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The optimization and effect of back panel structure on the performance of refrigerated display cabinet

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A B S T R A C T

In this paper, the effect of the back panel structure on the performance of fluid flow and heat transfer of vertical open refrigerated display cabinets (VORDC) is presented by experiments and numerical simulation. Experimental tests are performed to validate the accuracy of numerical predictions. The characteristics of heat transfer and fluid flow of VORDC are investigated at the different locations of the perforations of back panel at the same porosity and the different flow ratios between back panel and air curtain at different porosities. By comparing numerical results with experimental results, the predictive abilities of the computational model have been revealed. Further computational results have also shown that less than 3% porosities can provide a better performance in the VORDC; the location of perforations has a minor influence on the temperature distribution of products. Furthermore, the suitable porosities in the back panel among the shelves are more likely to improve the uniformity of products temperature in the VORDC. As a result, the overall uniformity of product temperature inside the refrigerated display cabinet and the maximum deviation values of product temperature have been improve to 41% and 49%, respectively. The present study can provide the theoretical guide for the design of the narrow VORDC.

1. Introduction

Keeping product temperature within an appropriate range is a challenging task for refrigerated display cabinets, so the cabinets must be properly designed (Morelli, Noel, Rosset, & Poumeyrol, 2012). For vertical open refrigerated display cabinets (VORDC), the infiltration of air curtain costs about 70% of the cooling load (Chen & Yuan, 2005; Cortella, Manzan, & Comini, 2001). Simultaneously, the results also show that the infiltration has had a significant effect on the temperature distribution of products which are located at the front of air curtain, and the problems of the uneven temperature distribution of products have existed widely in the open refrigerated display cabinets (Foster, Madge, & Evans, 2005). The results given by Evans et al. (Evans, Scarcelli, & Swain, 2007) also indicated that the highest temperature of products was situated in the front of VORDC. Therefore, a large number of researches had been focused on the heat transfer characteristics and the various factors on the influence of air curtain performance of VORDC. Gasper et al. (Gaspar, Carrilho Gonçalves, & Pitarma, 2011; Gaspar, Gonçalves, & Pitarma, 2008) analyzed the influence of ambient air temperature, humidity and convection on the thermal entrainment factor of air curtains in the VORDC. Chen (Chen, 2009) evaluated the thermal barrier performance of air curtain by adjusting its height/width radio and discharge angle. Ge et al. (Ge, Tassou, & Hadawey, 2010) studied the influence of ambient air temperature and humidity, and air flow velocity of air curtain on the temperature distribution in the VORDC. Laguerre et al. (Laguerre & Flick, 2010; Laguerre, Hoang, & Flick, 2012; Laguerre, Hoang, Osswald, & Flick, 2012) focused on the heat transfer and air flow characteristics and proposed a heat transfer model of refrigerated display cabinet, they also investigated the influence of operating conditions on the system performance of refrigerated display cabinet. Hammond et al. (Hammond, Quarini, & Foster, 2011) studied the thermal barrier performance by analyzing the air flow velocity of air curtain; while Gasper et al. (Gaspar, Carrilho Gonçalves, & Pitarma, 2012; Gaspar, Gonçalves, & Pitarma, 2012) investigated the performance of VORDC affected by some parameters, such as air flow rate through evaporator heat exchangers, air curtain behavior, hole dimensions and distribution on the back panel, discharge and return grilles angles and flow deflector locations inside the internal duct.

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Table 1
Variables, diffusion coefficients and source terms of each governing equations.

Equation	φ	Γ	s_φ
Mass equation	1	0	0
Momentum equation in *X*-direction	u	$\eta+\eta_t$	$-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left(\eta_{eff}\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\eta_{eff}\frac{\partial v}{\partial x}\right)$
Momentum equation in *Y*-direction	v	$\eta+\eta_t$	$-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left(\eta_{eff}\frac{\partial u}{\partial y}\right)+\frac{\partial}{\partial y}\left(\eta_{eff}\frac{\partial v}{\partial y}\right)$
K-equation	K	$\eta+\frac{\eta_t}{\sigma_k}$	$\rho G_K-\rho\varepsilon$
ε-equation	ε	$\eta+\frac{\eta_t}{\sigma_\varepsilon}$	$\frac{\varepsilon}{K}(C_1\rho G_K-C_2\rho\varepsilon)$
Energy	T	$\frac{\eta}{\Pr}+\frac{\eta_t}{\sigma_T}$	0
Species transport equation	C	$D+\frac{\eta_t}{Sc_t}$	0

NOTE:

$$G_K=\frac{\eta_t}{\rho}\left\{2\left[\left(\frac{\partial u}{\partial x}\right)^2+\left(\frac{\partial v}{\partial y}\right)^2\right]+\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2+\left(\frac{\partial w}{\partial x}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2\right\};\ \eta_{\text{eff}}=\eta+\eta_t;\ \eta_t=\rho C_\mu K^2/\varepsilon.$$

Although air curtain plays an important role in the resistance of infiltration, the effect of cold air from the back panel should not be ignored. The study by Axell et al. (Axell, 2002) showed that the unreasonable proportion of cold air from the back panel would increase the temperature difference and decrease the performance of the air curtain. Chen et al. (Chen et al., 2004) proved that the porosity of the back panel should normally be controlled at 1%–2%. The study by Navaz et al. (Navaz, Henderson, Faramarzi, Pourmovahed, & Taugwalder, 2005) indicated that about 70%–80% of air flow across the back panel would enter the return air grill. D'Agaro et al. (D'Agaro, Cortella, & Croce, 2006) observed that the unreasonable flow ratio between air curtain and back panel would increase the entrainment of the ambient air. Gray et al. (Gray et al., 2008) observed that the 70%–30% air flow ratio between discharge air grill and back panel obtained a favorable stability of air curtain and maintain the internal temperature of the cabinet; therefore, the air flow from back panel has a certain effect on the internal temperature of cabinet and the temperature of the return air grille (RAG).

From above literature reviews, it can be seen that a few researchers focused on the effect of back panel on the performance of refrigerated display cabinets, whereas some paid more attention on the wide vertical open refrigerated display cabinet which had two air curtains. Recently, owing to the limitations of supermarket area, some supermarketeers put forward that the display area of VORDC should be as large as possible, but the volume of VORDC should be as narrow as possible instead. Accordingly, this paper will focus on the effects of porosities and the locations of the perforations of back panel in a narrow vertical open refrigerated display cabinet with a single air curtain and a 400 mm-width shelf. The aim of the paper is to obtain a lower and uniform temperature of products in the refrigerated display cabinet.

2. Mathematical model

2.1. Mathematical model

In the present investigation, the cabinet model is simplified as a two-dimensional model, for the refrigerated display cabinet is larger in length than in width. The mathematical model of the refrigerated display cabinet is assumed to be steady, incompressible fluid flows, and neglect the viscous dissipation. The influence of external relative humidity is incorporated by the species transport model. The mathematical model is as follows:

Table 2
The coefficients in $K-\varepsilon$ model.

C_μ	C_1	C_2	σ_K	σ_ε	σ_T
0.09	1.4	1.92	1.0	1.3	0.8

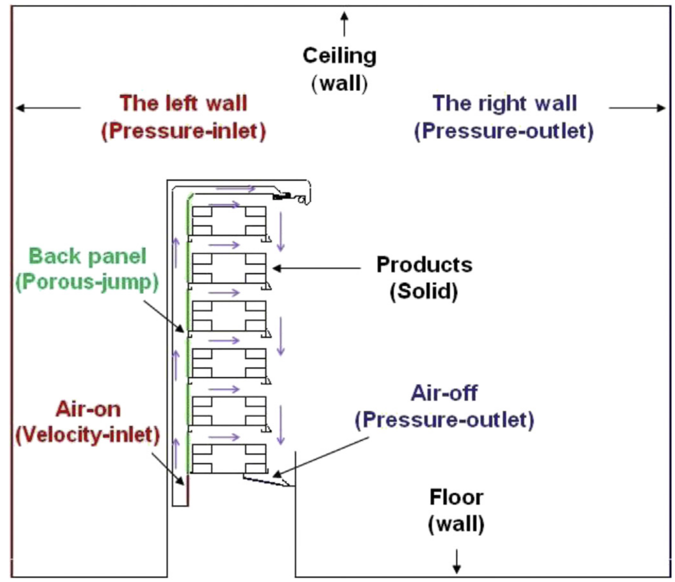

Fig. 1. Boundary conditions.

$$\text{div}(\rho V\varphi)=\text{div}(\Gamma_\varphi\text{grad}\varphi)+S_\varphi \tag{1}$$

where: φ is the common variable; Γ is the generalized diffusion coefficient; S is the generalized source term corresponding to φ; Table 1 gives the parameter used in the above equation. Classically recommended values are applied as the constants for the $k-\varepsilon$ model in Table 1, which are proposed by Tao (Tao, 2004) and shown in Table 2.

The heat gain by thermal radiation is one of the most important cooling load between food package and walls. The thermal radiation equation is as follows:

$$\frac{dI}{ds}+(a+\sigma_s)I=a\frac{\sigma T^4}{\pi}+\frac{\sigma_s}{4\pi}\int_0^{4\pi}Id\Omega \tag{2}$$

where: I is the directional radiation intensity; s is the optical path length; a is the absorption rate; σ_s is the scattering rate; T is the local temperature; σ is the Stefan–Boltzmann constant; Ω is the solid angle.

2.2. Boundary conditions

Fig. 1 shows the boundary conditions of the computational model. Table 3 give the boundary conditions. The left and right boundaries are defined as pressure inlet and pressure outlet respectively. Meantime, air temperature, humidity, and pressure

Table 3
Boundary conditions.

	Temperature	Velocity	Pressure	Relative humidity
Air-on	271.5 K	0.5 m/s		100%
Air-off	280 K		−1.1 Pas	75%
The left wall	298.15 K		0	60%
The right wall	298.15 K		0	60%