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# Prediction of $CO_2$ condensation heat transfer coefficient in a tube

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ABSTRACT

A prediction model of $CO_2$ condensation heat transfer coefficients was developed. The flow patterns in the present model were predicted by the Soliman model. The existing Thome et al. model was applied to predict heat transfer coefficients for the wavy-intermittent and annular flow. The modified Dittus–Boelter equation was utilized for the mist flow. Heat transfer coefficients in the annular-mist flow were calculated from the combination of models for the annular and the mist flow with the ratio of *We* number. The mean deviation of the present model with the experimental data of 589 points from seven different sources was 44.8%. The present model showed improved predictability compared to the existing models under high mass flux and high condensation temperature conditions, and with a small diameter tube.



## 1. Introduction

A vapor compression refrigeration system using $CO_2$ has a trans-critical cycle in applications of a domestic heat pump, a mobile air-conditioner, a vending machine, and a water heater. The utilization of $CO_2$ has been recently expanded to low temperature applications, such as food preservation and food processing, due to the increase of environmental concerns and superior thermo-physical properties of $CO_2$ at low temperature. In those systems, $CO_2$ is applied to the low stage cascade system or utilized as a secondary fluid for the low temperature applications, with a $CO_2$ condensation process that differs from the conventional trans-critical system (Park and Hrnjak [1], Gunawardane and Bansal [2]). Therefore, understanding the characteristics of $CO_2$ condensation heat transfer and precise prediction of the $CO_2$ condensation heat transfer coefficient is critical for the optimum design of a condenser for a $CO_2$ refrigeration system.

Many researchers have studied $CO_2$ condensation heat transfer characteristics and compared the measured heat transfer coefficients to the widely accepted models. Table 1 shows mean absolute deviations of prediction of the existing experimental data with the previous models. Kim et al. [3], Iqbal and Bansal [4], and Kang et al. [5] conducted experiments with a smooth tube, and Haui and Koyama [6], Park and Hrnjak [7], and Heo et al. [8,9] utilized a microchannel as a test tube. Their experiments were performed under various test conditions showed large differences among the researchers. The mass flux condition of Iqbal and Bansal [4] was 50–200 kg/m$^2$ s, much lower than that of other studies. The condensation temperatures of Kim et al. [3] and Park and Hrnjak [7] were −15 and −25 °C. However, the condensation temperature of Haui and Koyama [6] was 21–31 °C. The mass fluxes of Kang et al. [5], and Heo et al. [8,9] were high and condensation temperatures were from −10 to 5 °C. It is very hard to generalize the prediction ability of the existing models, because the test results from each source were obtained under significantly different test conditions. However, it was universally found that most models over-predicted the experimental data, and the deviation increased with increase of mass flux, vapor quality, and condensation temperature. Also, it was observed that the Thome et al. model [10] showed minimum deviations with experimental data among the existing prediction models. However, until now the detailed discussion of the low predictability of the existing models was insufficiently examined, and the prediction ability of the Thome et al. [10] is unsatisfactory. Previous studies of Kim et al. [3], and Iqbal and Bansal [4] attributed this large deviation to the thermophysical properties of $CO_2$ that differ from those of the conventional refrigerants. Kang et al. [5], Park and Hrnjak [7], and Heo et al. [8,9] discussed that the large deviation from the predicted heat transfer coefficients came from

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**Nomenclature**

| | |
|---|---|
| $A$ | total cross-sectional area of tube, $m^2$ |
| Abs | Absolute |
| $A_L$ | cross-sectional area of tube occupied by liquid, $m^2$ |
| Arith | Arithmetic |
| $C_p$ | specific heat of the liquid, J/kg s |
| $D$ | tube diameter, m |
| $D_h$ | hydraulic diameter, m |
| $f_i$ | interfacial roughness factor |
| $Fr$ | Froude number |
| $g$ | gravitational acceleration, $m/s^2$ |
| $G$ | mass flux, $kg/m^2$ s |
| $Ga$ | Galileo number |
| $h$ | heat transfer coefficient, $W/m^2$ K |
| $h_c$ | convective condensation heat transfer coefficient, W/$m^2$ K |
| $k$ | thermal conductivity, W/mK |
| $Pr$ | Prandtl number |
| $Re$ | Reynolds number |
| $R_{We}$ | Ratio of $We$ numbers |
| $u$ | velocity, m/s |
| $We$ | Weber number |
| $x$ | vapor quality |
| $X_{tt}$ | Martinelli parameter |

*Greek symbols*

| | |
|---|---|
| $\delta$ | liquid film thickness, m |
| $\varepsilon$ | void fraction of vapor |
| $\lambda$ | thermal conductivity, W/mK |
| $\mu$ | viscosity, $m^2/s$ |
| $\rho$ | density, $kg/m^3$ |
| $\sigma$ | surface tension, N/m |
| $\phi_v$ | two-phase pressure drop multiplier |

*Subscripts*

| | |
|---|---|
| c | condensation |
| i | interfacial |
| L | liquid film |
| l | liquid |
| ll | lower limit |
| sat | saturation |
| tp | two phase |
| ul | upper limit |
| v | vapor |

the flow complexity, relating to the flow pattern transition. It should be noted that a verifiable and trustworthy flow pattern model for $CO_2$ condensation does not exist. Therefore, it is necessary to develop a model of the condensation heat transfer coefficient of $CO_2$ with high prediction ability, considering the flow patterns and thermo-physical properties of $CO_2$.

The objective of this study is to develop a prediction model of the condensation heat transfer coefficient of $CO_2$ by considering the flow patterns, required to improve the prediction ability. The Soliman model [11,12] was applied to the prediction of the flow patterns of $CO_2$ condensation. Different prediction models of heat transfer coefficient for each flow pattern were provided.

**Table 1**
Models applied by several researchers for prediction of condensation heat transfer coefficients.

| Refs. | Applied models for prediction of condensation heat transfer coefficient | Mean abs. deviation |
|---|---|---|
| Kim et al. [3] | The Dobson and Chato model | 56.9% |
| | The Cavallini et al. model | 35.2% |
| | The Thome et al. model | 20.9% |
| Iqbal and Bansal [4] | Dobson and Chato | 32.5% |
| | Shah | 34.9% |
| | Li et al. | 70.3% |
| Kang et al. [5] | Shah model | 233.4% |
| | Cavallini and Zecchin | 264.4% |
| | Thome et al. | 172.2% |
| Haui and Koyama [6] | Koyama | −50% − more than 200% |
| Park and Hrnjak [7] | Dobson and Chato | 62.7% |
| | Cavallini et al. | 46.6% |
| | Thome et al. | 18.3% |
| Heo et al. [9] | Thome et al. | 47.7% |
| | Cavallini et al. | 147.7% |
| | Bandhauser | 154.7% |

## 2. Model development

### 2.1. Existing models

Flow pattern maps from Cavallini et al. [13], El Hajal et al. [14], and Soliman [11,12] have been considered to determine the flow patterns of $CO_2$ condensation. The flow pattern map by Cavallini et al. [13] was developed based on working fluids of R-22, R-134a, R-125, R-32, R-236ea, R-407C, and R-410A in an 8 mm inside diameter plain tube. Two phase flow regimes were divided into slug, wavy-stratified,

**Table 2**
Models for condensation flow regimes.

Refs.: Soliman [11,12]

Eqs. For the calculation of the flow pattern map:

$$Fr = \left(\frac{Re_l}{aGa^b}\left(\frac{\phi_v}{X_{tt}}\right)^c\right)^d \quad (1)$$

$$We = eRe_v^f\left(\frac{\mu_v^2}{\rho_v\sigma D}\right)^{0.3}\left[\left(\frac{\mu_v}{\mu_l}\right)\left(\frac{\rho_l}{\rho_v}\right)\right]^g\frac{X_{tt}^h}{\varphi_v^{0.4}} \quad (2)$$

where
$a = 10.18$, $b = 0.313$, $c = 0.938$, $d = 1.6$, $e = 2.45$, $f = 0.64$, $g = 0$, $h = 0$ for $Re_l \leq 1250$,
$a = 0.79$, $b = 0.481$, $c = 1.442$, $d = 1.04$, $e = 0.85$, $f = 0.79$, $g = 0.084$, $h = 0.157$ for $Re_l > 1250$
Wavy and intermittent: $Fr < 7$
Annular: $Fr > 7$ and $We < 20$
Annular-mist: $Fr > 7$ and $20 \leq We \leq 30$
Mist: $Fr > 7$ and $We > 30$

$$Re_l = \frac{GD(1-x)}{\mu_l} \quad (3)$$

$$Ga = gD^3\left(\frac{\rho_l}{\mu_l}\right)^2 \quad (4)$$

$$\phi_v = 1 + 1.09X_{tt}^{0.039} \quad (5)$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1} \quad (6)$$