International Journal of Heat and Mass Transfer 87 (2015) 685–692

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

A multi-scale approach for refrigerated display cabinet coupled with supermarket HVAC system-Part II: The performance of VORDC and energy consumption analysis

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ARTICLE INFO

Article history:
Received 22 July 2014
Received in revised form 15 January 2015
Accepted 3 April 2015
Available online 21 April 2015

Keywords:
System performance
Energy consumption
Multi-scale approach
HVAC
VORDC

ABSTRACT

A multi-scale approach has been formulated in a previous work. The influence of HVAC system in supermarket on the performance of vertical open refrigerated display cabinet (VORDC) is considered. The influence is evaluated by varying the supply air temperature and velocity at the diffusers of HVAC. The computational results show that there is a direct interaction between the HVAC system and VORDC in supermarket. When the supply air temperature of HVAC system decreases from 19 °C to 16 °C, the electricity energy input in VORDC system decreases by 6.36%, and the highest products temperature is reduced by 0.18 °C. When the diffuser supply air velocity of HVAC system increases from 2.0 m/s to 3.5 m/s, the energy input in VORDC system decreases by 23.4%, and the highest product temperature reduces by 0.4 °C.

1. Introduction

In the vertical open refrigerated display cabinets (VORDC), the air curtain not only provides cooling capacity but also establishes an aerodynamic barriers between conservation space and external environment [1]. The thermal insulation performance of the air curtain has a great influence on the perishable food storage and energy consumption for VORDC. The infiltration load through the air curtain is responsible for a major percentage, about 70% [2,3]. Meanwhile, the energy consumption spent on food preservation is about 50% of the supermarket's total energy consumption [4–6].

In recent years, many experiments and numerical studies have been conducted to improve the performance of air curtains. There are many interdependent factors which act simultaneously on of air curtains, such as discharged air parameters and ambient air conditions [7–10]. Field and Loth [11] pointed out that the entrainment of ambient air was primarily due to eddy engulfment which arose from shear layer interactions and enhanced by high initial turbulence levels in the curtains. D'Agaro et al. [12] and Foster et al. [13] developed a 3D CFD to predict refrigeration power within engineering accuracy. Yu et al. [14,15] proposed a two-fluid model to optimize the air curtain effectiveness. The studies carried out by Gray et al. [16] and Wu et al. [17] showed that a 70–30 distribution of flow between the air curtain and the rear duct perforations yielded a performance that satisfies the standards. Gaspar et al. [18,19] developed a detailed CFD VORDC model including the internal ducts, fans and evaporator to get a global performance improvement. Laguerre et al. [20] proposed a mathematical model of heat transfer to predict the products temperature in an VORDC under different operating conditions, taking into account of the conduction, convection and radiation. Chang et al. [21] developed a 2D CFD model of VORDC to evaluate the influence of light, velocity in air curtains, ambient temperature and relative humidity on the food package temperature.

Furthermore, if the store indoor environment maintains at a lower relative humidity, it will result in a higher operating cost of HVAC system, but it could reduce frost formation on the refrigerator evaporator coil and anti-sweat heater. When the store relative humidity was reduced from 55% to 35%, the majority of the refrigerated display cabinet would be saved 20–30% in compressor energy, 40–60% in defrost energy, and 19–73% in anti-sweat heater operation [22,23]. Orphelin et al. [24] developed a coupling model including display cabinets and the air-conditioning system to estimate impacts of temperature and humidity on the total energy balance of supermarkets. Bahman et al. [25] analyzed the energy saving in a supermarket refrigeration/HVAC system using MATLAB software.

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2015.04.003
0017-9310/

Nomenclature

A	area, [m^2]
Cp	specific heat, [J kg^{-1} K^{-1}]
h_{fg}	vaporation enthalpy, [kJ kg^{-1}]
h	enthalpy, [J kg^{-1} K^{-1}]
m	mass flow rate, [kg s^{-1}]
NIR_0	infiltration rate for air curtain without PBP
NIR_{PBP}	infiltration rate for perforated back panel airflow
Q, P	power, [W]
$P_{chiller}$	power input to chiller, [W]
P_{cond}	power requirement to condense supply air in chiller (latent heat), [W]
T	temperature, [°C]
v	average velocity, [m s^{-1}]
W	width, [m]

Superscripts and subscripts

amb	indoor ambient
BPB	perforated back panel airflow ratio
cli	climate conditions
diff	diffuser
out	outdoor air
sat	saturated air
sup	diffuser supply air

Greek symbols

α	constant in regression analysis, [gH_2O m^{-3} $°C^{-1}$]
β	perforated back panel airflow ratio, [–]
ρ	density, [kg m^{-3}]
ω	moisture content, [g kg^{-1}]
φ	relative humidity, [%]

Abbreviations

2D	two-dimensional
COP	coefficient of performance
DAG	discharge air grille
DSAT	diffuser supply air temperature
DSAV	diffuser supply air velocity
HVAC	heating, ventilation and air conditioning
NUM	number
RAG	return air grille
BPB	perforated back panel
SEN	sensible
LAT	latent
NIR	non-dimensional form of the infiltration rate
VORDC	vertical open refrigerated display cabinet

From the literature review, it can be seen that the indoor environment not only affects the performance of air curtains, but also determines the energy consumption of the supermarket HVAC system. However, few studies have gave comprehensive details on the heat transfer and fluid flow characteristics coupling the supermarket HVAC system and the VORDC, and the influence of the HVAC system on the thermal performance of VORDC. In fact, the heat transfer and fluid flow characteristics include multiple length scales from the supermarket HVAC system to the products in the VORDC [26], as is show in Fig. 1. Therefore, the work described in this paper intends to evaluate the influence of supermarket HVAC system (such as air temperature, relative humidity and velocity, et al.) on the thermal performance and energy consumption of VORDC based on a multi-scale approach.

The previous work has proved the feasibility of the multi-scale approach which couples the supermarket HVAC system and the VORDC. The fluid flow and heat transfer process for different scales models are assumed as a 3D steady-state model. The basic governing equations are continuity, momentum and energy equations. The turbulence is modeled by the standard κ–ε two-equation model. Considering the mixing fluid of dry bulb air and water vapor and the heat gain of products through radiation, a species transfer model and the discrete ordinates model are used, respectively. The SIMPLE algorithm is employed to deal with the pressure–velocity coupling. The convection term is discretized using the second order upwind scheme.

2. Parameter definitions

Due to the fluctuation in the air curtains, the ambient warm air is entrained throughout the entire height of the display cabinet opening and mixed with the air curtains. A portion of the mixed ambient air will enter the return air grille, which is defined as infiltration. The rest spills into ambient at the bottom of the opening. Generally, the non-dimensional form of the infiltration rate without perforated back papal (PBP) airflow, NIR_0, is defined as Eq. (1) [27].

$$NIR_0 = \frac{h_{RAG} - h_{DAG}}{h_{amb} - h_{DAG}} \approx \frac{T_{RAG} - T_{DAG}}{T_{amb} - T_{DAG}} \tag{1}$$

When the entire discharged air from the DAG is drawn back into the RAG, $T_{DAG} = T_{RAG}$, the infiltration air flow goes to zero, which represents the best insulating performance of air curtains.

Considering the influence of the PBP airflow, the NIR formula can be defined in Eq. (2)

$$NIR = (1 - \beta)NIR_0 + \beta NIR_0 NIR_{PBP} \tag{2}$$

where, β is the mass flow ratio from PBP which can be obtained based on Eq. (3) and NIR_{PBP} is the NIR for PBP airflow given by Eq. (4)

$$\beta = \frac{m_{PBP}}{m_{DAG} + m_{BPB}} \tag{3}$$

$$NIR_{BPB} = \frac{T_{BPB} - T_{DAG}}{T_{amb} - T_{DAG}} \tag{4}$$

The ambient measurement point of a VORDC is midway along the cabinet length in accordance with the EN 441-4. The point location is shown in Fig. 2.

The cooling load of refrigerated display cabinets mainly consists of the heat transfer between the fabric of the cabinet and the ambient air, radiation between the products and surrounding surfaces, heat gains from fan motor and lights and infiltration of ambient air across the air curtains [28]. The total cooling loads can be calculated simply by Eq. (5). It can be divided into sensible Eq. (6) and latent Eq. (7) components [2,29]. Thus, the electricity consumption can be obtained by cooling load divided by the coefficient of performance (COP) of refrigeration system. Commonly, the COP of a middle typical temperature VORDC is 2 [30].

$$Q_T = (\rho v A)_{DAG}(h_{RAG} - h_{DAG}) \tag{5}$$

$$Q_{T,SEN} = (\rho v A)_{DAG} c_{P,M}(T_{RAG} - T_{DAG}) \tag{6}$$

$$Q_{T,LAT} = (\rho v A)_{DAG} h_{fg}(\omega_{RAG} - \omega_{DAG}) \tag{7}$$