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Prevention of dew condensation on the case surfaces of ceiling-cassette indoor air conditioning units

Guangri Jin, Kwan-Soo Lee*

School of Mechanical Engineering, Hanyang University, 222 Wangsimni-ro, Seongdong-gu, Seoul 04763, Republic of Korea

HIGHLIGHTS

- A new method for preventing dew condensation on the case surface of air conditioning indoor unit was proposed.
- The effect of non-condensable gases was incorporated in the phase change model.
- The installation of the optimized blockage led to a 41% reduction in the condensation rate.

ARTICLE INFO

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Ceiling-cassette air conditioner
Dew avoidance
Flow blockage
Optimization

ABSTRACT

We proposed a new method for preventing dew condensation on the case surfaces of ceiling-cassette indoor air conditioning units. We investigated the dew condensation problem numerically using a diffusion boundary layer phase-change model that incorporates the effect of non-condensable gases. By installing a blocking device on the case surface, we significantly reduced the suction flow of warm, humid air and the total condensation rate on the surface. We optimized the installation location, height, and angle of the blockage in order to minimize dew condensation. When the optimized blockage device was installed, and the air conditioning unit was operated at low discharge speed, we observed that the total numerical and experimental condensation rate fell by 41 and 30%, respectively.

1. Introduction

Recently, as living standards have improved, the demand for thermal comfort of indoor environment has increased. As an effective means of controlling room temperature and humidity levels, interest in air conditioners has risen commensurately with this demand [1]. Because indoor air conditioning (AC) units are installed in spaces where people are active, they play a significant role in ensuring that indoor environments are comfortable [2,3]. In recent years, many consumers have opted for ceiling-cassette air conditioners, which provide thermal comfort with high spatial efficiency [4]. However, when this type of air conditioner is operated under very humid conditions, dew condensation forms on the case of the unit. Droplets of condensation then drip off when they become large enough, decreasing the level of comfort. This problem is particularly pronounced in subtropical areas during the rainy season. Prediction and avoidance of this problem will help to improve the level of comfort provided by air conditioning units.

In our previous study [5], we investigated the causes of dew condensation formation on ceiling-cassette indoor air conditioning units; the main cause of dew condensation was found to be warm, humid air being sucked into the vicinity of the discharging outlet and contacting the cold surface of the case. Although we determined the causes of dew condensation and investigated the effects of different operating conditions on dew condensation in the previous study, we did not provide a practical solution to this problem. Additionally, the phase-change model [6,7] used in the study failed to take into account the influence of non-condensable gases. Consequently, the accuracy of the computational results decreased as flow rate increased.

In this study, we used a more accurate diffusion boundary layer phase change model to simulate the dew condensation process [8–10]. We also suggest a novel method for avoiding dew condensation by installing a flow-blocking device. In addition, optimization study of the flow-blocking device was carried out to maximize the avoiding effect.

2. Numerical models

2.1. Physical description and computational domain

Fig. 1(a) shows a schematic diagram of a ceiling-cassette indoor air conditioning unit comprised of a turbo fan, heat exchanger, guide vane,

* Corresponding author.
E-mail address: ksleehy@hanyang.ac.kr (K.-S. Lee).

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Nomenclature

A	area of computational grid
h	height (m)
d	distance (m)
D	mass diffusion coefficient (m^2/s)
L_H	latent heat (kJ/kg)
T	temperature (K)
$\dot{m}''$	mass transfer rate ($kg/m^3{\cdot}s$)
w	absolute humidity (g/kg_{DA})
u	velocity (m/s)
x, y	coordinates
S_x	precision index
V	volume of computational grid

Greek symbol

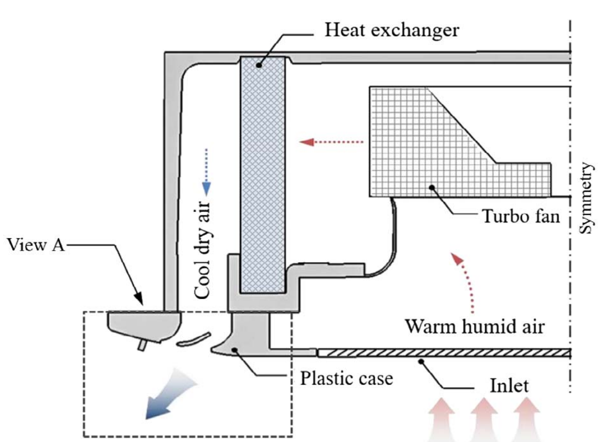
α	blockage angle (°)
β	guide vane angle (°)
ρ	density (kg/m^3)
δ_b	bias error

Subscripts

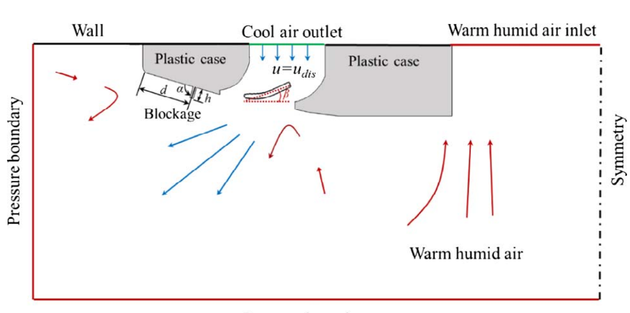
a	ambient
$cell$	computation grid
dis	discharge
ncg	non-condensable gas
H_2O	water vapor

discharging outlet, air inlet and plastic case. The air passes into the indoor unit by rotation of the turbo fan, is cooled in the heat exchanger, and is finally discharged through the outlet.

The dew condensation on the indoor case occurs at a specific location, we limited the computational domain, as shown in Fig. 1(b). The boundary conditions were the same as the experimental conditions. The case of the AC unit was constructed from plastic (HIPS-60-HRI) of thermal conductivity 0.4 W/m·K. The guide vane angle was fixed at $\beta = 20°$. As dew condensation was evident only at the lowest operating temperature (288 K), the temperature boundary condition of the discharged air was set to 288 K. The absolute humidity and temperature of the ambient air were $w = 17.6\ g/kg_{DA}$ (relative humidity = 78%) and $T_a = 300$ K, respectively.



(a) Schematic of indoor cassette AC



(b) Computational domain (Enlarged view A)

Fig. 1. Schematic of the indoor cassette air conditioner and the computational domain (dashed-dotted line: y-axis of symmetry).

2.2. Governing equations and condensation model

We based our numerical simulation of dew condensation on the following assumptions:

(1) The flow was turbulent, steady and two-dimensional (2D).
(2) Air is an ideal gas that, with the exception of the density, has constant properties.
(3) Radiation heat transfer and natural convection were neglected.

The discharge flow was turbulent, especially close to the discharge outlet. Accordingly, a turbulence model was selected. For an accurate simulation, the SST *k-w* model was selected, because it had the smallest deviation from the experimental data. The governing equations used in this study were as follows:

$$\frac{\partial}{\partial x_i}(\rho u_i \phi) = \frac{\partial}{\partial x_i}\left(\Gamma_\phi \frac{\partial \phi}{\partial x_i}\right) + S_\phi \tag{1}$$

where Γ_ϕ and S_ϕ are arbitrary scalar parameters used to describe continuity, momentum, turbulent kinetic energy, turbulent kinetic energy dissipation, and energy, as listed in Table 1.

Unlike our previous study, we improved the accuracy of our numerical results by using a diffusion boundary layer phase-change model [8] that incorporates the effect of non-condensable gases. In this model, the mass transfer resistance associated with non-condensable gases is calculated by considering the concentration gradients of both water vapor and non-condensable gases, as shown in Fig. 2. As the phase change model used in the present study is based on the finite volume method, mass transfer must be described by calculating volumetric source terms. The unit of such terms is $kg/m^3{\cdot}s$, and the mass transfer rate during the condensation process was calculated as follows [8]:

$$\dot{m}'' = \frac{1}{w_{ncg}}\rho D \frac{\partial w_{H_2O}}{\partial y}\frac{A_{cell}}{V_{cell}} \tag{2}$$

Table 1
Values of Γ_ϕ and S_ϕ in general transport equation.

	ϕ	Γ_ϕ	S_ϕ
Continuity	1	0	S_M
u_x-Momentum	u_x	$\mu + \mu_t$	$-\frac{\partial p}{\partial x}$
u_y-Momentum	u_y	$\mu + \mu_t$	$-\frac{\partial p}{\partial y}$
Energy	T	$\frac{\mu}{Pr} + \frac{\mu_t}{Pr_t}$	$\frac{1}{C_p}(\mu\Phi + \rho\varepsilon) + S_E$
Species	Y_N	J	S_N
k	k	$\mu + \frac{\mu_t}{\sigma_k}$	$G_k - \rho\varepsilon$
ε	ε	$\mu + \frac{\mu_t}{\sigma_\varepsilon}$	$C_1 G_k \frac{\varepsilon}{k} - C_2\rho\frac{\varepsilon^2}{k + \sqrt{\nu\varepsilon}}$