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Application of CFD in evaluation and energy-efficient design of air curtains for horizontal refrigerated display cases

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Abstract

Horizontal refrigerated display cases consume a large amount of energy and interact closely with the indoor environment in supermarkets. How to design a high energy-efficient display case has long been an important task of the industry and an important topic for research. This paper uses the CFD method to evaluate the energy performance of the air curtain for horizontal refrigerated display cases and to optimize its design. The CFD method is also validated by comparing the CFD calculation results with experimental results. Using the CFD method, the key factors that influence the cooling load of the air curtain in the display cases are studied. Qualitative designs are then proposed to make the air curtain more energy-efficient.

Keywords: Refrigerated display cases; Air curtain; CFD model; Key factors; Energy-efficient design

1. Introduction

Horizontal refrigerated display cases are used to merchandise perishable food and provide desirable storage temperatures. The refrigeration system controls product storage temperatures by removing all of the heat gain components of the display case. The air curtain, a chief constituent of the refrigeration system of display cases, provides a barrier of moving air across a door opening and reduces heat transfer by preventing the penetration of hot air. At the same time, the air curtains also provide environmental separation by preventing the infiltration of pollutants, dust, dirt and insects.

Each year, American supermarkets consume about 54.5 billion kW-hr of electrical energy, with refrigeration and air-conditioning systems accounting for as much as 75% of the load [1]. A significant portion of that energy is used to satisfy the cooling load requirements of refrigerated display cases, which are strongly dependent on the performance of the air curtain of them. In such a fiercely competitive industry, it is a continuing challenge for researchers to investigate the characteristics of the air curtain and develop energy-efficient designs for it. Nowadays, studies on the air curtain are conducted mainly in such fields as experimental testing and computational fluid dynamics (CFD) simulations.

Experimental testing is a direct research method. Howell [2] conducted tests of display cases at the ambient conditions of constant temperature and changing relative humidity (RH). The tests indicated that the cooling load of the air curtain is greatly influenced by the relative humidity of the ambient air when the temperature of the ambient air is constant. Sweetser [1] suggested that the energy impact was caused by the infiltration of ambient air through the air curtain, which constitutes about 80% of the total cooling load of the open vertical case. Lowering the RH from 55 to 35% reduces the latent load and compressor power demand of the open vertical dairy case by 53 and 21%, respectively. It also reduces the duration of the defrosting period by 40%. In addition, Faramarzi [3] pointed out that a newer display case, which is equipped with an improved single-band air curtain design, is more energy-efficient than its predecessor. However, what should be pointed out is that the flow of the air curtain is a cold turbulent jet, whose mechanisms of flow, heat and mass transfer are rather complicated. Therefore, the experiments that have been conducted for the evaluation of practical designs have been rather limited. It is difficult for designers to investigate the performance of the air curtain just by relying

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Nomenclature

b_0	width of air curtain	m
c_μ	a coefficient in K–ε two-equation model	
g	gravitational acceleration	$\mathrm{m \cdot s^{-2}}$
H	flux of enthalpy	$\mathrm{J \cdot s^{-1}}$
K	turbulent kinetic energy	$\mathrm{m^2 \cdot s^{-2}}$
L	characteristic dimension of air curtain	m
Q	cooling load or heat transfer rate	W
R_K	source term of K	$\mathrm{J \cdot kg^{-1}}$
R_ε	source term of ε	$\mathrm{m^2 \cdot s^{-3}}$
T_h	temperature of ambient hot air	K
T_c	average inlet temperature of air curtain	K
u_0	average inlet velocity of air curtain	$\mathrm{m \cdot s^{-1}}$
u	velocity component in x-direction	$\mathrm{m \cdot s^{-1}}$
UI	uniformity of velocity	
v	velocity component in y-direction	$\mathrm{m \cdot s^{-1}}$
x	horizontal coordinate	m
y	vertical coordinate	m

Greek symbols

ε	dissipation rate of turbulent kinetic energy	$\mathrm{m^2 \cdot s^{-3}}$
ρ	density of air	$\mathrm{kg \cdot m^{-3}}$
Γ	diffusion coefficient	
Δ	velocity difference	

Subscripts

cur	air curtain
con, ext	conduction through external walls
con, int	conduction through internal walls
fan	fan and light
inlet	inlet of air curtain
outlet	outlet of air curtain
rad	radiation

on experimental methods. A convenient and low-cost evaluation method is needed for optimizing designs.

Fortunately, with the rapid development of computer technology, CFD, a powerful theoretical calculation tool, is being ever more widely applied to investigating the air curtain of display cases. With the help of CFD, the work performances of the air curtain can be well simulated using various parameters and the time spent on research and design saved. Mu et al. [4] used the K–ε two-equation turbulence model and takes account of buoyancy, to describe the flow field of an air curtain in an open vertical display case. The calculation results tallied well with the experimental results in some measurement points. Bobbo et al. [5] used a two-dimensional turbulence finite-element method to simulate the temperature distribution of products in vertical display cases. The outcome of the calculation showed that the temperature of the products close to the outlet of the air curtain was highest in the case. Stribling et al. [6] presented a two-dimensional turbulence model of a vertical dairy display case that could be used to design and optimize such equipment. Cortella [7] used simple CFD models to predict the airflow pattern and the food temperature in the display cabinet. The numerical method was validated by comparison with experimental tests and showed to be reliable, and of valuable help to designers.

Although the CFD model showed good qualitative agreements with measured values and requires only fine-tuning to make it quantitatively accurate, there are shortcomings and restricting hypothesis in the current CFD investigations of the air curtain:

(a) *The influence of initial turbulence intensity on the development of the air curtain is not considered*;
(b) *The heat gain of the display case through radiation is ignored*; *however this is one of the most influential constituents of the cooling load for horizontal display cases* [8];
(c) *The changes in the properties of the fluid concerned, i.e., indoor wet air, are not taken into account. However, the definition of the boundary conditions has a great influence on the results of the calculation. At the same time, for a cold air jet in ambient hot air, the properties of the fluid are affected by its temperature to a great extent*;
(d) *The majority of the CFD investigations* [4–6] *place a great deal of emphasis on calculating and simulating the temperature and flow field of the air curtain, but they do not bring forward specific energy-efficient designs of the air curtain.*

In this study, with the aid of computational fluid dynamics software, Fluent 5.4, the authors use the K–ε turbulent model, radiation model and multiple species model to obtain numerical simulations of the air curtain in a horizontal display case. This paper presents the CFD model, the experimental validation, the approach of using the CFD method in optimizing the design of the air curtain and qualitative optimized designs.

2. Working principles and cooling load of horizontal display cases

Fig. 1 shows a sketch of the structure of the target display case in this paper. The display case is served by a refrigeration system using a CFC refrigerant, R12. It consists of a refrigeration system and an airflow system. The major portion of the cold load supplied by the refrigeration system is used to cool the circular airflow; the rest is used to keep the food in the case cool. In addition, the air in the circular

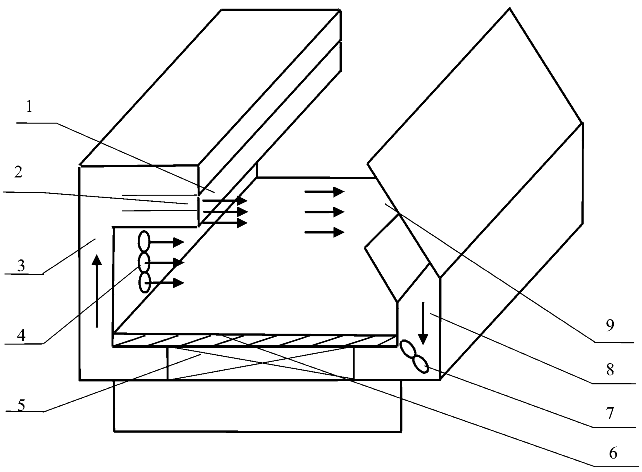

Fig. 1. Structure sketch of a horizontal refrigerated display case: (1. inlet, 2. inlet air honeycomb, 3. supplied air passage, 4. distributary holes, 5. evaporator, 6. insulator plate, 7. fan, 8. returned air passage, 9. outlet).

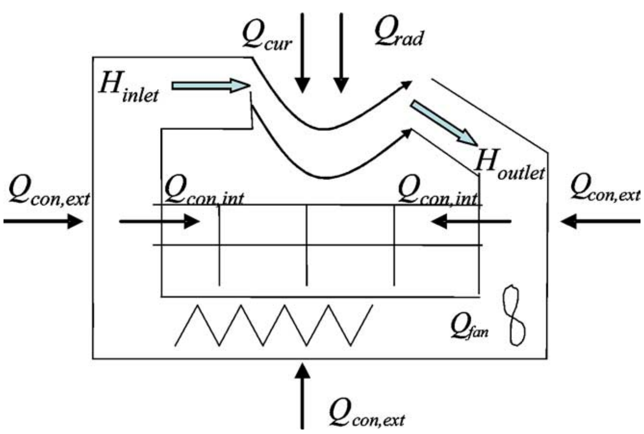

Fig. 2. Sketch of the cooling load balance of the display case.

flow can be divided into two parts: one flows through the distributary holes in the side wall to cool down the food and make the inside temperature distribution even, the other flows out of the inlet air honeycomb to form the cold air curtain. It is the air curtain that insulates the inside cold air from the outside warm air to maintain the appropriate temperature and humidity for the products inside. The power of the circular airflow comes from the fan below the case. Because the length of the case is much longer than its width, the flow of the air curtain can be investigated as a two-dimensional problem with the assumption that the effects of the two sidewalls on the air curtain could be negligible. In addition, the effects of the distributary holes on the air curtain are also assumed to be negligible as the ratio of the distributary flow rate to the air curtain flow rate is about 0.08 and the section area of the distributary holes is very small.

The heat transfer in the display case involves interactions between the products and the internal environment of the case, as well as the heat that enters the case from the surroundings. In steady state, the main cooling load components of medium-temperature display cases used to merchandise meat, deli products, dairy products, poultry and fish, etc., as shown in Fig. 2, are turbulent transport, radiation, fan motors and lights and external conduction [1], which can be formulated as follows:

$$Q = Q_{\text{cur}} + Q_{\text{rad}} + Q_{\text{fan}} + Q_{\text{con,ext}} \tag{1}$$

where, Q is the total cooling load of the display case, Q_{cur} is the turbulent transport load through the air curtain, Q_{rad} is the radiation load, Q_{fan} is the fan motor and light load, $Q_{\text{con,ext}}$ is the external conduction load.

The schematic diagram of energy balance in the display case is shown in Fig. 2. The corresponding equation of energy balance of air curtain is given by Eq. (2).

$$Q_{\text{cur}} = H_{\text{outlet}} - H_{\text{inlet}} + Q_{\text{con,int}} \tag{2}$$

where, $Q_{\text{con,int}}$ is the heat transfer rate through the internal walls. The H_{inlet} and H_{outlet} are the net flux of enthalpy of the air curtain at the inlet and outlet respectively.

For convenience, the internal walls of the display case are assumed to be adiabatic while the conditions of the air curtain inlet and the operating conditions of the display case are changed intentionally in CFD simulation to find energy-efficient designs. This assumption can be justified because the heat transfer through the internal walls is negligible as the internal bottom walls are insulator plates. It was observed also in the experiments. Thus, Eq. (2) can be rewritten into Eq. (3), which is used in the calculation of air curtain load and later CFD simulation for performance evaluation.

$$Q_{\text{cur}} \approx H_{\text{outlet}} - H_{\text{inlet}} \tag{3}$$

3. The CFD model and its validation

3.1. Simulation models

In this study, fluent 5.4, a multifunctional CFD computer software, was used. The standard K–ε two-equation model [9], as shown in Eqs. (4) and (5), was used as the turbulent model; and the discrete ordinates model was used as the radiation model. As the air curtain is wet air (i.e., a mixture of dry air and water vapor), the multiple species model was used. A uniform calculation grid (1 cm $\times$ 1 cm) was adopted. A few trial tests were conducted to verify the effects of grid size on the calculation accuracy and efficiency. It was observed that too long time was consumed in iteration if a smaller grid size was adopted. When a larger grid size was adopted the calculation accuracy was not desirable. A standard wall function was employed to deal with the flow field close to solid walls, and the SIMPLE algorithm [10] was employed to solve the coupling between the mass and momentum equations. With the CFD model, the main task of users is to define the boundary conditions. Fig. 3 illustrates the boundary and its nature.

$$\frac{\partial(\rho u K)}{\partial x}+\frac{\partial(\rho v K)}{\partial y} = \frac{\partial}{\partial x}\left(\Gamma_K \frac{\partial K}{\partial x}\right)+\frac{\partial}{\partial y}\left(\Gamma_K \frac{\partial K}{\partial y}\right)+R_K \tag{4}$$

$$\frac{\partial(\rho u \varepsilon)}{\partial x}+\frac{\partial(\rho v \varepsilon)}{\partial y} = \frac{\partial}{\partial x}\left(\Gamma_\varepsilon \frac{\partial \varepsilon}{\partial x}\right)+\frac{\partial}{\partial y}\left(\Gamma_\varepsilon \frac{\partial \varepsilon}{\partial y}\right)+R_\varepsilon \tag{5}$$

Because complicated mass and heat transfer occur at the opening of the display case, it is difficult to set reasonable boundary conditions there. In this study, the calculation area is extended left, right and up until the effect of the display case opening is negligible there. The extended boundaries (i.e., up, right-up and left-up, as illustrated in Fig. 3) are defined as isothermal walls whose temperatures are equal to the ambient air DB temperature. It should be mentioned that it is difficult to define the boundary conditions of the case internal walls and the outlet of the air curtain during the course of CFD simulation for predicting performances of the display case under other different conditions as no experiment data of the temperature distribution of the internal walls are available there. Therefore, the internal walls of the display case are defined to be adiabatic walls as stated in the previous section. The outlet of the air curtain, where the details of the flow velocity and pressure are not known prior to solution of the flow problem, is hereby defined as outflow boundary condition which assumes a zero normal gradient for all flow variables except pressure. Since the distribution of temperature and velocity at the inlet of the air curtain are not even, the inlet is divided into five identical sections, each of which is defined as a velocity inlet. The temperature and velocity at each section are also provided by experiments. The turbulent kinetic energy at each section, K, which is not measured here due to the lack of adequate experimental equipment, is set to be 1% of the average kinetic energy. The dissipation rate, ε, is determined by Eq. (6) according to Patankar et al. [11], who pointed out that the influence of the inlet K and ε on the final calculation results is little when the turbulent flow is intense in the calculation area.

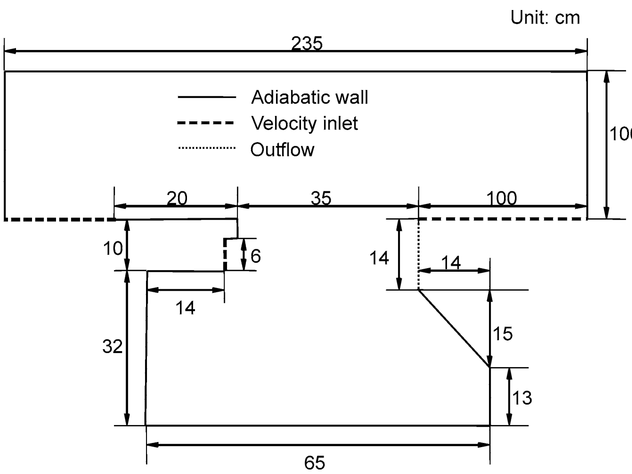

Fig. 3. Dimensions (non-scaled) and boundary types of the calculation area in Fluent 5.4.

$$\varepsilon=\frac{500K^2 c_\mu}{u_0 L} \tag{6}$$

where c_μ is a constant coefficient with a value of 0.09, u_0 is the average inlet velocity and L is the characteristic dimension of the air curtain (here it is the diameter of the comb). The left and right extended boundaries are defined as velocity inlets where the velocity, K and ε are all set to be zero, and the temperatures are all set to be the ambient temperature.

3.2. Experimental setup and experiments

The main aim of experiments is to detect the distribution of temperatures and the velocity uniformity at the inlet of the air curtain as well as the distribution of temperatures at the outlet under given operational conditions, e.g., ambient air dry-bulb (DB) temperature and relative humidity (RH). The experimental results not only provide necessary boundary conditions for later calculation, but also supply data that will later be compared with the results of simulation to assess the accuracy and viability of the established CFD model.

The experiments were carried out in an air-conditioned room with insulator covered walls and implemented on a horizontal refrigerated display case, with a chamber size of 300 liters and a cooling capacity of 0.3 kW. The indoor temperature and relative humidity of the room can be controlled. In most tests in this study, the ambient room conditions are 32.5 °C DB temperature and 45% RH. The velocity uniformity index at the inlet of the air curtain, *UI* (defined in 4.1), is 0.017 and the average inlet velocity is 0.26 $\text{m}\cdot\text{s}^{-1}$. Under these conditions of operation, air temperature and velocity at the inlet of the air curtain are measured as boundary conditions. Figs. 4 and 5 give the temperature and velocity profiles at the inlet of the air curtain, respectively. The upper temperature measurement points are affected by the ambient hot air and therefore assume higher temperatures, as shown in Fig. 4. The outlet temperatures of the air curtain are also measured in order to compare them with the simulated results.

The data acquisition systems include the temperature acquisition system equipped with special-grade type-T thermocouples accurate to $+/-0.1$ °C, a wind velocity acquisition system with hot-wire anemometers accurate to $+/-1\%$, and a relative humidity sensor accurate to $+/-3\%$. The sensing element of the relative humidity sensor is a thin film capacitor with membrane filter.

The setup of thermocouples in the experiment is illustrated in Fig. 6. In the test, one is placed 100 cm over the display case; six are evenly distributed at the inlet; eleven

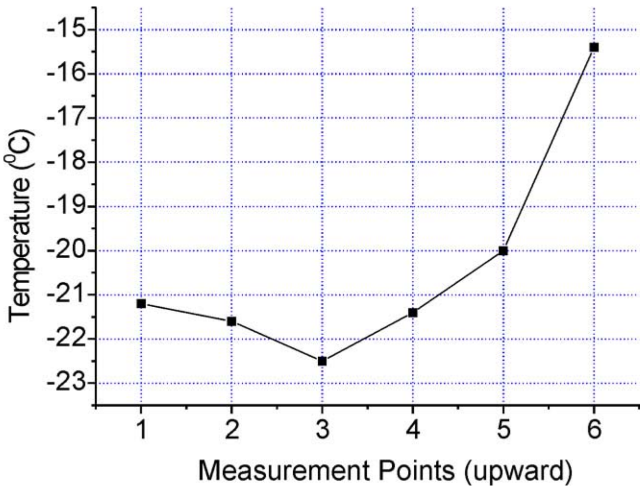

Fig. 4. Temperature profile at the inlet of the air curtain.

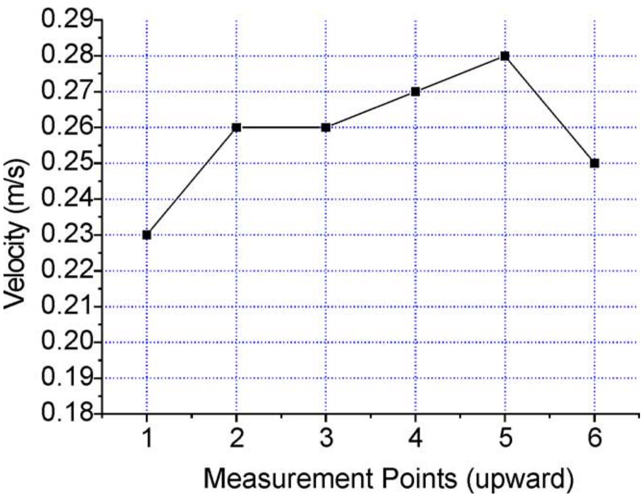

Fig. 5. Velocity profile at the inlet of the air curtain ($UI = 0.017$ and average value $= 0.26\ \mathrm{m{\cdot}s^{-1}}$).

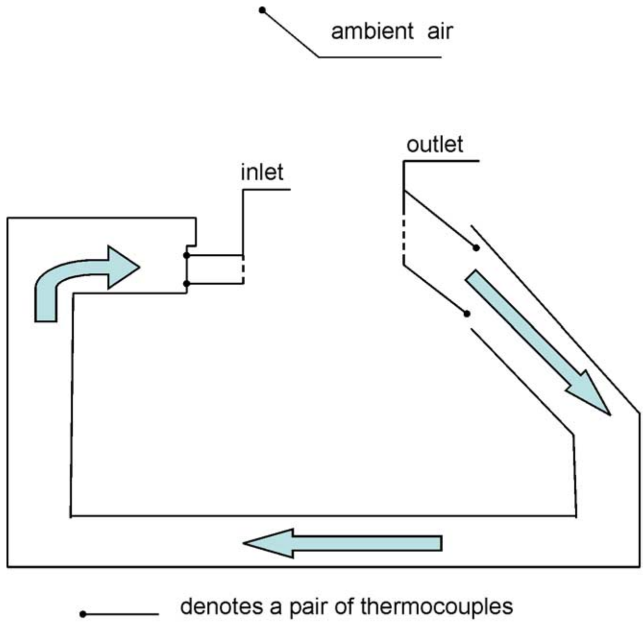

Fig. 6. The distribution of temperature measurement points.

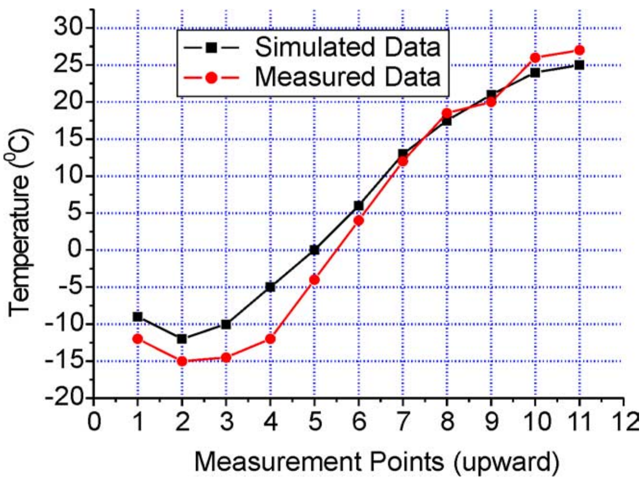

Fig. 7. Temperature profile at the outlet of the air curtain.

are evenly distributed at the outlet. The sampling interval is 2 s. The data visualization provided by the software can help determine the steady state of operation of the refrigeration system. Because the known directions are prerequisite to effective measurements with the velocity sensors and only the directions of the air velocity at the inlet is known, the velocity measurement points are those points where the inlet air temperatures are measured.

3.3. Comparisons between CFD predictions and experimental results

By virtue of the CFD model established previously, the temperature profile at the outlet of the air curtain was simulated and compared with the experimental results as shown in Fig. 7. The simulated results bear a great deal of similarity with the experimental results, whether quantitatively or qualitatively. However there is a difference of 4 °C at the bottom of the outlet. Fig. 8 presents the two-dimensional temperature isolines diagram with 2 °C temperature difference between each two isolines. Fig. 9 presents the velocity vectorgraph in the display case, which is divided into five areas.

Fig. 8 shows that the temperature isolines in the infiltration area, where forced convection and natural convection are mixed, are very dense. This means that there is a large temperature gradient there, which can induce significant transfers of heat and mass between the cold air curtain and the hot ambient air. On the contrary, in Fig. 8, the temperature isolines in the internal circumfluence area, the main circumfluence area and the external circumfluence area are rather sparse. It also can be observed in Fig. 8 that the principal axis of the air curtain, though very cold initially, is warmed by the ambient air and disappears gradually as the temperature isolines become increasingly sparse form the

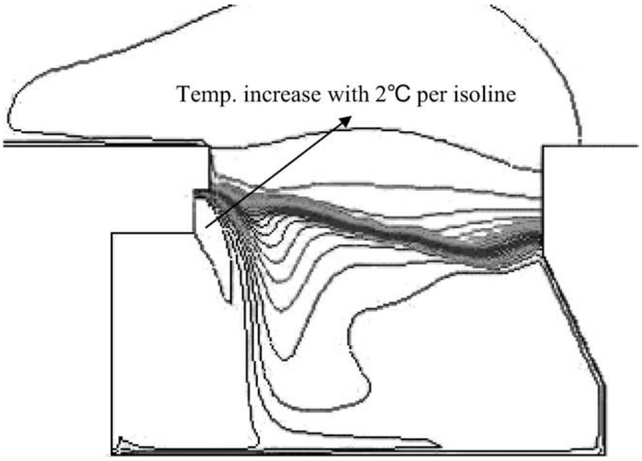

Fig. 8. Temperature isoline diagram in the display case (ambient temperature $= 32.5\,^{\circ}$C, RH $= 45\%$, $UI = 0.017$, and average velocity $= 0.26$ m·s^{-1}).

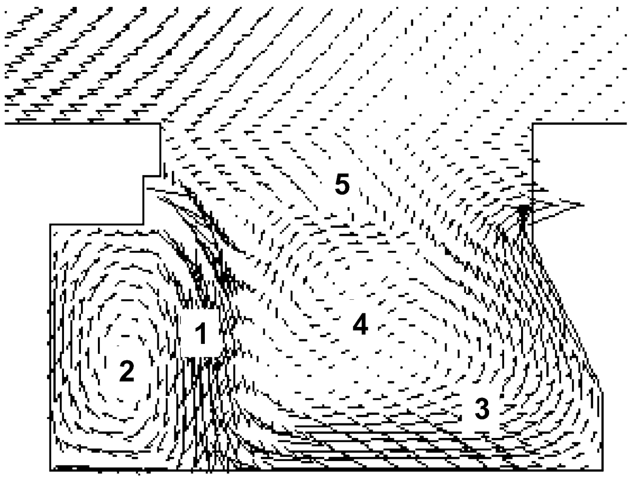

Fig. 9. Velocity vectorgraph in the display case (ambient temperature $= 32.5\,^{\circ}$C, RH $= 45\%$, $UI = 0.017$, and average velocity $= 0.26$ m·s^{-1}) *1. jet area, 2. internal circumfluence area, 3. main circumfluence area, 4. external circumfluence area, 5. infiltration area).*

nucleus of the principal axis to its brim. Fig. 9 illustrates, as we expect, a cold turbulent jet in limited space, which is heavily affected by gravity because its density is higher than that of ambient air. Affected by gravitational acceleration, the principal axial of the air curtain bends down quickly after it leaves the inlet; meanwhile, the magnitude of the velocity increases as demonstrated by the change of velocity vector. The air curtain jet wraps the ambient air during its development until it falls down to the bottom of the display case, then comes into two streams. One stream, mixing with the ambient air, flows towards the outlet at low pressure and forms the main circumfluence area and the external circumfluence area. The other stream flows back there and eventually forms an internal circumfluence area in the display case.

The illustrations here not only agree with the characteristics of gravity-driven jet, but also tally well with the related calculations and theoretical analyses in references [4] and [6]. The established CFD model and its solution algorithms can therefore be considered as validated.

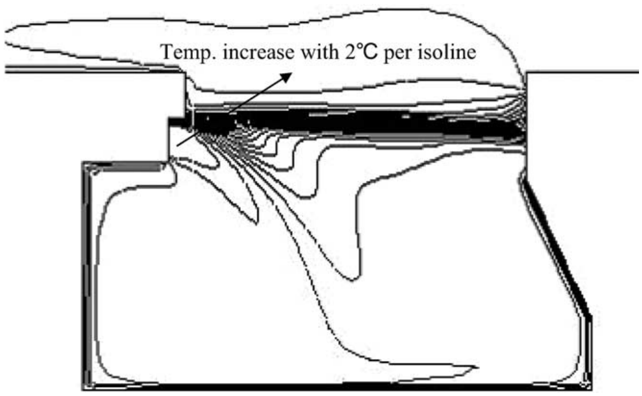

Fig. 10. Temperature isoline diagram in the display case (ambient temperature $= 32.5\,^{\circ}$C, RH $= 45\%$, $UI = 0$, and average velocity $= 0.26$ m·s^{-1}).

4. Evaluation of key factors and energy-efficient designs of the air curtain

As analyzed previously, infiltration through the air curtain is the largest constituent of the case cooling load [8]. The infiltration load of the display case refers to the entrainment of warm, moist air from the room, across the air curtain and into the internal refrigerated space. The total performance of the air curtain and the amount of heat transferred across it depends on several factors, including:

(a) *the uniformity of inlet velocity of the air curtain*;
(b) *the magnitude of inlet velocity of the air curtain*;
(c) *the height and shape of products inside the case*;
(d) *the temperature difference between the air curtain and the ambient hot air*;
(e) *the relative humidity of the ambient hot air*.

In the following sections, with the aid of the validated CFD model, the above five factors will be analyzed in detail and corresponding energy-efficient designs will be proposed.

4.1. The uniformity index of inlet velocity of the air curtain

The uniformity index, *UI*, which here describes the uniformity of the distribution of the inlet velocity of the air curtain, is defined as follows:

$$UI = \frac{\sqrt{\sum \Delta u^2/n}}{u_0} \tag{7}$$

where Δu is the difference between the velocity of a measured point and the average velocity in the inlet, u_0 is the average velocity in the inlet, and n is the number of measured points.

At ambient room conditions of $32.5\,^{\circ}$C DB temperature and 45% RH, Fig. 10 represents the temperature isoline

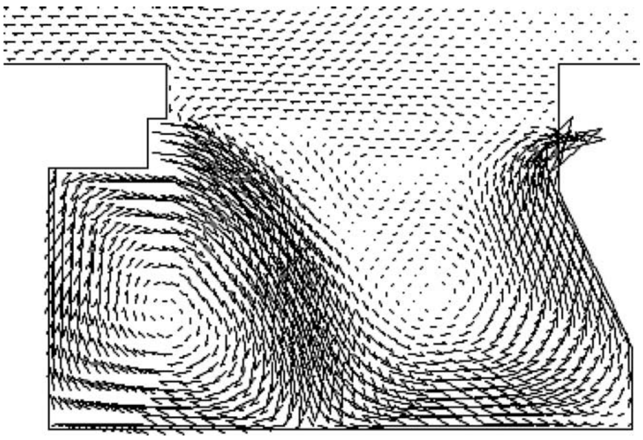

Fig. 11. Velocity vectorgraph in the display case (ambient temperature $= 32\,^{\circ}\mathrm{C}$, RH $= 45\%$, $UI = 0$, and average velocity $= 0.26\ \mathrm{m \cdot s^{-1}}$).

diagram with a temperature difference of $2\,^{\circ}\mathrm{C}$ between each two isolines in a display case when the inlet *UI* is zero and the average inlet velocity is $0.26\ \mathrm{m \cdot s^{-1}}$, and Fig. 9 represents the velocity vectorgraph with the same operating conditions. Comparing Fig. 8 with Fig. 10 and Fig. 9 with Fig. 11, it is clear that lowering the inlet *UI* from 0.017 to 0 can shorten the principal axis of the air curtain jet and enlarge the internal circumfluence area where the distribution of the temperature is very even.

Because the principal axis is the place where substantial heat and mass transfer occur, it can be assumed that the heat and mass transfer through the air curtain will diminish and the cooling load of the display cases will decrease as the principal axis is shorten. According to this comparison, a low inlet *UI* value is preferred in energy-efficient designs for display cases.

The following are two methods to decrease the inlet *UI* aimed at improving the performance of the air curtain:

(a) Improvement of the air passage. Because the evaporator of the display case is a cooling coil evaporator, the air in the air passage is inevitably very turbulent after flowing through the evaporator. However, the elbow of the air passage can be modified with a cambered transition in order to decrease the inlet *UI*.

(b) Improvement of the discharge honeycomb. As mentioned in [12], if the discharge honeycomb is a combination of big meshes and small meshes, the honeycomb can effectively reduce the inlet *UI*.

4.2. The magnitude of inlet velocity of the air curtain

The air curtain's capacity to minimize the penetration of the ambient hot air depends on its initial momentum and the adverse pressure difference, which is due to the density difference among the jet area of the air curtain. According to the analysis in Ref. [8], the mixed flow (forced convection and natural convection) of the air curtain is the main contributor of its cooling load. Fig. 12 and Fig. 13 show the simulated temperature isolines diagram with a temperature difference of $2\,^{\circ}\mathrm{C}$ between each two isolines and velocity vectorgraph of the air curtain at ambient room conditions of $32.5\,^{\circ}\mathrm{C}$ and 45% relative humidity (RH) with an inlet velocity of $0.4\ \mathrm{m \cdot s^{-1}}$ and an inlet *UI* at zero.

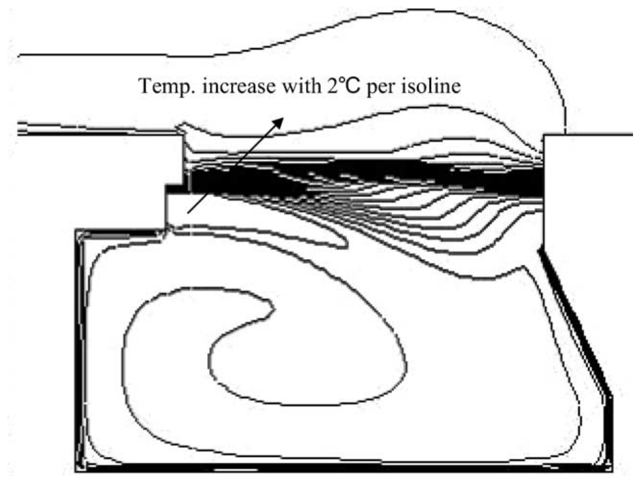

Fig. 12. Temperature isoline diagram in the display case (ambient temperature $= 32.5\,^{\circ}\mathrm{C}$, RH $= 45\%$, $UI = 0$, and average velocity $= 0.4\ \mathrm{m \cdot s^{-1}}$).

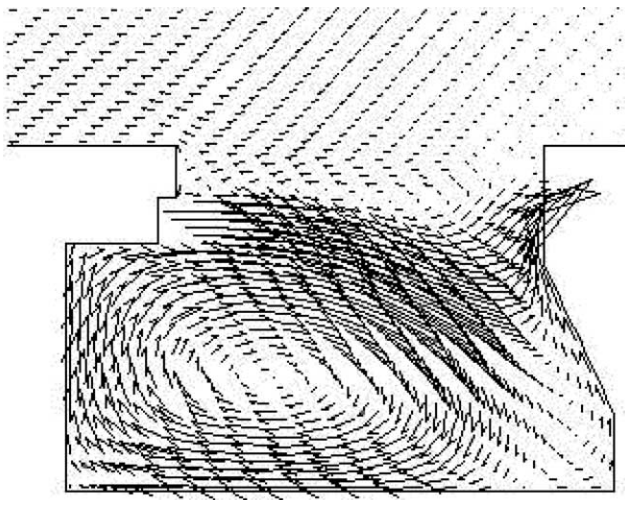

Fig. 13. Velocity vectorgraph in the display case (ambient temperature $= 32.5\,^{\circ}\mathrm{C}$, RH $= 45\%$, $UI = 0$, and average velocity $= 0.4\ \mathrm{m \cdot s^{-1}}$).

Comparing Fig. 10 with Fig. 12 and Fig. 11 with Fig. 13, it can be concluded that the principal axial of the air curtain jet straightens, the core of the jet enlarges and the internal circumfluence area expands with an increase in the inlet velocity of the air curtain, while the inlet *UI* remains constant at zero. Meanwhile, the proportion of forced convection in the mixed flow of the curtain increases gradually, and natural convection decreases. Hence, it is inevitable that the heat and mass transfer through the infiltration of the air curtain will increase, which will finally lead to an increase in the cooling load of the display. Therefore, a relatively small inlet velocity is recommended for the design of the air curtain of the display case. However it is very difficult to satisfy the demand of the design if the inlet velocity is too small, because the excessively small initial momentum cannot ef-

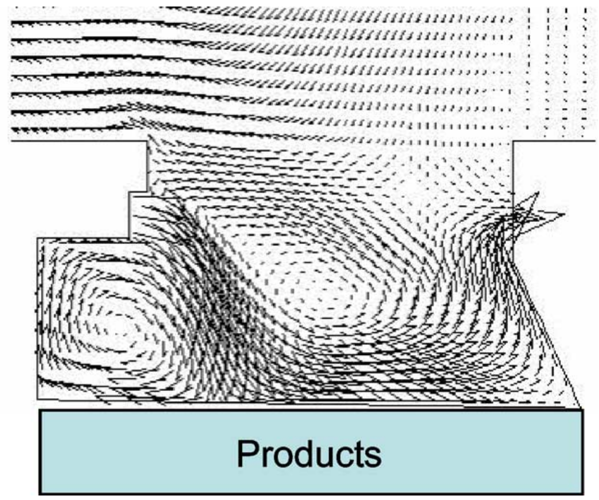

Fig. 14. Velocity vectorgraph—partially filled (ambient DB temperature $= 32.5\,^\circ\mathrm{C}$, RH $= 45\%$, $UI = 0$, and average velocity $= 0.26\ \mathrm{m\cdot s^{-1}}$).

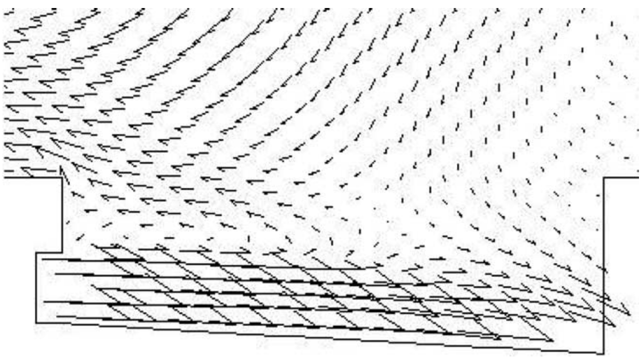

Fig. 15. Velocity vectorgraph—fully filled (ambient DB temperature $= 32.5\,^\circ\mathrm{C}$, RH $= 45\%$, $UI = 0$, and average velocity $= 0.26\ \mathrm{m\cdot s^{-1}}$).

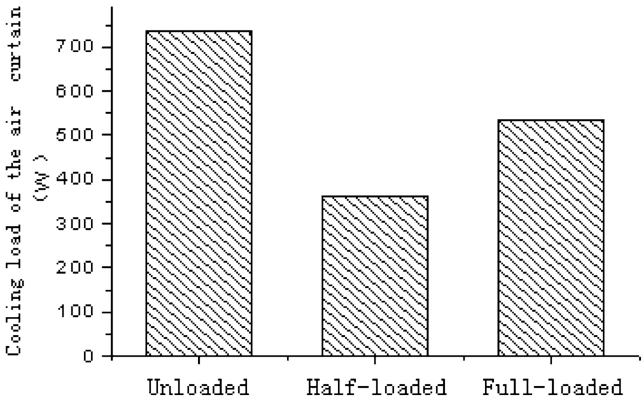

Fig. 16. Cooling load of the air curtain with different filling levels.

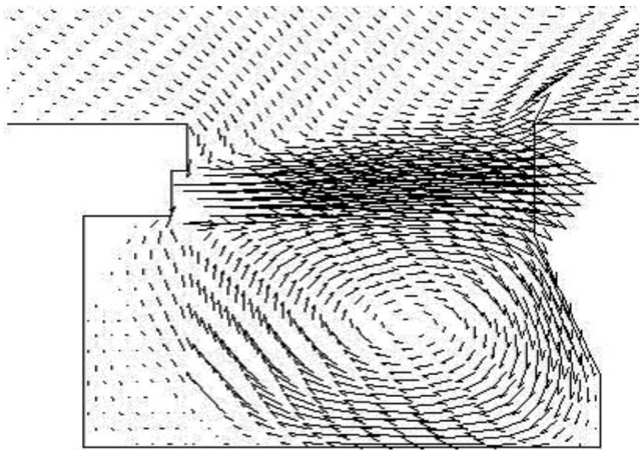

Fig. 17. Velocity vectorgraph in the display case when $T_h - T_c = 0$ (compressor shut down, ambient DB temperature: $32.5\,^\circ\mathrm{C}$, RH: 45%, UI: 0, average velocity: $0.26\ \mathrm{m\cdot s^{-1}}$).

fectively prevent the penetration of hot air from outside and ensure enough internal circumfluence area for storing products. From the above discussion, it can be inferred that there is an optimum value for the inlet velocity of the air curtain although its actual value depends on the design and operating conditions of a display case, and need to be further investigated.

4.3. Height and shape of products inside the case

The shape and height of the products in a display case change the flow of the air curtain and thereby change the air curtain's mechanism of heat and mass transfer. By comparing Figs. 11 and 14, it can be seen that the principal axial of the air curtain will rise, and the principal axis of the air curtain will shorten when the height of product inside the display case increases. As a result, it can be assumed that the heat transferred through the air curtain decreases, which can finally lower the cooling load of the display case. When the height of product is close to the inlet height of the air curtain (Fig. 15), the flow of the air curtain is almost horizontal, and the principal axis is shorter than ever. However, here, the heat and mass transfer through the air curtain is strong and the cooling load of the air curtain of the display case is large. Fig. 16 provides the cooling load of the air curtain of the display case with different product loads (calculated by Eq. (3)). Obviously, it can also be inferred that there is an optimum value for the height of products in the display case.

Moreover, another matter of great concern is that the shape of the product in the display case can affect the distribution of temperature and velocity in the display case. For example, stacking products in a convex shape can uplift the air curtain and shorten its principal axis, which will undoubtedly reduce the effects of the ambient hot air on the air curtain and, consequently, lower its cooling load.

4.4. Temperature difference between the air curtain and the ambient hot air

Fig. 17 illustrates the vectorgraph in the display case when the temperature difference between the air curtain (T_c) and the ambient hot air (T_h) is set to be zero. When compared with Fig. 11, it can be seen that the temperature difference affects the flow characteristics of the air curtain and then its energy performance. The reasons are provided as follows:

(a) When the temperature difference, $T_h - T_c$, decreases, the principal axis of the air curtain will uplift and shorten due to the decrease of the gravitational acceleration. The internal circumfluence area expands significantly. The total mass and heat infiltration may decrease due to the shrinking of the infiltration area, and vice versa.

(b) Since the temperature difference, $T_h - T_c$, is the driving force of transfers of heat and mass between the cold air curtain and the hot ambient air, the difference determines the cooling load of the air curtain to a great extent. The smaller the temperature difference is, the lower the cooling load of the air curtain.

Consequently, during the design of the air curtain, it is recommended that $T_h - T_c$ should be kept as small as possible. There are two ways to achieve this goal. One is to set a higher inlet air curtain temperature on condition that the inlet temperature is low enough to ensure the quality of the products. The other is to use the store air-conditioning system to lower the indoor air temperature, i.e., T_h, to a suitable degree. However, the set-points of the air curtain inlet temperature and the indoor air temperature should be determined from the total energy point of view and not only from minimizing the display case energy consumption.

4.5. Relative humidity of the ambient hot air

When the cold air jets out of the inlet, it will meet the ambient hot air and cause the water vapor in the hot air condense, which discharges a great amount of latent heat to the air curtain. Along with the condensation, a great amount of latent heat is added to the cooling load of the air curtain. The higher the relative humidity of the outside hot air and the colder the air curtain, the greater the latent heat load is. The infiltration load has two components: a sensible load and a latent load. The former refers to the direct temperature-driven heat added to the display case. The latter refers to the heat content of the moisture added to the display case by the air of the room drawn into the case through the air curtain. Because the circulated air in the air passage is dehumidified when it passes through the evaporator, the main source of latent load for a display case is the moisture content of the ambient air that is entrained into the display case across the air curtain.

It is indicated in [2] that as far as single-shelf display cases for dairy products is concerned, when the relative humidity of the ambient hot air increases from 25 to 65%, the relative coefficient of latent heat, defined as the ratio of the current latent heat to the latent heat at standard relative humidity (the standard relative humidity is 60%), will increase from 0.011 to 1.167, and the relative coefficient of the cooling load, defined as the ratio of the current cooling load to the cooling load at standard relative humidity, will increase from 0.582 to 1.14.

When the relative humidity is reduced from 55 to 35%, display cases can save 20 to 30% in compressor energy, 40 to 60% in defrosting energy, and 19 to 73% in anti-sweat heater operation [13]. Meanwhile, the operating costs of the store air-conditioning system will increase by 4–8%. Therefore, the relative humidity of ambient hot air has a great influence on the cooling load of display cases. The store air-conditioning system can pre-dehumidify indoor air by special measures. 'Transferring' the latent load from the display cases to the store air-conditioning system can save energy [1]. Furthermore, this load assignment can reduce the demand on the defrosting and anti-condensate operation.

In order to reduce the influence of the relative humidity of ambient hot air on the air curtain, the following measures can be adopted:

(a) On the condition that the required temperature of the air curtain and the ambient hot air be kept, the temperature difference between the air curtain and the ambient hot air should be reduced to as low a level as possible.
(b) Independent dehumidification fixtures should maintain a proper level of relative humidity in the ambient hot air to reduce the energy consumed by the display cases and the store air-conditioning system.
(c) In the evening, baffles are recommended for display cases. Engineers and display case manufacturers have concluded that reflective aluminum baffles can prevent 70% of the heat gained when a refrigerated display case is opened [14].

5. Conclusions

The air curtain is the main factor influencing the overall performance of horizontal display cases and energy consumption in typical supermarkets. Using the experimental results, the established two-dimensional CFD models and algorithms have been validated. Assisted by the CFD models, the key factors affecting the energy performance of the air curtain, i.e., the inlet *UI* of the air curtain, the inlet velocity of the air curtain, the height and shape of the products inside, the temperature difference between the inlet of the air curtain and the ambient hot air and the relative humidity of the ambient hot air, have been analyzed and discussed in.

The inlet *UI* and velocity magnitude of the air curtain determine the characteristics of the initial momentum and then affect the development of the air curtain. A lower inlet *UI* can effectively shorten the principal axial of the air curtain and reduce its cooling load. There is an optimum value for the inlet velocity of the air curtain, while other designs remain unchanged. With respect to the stacking of products in display cases, there is also an optimum value for the height of the product. Furthermore, the air curtain is heavily affected by both the inlet air temperature and the RH of ambient air. The larger the temperature difference or the RH difference between the inlet air and the ambient air, the more intensive will be the heat and mass transfer between the air curtain and the ambient air. Therefore, properly controlled indoor thermal conditions (dry-bulb temperature and RH) could well balance the cooling load of the store against that of the display cases and help achieve overall energy efficiency.

From the experimental and analytical studies, it could be concluded that the CFD method presented is an effective and feasible tool for the optimal design and performance evaluation of air curtain of horizontal refrigerated display cases.

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