Title:

Condensation from superheated vapor flow of R744 and R410A at subcritical pressures in a horizontal smooth tube

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Abstract

This paper presents experimentally determined heat transfer coefficients for condensation from a superheated vapor of CO_2 and R410A. The superheated vapor was flowed through a smooth horizontal tube with 6.1 mm ID under almost uniform temperature cooling at reduced pressures from 0.55 to 0.95, heat fluxes from 3 to 20 kWm⁻², and superheats from 0 to 40 K. When the tube wall temperature reaches the saturation point, the measured results show that the heat transfer coefficient gradually starts deviating from the values predicted by a correlation valid for single-phase gas cooling. This point identifies the start of condensation from the superheated vapor. The condensation starts earlier at higher heat fluxes because the tube wall temperature reaches the saturation point earlier. The heat transfer coefficient reaches a value predicted by correlations for condensation at a thermodynamic vapor quality of 1. The measured heat transfer coefficient of CO_2 is roughly 20 to 70% higher than that of R410A at the same reduced pressures. This is mainly because the larger latent heat and liquid thermal conductivity of CO_2 , compared to that of R410A, increase the heat transfer coefficient.

Keywords

Carbon dioxide, R410A, Condensation, Horizontal tube, Heat transfer, Refrigeration

Nomenclature

- Cp isobaric heat capacity, J kg⁻¹K⁻¹
- $d_{\rm i}$ inner diameter of test tube, m
- f friction factor
- $F_{\rm a}$ correction factor for radial property change

- g gravitational acceleration, m s⁻²
- G mass flux, kg m⁻²s⁻¹
- h specific enthalpy, J kg⁻¹
- $\Delta h_{\rm LV}$ latent heat, J kg⁻¹
- $J_{\rm G}$ dimensionless gas velocity in Eq. (34)
- J_{G}^{T} transition dimensionless gas velocity in Eq. (34)
- \dot{m} mass flow rate, kg s⁻¹
- *n* number of data point
- *Nu* Nusselt number
- P pressure, Pa
- *Pr* Prandtl number
- q heat flux, W m⁻²
- \dot{Q} heat transfer rate, W
- *r* radial position, m
- *R* radius of test tube (= $d_i/2$), m
- s cross sectional free flow area, m²
- T temperature, °C
- u refrigerant velocity in axial direction, m s⁻¹
- x_{a} actual vapor quality
- *x*_b thermodynamic equilibrium vapor quality
- ΔZ_{α} active cooling length of test tube, m

Greek symbols

α	heat transfer coefficient, W m ⁻² K ⁻¹
Xtt	Lockhart-Martinelli parameter for turbulent flow
Е	relative deviation, %
λ	thermal conductivity, W m ⁻¹ K ⁻¹
μ	viscosity, Pa·s
ν	kinematic viscosity, m ² s ⁻¹
ρ	density, kg m ⁻³
σ	standard deviation, %
ξa	actual void fraction

Subscripts

crit

0	uniform property in radial direction
amb	ambient
A	annular
b	evaluated at bulk temperature
cal	calculated value
center	at the center of test tube
cond	conduction heat
crit	critical point

exp	experimental value
f	evaluated at film temperature
FC	free convective film-wise condensation
gain	heat gain or leak
H2O	water
i	inlet
L	liquid
latent	latent heat
LO	liquid only
МС	mixer
0	outlet
PC	pre-cooler
r	refrigerant
sat	saturation temperature
SC	subcool
SH	superheat
strat	stratified
total	total
TP	two-phase
TS	test section

i temperature of the interior tube wall or evaluated at interior tube wall

1. Introduction

Refrigerants are usually superheated at the inlet of condensers, and the heat is rejected in a sensible way along a length of tube typically called the superheat zone or de-superheating zone. Superheated vapor begins to condense at a certain point where the temperature of the fluid close to the tube wall reaches the saturation value. From that point on, condensate will be present at the wall while the core flow is still superheated. This paper focuses on the zone where condensation occurs in the presence of superheated vapor and the processes involved. Saturated condensation occurs when the bulk enthalpy reaches a thermodynamic quality of 1, where the bulk temperature is nearly equal to the saturation temperature. The tube wall mostly absorbs latent heat and minute sensible heat from the subcooled condensate. Although scarce, some literature on condensation from superheated vapor is available. However, the authors are not aware of any studies describing fluids and conditions relevant for vapor compression cycles.

Minckowycz and Sparrow [1][2] considered effects of superheat on condensation heat transfer to an isothermal vertical cooling plate. To the best of our knowledge, these are the first fundamental studies on the subject. Following these studies, Winker et al. [3] concluded that the effect of superheat causes a negligible increase in the wall heat flux for pure vapors. However, the effect is more significant when the flow includes non-condensable gas. Mitrovic [4] approached the same issue theoretically and concluded that vapor convection reduces the condensate film thickness and lowers transport resistance.

Stern and Votta [5] experimentally investigated the overall heat transfer coefficient (HTC) of down flow condensation in a 1.07 m long water-cool vertical tube with 25.3 mm ID. Recently, Hsu [6] numerically

investigated the effect of superheat on condensation onto a sphere. Longo [7] experimentally investigated the effect of superheat on the HTC of water-cool brazed plate heat exchangers. The results of this study showed that superheat more significantly affects inner condensation flows compared to exterior condensation flows. In the case where the temperature of the cooling wall changes along the flow direction, such as during in-tube flow, condensation is expected to start from a certain point, and superheat would have a clear impact on condenser design and the total HTC.

Balekjian and Katz [8] experimentally investigated condensation from superheated vapors of R114 and steam on the outer surface of a horizontal tube. Their experimental HTC suggested that lowering the cooling surface temperature below the saturation point generates a condensate from superheated vapor.

Altman et al. [9] provided six points of experimentally determined HTC, which were averaged from various superheated inlets to a saturated outlet in a 1.22 m long test section with an ID of 8.71 mm. With these six data points measured at various degrees of superheat at the test section inlet, an approximate 30 to 70% decrease in HTC from a two-phase zone condensation correlation was confirmed. They also hypothesized about very thin ridges and intermittent droplets on the inner surface of the glass tube for high-velocity superheated vapor.

Bell [10] also accepted that the criterion for condensation to occur in a de-superheating zone, i.e., superheat zone, is having a tube wall temperature below the saturation point and cautioned that the simple use of the LMTD (logarithmic mean temperature difference) method to calculate the overall coefficient of condensers could be invalid.

Miropolskiy et al. [11] provided experimental data for the quasi-local HTC of superheated steam vapor flowing downward in a cooled vertical smooth tube. The cooling length of the test tube was sectionalized so that the enthalpy change through one section was relatively small. The results experimentally verified the criterion for the start of condensation, which occurs when the tube wall temperature falls below the saturation point. Furthermore, their data showed the behavior of superheat-zone condensation for various reduced pressures until $P/P_{crit} = 0.82$.

Fujii et al. [12] experimentally investigated the condensation of R11 and R113 flow in horizontal smooth tubes. From the temperature distribution in the radial direction of the horizontal middle plane of the tube, they proved the coexistence of superheated vapor and subcooled liquid in condensation flow. They varied the vapor mass quality, which indicates the actual vapor and liquid mass flow rate in nonequilibrium conditions, to analyze the mass transport process. They also established a semi-empirical prediction method for HTC.

Lee et al. [13] experimentally investigated condensation in superheated R22 vapor and proposed a physical model to account for the sensible heat in condensation heat transfer. Their model defines the HTC bulk temperature as the reference temperature. In response, Webb [14] reported that the sensible heat is negligible, and a simplified model defining HTC using the saturation temperature gives the same results as the model presented by Lee.

Karwacki et al. [15] visualized the heat rejection process of superheated R507 and isobutene (R600a) with oil flow in an air-cool glass tube. According to their observations, small droplets of accumulated oil were observed when the degree of vapor superheat was sufficiently high. When condensation started, a very thin liquid film of the condensate, formed in presence of highly superheated vapor, was observed.

Typical condenser models divide the heat rejection process into three zones: de-superheating, condensation, and subcooling. Heat transfer in the superheat zone is typically modeled by assuming single-phase flow with the justification that the rejected heat is insignificant when compared to heat rejected during õcondensationö. The development of various systems and applications drove operating conditions closer to the critical point, at which these assumptions were less justifiable. Often, the rejected heat in the superheat zone is greater than in the two-phase zone, specifically as pressure increases toward the critical point. Conditions just below the critical point are increasingly common for both R410A water heaters and CO_2 commercial refrigeration systems operating between autumn and spring. The authors [16] presented experimental results for a CO_2 heat rejection (cooling/condensation) flow in a 6.1 mm ID smooth tube at pressures ranging from 5.0 to 7.5 MPa. The experimental results showed condensation in the superheated vapor. To further validate these experimental results, one additional refrigerant, R410A, was explored. This particular refrigerant was chosen because of its importance for various applications and the effect of the properties of the refrigerant on condensation heat transfer.

2. Definition of the heat transfer coefficient

Refrigerant flow is normally superheated in the superheat zone at the beginning. As a result of the high refrigerant temperature, for a constant coolant (air or water) temperature (high heat flux), the tube wall is still above the saturation temperature for a given pressure and the refrigerant rejects sensible heat. At some point, the tube wall reaches the saturation temperature and conditions for condensation are met. The subcooling of the tube wall is small, and the latent heat transfer rate can still be small relative to the sensible heat. Thus, this heat rejection process is envisioned as a simultaneous heat transfer (latent and sensible), where the ratio of the sensible to total heat exchange starts from 1 and eventually decreases 0. Due to the difficulties in precisely quantifying this ratio, the heat balance and definition of HTC in this study are described below.

2.1 Superheat, subcool degree and bulk mean enthalpy

The bulk mean specific enthalpy of flow passing through a cross section is described as

$$h_{\rm b} = \frac{\int (h\rho u)ds}{\int (\rho u)ds} = \frac{\int (C\rho T_{\rm r}\rho u)ds}{\int (\rho u)ds}$$
(1)

where Cp, ρ , and u are the local specific heat, density, and velocity of the refrigerant at the small cross-sectional free flow area, ds, respectively. Similarly, the average vapor and liquid specific enthalpies \overline{h}_V and \overline{h}_L , are described as functions of the degree of superheat $\Delta T_{\rm SH}$ and the degree of subcool $\Delta T_{\rm SC}$.

$$\overline{h}_{V} = \frac{\int (h_{V} \cdot \rho_{V} u_{V}) ds}{\int (\rho_{V} u_{V}) ds} = h_{Vsat} + \frac{\int (Cp_{V} \Delta T_{SH} \cdot \rho_{V} u_{V}) ds}{\int (\rho_{V} u_{V}) ds}
\overline{h}_{L} = \frac{\int (h_{L} \cdot \rho_{L} u_{L}) ds}{\int (\rho_{L} u_{L}) ds} = h_{Lsat} - \frac{\int (Cp_{L} \Delta T_{SC} \cdot \rho_{L} u_{L}) ds}{\int (\rho_{L} u_{L}) ds} = h_{Vsat} - \Delta h_{LV} - \frac{\int (Cp_{L} \Delta T_{SC} \cdot \rho_{L} u_{L}) ds}{\int (\rho_{L} u_{L}) ds}$$
(2)

Here, h_{Vsat} and h_{Lsat} are the specific enthalpy of the saturated vapor and liquid. Δh_{LV} is the latent heat ($h_{Vsat} - h_{Lsat}$). Denominators in Eqs. (1) and (2) are the total, vapor and liquid mass flow rates, \dot{m}_{total} , \dot{m}_{V} , and \dot{m}_{L} .

$$\dot{m}_{\text{total}} = \int (\rho u) ds = \dot{m}_{\text{v}} + \dot{m}_{\text{L}}$$

$$\dot{m}_{\text{v}} = \int (\rho_{\text{v}} u_{\text{v}}) ds$$

$$\dot{m}_{\text{L}} = \int (\rho_{\text{L}} u_{\text{L}}) ds$$
(3)

The average heat rate carried by the superheated vapor, and the subcooled liquid simplifies Eq. (2) to Eq. (4)

$$\frac{\int (Cp_{\rm v} \Delta T_{\rm SH} \cdot \rho_{\rm v} u_{\rm v}) ds}{\int (\rho_{\rm v} u_{\rm v}) ds} = \overline{Cp_{\rm v} \Delta T_{\rm SH}} = \Delta \overline{h}_{\rm SH}, \quad \frac{\int (Cp_{\rm L} \Delta T_{\rm SC} \cdot \rho_{\rm L} u_{\rm L}) ds}{\int (\rho_{\rm L} u_{\rm L}) ds} = \overline{Cp_{\rm L} \Delta T_{\rm SC}} = \Delta \overline{h}_{\rm SC}$$

$$\overline{h}_{\rm v} = h_{\rm v_{sat}} + \overline{Cp_{\rm v} \Delta T_{\rm SH}} = h_{\rm v_{sat}} + \Delta \overline{h}_{\rm SH}$$

$$\overline{h}_{\rm L} = h_{\rm v_{sat}} - \Delta h_{\rm Lv} - \overline{Cp_{\rm L} \Delta T_{\rm SC}} = h_{\rm v_{sat}} - \Delta h_{\rm Lv} - \Delta \overline{h}_{\rm SC}$$
(4)

The bulk enthalpy h_b in Eq. (1) is transcribed as

$$h_{\rm b} = \left(\overline{h}_{\rm v}\dot{m}_{\rm v} + \overline{h}_{\rm L}\dot{m}_{\rm L}\right) / \dot{m}_{\rm total} = \left(h_{\rm vsat} + \overline{Cp_{\rm v}\Delta T_{\rm SH}}\right) \left(\dot{m}_{\rm v} / \dot{m}_{\rm total}\right) + \left(h_{\rm vsat} - \Delta h_{\rm Lv} - \overline{Cp_{\rm L}\Delta T_{\rm SC}}\right) \left(\dot{m}_{\rm L} / \dot{m}_{\rm total}\right) = \left(h_{\rm vsat} + \Delta \overline{h}_{\rm SH}\right) \left(\dot{m}_{\rm v} / \dot{m}_{\rm total}\right) + \left(h_{\rm vsat} - \Delta h_{\rm Lv} - \Delta \overline{h}_{\rm SC}\right) \left(\dot{m}_{\rm L} / \dot{m}_{\rm total}\right)$$
(5)

When the superheated vapor starts to condense at the tube wall, the condensate (liquid) starts flowing, and the resulting vapor quality based on the actual vapor and liquid mass flow rates is defined as

$$x_{\rm a} = \dot{m}_{\rm V} / \dot{m}_{\rm total}, \quad (1 - x_{\rm a}) = \dot{m}_{\rm L} / \dot{m}_{\rm total} \tag{6}$$

The actual vapor quality x_a is not equal to the thermodynamic vapor quality x_b , which is based on the equilibrium state under diabatic conditions,

$$x_{\rm a} \neq x_{\rm b} \approx f_{equilibrium}\left(h_{\rm b}, P\right) \tag{7}$$

2.2 Heat balance of condensing superheat zone

Figure 1 (b) illustrates the temperature profile and heat flow in superheat zone condensation. According to Solimanøs flow regime [17], condensation begins as mist flow and then changes into annular flow. According to Altman [9], thin ridges or droplets flow on the interior tube surface. Figure 1 illustrates heat exchange simplified using the annular flow model. According to continuity, the total mass flow rate \dot{m}_{total} of the vapor and liquid refrigerant is

$$\dot{m}_{\text{total}} = \dot{m}_{\text{V, i}} + \dot{m}_{\text{L, i}} = \dot{m}_{\text{V, o}} + \dot{m}_{\text{L, o}} \tag{8}$$

The amount of condensate $\Delta \dot{m}_{\rm L}$ generated through a segment is expressed using continuity as

$$\Delta \dot{m}_{\rm L} = \dot{m}_{\rm L,o} - \dot{m}_{\rm L,i} = \dot{m}_{\rm V,i} - \dot{m}_{\rm V,o} = \Delta \dot{m}_{\rm V} \tag{9}$$

The total inlet enthalpy at the entrance of a segment is

Likewise, the total outlet enthalpy at the exit of a segment is

$$\begin{split} h_{\mathrm{b,o}} \dot{m}_{\mathrm{total}} &= \overline{h}_{\mathrm{V,o}} \dot{m}_{\mathrm{V,o}} + \overline{h}_{\mathrm{L,o}} \dot{m}_{\mathrm{L,o}} \\ &= \overline{h}_{\mathrm{V,o}} \dot{m}_{\mathrm{V,o}} + \overline{h}_{\mathrm{L,o}} \left(\dot{m}_{\mathrm{L,i}} + \Delta \dot{m}_{\mathrm{L}} \right) \\ &= \left(h_{\mathrm{Vsat}} + \Delta \overline{h}_{\mathrm{SH,o}} \right) \dot{m}_{\mathrm{V,o}} + \left(h_{\mathrm{Vsat}} - \Delta h_{\mathrm{LV}} - \Delta \overline{h}_{\mathrm{SC,o}} \right) \left(\dot{m}_{\mathrm{L,i}} + \Delta \dot{m}_{\mathrm{L}} \right) \end{split}$$
(11)

Subtracting Eq.(11) from Eq.(10) yields the enthalpy change through a segment, which can be written as

$$\begin{pmatrix} h_{\mathrm{b,\,i}} - h_{\mathrm{b,\,o}} \end{pmatrix} \dot{m}_{\mathrm{total}} = \left(\varDelta \bar{h}_{\mathrm{SH,i}} - \varDelta \bar{h}_{\mathrm{SH,o}} \right) \dot{m}_{\mathrm{V,o}} + \left(\varDelta \bar{h}_{\mathrm{SC,o}} - \varDelta \bar{h}_{\mathrm{SC,i}} \right) \dot{m}_{\mathrm{L,i}} + \left(\varDelta \bar{h}_{\mathrm{SH,i}} + \varDelta h_{\mathrm{LV}} + \varDelta \bar{h}_{\mathrm{SC,o}} \right) \varDelta \dot{m}_{\mathrm{L}}$$

$$= \left(\varDelta \bar{h}_{\mathrm{SH,i}} - \varDelta \bar{h}_{\mathrm{SH,o}} \right) \dot{m}_{\mathrm{V,o}} + \varDelta \bar{h}_{\mathrm{SH,i}} \left(\dot{m}_{\mathrm{V,i}} - \dot{m}_{\mathrm{V,o}} \right) + \varDelta h_{\mathrm{LV}} \varDelta \dot{m}_{\mathrm{L}}$$

$$+ \left(\varDelta \bar{h}_{\mathrm{SC,o}} - \varDelta \bar{h}_{\mathrm{SC,i}} \right) \dot{m}_{\mathrm{L,i}} + \varDelta \bar{h}_{\mathrm{SC,o}} \left(\dot{m}_{\mathrm{L,o}} - \dot{m}_{\mathrm{L,i}} \right) (12)$$

$$= \underbrace{\left(\varDelta \bar{h}_{\mathrm{SH,i}} \dot{m}_{\mathrm{V,i}} - \varDelta \bar{h}_{\mathrm{SH,o}} \dot{m}_{\mathrm{V,o}} \right)}_{\mathrm{SH}} + \underbrace{\varDelta h_{\mathrm{LV}} \varDelta \dot{m}_{\mathrm{L}}}_{\mathrm{Iatent}} + \underbrace{\left(\varDelta \bar{h}_{\mathrm{SC,o}} \dot{m}_{\mathrm{L,o}} - \varDelta \bar{h}_{\mathrm{SC,i}} \dot{m}_{\mathrm{L,i}} \right)}_{\mathrm{SC}}$$

On the right side of Eq. (12), the first, second, and third terms indicate the heat transfer rate caused by de-superheating of the vapor flow, latent heat rejection to the generate condensate, and subcooling of the condensate, respectively. Thus, Eq. (12) can be summarized as the following heat balance equation

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{SH}} + \dot{Q}_{\text{latent}} + \dot{Q}_{\text{SC}}$$
(13)

2.3 Driving temperature difference in condensing superheat zone ($T_{wi} < T_{sat} < T_{rb}$)

A bulk mean temperature, i.e., mixing-cup temperature, flow through a cross section is defined the same way as in Eq. (1),

$$T_{\rm rb} = \frac{\int (T_{\rm r} \rho u) ds}{\int (\rho u) ds}$$
(14)

This bulk mean temperature is calculated from the bulk mean enthalpy and pressure with an equilibrium correlation. In this study, Refprop. Ver. 8.0 [18] was used to calculate the correlation of equilibrium state:

$$T_{\rm rb} \approx f_{equilibrium} \left(h_{\rm b}, P \right) \tag{15}$$

In the superheat zone, the bulk enthalpy, h_b , exceeds the saturated vapor enthalpy, h_{Vsat} . The bulk mean temperature, obtained from the bulk enthalpy, also exceeds the saturation temperature, and the average superheat in vapor flow $\Delta \overline{T}_{SH}$ should exceed zero. From the point when the tube wall reaches the saturation temperature, superheated vapor begins to condense, and a small amount of condensate (liquid) is generated. Then, the liquid mass flow rate \dot{m}_L exceeds zero, and the actual vapor quality x_a , decreases from 1. The flowing liquid has some

degree of subcool ΔT_{SC} . Hence, the heat balance of the condensing superheat zone is expressed as follows,

$$\begin{cases} \Delta \overline{T}_{SH} > 0, \quad T_{rb} > T_{sat}, \text{ and } h_b > h_{Vsat} \\ \Delta \overline{T}_{SC} > 0, \quad \dot{m}_L > 0, \quad \text{and } x_a < 1.0 \\ h_b = h_{Vsat} + \overline{Cp_V \Delta T_{SH}} x_a - \left(\Delta h_{LV} + \overline{Cp_L \Delta T_{SC}}\right) (1 - x_a) > h_{Vsat} \\ \therefore \underbrace{\overline{Cp_V \Delta T_{SH}} x_a}_{SH} > \underbrace{\left(\Delta h_{LV} + \overline{Cp_L \Delta T_{SC}}\right) (1 - x_a)}_{\text{latent} + SC} > 0 \end{cases}$$
(16)

This heat balance explains that vapor and liquid flow coexist in this zone and the superheated vapor flow carries greater heat than is rejected by the heat condensate. Meanwhile, heat fluxes are obtained when both sides of Eq. (13) are divided by the same corresponding heat transfer area of the interior tube wall.

$$q_{\text{total}} = q_{\text{SH}} + (q_{\text{latent}} + q_{\text{SC}}) \tag{17}$$

The driving temperature difference for the heat flux caused by de-superheating q_{SH} is probably the difference between the bulk and saturation temperatures on the liquid film surface, $(T_{rb} - T_{sat})$. The remaining summation of heat fluxes $(q_{latent} + q_{SC})$ is manipulated as a heat flux by saturated condensation. Because condensation requires a degree of subcool of the cooling surface, the HTC of saturated condensation α_{TP} always includes both heat fluxes. This driving temperature difference is normally calculated as the difference between the saturation and wall temperatures $(T_{sat} - T_{wi})$. Thus, Eq. (17) can be converted using these HTCs and the driving temperature differences to the following,

$$\alpha (T_{\rm rb} - T_{\rm wi}) = \alpha_{\rm SH} (T_{\rm rb} - T_{\rm sat}) + \alpha_{\rm TP} (T_{\rm sat} - T_{\rm wi})$$
⁽¹⁸⁾

$$\alpha = \frac{q_{\text{total}}}{T_{\text{rb}} - T_{\text{wi}}} \quad \text{at} \quad T_{\text{wi}} < T_{\text{sat}} \quad \text{and} \quad h_{\text{b}} > h_{\text{Vsat}}$$
(19)

2.4 Heat balance and driving temperature difference in the single-phase superheat zone ($T_{sat} \leq T_{wi} < T_{rb}$)

Figure 1 (c) illustrates temperature distribution at the very beginning of condensation as superheated vapor

flows into a cooling segment. The condensation in the horizontal cooling tubes starts where the tube wall temperature T_{wi} reaches saturation point T_{sat} . Up to this point, the heat rejection process is single-phase cooling because the wall temperature is above saturation, and the mass transfer rate of the condensate is equal to zero. In the following heat balance equation, only heat that is carried by the superheated vapor flow remains

$$\begin{cases} \Delta \overline{T}_{SH} > 0, \quad \dot{m}_{V} = \dot{m}_{total}, \quad T_{rb} > T_{sat}, \text{ and } h_{b} > h_{Vsat} \\ \Delta \overline{T}_{SC} = 0, \quad \dot{m}_{L} = 0 \\ \therefore \overline{Cp_{V}\Delta T_{SH}} > \left(\Delta h_{LV} + \overline{Cp_{L}\Delta T_{SC}}\right) \left(\dot{m}_{L} / \dot{m}_{total}\right) = 0 \end{cases}$$

$$(20)$$

Consequently, the heat fluxes caused by latent heat rejection and subcooling are equal to zero, and the total heat flux is caused only by de-superheating,

$$q_{\text{latent}} = q_{\text{SC}} = 0$$

$$\therefore q_{\text{total}} = q_{\text{SH}}$$
(21)

Hence, the driving temperature difference can be expressed as $(T_{rb} - T_{wi})$.

$$\alpha \left(T_{\rm rb} - T_{\rm wi} \right) = \alpha_{\rm SH} \left(T_{\rm rb} - T_{\rm wi} \right) \tag{22}$$

This condition is the most commonly used driving temperature for single-phase cooling heat transfer in tubes. The HTC under this condition, $T_{sat} < T_{wi}$, is simply predicted with the following correlations proposed for single-phase heat rejection,

$$\alpha = \frac{q_{\text{total}}}{T_{\text{rb}} - T_{\text{wi}}} \quad \text{at} \quad T_{\text{sat}} < T_{\text{wi}} \quad \text{and} \quad h_{\text{b}} > h_{\text{vsat}}$$
(23)

2.5 Heat balance and driving temperature difference in the two-phase zone ($T_{wi} < T_{sat} \notin T_{rb}$)

Figure 1 (a) illustrates the temperature distribution during condensation at the beginning of the two-phase zone. When the bulk enthalpy h_b is equal to the enthalpy of saturated vapor $h_{V_{sat}}$, the thermodynamic vapor quality x_b is 1, but the actual vapor quality x_a is below 1. Because condensation starts ahead, the heat balance at the

beginning of two-phase zone ($x_b = 1$) is expressed as

$$\begin{cases} T_{\rm rb} \approx T_{\rm sat}, \ h_{\rm b} = h_{\rm Vsat}, \ \text{and} \quad x_{\rm b} = 1\\ \Delta \overline{T}_{\rm SC} > 0, \ \dot{m}_{\rm L} > 0, \ \text{and} \quad x_{\rm a} < 1\\ \overline{Cp_{\rm V} \Delta T_{\rm SH}} x_{a} = \left(\Delta h_{\rm LV} + \overline{Cp_{\rm L} \Delta T_{\rm SC}}\right) (1 - x_{a}) > 0 \end{cases}$$
(24)

At the point where the bulk enthalpy determines the thermodynamic vapor quality $x_b = 1$, the enthalpy in the condensate, $(\Delta h_{LV} + \overline{Cp_L \Delta T_{SC}})(1-x_a)$, is equal to the enthalpy in the superheated vapor, $\overline{Cp_V \Delta T_{SH}} \cdot x_a$. Here, bulk mean refrigerant temperature T_{rb} can be slightly higher than saturation point T_{sat} . The exact bulk temperature under diabatic conditions is not equal to that in the equilibrium state and must be obtained from temperature and velocity profiles. However, most of the condensation HTC correlations in the two-phase zone use the saturation temperature as the refrigerant temperature. To compare the correlations in this study, it is assumed that $T_{rb} = T_{sat}$ when using Eq. (15). An example of the calculation results of the bulk mean temperature T_{rb} , for a thermodynamic vapor quality $x_b = 1$, is given in the Appendix.

In the two-phase zone, the thermodynamic vapor quality x_b varies from 1 to 0, and the heat balance is expressed as follows

$$\begin{cases} T_{\rm rb} \approx T_{\rm sat}, \ h_{\rm Lsat} \leq h_{\rm b} \leq h_{\rm Vsat}, \ \text{and} \ 0 \leq x_{\rm b} \leq 1 \\ \Delta \overline{T}_{\rm SC} > 0, \ \dot{m}_{\rm L} > 0, \ \text{and} \ 0 \leq x_{\rm a} < 1 \\ 0 < \left(\Delta h_{\rm LV} + \overline{Cp_{\rm L}} \Delta T_{\rm SC} \right) (1 - x_{\rm a}) - \overline{Cp_{\rm V}} \Delta T_{\rm SH} x_{\rm a} < \Delta h_{\rm LV} \end{cases}$$

$$(25)$$

Using Eq. (15), the driving temperature $(T_{rb} - T_{wi})$ eventually becomes the subcool degree of the tube wall temperature $(T_{sat} - T_{wi})$ at $x_b=1$, which is the most commonly used as driving temperature difference for saturated condensation. Thus, the definition of HTC also becomes

$$\alpha = \frac{q_{\text{total}}}{T_{\text{sat}} - T_{\text{wi}}}, \text{ at } T_{\text{wi}} < T_{\text{sat}} \approx T_{\text{rb}} \text{ and } h_{\text{Lsat}} < h_{\text{b}} < h_{\text{Vsat}}$$
(26)

This definition should be valid for the entire two-phase zone, where the bulk enthalpy h_b is between the saturated vapor h_{Vsat} and saturated liquid h_{Lsat} and the bulk temperature T_b is equal to T_{sat} , as shown in Eq. (15).

3. Experimental setup and method

3.1 Experimental setup

Figure 2 shows a schematic diagram of the experimental apparatus. The CO_2 and R410A loop consists of a variable speed gear pump, a Coriolis-type mass flow meter, an electric pre-heater, a mixer, a pre-cooler, a test section, two after-coolers, and a receiver tank. The mixer is placed at the entrance of the pre-cooler and used to measure the pressure and bulk-mean temperature of the superheated vapor. The system pressure is roughly adjusted by varying the amount of refrigerant charge and precisely adjusted by changing the inlet temperature and flow rate of the cooling water flowing through the after-coolers.

3.2 Test section and test tube

Figures 3 (a) and (b) show a structure of the test section and dimensions of the test tube. The test tube, which is a smooth copper tube with an ID of 6.1 mm and OD of 9.53 mm, is placed horizontally and covered with a thick brass jacket. The halved brass jacket is pressed over the test tube, and the small gap between them is filled with a thermal paste. On the outside of the brass jacket, copper tubes are attached with solder, which allows cooling water to flow through them. This structure yields cooling conditions with an almost uniform temperature. Twelve thermocouples are embedded into the top, bottom, right, and left of the test tube wall at three positions in the axial direction. The active cooling length due to the brass jacket is 150 mm, which is relatively short for measuring the quasi-local HTC in the axial direction.

3.3 Experimental Procedure

When measuring heat transfer in the superheat zone, the refrigerant superheat at the test section inlet is controlled from 5 to 40 K by the pre-heater; meanwhile, the water flow of the pre-cooler is shut down. During measurements in the two-phase and superheat zones below 5 K of superheat, the superheated fluid flow through the mixer is maintained at approximately 5 K to determine the bulk enthalpy. The inlet conditions of the test section are controlled by the flow rate and inlet temperature of the cooling water flowing through the pre-cooler.

3.4 Data Reduction Method

Figure 3 (c) presents the data reduction method. The measured values are the refrigerant mass flow rate, $\dot{m}_{\rm r}$; bulk-mean temperature, $T_{\rm rb,MC}$; absolute pressure in the mixer, $P_{\rm MC}$; bulk water temperature of the pre-cooler inlet, $T_{\rm H2O,PCi}$, and outlet $T_{\rm H2O,PCo}$; test section inlet, $T_{\rm H2O,TSi}$ and outlet, $T_{\rm H2O,TSo}$; and the water mass flow rate of pre-cooler, $\dot{m}_{\rm H2O,PC}$ and test section, $\dot{m}_{\rm H2O,TS}$. The bulk-mean enthalpy in the mixer $h_{\rm rb,MC}$ can be calculated using $T_{\rm rb,MC}$ and $P_{\rm MC}$ under assumption of equilibrium by Refprop Ver. 8.0 [18]. The enthalpy changes through the pre-cooler $\Delta h_{\rm PC}$ and the test section $\Delta h_{\rm TS}$ are determined by the water side heat balances and as below.

$$\Delta h_{\rm PC} = \dot{Q}_{\rm PC} / \dot{m}_{\rm r} = \left(\dot{Q}_{\rm H2O, PC} - \dot{Q}_{\rm gain, PC} \right) / \dot{m}_{\rm r} = \left[\left(T_{\rm H2O, PCo} - T_{\rm H2O, PCi} \right) \dot{m}_{\rm H2O, PC} C p_{\rm H2O} - \dot{Q}_{\rm gain, PC} \right] / \dot{m}_{\rm r}$$
(27)

$$\Delta h_{\rm TS} = \dot{Q}_{\rm TS} / \dot{m}_{\rm r} = \left(\dot{Q}_{\rm H2O,TS} - \dot{Q}_{\rm gain,TS} \right) / \dot{m}_{\rm r} = \left[\left(T_{\rm H2O,TSo} - T_{\rm H2O,TSi} \right) \dot{m}_{\rm H2O,TS} C p_{\rm H2O} - \dot{Q}_{\rm gain,TS} \right] / \dot{m}_{\rm r}$$
(28)

Here, $\dot{Q}_{gain,PC}$ and $\dot{Q}_{gain,TS}$ represent the heat leakage from the ambient air through the insulators, which are preliminarily measured by water side heat balances as the refrigerant loop is vacuumed and correlated with the temperature difference between the ambient T_{amb} and water T_{H2O} . While the temperature difference $(T_{H2O,PC} - T_{amb})$ changes from -14 to 42 K, the heat gain $\dot{Q}_{gain,PC}$ changes from 5 ±0.84 to -16 ±0.84 W in the pre-cooler. Likewise, $\dot{Q}_{gain,TS}$ changes from 0.9 ±0.17 to -1.9 ±0.17 W in the test section while the temperature difference $(T_{H2O,TS} - T_{amb})$ changes from -14 to 30 K. When the ambient temperature T_{amb} is 20 °C and water temperature $T_{\text{H2O,TS}}$ is 10 °C, $\dot{Q}_{\text{gain,TS}}$ is 2% of the total heat transfer rate for a heat flux q_{wi} of 10 kWm⁻². For a heat flux q_{wi} of 3 kWm⁻², the $\dot{Q}_{\text{gain,TS}}$ is approximately 7% of the total heat transfer rate.

The bulk mean temperature at the test section $T_{\rm rb}$ is obtained from the bulk enthalpy and pressure with the equilibrium state function in Refprop Ver. 8.0 [18].

$$T_{rb} = (T_{rb,i} + T_{rb,o})/2$$

$$T_{rb,i} = f_{equiblium,Refprop8}(h_{b,i}, P_{TS,i}), \quad T_{rb,o} = f_{equiblium,Refprop8}(h_{b,o}, P_{TSo})$$

$$\begin{cases}
h_{b,i} = h_{b,MC} - \Delta h_{PC} \\
P_{TS,i} = P_{MC} - \Delta P_{PC} \\
h_{b,o} = h_{b,MC} - \Delta h_{PC} - \Delta h_{TS} \\
P_{TSo} = P_{MC} - \Delta P_{PC} - \Delta P_{TS}
\end{cases}$$
(29)

The average heat flux of the test section at the interior tube wall q_{wi} can be expressed as

$$q_{\rm wi} = \left(\dot{Q}_{\rm H20,TS} - \dot{Q}_{\rm gain,TS} - \dot{Q}_{\rm cond}\right) / \left(d_{\rm i} \cdot \pi \cdot \Delta Z_{\alpha}\right) \tag{30}$$

where \dot{Q}_{cond} is the conduction heat from outside the cooling brass jacket, which is estimated numerically for each condition. The numerical analysis procedure and typical results for \dot{Q}_{cond} are specified in Ref. [16].

The definition of the average heat transfer coefficient α can be written as

$$\alpha = \frac{q_{\rm wi}}{\Delta T}, \qquad \Delta T = T_{\rm rb} - T_{\rm wi} \tag{31}$$

where T_{wi} is the average temperature of the 12 points on the tube wall. The reference refrigerant temperature is defined as the arithmetic mean of the inlet and outlet bulk temperatures, $T_{rb,i}$ and $T_{rb,o}$, which are calculated with pressures, P_{TSi} and P_{TSo} , and enthalpies, $h_{b,i}$ and $h_{b,o}$. Using this method, the driving temperature difference ΔT in the superheat zone is defined as the difference between the tube wall and bulk refrigerant temperature. This continuously changes to the difference between the tube wall and saturation temperatures, at a thermodynamic vapor quality of 1, for the two-phase zone.

Table 1 lists the measurement uncertainties obtained from the results of two calibration standard deviations,

resolution of data loggers and calibration tools, and stability of the excitation voltages. The combined measured uncertainties are calculated from the uncertainties in conformity described in Refs. [19] and [20]. Table 2 lists the test conditions used in this paper. Table 3 compares the properties of the tested refrigerants, CO₂ and R410A, at the typical test reduced pressure, $P/P_{crit} = 0.81$.

4. Results and discussion

4.1 Comparison between experimental results and correlations

Figure 4 shows the experimental results of R410A condensation flow in a 6.1 mm ID horizontal tube at 2.7 MPa and 200 kg m⁻²s⁻¹. The reduced pressure P / P_{crit} is 0.55 and saturation temperature, T_{sat} , is approximately 44.5 °C, which is the typical condition for air-cooled condensers in air conditioners during the summer.

The horizontal axes show the bulk enthalpy h_b and top axes in the upper graphs show the thermodynamic vapor quality x_b . The upper graphs show the bulk mean refrigerant temperature in the test section inlet $T_{rb,i}$, and outlet $T_{rb,o}$, and the averaged tube wall temperatures T_{wi} . The center graphs show the averaged heat flux q_{wi} and driving temperature difference ΔT in Eq. (31). The bottom graphs show the average HTC of the test section α . The vertical bars attached to symbols show the respective uncertainties and the horizontal bars show the enthalpy changes through the test section. The blue solid lines in the middle graphs show the value of the heat flux q_{wi} used in the calculations. The solid lines of T_{wi} , ΔT , and α in Fig. 4, except for the q_{wi} lines, show the calculation results from the following selected correlations.

The experimental data from the superheat zone ($h_b > h_{Vsat}$) are compared with that from the Gnielinskiøs correlation [21] with the Petukovøs correction factor F_a [22], which is valid for single-phase turbulent flow in smooth round tubes.

$$f_{b} = \left[1.82 \cdot \log_{10} \left(G_{r}d_{i}/\mu_{b}\right) - 1.64\right]^{-2} \\ Nu_{0} = \left[\left(f_{b}/8\right)\left(G_{r}d_{i}/\mu_{b} - 1000\right)Pr_{b}\right] / \left[1 + 12.7\left(f_{b}/8\right)^{1/2}\left(Pr_{b}^{-2/3} - 1\right)\right] \right\}$$

$$\alpha_{SH} = Nu_{0}F_{a}\left(\lambda_{b}/d_{i}\right), \quad F_{a} = \left[\left(T_{wi} + 273.15\right)/\left(T_{rb} + 273.15\right)\right]^{-0.36} \right]$$
(32)

In the two-phase zone ($h_{Lsat} > h_b > h_{Vsat}$ and $0 < x_b < 1$), a modification of the correlations presented in [23], based on the Cavalliniøs correlation [24], is used. This correlation was developed for near critical conditions with the liquid properties evaluated at the film temperature. This modification method is along the way proposed by Fujii et al. for free convective condensation in the subcritical region [25][26].

Figure 5 shows the CO₂ liquid properties change from the liquid surface to tube wall at pressures of 5, 6, 7 and 7.37 MPa. These measurements are obtained by assuming a linear distribution of liquid temperature from the saturation temperature at liquid surface to the degree of subcool 5 K at the tube wall. The Cavallini¢s correlation [24], which is valid for saturated condensation in horizontal tubes at lower reduced pressures, evaluates liquid properties at the saturation temperature, as shown in black circles in Fig. 5. Fujii et al. verified the Nusselt¢s correlation [27] at reduced temperatures T / T_{crit} from 0.994 to 0.9998 in steam and from 0.980 to 0.997 in CO₂ and found that the theory overestimates condensation HTC on the vertical cooling plate in the subcritical region. To extend the Nusselt¢s correlation to the subcritical region, they suggested using liquid properties evaluated at the film temperature. The white circles in Fig. 5 show the properties evaluated at the film temperature, $T_f = (T_{sat} + T_{wi}) / 2$. In the specific heat Cp_{L} , the white circles show the average integral value of liquid specific heat, which can be written as

$$\overline{Cp}_{\rm L} = \left(h_{\rm Lsat} - h_{\rm L,wi}\right) / \left(T_{\rm sat} - T_{\rm wi}\right)$$
(33)

This averaging method was proposed by Baskov et al. [28] for calculating the cooling HTC at super-critical conditions where fluid properties drastically distribute in the radial direction. As shown with the white circles in Fig. 5, the newly evaluated values more accurately reflect the values evaluated at saturation temperatures for

pressures above 6 MPa. This modification extends the validity of the Cavalliniøs correlation at reduced pressures up to 0.975 [23].

$$J_{G} = x_{b}G_{r}/\sqrt{gd_{i}\rho_{Vsat}(\rho_{Lsat} - \rho_{Vsat})}, \quad J_{G}^{T} = \left\{ \left(7.5/4.3X_{u}^{1.111} + 1\right)^{-3} + 2.6^{-3} \right\}^{-1/3}$$

$$X_{u} = \left[\left(1 - x_{b}\right)/x_{b} \right]^{0.9} \left(\rho_{Vsat}/\rho_{Lsat}\right)^{0.5} \left(\mu_{Lsat}/\mu_{Vsat}\right)^{0.1}, \quad Pr_{L_{f}} = \left(\overline{Cp}_{L}\mu_{L_{f}}\right)/\lambda_{L_{f}}$$

$$\alpha_{LO_{f}} = 0.023 \left(G_{r}d_{i}/\mu_{L_{f}}\right)^{0.8} Pr_{L_{f}}^{0.4} \left(\lambda_{Lsat}/d_{i}\right)$$

$$\alpha_{strat} = 0.725 \left\{ 1 + 0.741 \left(\frac{1 - x_{b}}{x_{b}}\right)^{0.321} \right\}^{-1} \left[\frac{\lambda_{L_{f}}^{3}\rho_{L_{f}}\left(\rho_{L_{f}} - \rho_{Vsat}\right)g\Delta h_{LV}}{\mu_{L_{f}}d_{i}(T_{sat} - T_{wi})} \right]^{0.25} + \left(1 - x_{b}^{0.087}\right)a_{LO_{f}} \left\{ 34 \right\}$$

$$\alpha_{Cavallini_{f}} = \left\{ J_{G} > J_{G}^{T} : \alpha_{A} = \alpha_{LO_{f}} \left[1 + 1.128x_{b}^{0.8170} \left(\frac{\rho_{Lsat}}{\rho_{Vsat}}\right)^{0.3685} \left(\frac{\mu_{Lsat}}{\mu_{Vsat}}\right)^{0.2363} \left(1 - \frac{\mu_{Vsat}}{\mu_{Lsat}}\right)^{2.144} Pr_{L_{f}}^{-0.1} \right] \right\}$$

$$(34)$$

As shown in Fig. 4, shows a high degree of overlap between the symbols, which correspond to the experimental results, and solid lines, which correspond to Eq. (32), when the refrigerant is sufficiently superheated. When the tube wall temperature is lower than the saturation point, the discrepancy is not negligible. In the two-phase zone, there is satisfactory agreement between the experimental results and modified Cavalliniøs correlation (Eq. (34)), as shown with the solid lines and symbols.

Figure 6 compares 39 points of experimentally obtained HTCs in the single-phase superheat zone, specifically where the tube wall temperature T_{wi} is at least 0.8 K above the saturation temperature T_{wi} , to the Gnielinskiøs correlation (Eq. (32)) in Fig. 5 (a). The other 432 points in the two-phase zone, where the thermodynamic vapor quality x_b varies from 0.02 to 0.98, are compared to the modified Cavalliniøs correlation (Eq. (34)) in Fig. 5 (b). In these figures, the values of the following deviation are noted. The mean relative deviation quantifies the average bias error between the experimental HTC and calculated HTC as

$$\overline{\varepsilon} = \frac{1}{n} \sum_{j=1}^{n} \varepsilon_j = \sum_{j=1}^{n} \left(\frac{\alpha_{\exp} - \alpha_{cal}}{\alpha_{cal}} \times 100 \right) \quad [\%]$$
(35)

while the standard deviation quantifies the dispersion of the relative deviation as

$$\sigma = \sqrt{\frac{1}{n-1} \sum_{i=1}^{n} \left(\varepsilon_i - \overline{\varepsilon}\right)^2} \quad [\%]$$
(36)

At the single-phase superheat zone, the mean relative deviation is +9% and standard deviation is 21%. The bias error of the experimental results is higher than the correlation, but acceptable for determining the start of condensation. For the two-phase zone, the mean relative deviation is -2.1% and standard deviation is 18%. Here, the deviations are lower. In particular, the experimental HTC of the lower heat flux is lower, as shown with the white symbols in Fig. 6 (b). As shown in Fig. 4 (a), the combined uncertainty of HTC grows significantly with small temperature differences ΔT , at lower heat fluxes. When the data at heat flux q_{wi} below 8 kW m⁻² are excluded, as shown with the gray symbols in Fig 5 (b), the mean relative deviation is +5% and experimental HTC is slightly higher than the correlation.

4.2 Identification of the condensing superheat zone

The horizontal green dashed lines in the top row of graphs in Fig. 4 show a saturation temperature T_{sat} of approximately 44.5°C. As shown with the crossing point of the horizontal and vertical green dashed lines on the right side, the starting point of the discrepancy between the experimental HTC and Gnielinskiøs correlation (Eq. (32)) occurs when the tube wall temperature, T_{wi} reaches the saturation point, T_{sat} . All of the experimental results confirmed that the starting point of the discrepancy is always determined by the threshold of the tube wall temperature. Furthermore, the experimental HTC merges into the modified Cavalliniøs correlation (Eq. (34)) at a thermodynamic vapor quality of 1, which identifies condensation at the superheat zone and undertakes the criterion of the beginning of condensation assumed by Bell [10]. The discrepancy in the HTC, compared to the Gnielinskiøs correlation, shows the effect of latent heat rejection and subcooling.

4.3 Categorization of the cooling process in condensers

The vertical green dashed lines in Fig. 4 divide the heat rejection process into superheat and two-phase zone (III) categories according to the bulk temperature. The superheat zone is further subdivided into the single-phase superheat zone (I) and condensing superheat zone (II) according to the tube wall temperature. In the single-phase superheat zone (I), where the tube wall temperature is above the saturation point, the superheated vapor releases only sensible heat, which corresponds to the exact de-superheating zone. In the condensing superheat zone (II), where the tube wall temperature is below the saturation point, condensation occurs in the superheated vapor, and the HTC increases by latent heat rejection and a small amount of subcooling.

4.4 Effect of heat flux at the beginning of condensation

Figures 4 (a), (b), (c) show the results of heat flux values q_{wi} of 3, 10, and 20 kW m⁻², respectively. The exact values of the average heat fluxes at each data point are shown with blue triangles in the middle graphs. The start of condensation at 3, 10, and 20 kW m⁻² correspond to bulk enthalpies values h_b of 435, 453, 469 kJ kg⁻¹. Likewise, condensation starts at higher bulk enthalpies as the heat flux increases. With increasing heat flux, the tube wall temperature T_{wi} tends to decrease. Thus, the tube wall temperature T_{wi} reaches the saturation point earlier and condensation starts at a higher bulk enthalpy or superheat.

4.5 HTC comparison between CO₂ and R410A

Figure 7 shows the HTC of CO₂ on the left and R410A on the right at 100 kg m⁻²s⁻¹ and 10 kW m⁻², and reduced pressures P / P_{crit} of 0.68, 0.81 and 0.95. The saturation temperature T_{sat} and latent heat Δh_{LV} are shown

with the corresponding conditions in each graph. The vertical green dashed line separates the single-phase superheat, condensing superheat, two-phase, and subcool zones from right to left. The red symbols show the experimental HTC, and the blue solid lines represent the HTC calculated with Eqs. (32) and (34) for the superheat and two-phase zones. Because Eq. (32) is invalid in the condensing superheat zone, the calculated value is shown with dashed blue lines. In the subcool zone, where the bulk refrigerant temperature is below the saturation point, the HTC is calculated with the Gnielinskiøs correlation [21] with the Sieder and Tateøs correction factor F_a [29] as below

$$\alpha_{\rm SC} = N u_0 F_{\rm a} \left(\lambda_{\rm b} / d_{\rm i} \right), \qquad F_{\rm a} = \left(\mu_{\rm b} / \mu_{\rm wi} \right)^{0.14} \tag{37}$$

In the single-phase superheat zone, the HTC of CO₂ is barely higher than that of R410A at the same reduced pressure. In the two-phase zone, the HTC of CO₂ exceeds that of R410A. The HTC is almost always at its maximum at the border between the two-phase and superheat zones ($x_b = 1$). The maximum HTC of CO₂ is approximately 120 to 170 % that of R410A at reduced pressures between 0.68 and 0.95.

To analyze the differences between CO₂ and R410A, Table 4 compares effect of the fluid properties evaluated with the film temperature on the HTC for a reduced pressure of $P / P_{crit} = 0.81$ and degree of subcool of $T_{sat} - T_{wi} = 5$ K. The terms listed are described in detail in the following. Using the Nusseltøs theory of film-wise free convective condensation, the following correlation can be derived:

$$\alpha_{\rm FC} = 0.725 \left(\frac{Ga \cdot Pr_{\rm L}}{Ph}\right)^{\frac{1}{4}} \left(\frac{\lambda_{\rm L}}{d_{\rm i}}\right) = 0.725 \left[\frac{g\lambda_{\rm L}^{3}\rho_{\rm L}\left(\rho_{\rm L}-\rho_{\rm Vsat}\right)\Delta h_{\rm LV}}{\mu_{\rm L}d_{\rm i}\left(T_{\rm sat}-T_{\rm wi}\right)}\right]^{\frac{1}{4}}$$

$$= 0.725 \left[\frac{g}{d_{\rm i}\left(T_{\rm sat}-T_{\rm wi}\right)}\right]^{\frac{1}{4}} \left\{\lambda_{\rm L}^{\frac{3}{4}} \cdot \left[\rho_{\rm L}\left(\rho_{\rm L}-\rho_{\rm Vsat}\right)\right]^{\frac{1}{4}} \cdot \Delta h_{\rm LV}^{\frac{1}{4}} \cdot \mu_{\rm L}^{-\frac{1}{4}}\right\}$$

$$\therefore Ga = \frac{gd_{\rm i}^{3}\Delta\rho}{\nu^{2}} = \frac{gd_{\rm i}^{3}}{\left(\mu_{\rm L}/\rho_{\rm L}\right)^{2}} \frac{\rho_{\rm L}-\rho_{\rm V}}{\rho_{\rm L}}, \quad Pr_{\rm L} = \frac{\nu}{\alpha} = \frac{Cp_{\rm L}\mu_{\rm L}}{\lambda_{\rm L}}, \quad Ph = \frac{\Delta T_{\rm SC}Cp_{\rm L}}{\Delta h_{\rm LV}} = \frac{(T_{\rm sat}-T_{\rm wi})Cp_{\rm L}}{\Delta h_{\rm LV}}$$

$$(38)$$

This correlation expresses the condensation HTC of quiescent vapor filled in circular tube when the condensate

does not accumulate in the tube and the entire circumference acts as an active heat transfer area. The multipliers on each property in the second line of Eq. (38) show sensitivities of those properties on the free convective condensation part in total HTC. Similarly, the sensitivities of fluid properties on the liquid film forced convective part in total HTC are expressed as

$$\alpha_{\rm LO} = 0.023 R e_{\rm LO}^{0.8} P r_{\rm LO}^{0.4} \left(\frac{\lambda_{\rm L}}{d_{\rm i}}\right) = \left(\frac{G_{\rm r} d_{\rm i}}{\mu_{\rm L}}\right)^{0.8} \left(\frac{C p_{\rm L} \mu_{\rm L}}{\lambda_{\rm L}}\right)^{0.4} \left(\frac{\lambda_{\rm L}}{d_{\rm i}}\right) = 0.023 \left(\frac{G_{\rm r}}{d_{\rm i}^{0.2}}\right) \left(\mu_{\rm L}^{-0.4} \cdot C p_{\rm L}^{0.4} \cdot \lambda_{\rm L}^{0.6}\right)$$
(39)

As listed in Table 4, the free convective film-wise condensation HTC, α_{FC} , of R410A is 20% lower than that of CO₂, which is mainly a result of the lower thermal conductivity and latent heat. The forced convective liquid HTC α_{LO} of R410A is 21% lower than that of CO₂, which is mainly a result of the lower specific heat and thermal conductivity. Although the actual heat transfer phenomenon is the combination of forced convection of liquid and vapor flow and phase-change on the complex interface between them, it is speculated that the lower R410A thermal conductivity increases the heat resistance of the liquid film. In addition, the smaller R410A latent heat and specific heat decreases the effect of phase-change and forced convective heat transfer.

Although the experimental and calculated HTC generally agree, the experimental HTC of R410A slightly deviates from the calculated values when approaching the critical point and is approximately 25% below the correlation at a reduced pressure of 0.95. The critical pressure of R125, which is one of component in R410A, is 3.61 MPa. Possibly, supercritical R125 most likely behaves as an incondensable gas and decreases the condensation HTC more than the correlation prediction at a reduced pressure of 0.95.

4.6 Effect of reduced pressure on HTC

As shown in Fig. 7, HTC is almost unchanged as the pressure is reduced until condensation begins (in the

single phase superheat zone). The effect of decreasing the vapor velocity due to increasing vapor density seems to compensate for the effect of increasing the heat capacity. In the two-phase zone, HTC clearly decreases with increasing reduced pressure in the range presented. Jiang et al. [30] measured the HTC of R410A and R404A in horizontal tubes 6.2 and 9.4 mm ID at reduced pressures of 0.8 and 0.9 and mass fluxes from 200 to 800 kg m⁻²s⁻¹. They concluded that the HTC is almost unchanged at reduced pressures as a result of compensating properties. The results of this study are not in agreement with their conclusion in the presented range of mass fluxes; however, the experimental and calculated HTC of CO₂ and R410A both show the same tendency. This can be attributed to the decreasing latent heat of CO₂ and R410A, although there are other factors, such as increasing *Pr*, decrease in liquid viscosity, which may also have affected the observations. The results could also show the same conclusions that Jiang et al. stated at the mass fluxes above 300 kg m⁻²s⁻¹.

In real applications, the enthalpy change in the condensing superheat zone increases with increasing reduced pressure, which suggests that the condensing superheat zone tends to be longer and two-phase zone tends to be shorter as the operating pressure increases.

Figure 8 shows the ratio of the heat flux only driven by superheat q_{SH} as shown in Eq. (17) to the total heat flux q_{wi} at a CO₂ HTC of 100 kg m⁻²s⁻¹ and total heat flux q_{wi} of 10 kWm⁻². In the graphs at the top, the symbols represent the ratio of heat fluxes q_{SH}/q_{wi} . The heat flux of superheat q_{SH} is obtained with the following equation and the experimentally determined bulk temperature T_{rb} , which is given by Eq. (18):

$$q_{\rm SH} = \alpha_{\rm SH} \left(T_{\rm rb} - T_{\rm sat} \right) \tag{40}$$

Although the existence of the condensate probably affects the vapor flow and enhances de-superheating, it is assumed that Gnielinskiøs correlation (Eq. (32)) is applicable for calculating the HTC between the superheated vapor and saturated condensate surface a_{SH} in the condensing superheat zone (zone III in Fig. 4). In the bottom

graph, the symbols represent the experimental HTC. Blue circles, green squares, and red triangles in both graphs show the results at 5, 6, and 7 MPa, respectively. The solid lines represent the correlation of HTC for the single-phase superheat and two-phase zones. The dashed curves are the expected curves for the experimental results, and the vertical dashed lines indicate the border between the superheat and two-phase zones.

If the heat flux from the superheated vapor q_{SH} is negligible compared to the total heat flux q_{wi} , then the heat transfer process in condensing superheat zone can simply be treated as two-phase zone condensation. The correlation for HTC in the two-phase zone condensation should generate the same value as in the condensing superheat zone for $x_b = 1$. HTC could be defined by the saturation temperature but not the bulk temperature in that zone.

The top graph in Fig. 7 shows the rate of decreasing sensible superheat heat flux $q_{\rm SH}$ in terms of the total heat flux $q_{\rm wi}$. The ratio, $q_{\rm SH} / q_{\rm wi}$, decreases almost linearly from the single-phase superheat zone to two-phase zone, but the decline becomes more moderate at higher pressures. The effect of pressure on HTC is also smaller at higher pressures, as shown in the bottom graph, which can be explained with the decreasing ratio of latent heat to sensible heat in the superheated vapor, $(\Delta h_{\rm LV}/\overline{Cp_{\rm V}\Delta T_{\rm SH}})$, as pressure increases.

The results above show that the heat transfer process in the condensing superheat zone can be modeled as simultaneous heat transfer of latent and sensible heat. According to the Nusseltøs theory, the HTC indicates that the latent and subcooling heat transfers are at a maximum due to the thin liquid film when the degree of subcool on the interior tube wall is close to zero for actual vapor quality of $x_a=1$. Meanwhile, the de-superheating heat transfer is still dominant at $x_a=1$. De-superheating requires a much larger driving temperature difference while the de-superheating HTC is much lower, which cancels out much of the higher HTC through condensation. As the de-superheating heat flux of the sensible superheat q_{SH} decrease, the portion of HTC that depends on condensation

HTC increases. Meanwhile, the condensation HTC decreases as the liquid film thickness grows. As a result of this conjugate heat transfer, the maximum point of the total HTC appears around a thermodynamic vapor quality of x_b = 1. The extremum in HTC tends to be moderate at a higher reduced pressure of 7 MPa, where the de-superheating HTC increases near the saturation point due to the drastically increasing Prandtl number and relatively lower condensation HTC due to smaller latent heat and lower liquid thermal conductivity.

5. Conclusions

The cooling and condensation heat transfer process from the superheat to two-phase zones in a horizontal smooth tube was experimentally investigated using CO_2 and R410A to determine the heat transfer coefficient. The deviation between the experimentally determined HTC and the correlation valid for single-phase cooling gradually increases from the point where the tube wall temperature reaches the saturation point, which is a result of the start of condensation in the presence of superheated vapor. With increasing the heat flux at a fixed pressure, condensation starts earlier due to the lower tube wall temperature. When approaching the critical point (increasing reduced pressures from 0.68 to 0.95), the HTC is reduced, which is mainly attributed to decreasing latent heat.

The HTC for CO_2 is approximately 20 to 70% higher than that for R410A at a thermodynamic vapor quality of 1, which is mainly due to lower thermal conductivity and larger latent heat in CO_2 . The heat flux, which is caused by superheat, linearly decreases through the condensing superheat zone. Thus, the sensible heat flux from the superheated vapor is not negligible, and the heat transfer process of the condensing superheat zone should be modeled as a simultaneous heat transfer of sensible and latent heat.

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Appendix

The sample calculation of the bulk mean temperature, T_{rb} , of CO₂ at 5 MPa, 100 kg m⁻²s⁻¹, and a vapor quality of thermodynamic equilibrium of $x_b = 1$ are shown here. The calculation using Eq. (14) requires temperature and velocity profiles. In these profiles, the flow regime is assumed to be in completely separated annular flow. The 7th power law is valid in the vapor phase, and a linear profile is used in the liquid phase. Under these assumptions, the vapor velocity in the vapor phase can be written as

$$\begin{cases} \Delta u_{\rm v}(r) = u_{\rm v}(r) - u_{\rm L,max} = (u_{\rm center} - u_{\rm L,max})(1 - r / R_{\rm v})^{\frac{1}{7}} \\ \overline{\Delta u}_{\rm v} = 0.816(u_{\rm center} - u_{\rm L,max}) = \left(\frac{G_{\rm r}x_{\rm a}}{\rho_{\rm Vat}\overline{\tau}_{\rm v}}\right) / \xi_{\rm a} \\ R_{\rm v} = (d_{\rm i}^2 \cdot \xi_{\rm a})^{0.5}, \quad \xi_{\rm a} = 0.98 \end{cases}$$

$$(41)$$

where, $\Delta u(r)$, u_{center} , and $\overline{\Delta u}_{V}$ are the vapor velocity relative to the maximum liquid velocity, $u_{\text{L,max}}$, at a radial position, r [m]; vapor velocity at the tube center, r = 0 [m]; and average relative vapor velocity, respectively. R_{v} and ξ_{a} are the radius of vapor core flow and actual void fraction (assumed to be 0.98), respectively. $\rho_{\text{Vat}\overline{T}v}$ is the density evaluated at the average vapor temperature. In the liquid phase, the velocity profile is assumed to be linear,

$$\begin{cases} u_{wi} = 0 & \text{at } r = R \\ \overline{u}_{L} = \left[\frac{G_r \left(1 - x_a \right)}{\rho_{Lat\overline{T}_L}} \right] / (1 - \xi_a) \\ u_{L,max} = 2\overline{u}_L & \text{at } r = R_V \end{cases}$$

$$\tag{42}$$

where, u_{wi} , \overline{u}_{L} , and $u_{\text{L,max}}$ are the liquid velocity on the interior tube wall, average liquid velocity, and the maximum liquid velocity at the liquid surface, respectively. $\rho_{\text{Lat}\overline{r}_{\text{L}}}$ is the density evaluated at the average liquid

temperature. Meanwhile, the vapor temperature profile in the vapor phase can be written as

$$\begin{cases} \Delta T_{\rm SH}(r) = T(r) - T_{\rm sat} = (T_{\rm center} - T_{\rm sat})(1 - r / R_{\rm V})^{\frac{1}{7}} \\ \overline{\Delta T}_{\rm SH} = 0.833(T_{\rm center} - T_{\rm sat}) = q'_{\rm SH}/\alpha_{\rm SH} \\ \overline{T}_{\rm V} = T_{\rm sat} + \overline{\Delta T}_{\rm SH} \\ \alpha_{\rm SH} = 0.023 \left(\frac{\overline{u} \cdot 2R_{\rm V}}{\nu_{\rm Vsat}}\right)^{0.8} Pr_{v_{\rm at}\overline{T}_{\rm V}}^{\frac{1}{3}} \left(\frac{\lambda_{\rm Vat}\overline{T}_{\rm V}}{2R_{\rm V}}\right) \end{cases}$$

$$\tag{43}$$

where, $T_{\rm SH}(r)$, $T_{\rm center}$, and $\overline{T}_{\rm V}$ and are the local vapor temperature at a radial position, r [m]; tube center r = 0 [m]; and average vapor temperature, respectively. $Pr_{\rm Vat}\overline{T}_{\rm V}$ and $\lambda_{\rm Vat}\overline{T}_{\rm V}$ are the vapor Prandtl number and thermal conductivity evaluated at the average vapor temperature, $\overline{T}_{\rm V}$. The ratio of heat flux caused by de-superheating, $q'_{\rm SH}$, defined using the actual bulk mean refrigerant temperature to total heat flux, $q_{\rm wi}$, is calculated with a modified Cavallini correlation. This gives total HTC, which defined from saturation temperature to tube wall temperature as

$$\frac{q_{\rm SH}'}{q_{\rm wi}} = \frac{\alpha_{\rm SH} \,\overline{\Delta T}_{\rm SH}}{\alpha_{\rm Cavallini_f}} \left(T_{\rm sat} - T_{\rm wi} \right) \tag{44}$$

This ratio of heat fluxes is assumed to be 0.1 in this sample calculation. In the liquid phase, the temperature profile is also assumed to be linear:

$$\begin{cases} T_{\text{L,wi}} = T_{\text{wi}} & \text{at } r = R \\ T_{\text{L}} = T_{\text{sat}} & \text{at } r = R_{\text{V}} \\ \overline{T}_{\text{L}} = (T_{\text{sat}} - T_{\text{wi}})/2 \\ T_{\text{wi}} = T_{\text{sat}} - (q_{\text{wi}}/\alpha_{\text{Cavallini}_{\text{f}}}) \end{cases}$$

$$(45)$$

Furthermore, the actual vapor quality, x_{a} is calculated from Eq. (24) at a thermodynamic equilibrium vapor quality of $x_{b} = 1$:

$$\begin{cases} \frac{\int (Cp_{\rm v} \Delta T_{\rm SH} \cdot \rho_{\rm v} u_{\rm v}) ds}{\int (\rho_{\rm v} u_{\rm v}) ds} \cdot x_a = \left[\Delta h_{\rm Lv} + \frac{\int (Cp_{\rm L} \Delta T_{\rm SC} \cdot \rho_{\rm L} u_{\rm L}) ds}{\int (\rho_{\rm L} u_{\rm L}) ds} \right] (1 - x_a) \\ ds = 2r \cdot dr \end{cases}$$
(46)

As a result, the actual vapor quality, x_{a} , to fulfill the aforementioned condition is 0.97 at heat fluxes of 10 kW m⁻² and

0.95 at 20 kW m⁻². In the regressive approach, the ratio of heat flux q'_{SH}/q_{wi} is calculated with this actual vapor quality. The calculated ratio of heat flux q'_{SH}/q_{wi} is 0.095 at heat fluxes of 10 kW m⁻² and 0.098 at 20 kW m⁻², and it was confirmed that these values are relatively close to the hypothesized value.

Figure 9 (a) and (b) show the calculated velocity and temperature profile of CO₂ at 5 MPa, 100 kg m⁻²s⁻¹, and a thermodynamic equilibrium vapor quality of $x_b = 1$. The average degree of superheat in the vapor phase $\overline{\Delta T}_{SH}$ is 1.8 K at a heat flux of 10 kW m⁻² and 3.9 K at 20 kW m⁻². The degree of subcool on the tube wall ΔT_{wi} is 3.3 K at a heat flux of 10 kW m⁻², and 7.5 K at 20 kW m⁻². The calculated bulk mean temperature is 1.5 K above saturation temperature at a heat flux of 10 kW m⁻², and 3.4 K at 20 kW m⁻².

Figure 10 compares the calculated and assumed bulk mean temperatures at the equilibrium state of CO₂ at 5 MPa, 100 kg m⁻²s⁻¹. The vertical dashed and dashed-dotted lines show the start of condensation: actual vapor quality x_a = 1 at heat fluxes of 10 and 20 kW m⁻². These are the points where the tube wall temperature reaches the saturation point $T_{wi} = T_{sat}$, which are found using Eq. (32). As calculated, the bulk mean temperature is 1.5 and 3.4 K above from saturation temperature with heat fluxes of 10 and 20 kW m⁻² at the thermodynamic equilibrium vapor quality of $x_b = 1$. At some point, as decreasing degree of superheat in vapor phase, the bulk mean temperature should reach saturation point.

However, it should be noted that this calculation requires some hypotheses. For further understanding of the exact bulk mean temperature, the correlation between actual void fraction and actual vapor quality, actual temperature and velocity profiles should be investigated.

Caption list

- Table 1
 Measurement uncertainties
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Measurement points			Instrument	Uncertainty
Refrigerant and water temperature		T _{rb} T _{H2O}	Sheathed T type Thermocouple	±0.05 K
Tube wall tem	perature	$T_{ m wi}$	Twisted T type Thermocouple	±0.10 K
Absolute pressure in the mixer		$P_{\rm MC}$	Diaphragm absolute pressure transducer	±0.05 MPa
Differential pressure		ΔP	Diaphragm differential pressure transducer	±0.26 kPa
Refrigerant and	Test Section	ṁ _{н20, тs} ṁ _r	Coriolis mass flow meter	$\pm 0.1 \text{ g s}^{-1}$
	Pre-cooler	$\dot{m}_{ m H2O,\ PC}$	Coriolis mass flow meter	±0.5 g s ⁻¹
Heat flux		$q_{ m wi}$	-	± 0.18 to ± 1.86 kW $m^{\text{-}2}$
НТС		α	-	±3 to ±48 %
				of measured value

Table 1 Measurement uncertainties

Table 2 Test conditions

Refrigerant		CO ₂ (R744), R410A
Reduced pressure	P/P _{crit}	0.55 to 0.998
Tube inner diameter	$d_{ m i}$	6.1 mm
Refrigerant mass flux	Gr	100 to 300 kg m ⁻² s ⁻¹
Heat flux	$q_{ m wi}$	3 to 30 kWm ⁻²
Thermodynamic equilibrium vapor quality	x _b	0.02 to 0.98
Degree of superheat		0 to 30 K

Properties	5	Unit	R744 (C0	D ₂)	R410A	L
Temperature		°C	21.98		61.69	
Pressure		MPa	6.00		3.99	
Densita	Liquid	kg m ⁻³	751.03	(100%)	795.48	(106%)
Density	Vapor	kg m ⁻³	210.88	(100%)	215.74	(102%)
Latent heat		kJ kg ⁻¹	140.47	(100%)	98.38	(70%)
Succific heat	Liquid	kJ kg ⁻¹ K ⁻¹	4.84	(100%)	3.50	(72%)
Specific fleat	Vapor	kJ kg ⁻¹ K ⁻¹	5.51	(100%)	4.53	(82%)
Thermal and the first	Liquid	$W m^{-1} K^{-1}$	0.08342	(100%)	0.07011	(84%)
Thermal conductivity	Vapor	$W m^{-1} K^{-1}$	0.03747	(100%)	0.03466	(92%)
Viccosity	Liquid	Pa•s	6.27E-05	(100%)	6.43E-05	(102%)
viscosity	Vapor	Pa•s	1.88E-05	(100%)	1.85E-05	(98%)
Daraki	Liquid	-	3.637	(100%)	3.210	(88%)
Frandu	Vapor	-	2.768	(100%)	2.423	(88%)
Surface tension		N m ⁻¹	9.33E-04	(100%)	7.51E-04	(81%)

Table 3 Comparison of properties between CO₂ and R410A at reduced pressure 0.81

	R744 (CO ₂)		R410A	
${\lambda_{ m f}}^{3/4}$	1.68E-04	(100%)	9.17E-05	(55%)
$\left[ho_{\mathrm{L_f}}(ho_{\mathrm{L_f}}$ - $ ho_{\mathrm{Vsat}}) ight]^{1/4}$	26.0	(100%)	26.8	(103%)
$\Delta h_{ m LV}{}^{1/4}$	19.4	(100%)	17.7	(91%)
$\mu_{ m L_f}$ -1/4	11.2	(100%)	11.2	(99%)
Free convective film-wise condensation HTC: α_{FC}	2.87	(100%)	2.30	(80%)
$\mu_{ m L_f}$ -0.4	46.2	(100%)	45.9	(99%)
$Cp_{ m L}{}^{0.4}$	27.0	(100%)	23.8	(88%)
$\lambda_{ m L_f}^{0.6}$	0.23	(100%)	0.21	(89%)
Forced convective HTC of liquid film: α_{LO}	0.76	(100%)	0.60	(79%)

Table 4 Analysis of the fluid property effect on HTC at $P/P_{crit} = 0.81$

Caption list

Figure 1 Illustration of heat balance in a cooling segment

(a) beginning of the two-phase zone(b) superheat zone condensation(c) beginning of condensationFigure 2. Schematic diagram of the experimental apparatus.

Figure 3. Dimensions of test section and data reduction method.

(a) structure of the test section (b) dimensions of the test tube (c) data reduction procedure

Figure 4. Experimental results of R410A at 2.7 MPa and 200 kg m⁻²s⁻¹.

(a) 3 to 8 kW m⁻² (b) 10 kW m⁻² (c) 20 kW m⁻²

Figure 5. Property change in CO₂ liquid film near the critical pressure

(--- evaluated at saturation point, --- evaluated value in modified correlation Eq. (34))

Figure 6. Comparison between experimental HTC and calculated HTC.

(a) single-phase superheat zone (b) two-phase zone

Figure 7. Comparison on HTC between R410A and CO₂ at reduced pressures 0.68, 0.81, and 0.95.

Figure 8. Effects of pressure on sensible heat flux from superheat.

Figure 9. Velocity and temperature profiles of CO₂, at 5 MPa, 100 kg m⁻²s⁻¹, and thermodynamic equilibrium vapor quality $x_b = 1$.

(a) 10 kW m^{-2} (b) 20 kW m^{-2}

Figure 10. Effect of heat flux on the bulk mean temperatures and actual vapor qualities $x_a = 1$.



Figure 1 Illustration of heat flow and temperatures in a cooling segment



Figure 2 Schematic diagram of the experimental apparatus.





(c) data reduction procedure

Figure 3. Dimensions of test section and data reduction method.



Figure 4. Experimental results of R410A at 2.7 MPa and 200 kg $m^{\text{-2}}\text{s}^{\text{-1}}.$



Figure 5. Property change in CO_2 liquid film near the critical pressure

(--- evaluated at saturation point, --- evaluated value in modified correlation Eq. (34))



Figure 6. Comparison between experimental HTC and calculated HTC.



Figure 7. Comparison on HTC between R410A and CO₂ at reduced pressures 0.68, 0.81, and 0.95.



Figure 8. Effects of pressure on sensible heat flux from superheat.



Figure 9. Velocity and temperature profiles of CO_2 , at 5 MPa, 100 kg m⁻²s⁻¹,

and thermodynamic equilibrium vapor quality $x_b = 1$.



Figure 10. Effect of heat flux on the bulk mean temperatures and actual vapor qualities $x_a = 1$.