling film Reynolds number (b) at $v = 0.9\ m/s$[5]

Fig. 2 (a) shows the inner heat transfer coefficient in the tube as a function of the Reynolds number (Re). The enhanced tube, which has a ridge structure inside tube, demonstrates an inner heat transfer performance that is 2.4 times higher than the inner heat transfer coefficient of the smooth tube predicted by the Dittus-Boelter correlation.

Fig. 2 (b) shows the overall heat transfer coefficient (U) for the three enhanced tubes with respect to Re_f at water velocity 0.9 m/s ($v = 0.9\ m/s$). The outer heat transfer coefficient for all three types of tubes remains constant above Re_f of 220. This is because a uniform refrigerant film forms on the outer surface of the enhanced tube at Re_f above 220, and the incidence of dryout regions is extremely low. In actual industrial heat pump evaporators, the heat transfer performance may decrease because of fouling when the water velocity is 0.9m/s. Therefore, the heat transfer performance of the enhanced tube is investigated under conditions of higher flow velocity.

Fig. 3 (a) illustrates the outer heat transfer coefficient for 45 FPI and 52 FPI tube with respect to Re_f at water velocity 2.0 m/s ($v = 2.0\ m/s$). The 52FPI tube achieves an outer heat transfer coefficient of up to $12kW/m^2 \cdot K$. At the same falling film Reynolds number, 52 FPI tube shows approximately 12% improvement in outer heat transfer performance compared to 48 FPI tube and 30% improvement in outer heat transfer performance compared to 45 FPI.

Fig. 3. (b) illustrates the overall heat transfer coefficient (U) for 45FPI and 52FPI tubes with respect to the Re_f at water velocity of 2.0 m/s. Compared to the Fig. 2 (c), The overall heat transfer coefficient of the 45 FPI tube and 48 FPI tube improved by 12% as the water velocity increased from 0.9 m/s to 2.0 m/s. Similarly, for the 52 FPI tube, the overall heat transfer coefficient showed an 11% enhancement as the water velocity increased from 0.9 m/s to 2.0 m/s.

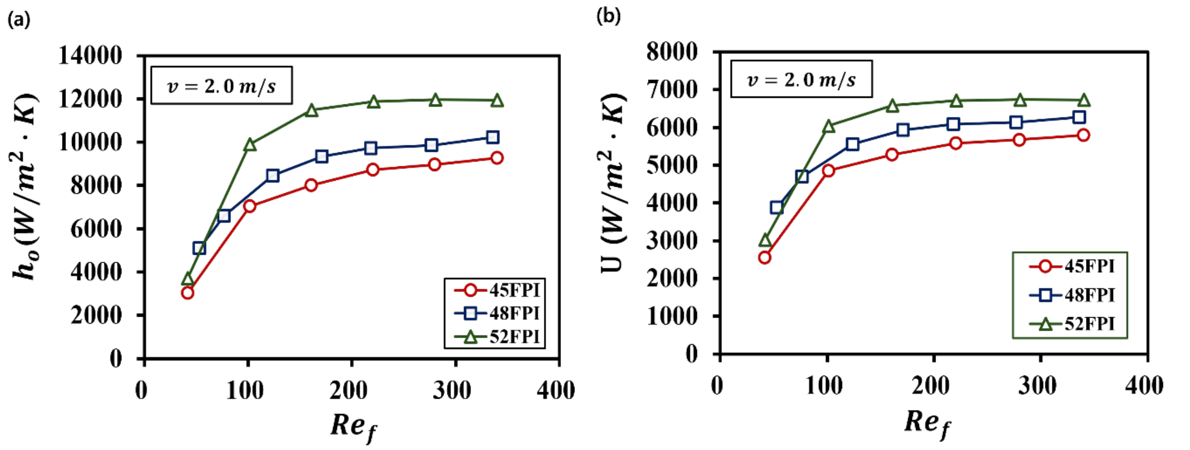

Fig. 3 outer heat transfer coefficient (a), overall heat transfer coefficient with respect to falling film Reynolds number (b) at $v = 2\ m/s$

4. Conclusion

This study evaluated the heat transfer performance of enhanced tubes applied to a falling film evaporator using R1233zd(E) as the working fluid. The following conclusions were drawn from the present study.

1. Internal surface enhancement applied in this study improves the heat transfer performance by 2.4 times compared to smooth tube.
2. As the number of FPI increased, the heat transfer performance of the enhanced tubes also improved. Under conditions where a sufficiently film is fully developed, the outer heat transfer coefficient (h_o) of the 52 FPI tube is 12% higher than 48 FPI tube and 30% higher than 45 FPI tube.
3. Increasing the internal flow velocity enhances the overall heat transfer coefficient (U) by intensifying the heat transfer rate within the enhanced tube. However, the optimal operating condition must be determined by comprehensively comparing the resulting thermal performance enhancement against the corresponding increase in pumping power consumption.

5. Nomenclature

A : surface area of tube (m^2)
C : specific heat ($J/kg \cdot K$)
d : diameter (m)
h : heat transfer coefficient (W/$m^2 \cdot K$)
k : thermal conductivity (W/$m \cdot K$)
L : tube length (m)
$\dot{m}$: mass flow rate (kg/s)
Nu : Nusselt number
Pr : Prandtl number
$\dot{Q}$: heat transfer rate (W)
q": heat flux (W/m^2)
R : thermal resistance (K/W)
Re : Reynolds number
T : temperature (K)
U : Overall heat transfer coefficient (W/$m^2 \cdot K$)
μ : viscosity ($Pa \cdot s$)
Γ : mass flow rate per unit length ($kg/s \cdot L$)

Subscripts
a : air
f : falling film
i : inner
liq : liquid
LMTD : Log mean temperature difference
o : outer
ov : overall
P : constant pressure
ref : refrigerant
s : surface
w : water

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