Handbook of Research on Advances and Applications in Refrigeration Systems and Technologies

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Chapter 16
Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers: Application to Refrigerated Display Cases

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ABSTRACT

This chapter reports an overview about experimental studies concerning the thermal performance of air curtains and heat exchangers installed in vertical open refrigerated display cases. The air curtain analysis shows the influence on the thermal performance by varying the width of the discharge air grille and the perforation density of the back panel by a mathematical model. The variation on the perforation density of the back panel and the width of discharge air grille alter significantly the thermal entrainment factor and the energy consumption of the equipment. Focusing the influence of environmental conditions on the performance of the heat exchanger, a second mathematical model was developed to evaluate the total heat load, its partial components and the condensate water mass. This analysis provides valuable information to the design of the air curtain and heat exchanger based on in-store environmental conditions and airflow efficiency.

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INTRODUCTION

In order to preserve food for a longer period, man has developed cooling forms since prehistoric times. The effects of low temperatures on food preservation were already known before 2000 B.C. The older methods of producing cold made use of natural ice or mixtures of salt and snow. Later, the mechanical refrigeration cycle has emerged consisting essentially of four main components: evaporator, compressor, condenser and expansion device.

With the use of refrigeration machines and the development of new technologies, storing and transporting perishable foods over long distances and for longer periods became possible. Therefore, commerce and distribution of perishable food began to increase. Perishable foods, since production to the final consumer, are maintained and distributed through the system called as cold chain, which is composed of five main links (Rigot, 1991).

• Cold production
• Cold storage
• Refrigerated transport
• Cold distribution
• Domestic cold.

The fourth link in the cold chain, which will be described along this chapter, is commonly referred as commercial refrigeration. Refrigerated and frozen products are separate on the production phase and follow in environments with different temperatures through the storage, transportation and distribution. At the distribution phase, foods are expose for sale at convenience stores, small and large supermarkets through refrigerated equipment of various designs and levels of temperature according to the industry regulations and the growing demand for quality by consumers. The refrigerated or freezing display cases can be open or closed to the external environment.

Generally, refrigerated display cases are designed to operate in supermarkets that have ambient air conditioning. The global industry considers climatic condition of 25 °C with 60% of relative humidity (climate class n.º 3 (ISO 23953-2, 2005) as the standard to certify this type of equipment - critical summer condition).

Refrigerated display cases can be classified based on several criteria, however the standard ISO 23953-1 (2005) divides them into vertical, semi-vertical, horizontal and combined; which may be open or closed to the store environment; for self-service or not. The specifications of each display case type are:

(1) Vertical refrigerated display cases with multiple shelves have height exceeding 1.5 m. Semi-vertical refrigerated display cases are whose overall height does not exceed 1.5 m and the display opening may be vertical or inclined. Horizontal refrigerated display cases have horizontal display opening on its top. Open refrigerated display cases (ORDC) are those that access to the products takes place in a direct way without the need of opening doors or lids. Closed refrigerated display cases (CRDC) has the access to the products by opening of a door or lid. Combined refrigerated display cases are those that combine at least two of the above characteristic. Each type of display case has its characteristic operating temperature, from preserving ice cream, frozen foods, fresh meat, dairy/deli to fruits/vegetables.
The ORDC are more sensible to changes in environmental conditions. According to Faramarzi (1999) and Gaspar et al. (2011a), the ambient air infiltration load corresponds to 67%–81% of the total thermal load of this type of display case. This is due to the low efficiency of the air curtain, which forms a physical barrier between the internal and the external environment. The relative contribution of each heat load in an ORDC is shown in Figure 1.

The energy consumption reduction due to infiltration depends directly of the cool airflow inside the ORDC. In the case of CRDC, infiltration is proportional to the frequency of door opening by the consumers and the accumulated time for product replacement.

There is a global tendency to use closed refrigerated or freezing display cases. The use of CRDC with glass doors is already a reality in countries with tight control of energy consumption and food safety. However, in other countries, this type of equipment still is restricted to large retailers. The cost of energy and competitiveness, should pull new technologies to less developed countries with the use of LED lighting, electronic motors and CRDC.

Therefore, the performance and energy efficiency optimization of refrigeration equipment is very important. This optimization is conditioned by several factors, so, it is essential to ensure that the means and instruments of calculation and design are appropriate to improve the energy performance ensuring proper storage of perishable foods and the lowest environmental impact. The following sections describe several experimental studies developed to improve the thermal performance of vertical ORDC (VORDC) through the improvement of the air curtain and of the airflow through the evaporator.

**GENERAL CONCEPTS**

**Introduction**

The following sections describe a set of works experimental for the cold air jet that form the thermal barrier between the external and internal environment for this type of equipment. The methodology is
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applicable to ORDC and CDRC, whereas the airflow is subject to the same physics laws. This section provides an overview concerning the fundamental concepts about thermal performance and airflow in this type of equipment.

Problem

ORDC are used extensively in convenience stores and supermarkets in order to display perishable products meeting the criteria of food security. In Brazil, there are about 81 thousand stores using open or closed refrigerated display cases for products sale.

The VORDC is the model that consumes more energy. About 81% of its total consumption is due to the infiltration of outside air through the air curtain. The air curtain performance can be influenced by several factors, among which, the customers’ movement near the equipment frontal opening causing a perturbation in the air jet and increasing its energy consumption.

Physical Characteristics

VORDC typically comprises (1) an insulating body (IB) surrounding all the equipment; (2) tube and fins heat exchanger (HX); (3) discharge air grille (DAG); (4) return air grille (RAG); (5) perforated back panel (PBP) and shelves (SH) as shown in Figure 2 (a). This type of equipment is used to display chilled products for sale in a more pleasant and convenient way for consumers. The temperature of the refrigerated compartment is provided by the cold air mass flow that exits DAG and PBP and returns to RAG to be cooled again in the HX. The airflow exiting DAG forms an air curtain, which protects the inner refrigerated compartment.

The specific device tested has four fans with 53 W each to supply a flow rate of 0.4 m³s⁻¹ to DAG and to the PBP. The air, before reaching the DAG, passes through an HX with dimensions 2.20x0.13x0.35 m³ constituted by 222 fins and three rows of tubes in the airflow direction and 8 rows of tubes perpendicular to it. The dual DAG has a total width, \( b \), of 140 mm, which is equally distributed to form the Primary Air Curtain (PAC) with a width of \( b_{\text{PAC}} = 70 \text{ mm} \) and the Secondary Air Curtain (SAC) \( b_{\text{SAC}} = 70 \text{ mm} \).

The air curtain is the protective thermal barrier between internal and external environments. Internal temperatures can reach -25°C while the outside can reach 40°C, that is why the thermo-aerodynamic flow needs to be sufficient efficient to minimize the thermal interaction between environments. Figure 2 (b) shows an example of the visualization of the air curtain flow using fog.

The critical time for VORDC is the doors opening either by the shoppers or by the merchandise stockers. The higher is the frequency of doors openings, higher will be the interaction with the external ambient. The air curtain needs to be sufficient efficient to maintain the refrigeration at a minimum level while the door remains open and re-establish the refrigeration at a maximum level when the doors are closed.

Physical-Mathematical Formulation

The ORDC total heat load is the sum of several partial loads, which can be described as follows (Faramarzi, 1999):

- **Transmission Load**: Heat conduction through walls and convection load of the ORDC, \( Q_{\text{TRANS}} \);
- **Radiation Load**: Thermal radiation exchange with the surrounding surfaces, \( Q_{\text{RAD}} \);
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Figure 2. Vertical open refrigerated display case

- **Pull-Down Load**: Decrease of the initial temperature of food products, $Q_{PD}$;
- **Defrost Load**: Heat dissipated by the defrost mechanism, $Q_{DEF}$;
- **Anti-Sweat Heaters Load**: Heat dissipated by the anti-sweat heaters, $Q_{ASH}$;
- **Internal Loads**: Heat dissipated by motors and lights, $Q_{MOT} + Q_{LGT}$;
- **Infiltration Load**: Thermal entrainment through the air curtain (ORDC) or during door opening (CRDC), $Q_{INF,SEN} + Q_{INF,LAT}$;
- **Products Respiration Load**: Heat dissipated by the respiration of horticultural products, $Q_{PB}$.

The total heat load, $Q_{TOT}$, is described by the sum of sensible and latent heat loads as expressed in Equation 1 and Equation 2:

$$Q_{TOT} = Q_{SEN} + Q_{LAT}$$  \hspace{1cm} (1)

$$Q_{TOT} = \left( Q_{TRANS} + Q_{RAD} + Q_{PD} + Q_{DEF} + Q_{ASH} + Q_{MOT} + Q_{LGT} + Q_{INF} \right)_{SEN} + \left( Q_{INF} + Q_{PB} \right)_{LAT}$$  \hspace{1cm} (2)
The total heat load can be determined experimentally. According to ISO 23953-2 (2005), the total heat load should be calculated with experimental data of refrigerant mass flow, \( \dot{m}_c \), and enthalpy difference between the heat exchanger (evaporator) output, \( h_{c,\text{OUT}} \), and input, \( h_{c,\text{IN}} \), according to Equation 3.

\[
Q_{\text{TOT}} = \dot{m}_c \cdot (h_{c,\text{OUT}} - h_{c,\text{IN}})
\]  

(3)

The refrigerant circulating inside the heat exchanger absorbs the total heat load. The cooling load can be determined by summing all heat load components.

**Components of Cooling Load**

**Transmission Load**

The conduction heat transfer that occurs through the walls of the equipment and by the convection heat transfer between the inner and outer surface and it respective surrounding air composes the transmission load. This is the smaller portion of the total cooling load since the walls are constructed with insulating materials. The heat gain due to conduction is given by Equation 4.

\[
Q_{\text{TRANS}} = U \cdot A \cdot (T_{\text{amb}} - T_{\text{ORDC}})
\]  

(4)

where,

- \( Q_{\text{TRANS}} \) : Transmission heat transfer, [W];
- \( U \) : Overall heat transfer coefficient, [Wm\(^{-2}\)K\(^{-1}\)];
- \( A \) : Area of thermal exchange, [m\(^2\)];
- \( T_{\text{amb}} \) : Room ambient air temperature, [K];
- \( T_{\text{ORDC}} \) : Display case temperature, [K].

**Radiation Load**

The heat gain due to thermal radiation is a function of the conditions of the internal surfaces of the equipment (temperature, area and emissivity) and external surfaces (walls, floor, ceiling and other objects and their corresponding areas and emissivities) as well as of the view factor between internal and external surfaces. The thermal radiation exchange can be modelled as two gray surfaces, one surface being the total surface area of the surroundings (walls, floor, and ceiling) and the other being an imaginary plane that covers the opening of the ORDC. All the radiation leaving the surroundings surface arrives at the imaginary plane. This imaginary plane at the ORDC opening will exchange the radiation reaching it with the interior surfaces of the ORDC. The thermal radiation load can be determined with the Equation 5 in its simplest way.
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\[
Q_{RAD} = \frac{\sigma \left( T_{SURR}^4 - T_{ORDC}^4 \right)}{1 - \varepsilon_{SURR}} + \frac{1}{A_{SURR}} + \frac{1 - \varepsilon_{ORDC}}{\varepsilon_{ORDC} A_{ORDC}}
\]  \tag{5}

where,

\(Q_{RAD}\): Heat gain due to radiation, [W];

\(\sigma\): Stefan-Boltzmann constant \((\sigma = 5.670373 \times 10^{-8} \text{ kg s}^{-3} \text{ K}^{-4})\);

\(\varepsilon\): Emissivity;

\(A\): Area, [m²];

\(F\): View factor from the ORDC to surroundings walls/objects.

**Pull Down Load**

The pull-down load comprises two components, (1) when the product is stocked on the ORDC at higher temperature than the refrigeration temperature, \(Q_{PD,STK}\), and (2) immediately after the defrost time when products temperature increase above the refrigeration temperature, \(Q_{PD,DEF}\). The heat load are given by Equation 6 and Equation 7.

\[
Q_{PD} = Q_{PD,STK} + Q_{PD,DEF}
\]  \tag{6}

\[
Q_{PD} = \frac{m_{STK} \cdot c_{p,STK} \cdot (T_{STK} - T_{REF})}{\Delta t} + \frac{m_{P} \cdot c_{P,P} \cdot (T_{Pot,DEF} - T_{REF})}{\Delta t}
\]  \tag{7}

where,

\(m\): food product mass, [kg];

\(c_p\): Specific heat, [J kg⁻¹ K⁻¹];

\(t\): Time, [sec];

**Defrost Load**

Moist air passing through the heat exchanger allows ice formation on the surface of the evaporator when its temperature is below the dew point temperature. Regular stops in cooling are necessary to defrost the evaporator. The defrost methods applied on commercial refrigeration can be electric, hot gas or natural. Generally, medium temperature display cases use natural defrost and low temperature display cases use electric or hot gas defrost. The heat load by electric defrost is given by Equation 8 and the heat load by hot gas is given by Equation 9.

\[
Q_{DEF,E} = Q_E - Q_{FROST}
\]  \tag{8}
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\[ Q_{\text{DEF.HG}} = Q_{\text{HG}} - Q_{\text{FROST}} \]  \hspace{1cm} (9)

where,

\( Q_E \): Heat dissipated by electric resistors, [W];
\( Q_{\text{HG}} \): Heat dissipated by hot gas (function of refrigerant mass flow and refrigerant enthalpies entering and leaving the coil), [W];
\( Q_{\text{FROST}} \): Heat required to melt the frost as a function of frost mass, heat of fusion and time, [W].

**Anti-Sweat Heaters Load**

The anti-sweat heaters are used on surfaces where the temperature is below the environmental dew point, generally heated glass or plate surfaces. The heat generated by these electrical heaters is transferred to the cold air and to the products. The anti-sweat heaters load is given by Equation 10.

\[ Q_{\text{ASH}} = k_{\text{ASH}} \cdot P_{\text{ASH}} \]  \hspace{1cm} (10)

where,

\( P_{\text{ASH}} \): Power of electric load (anti-sweat heaters), [W];
\( k_{\text{ASH}} \): Fraction of heat dissipated into air and food products, [%];

**Internal Load**

The internal loads of refrigerated display cases include the heat dissipation from fans motors and lighting that are given by Equation 11 and Equation 12.

\[ Q_{\text{MOT}} = k_{\text{MOT}} \cdot P_{\text{MOT}} \]  \hspace{1cm} (11)

\[ Q_{\text{LGT}} = k_{\text{LGT}} \cdot P_{\text{LGT}} \]  \hspace{1cm} (12)

where,

\( P_{\text{MOT}} \): Power of fan motors, [W];
\( k_{\text{MOT}} \): Fraction of heat dissipated by the fan motors, [%];
\( P_{\text{LGT}} \): Power of lighting, [W];
\( k_{\text{LGT}} \): Fraction of heat dissipated by lighting, [%];

**Infiltration Load**

The ambient air inflow to the refrigerated space through the air curtain provides sensible and latent heat to the refrigerated display case. The intensity of this interaction depends on the characteristics of the jet outflow at DAG.
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- Air profile velocity;
- Air temperature;
- Number of jets;
- Air jet thickness, width and height;
- DAG physical characteristics;
- Ambient temperature and humidity;
- Rate of air curtain perturbation (ORDC);
- Rate of door opening (CRDC).

The sensible infiltration load, \( Q_{INF.SEN} \), is from the environmental air temperature, which is cooled through the heat exchanger. The latent infiltration load, \( Q_{INF.LAT} \), is from the environmental air moisture, which condenses and freezes at evaporator surface. The sensible infiltration heat is given by Equation 13 and the latent infiltration heat is given by Equation 14.

\[
Q_{INF.SEN} = \rho \cdot \dot{V}_{RAG} \cdot c_p \cdot \left( T_{amb} - T_{ORDC} \right)
\]  (13)

\[
Q_{INF.LAT} = \rho \cdot \dot{V}_{RAG} \cdot \left( \omega_{amb} - \omega_{ORDC} \right) \cdot h_{fg}
\]  (14)

where,

\( \rho \): Air density, \([\text{kg m}^{-3}]\);

\( \dot{V}_{RAG} \): Volumetric flow rate of air entrained through RAG, \([\text{m}^3 \text{s}^{-1}]\);

\( c_p \): Specific heat of air, \([\text{J kg}^{-1} \text{K}^{-1}]\);

\( \omega \): Humidity ratio, \([\text{kg kg}^{-1} \text{air}]\);

\( h_{fg} \): Latent heat of vaporization of water, \([\text{J kg}^{-1}]\)

**Products Respiration Load**

The products respiration (fresh fruits and vegetables) is another component of the latent load. This type of products lose moisture by respiration. The respiration latent load is given by Equation 15.

\[
Q_{pq} = \dot{m}_v \cdot A_p \cdot n \cdot h_{fg}
\]  (15)

where,

\( \dot{m}_v \): Mass transfer rate of water vapour leaving the product’s skin, \([\text{kg s}^{-1} \text{m}^{-2}]\);

\( A_p \): Surface area of the product, \([\text{m}^2]\);

\( n \): Number of products.
Dimensionless Parameters to Optimize the Air Curtain

The air jet from DAG deflect to the cooler side, in this case, inside the display case. Hayes & Stoecker (1969) developed a correlation that describes the ability of the air curtain to provide a proper separation between environments. The correlation is given by a dimensionless parameter named as deflection modulus, $D_m$, which is the ratio between the air curtain momentum and the modulus of the transverse forces caused by temperature difference between the contiguous environments (see Equation 16).

$$D_m = \frac{(\rho \cdot b \cdot \nu^2)_{DAG}}{g \cdot H^2 \cdot (\rho_{ORDC} - \rho_{amb})}$$

(16)

where,

$b$: DAG width, [m];
$
u$: Air velocity at DAG, [m s$^{-1}$];
$g$: Gravity, = 9.81 [m s$^{-2}$];
$H_j$: Height of the air curtain, [m].

Chen et al. (2005, 2009, 2011) developed studies using Computational Fluid Dynamics (CFD) codes to evaluate the thermo-physical parameters of the air curtain in ORDC. The performance of air curtain was evaluated by the following dimensionless numbers/parameters: Reynolds number, Grashof number, Richardson number and dimensionless temperature, given by Equation 17 to Equation 20 respectively, for different aspect ratios (height/width) of the air curtain.

Reynolds number on the discharge air grille:

$$Re = \frac{v_{DAG} \cdot H}{\nu}$$

(17)

where,

$\nu$: Kinematic viscosity, [m$^2$ s$^{-1}$];

Grashof number on the discharge air grille:

$$Gr = \frac{g \cdot \beta \cdot (T_{DAG} - T_{amb}) \cdot H^3}{\nu^2}$$

(18)

where,

$\beta$: Volumetric thermal expansion coefficient, [K$^{-1}$];
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\[
\text{Ri} = \frac{\text{Gr}}{\text{Re}^2} = \frac{g \cdot \beta \cdot (T_{\text{DAG}} - T_{\text{amb}}) \cdot H}{v_{\text{DAG}}^2} \tag{19}
\]

Dimensionless temperature:

\[
\theta = \frac{T - T_{\text{ref}}}{T_{\text{amb}} - T_{\text{ref}}} \tag{20}
\]

The results provided the following conclusions: an optimal thermal insulation developed by the cold air curtain jet is obtained for small height/width ratios yield that increase the critical Richardson numbers (thus decrease critical Reynolds numbers) and reduce the volumetric infiltration rate. As the Grashof number provides the fluctuation proportion of the buoyancy force that acts on a viscous fluid in situations involving heat transfer by natural convection while the Richardson number provides the information on the influence of natural convection in relation to forced convection, it is possible to use them to describe the flow. Thus, it can be stated that air curtains with small height/width ratio provide a good thermal performance.

Navaz et al. (2005) developed further studies using Digital Particle Image Velocimetry (DPIV), focusing mainly on the study of the effectiveness of the air curtain maintaining the temperature of food products to a predetermined value. The results indicate that the Reynolds number has direct effect on the ambient air entrainment into the refrigerated equipment due to its role in the turbulence development. According to Navaz et al. (2005), the best range of values for the Reynolds number in DAG is in the range 3200-3400. In that study, the authors defined the Thermal Entrainment Factor, TEF, to quantify the thermal entrainment of the air curtain with the ambient air, ranging between 0 < TEF < 1. The TEF is calculated based on the temperature values of DAG, RAG and ambient air, as shown in Equation 21. Gaspar et al. (2011a) used enthalpies instead of temperatures on Equation 21 to account the influence of the relative humidity on the thermal entrainment.

Thermal Entrainment Factor (TEF):

\[
\text{TEF} = \frac{T_{\text{RAG}} - T_{\text{DAG}}}{T_{\text{amb}} - T_{\text{DAG}}} \tag{21}
\]

The analysis of Equation 21 shows that as closer to 0 is TEF, lower is the thermal entrainment with the ambient air. The correlation described by Navaz et al. (2005) does not take into account the airflow through the perforated back panel (PBP). Yu et al. (2009) developed the TEF equation considering this airflow component. The new correlation is given by Equation 22 to Equation 25.

The Thermal Entrainment Factor considering the PBP airflow is given by Equation 22:

\[
\text{TEF} = (1 - \beta) \cdot X - \beta \cdot X \cdot X_{\text{PBP}} \tag{22}
\]

where,

\[X: \text{Thermal entrainment factor for air curtains without considering the PBP airflow (Equation 21);}\]
The airflow ratio between PBP and DAG is given by Equation 23:

\[
\beta = \frac{\dot{m}_{PBP}}{\dot{m}_{PBP} + \dot{m}_{DAG}}
\] (23)

The dimensionless temperature with PBP (thermal entrainment factor for PBP airflow) is given by Equation 24:

\[
X_{PBP} = \frac{T_{PBP} - T_{DAG}}{T_{amb} - T_{DAG}}
\] (24)

The results obtained by Yu et al. (2009) show a good approximation for TEF and temperature value at the RAG with deviations of 0.9% and 0.1 °C respectively. These deviations indicate that the correlation has a good approximation at the engineering level and can be applied in the design of VORDC.

Gaspar et al. (2007, 2008, 2009, 2010a, 2010b, 2011a, 2011b) evaluated the stability of the air curtain for climatic classes n.º 1, n.º 2 and n.º 3 according to EN-ISO 23953 (2005) and other classes beyond the standard. The evaluation was made by experimental testing and numerically using CFD models. The results showed that the equipment performance strongly depends on the ambient air conditions such as temperature, humidity, velocity and direction of ambient airflow in relation to the display case frontal opening.

These authors showed that (1) the cooling load increases with the air temperature and relative humidity of the external environment; (2) the increase of the ambient air velocity increases more significantly the power consumption of the equipment than the airflow direction change from parallel to perpendicular in relation the frontal opening of the VORDC; (3) the magnitude of deflection modulus \( D_m \) related with minimum momentum required to maintain a stable curtain of air is between 0.12 and 0.25; (4) the cooling load due to air infiltration is 78% - 81%, which range is closer to the value (73.5%) obtained by Faramarzi (1999); (5) Due to the influence of relative humidity, TEF should be determined by dimensionless enthalpy differences than by dimensionless temperature differences; and (6) TEF is not constant along the equipment length for parallel air flow. Furthermore, the TEF value increases when the ambient airflow goes from parallel to perpendicular, being the worst case for \( \theta_{amb} = 45^\circ \). In the case study, \( TEF = 0.25, 0.32, 0.3 \) for \( \theta_{amb} = 0^\circ, 45^\circ, 90^\circ \) respectively.

Laguerre et al. (2012) developed a simplified analytical model based on heat transfer equations to determine the values of air and product temperatures at various locations of an ORDC. The heat gain by radiation is more significant for products located on the front (top and bottom) and the heat gain by air infiltration is more significant for the products located in the rear (front and rear). Cao et al. (2010, 2011) developed a new strategy for conception and optimization in the air curtains design for VORDC. The strategy is based on the heat transfer model between two fluids (two-fluid of cooling loss - CLTF) developed based on a Support Vector Machine (SVM) algorithm. Mouset and Libsig (2011) developed the correlation described by Equation 25 that quantifies, for any ambient air condition, the cooling load increment relatively to the cooling load in the climate class n.º 3 (\( T_{amb}=25^\circ \mathrm{C}; \phi_{amb}=60\% \)) of ISO 23953 (2005).
\[ \Phi_x = \Phi_3 \frac{h_x}{h_3} \]  

(25)

where:

\( \Phi_x \): Heat extraction rate for the ISO climate class \( x \), [W];

\( \Phi_3 \): Heat extraction rate for the ISO climate class \( n.º 3 \), [W].

\( h_x \): Enthalpy of the humid air determined using the air temperature and relative humidity of the class \( x \), [J kg\(^{-1}\)];

\( h_3 \): Enthalpy of the humid air determined using the air temperature (25°C) and relative humidity (60%) of the climate class \( n.º 3 \), [J kg\(^{-1}\)].

Conclusive Remarks

With the description of the state of the art on the research concerning fluid flow and heat transfer in refrigerated equipment, namely on ORDC, can be detected a strong trend to analyse and quantify the thermo-physical parameters of the flow of cold air (air curtain) separating two environments. Thus, the mathematical equations provided along this section are the basis for the development of equipment that provide a better performance, and thus ensuring the food safety, while reducing the energy consumption.

EXPERIMENTAL STUDIES WITH REFRIGERATED DISPLAY CASES

In the following sections, some experimental studies with the aim of improving the thermal performance and energy efficiency of VORDC will be presented. These experimental studies focus mainly on the efficacy of the air curtain and analysis of the airflow distribution on the heat exchanger. The studies are developed for different ambient air conditions and following several standards. The experimental tests involved in the different studies made use of the following facilities, systems, devices and measuring equipment:

Climatic Chamber

The experimental tests were conducted in an environmental test room constructed according to ISO 23953 (2005) as shown Figure 3. The details about the devices and equipments used to control the environment conditions inside the test room are shown in Figure 4.

The VORDC tested has dimension of 1.10m×2.62m×2.12m and possesses a dual air curtain as shown in Figure 5. The equipment is used for chilling and displaying meat products, which temperature should be maintained between -1 °C and 5 °C (ISO 23953-2:2005 class M1).
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Figure 3. Test room (A=0.98m; B=1m; C=0.65m; X=2m; Y=1.85m) (EN ISO 23953-2, 2005)

Figure 4. Environmental test chamber
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Figure 5. Main dimensions and details of the VORDC under testing: (a) frontal view; (b) lateral view

Measuring Devices

In DAG, RAG and ambient were placed temperature and humidity sensors Super MT 530. Temperature sensors type PT1000 were placed in the test packages (product simulators). A Coriolis flowmeter MASSFLO 2100 DI 6 was installed at the liquid refrigerant line. Table 1 shows the experimental techniques and probes/experimental measuring devices used to collect the relevant physical properties. Additionally, the measuring range and accuracy of the devices is included.

General Experimental Testing Procedure

Each test period lasted a minimum of 24 hours after steady state conditions inside the test room were accomplished. Data acquisition related to air temperature, relative humidity, velocity and pressure drop was conducted at 1-minute intervals in order to analyse variations of these values over time. To measure physical parameters on the R22 refrigerant side, data was collected according to the recommendations provided by ISO 23953 (2005). However, each experimental study made use of specific testing procedures that will be described hereinafter.

Table 1. Experimental techniques and probes/experimental measuring devices

<table>
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<tr>
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<th>Model</th>
<th>Measuring Range</th>
<th>Accuracy</th>
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<tbody>
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<td>Thermometry</td>
<td>test packages</td>
<td>PT 1000</td>
<td>-40°C to +80°C</td>
<td>± 0.3 °C</td>
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<td></td>
<td>DAG, PBP, RAG, Amb</td>
<td>MT 530 Super</td>
<td>-10°C to 70°C</td>
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<td>MT 530 Super</td>
<td>20% to 85%</td>
<td>± 5%</td>
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<td>0.3 m/s⁻¹ to 34 m/s⁻¹</td>
<td>± 1%</td>
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<td>Flowmetry</td>
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<td>MASSFLO 2100</td>
<td>0 to 1000 kg.h⁻¹</td>
<td>0.1%</td>
</tr>
<tr>
<td>Barometry</td>
<td>liquid refrigerant line</td>
<td>AKS 32</td>
<td>0 to 200 psig</td>
<td>± 0.3%</td>
</tr>
</tbody>
</table>
EXPERIMENTAL EVALUATION OF THERMAL LOAD DUE TO BREAKAGE OF THE AIR CURTAIN

Experimental tests were conducted in open and closed hours of a food store. The results comparison allows evaluating the thermal performance of VORDC test cases for a real scenario (Nascimento et al., 2013a). The performance benchmark of the VORDC was conducted with air temperature and humidity data collected during six months. The VORDC was installed in a store with 532 m² of sales area without air conditioning. The supermarket is located in São Paulo - SP, Brazil.

Materials and Methods

Data acquisition was performed every 30 seconds, every day during 24 hours. The periods of interest for the evaluation and comparison of performance of the VORDC take into account the opening hours of the commercial establishment for consumer attendance. The air temperature and humidity values were collected near the DAG and RAG of an ORDC for packaged meat. The probes were located in the VORDC as shown in Figure 6.

Analysis and Discussion of Results

During the period that the store is open to public, the cold air curtain of the ORDC that provides a thermal sealing ability between the humid and hot ambient and the refrigerated one inside the ORDC is subject to food product handling by clients and repositories. According to Faramarzi (1999), this condi-

Figure 6. Air temperature and humidity probes location in the VORDC
tion causes the temperature difference between DAG and RAG to increase, and consequently the energy consumption. The air curtain efficiency can be evaluated by the TEF value calculated by Equation 22.

Results

The annual average variation of the air temperature and relative humidity in DAG, RAG and store environment are shown in Figure 7 and Figure 8. These figures show the annual average values collected on days when the store was closed (Store Closed - SC) and when it was open (Store Open - SO) to consumers, respectively.

Figure 9 shows the average variation of the air temperature and humidity values collected in laboratory tests (LAB) according to EN ISO 23953 (2005). The TEF values for the different scenarios as well as their linear trend are shown in Figure 10.

Figure 11 shows the percentage increase of the cooling load calculated according to Equation 25 and its linear trend.

Discussion

The cooling load increase due to the air curtain breakage is a condition accepted by all researchers, but the increased load due to the disturbance of the airflow caused by consumers and stockers is a small and difficult component to measure in field tests. As shown in Figure 7 and Figure 8, the period in when the temperature difference between DAG and RAG is greater coincides with the period when the ambient air temperature increases. Figure 9 shows the air temperature variation of test conducted according to EN ISO 23953 (2005). In Figure 9, it is shown a cyclic operation of the VORDC, period to period, since the environment of the test room is controlled. With these results, it is concluded that to assess the air curtain perturbation due to the extraction of food items and movement of people in the store near the frontal opening of the VORDC, it is necessary to develop standard tests in a controlled environment.

Figure 7. Daily average air temperature and relative humidity - Case study: Store closed
Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers

Figure 8. Daily average air temperature and relative humidity - Case study: Store open

![Graph showing daily average air temperature and relative humidity for a store open.]

Figure 9. Daily average experimental results of air temperature and relative humidity - Case study: Climate class n.º 3 (25°C; 60%RH) (EN ISO 23953, 2005)

![Graph showing daily average experimental results of air temperature and relative humidity for a climate class n.º 3.]

In Figure 10 it is possible to evaluate the air curtain performance by the TEF values. There are fluctuations in the TEF values when the VORDC is exposed to real conditions in a store without air conditioning, either closed or open. Additionally, the linear trend of TEF increases when the store is open to public. The period when the TEF value is greater also coincides with the period when the temperature value is higher.
Figure 10. TEF values and its linear trend for the different case studies: Store closed (SC), Store open (SO) and laboratory (LAB)

Figure 11. Percentage increase in the cooling load of case study: Store closed (SC) and Store open (SO) in relation to case study: Laboratory (LAB)

Figure 11 shows the percentage increase of cooling load when the store is open and closed to public. The two curves show oscillations throughout the period. When the store is opened to consumers, the cooling load increases 20% in relation to the value determined by laboratory test conducted according to EN ISO 23953 (2005). In both case studies, store closed or open, the cooling load increases during the period. Thus, the power consumption of the VORDC is very dependent on ambient air conditions and the use of HVAC systems to control the environment of the store or cold foods sector is crucial for reducing the cooling load.
Conclusive Remarks

In Brazil, due to its territorial extension, there are very significant different climate regions. Even with this climate difference, small supermarkets or store, mostly do not have any kind of air conditioning system. The store chosen for the development of this work is located in the Southeast of Brazil. With these results was possible to evaluate and compare the performance of a vertical open refrigerated display case, operating in a controlled environment and in situ operation. From the results analysis, it can be stated that the environment conditions, air temperature and humidity, increase significantly the energy consumption of open refrigerated display cases and it is possible to save energy when the environment is controlled. Additionally, the evaluation of the air curtain performance due to the extraction of products and customer movement inside the store and particularly in the front of the equipment’s opening is only possible when the environment is controlled.

EXPERIMENTAL STUDY OF THE INFLUENCE OF THE AIR CURTAIN THICKNESS AND POROSITY OF THE BACK PANEL ON THERMAL PERFORMANCE

This section reports the results of experimental tests in the VORDC to assess the impact on the thermal performance by varying the width of the discharge air grille (DAG) and the perforation density of the perforated back panel (PBP). The experimental laboratory tests were conducted for climate class n.° 3 ($T_{amb} = 25^\circ C$ and $\phi_{amb} = 60\%$).

Materials and Methods

Experimental tests (ET) were performed for initial evaluation of the VORDC. The PBP airflow ratio, $\beta$, is calculated by Equation (23) considering constant air density. Considering this assumption and since the characteristic length of DAG influence the TEF and overall energy consumption, experimental tests were conducted to four different levels of total airflow, i.e. to four different levels of mass airflow distributed by the primary air curtain (DAG$_{PAC}$) and PBP $\dot{m} = \dot{m}_{PAC} + \dot{m}_{PBP}$. A frequency inverter connected to the fans of DAG$_{PAC}$ provides the system control. After identifying the best configuration for the PBP distribution i.e. its porosity, the fans of the secondary air curtain (DAG$_{SAC}$) were also connected. With this procedure, the air curtain thickness was doubled. The air velocity in DAG$_{SAC}$ outlet was modulated to five different levels via control of the frequency inverter connected to the fans of DAG$_{SAC}$. The PBP was initially closed with tape as shown in Figure 12 for the first four tests (ET1 to ET4). The tapes were partially withdrawn at each series of ET with four levels of fans rotation velocity. The tapes were taken by its numerical order (as shown in Figure 12) to obtain a better distribution of air over the shelves. The procedure consists in firstly to take out all tapes n.º 1, and then take out tape n.º 2, and so on. After identifying the best configuration for the airflow ratio between PBP and DAG, $\beta$, the DAG width was increased from $b_{PAC} = 70$ mm to $b_{PAC+SAC} = 140$ mm for EE17 to EE21.
Analysis and Discussion of Results

This section describes the experimental results and discuss them in order to analyse the thermal performance and the energy consumption with the different setups of airflow distribution between DAG and PBP. Additional details can be found in Nascimento et al. (2013b).

Results

The air velocity in DAG (PAC and SAC), RAG and PBP were measured with the propeller type anemometer model HTA4200 in twelve points along the opening perpendicular to flow. The values of the airflow rate for each ET are shown in Figure 13. This figure also includes the value of $\beta$ for each ET.

The air temperature and relative humidity values for each ET (ET1 to ET21) are shown in Figure 14. The air velocity measured in each ET (ET1 to ET21) for PAC, SAC and RAG locations are shown in Figure 14. This figure also includes the airflow ratio between PBP and DAG, $\beta$.

Figure 16 shows the total thermal load and the TEF values for each of the experimental tests.

Discussion

The analysis of the experimental results aim to determine an airflow distribution between the DAG and PBP that improve the thermal performance and reduce the energy consumption in comparison to commercial equipment (results provided by given by ET15). The criteria for evaluating the performance of the VORDC was, (1) lowest TEF, (2) test packages temperature below 5 °C, and (3) lowest cooling load.

The combined analysis of the tests (EE1 the EE21) results shown in Figure 13 to Figure 16 show that: (1) there is an optimal value for $\beta$, (2) increasing the DAG thickness reduces TEF and the cooling load. The values of TEF, total cooling load ($Q_{\text{tot}}$) and the average and maximum test packages temperatures ($T_{\text{sim}}$) for the EE are shown in Figure 14 and Figure 16. The maximum value of test packages temperature is measured in the well tray for all EE. $Q_{\text{tot}}$ values were determined by Equation 3 whereas the TEF values were determined by Equation 22.
In Figure 14, it is possible to see that the air curtain efficiency is a relevant factor to reduce energy consumption and maintenance of product temperature. The air velocity in DAG, RAG e β for each ET is shown in Figure 15. In EE19 was obtained a $v_{\text{DAG}} = 0.92 \, \text{m/s}$. Gaspar et al. (2011a) determined the best performance at $v_{\text{DAG}} = 1.5 \, \text{m/s}$. Cao et al. (2011) obtained the best performance at $v_{\text{DAG}} = 0.8 \, \text{m/s}$ to 1 m/s, while Yu et al. (2009) found an optimum DAG velocity from $v_{\text{DAG}} = 0.7 \, \text{m/s}$ to 0.8 m/s. Comparing the results with these obtained by other authors, we can be stated that the optimal DAG velocity depends on the physical characteristic of the air curtain and the PBP airflow ratio, so for every height/width ratio exists an optimum value for the DAG velocity.
An efficient sealing ability provided by DAG can be identified by analysing the TEF values. It is intended to obtain a more efficient equipment considering the previously defined evaluation criteria (lowest TEF, $Q_{tot}$ and $T_{sim}$), without making major changes in the current design. The best configuration shown in Figure 16 is given by EE19, where $\beta = 0.46$, considering $v_{PAC} = 0.20 \text{ m}^3\text{s}^{-1}$, $v_{SAC} = 0.11 \text{ m}^3\text{s}^{-1}$ and $V_{PBP} = 0.19 \text{ m}^3\text{s}^{-1}$. The mass flows on other ET are different due to the different pressure drop caused in the different configurations.

With the experimental study, it was possible to adjust this particular VORDC and improve its performance by reducing 10% the energy consumption (ET19) as compared to commercial design (ET15).

**Conclusive Remarks**

This section includes an experimental optimization technique for VORDC. This is a test method of great value to industry, since usually it does not have easily the computational resources or expertise to develop
CFD modelling, but it can develop experimental studies measuring the thermo-physical parameters to quantify the air curtain flow. With the experimental results it was possible to identify a value for the airflow rate and its distribution between the PBP and DAG that provides a design with better performance. The optimum values are given by the experimental test where \( v_{\text{PAC}} = 0.92 \text{ m/s} \); \( v_{\text{SAC}} = 0.72 \text{ m/s} \); \( v_{\text{RAG}} = 2.22 \text{ m/s} \); \( b_{\text{DAG}} = 140 \text{ mm} \); and \( \beta = 0.46 \). With the development of this work, it is evident that for each type of VORDC case with a height/width ratio different from the model here presented, there are optimum values for \( \beta \) and DAG and RAG velocities.

**COMPARISON OF PERFORMANCE FOR IN-SITU OPERATION AND TESTING ACCORDING TO ISO AND ASHRAE STANDARDS**

Manufacturers seek during the design phase of the equipment to certify its suitability to the testing standards with the lowest energy consumption and ensuring food safety. With the aim to assist the development of new equipment that meet the test conditions established by testing standards and market needs, experimental tests were conducted comparing real operation conditions and test conditions following the testing standards ISO 23953/2005 and ASHRAE 72-2005. The experimental results show that test conditions required by standards are stricter than *in-situ* operation, providing higher conservation temperatures (56% increase) and energy consumption (17% increase).

The experimental study considers a VORDC that demands high electrical energy consumption due to its characteristics. The tests performed in the laboratory using a test room determine approximately 17% higher thermal load values than those of *in-situ* operation with remote refrigeration system. This condition arises from the different airflow conditions that are found inside stores when comparing with the conditions and patterns imposed by standards for equipment testing. This difference has a negative impact in the commercial scope since equipment manufacturers relate lower thermal load values than those determined in tests of similar models according to standards as stated by Nascimento *et al.* (2014a).

**Materials and Methods**

**Experimental Procedures**

The calibration of the test room was performed by two different procedures, EE1 and EE2. EE1 followed the EN ISO 23953 (2005) that suggests an experimental evaluation of the test room performance to comply with air velocity from 0.1 m/s to 0.2 m/s. For tests conducted in VORDC, the standard suggests an average value of the air movement parallel to the frontal opening of the VORDC switched off set to 0.2 m/s (±10%) in each of the three air flow measuring points (P1, P2 and P3) shown in Figure 17a, with the VORDC switched off. After reaching steady state conditions, the VORDC is switched on, and the experimental test runs for 24 hours.

In EE2 was followed the method used by some equipment manufacturers. It consists in following the ISO suggestions, but with the VORDC switched on, the air movement parallel to the frontal opening of the VORDC is adjusted to meet a value of 0.2 m/s (±10%) in each of the three air flow measuring points (P1, P2 and P3). Then, after reaching steady state conditions, the experimental test runs for 24 hours.

Data was collected over 24 hours. Each EE was carried out three times to minimize uncertainty.
Numerical Study

A numerical study based on Computational Fluid Dynamics was developed to assist the calibration procedure of the air velocity in front of the VORDC. The numerical results were used to fit the airflow settings in order to ensure the same air velocity value in the three points shown in Figure 17a.

The numerical study was developed for a simplified 3D geometry of the VORDC. The geometry was developed in CAD software - SolidWorks and transferred to Gambit software to generate the computational mesh. The Fluent CFD code was used to simulate the air flow in the test room for two different settings (1) Fixed air velocity of $v_{\text{amb}} = 0.16 \text{ m.s}^{-1}$ at the inlet face of the climate chamber with VORDC switched off; (2) Fixed air velocity of $v_{\text{amb}} = 0.16 \text{ m.s}^{-1}$ at the inlet face of the climate chamber with VORDC switched on.

The 3D geometry for the CFD models closely followed the real one. It was used the automatic orthogonal unstructured mesh generator included in Gambit software. The control volume discretization of the VORDC and the external surroundings required a computational grid with 1,941,924 cells and 373,132 nodes. The models require such high number of control volumes due to the geometrical distances variations near the end walls of the equipment. This mesh refinement allowed the development of a high quality grid without high skewness levels and aspect ratios.

Nascimento et al. (2013b) obtained the experimental data of air velocity and relative humidity for the same VORDC, which were considered as boundary conditions (BC) in the numerical models. The air velocity at the test room inlet was calculated to allow an air velocity of 0.2 m s$^{-1}$ parallel to the frontal opening of the VORDC. Due to free area reduction at the lateral flow head of the VORDC and consequent air velocity increase, the air velocity at the inlet face of the test room was imposed to $v_{\text{amb}} = 0.16 \text{ m.s}^{-1}$. In EE1, the areas of DAG and RAG were considered as walls (VORDC switched off). In EE2 (VORDC switched on), the PAC mass flow rate was imposed non-uniformly along length between 0.021 kg s$^{-1}$ to 0.031 kg s$^{-1}$. The mass flow rate at SAC was imposed at a constant value of 0.027 kg s$^{-1}$. The air temperature at PAC and SAC were imposed at $T_{\text{PAC}} = 0 ^\circ\text{C}$ and $T_{\text{SAC}} = 18 ^\circ\text{C}$ respectively. The RAG area
Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers

Figure 18. Path lines of air temperature [K] for the numerical case studies with ORCD switched off

Figure 19. Path lines of air temperature [K] for the numerical case studies with ORCD switched on

was fixed to a pressure outlet BC. Figure 17b shows the vertical plane where the measured air velocity should be 0.2 m s⁻¹. The temperature and relative humidity of air at the inlet face of the test room were fixed to $T_{\text{amb}} = 25$ °C and $\phi_{\text{amb}} = 60\%$ as defined in EN ISO 23953 (2005) for climate class n.°3.

The numerical predictions of temperature path lines for EE1 (VORDC switched off) are shown in Figure 18. The predictions show that the airflow inside the test room tends to concentrate in its lower part due to the difference of air density in result of the heat generated by the fluorescent lamps mounted in the ceiling of the climatic chamber. This result indicates that the climatic chamber was empty when the airflow was calibrated and the airflow may not have the same direction when the VORDC is switched on. Figure 19 shows the temperature path lines for EE2 (ORDC switched on). In this case study, the air
flows to the upper part of the climatic chamber. Again, this condition is due to differences in air density throughout the chamber. The heat exchange between the air curtain and external air (RAG to DAG height), is predicted to increase the air temperature at the floor level close to 18 °C.

The numerical simulation was necessary for a better understanding of the airflow inside the climatic chamber and to predict the changes of air velocity value parallel to the air curtain. The numerical simula-
tions help to set the climatic chamber airflow that fits the 0.2 m s\(^{-1}\) in front of the VORDC opening, due to the difficulty to obtain experimentally a uniform flow along the plane perpendicular to the air curtain. The air velocity value that meets EN ISO 23953 (2005) was only accomplished with auxiliary ventilation: five centrifugal fans positioned on a vertical line at 1.5 m from the head of VORDC shown in Figure 20.

### Analysis and Discussion of Results

This section includes the analysis of the experimental results of tests performed considering the different initial procedures to set the external air velocity: EE1 (VORDC switched off) and EE2 (VORDC switched on).

Figure 21 shows the air velocities values during the initial procedure of setting the external air velocity for EE1. The values of the average air velocity over 7 hours are \(v_{P1} = 0.18\) m s\(^{-1}\); \(v_{P2} = 0.21\) m s\(^{-1}\) and \(v_{P3} = 0.19\) m s\(^{-1}\) in positions P1 to P3 respectively. Figure 22 shows the air velocity distribution for EE1 during the 24-hour tests to the VORDC. It is shown as increase in air velocity sensors P1 and P2 to \(\Delta v_{P1} = 0.52\) m s\(^{-1}\) and \(\Delta v_{P2} = 0.39\) m s\(^{-1}\), respectively. In sensor P3 is measured a decrease of \(\Delta v_{P3} = -0.15\) m s\(^{-1}\). The numerical simulations also predicted this condition. In EE2, the external air velocity

Table 2. Thermal cooling load and M-packages temperature at different heights (shelves) for the case studies

<table>
<thead>
<tr>
<th>Case Study</th>
<th>Heat Load [kW]</th>
<th>M-Packages Temperature [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Shelf 4 (SH4)</td>
</tr>
<tr>
<td>EE1</td>
<td>6.4</td>
<td>9.8</td>
</tr>
<tr>
<td>EE2</td>
<td>5.3</td>
<td>4.1</td>
</tr>
</tbody>
</table>
Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers

was adjusted to meet a mean value of $v_{px} = 0.2 \text{ m s}^{-1}$ of the horizontal air velocity in sensor P1, P2 and P3. After accomplishing this condition, a 24-hour test to the VORDC was also performed. In this case study, the external air velocity in the measuring points was around $0.2 \text{ m s}^{-1}$ during the 24-hour test.

Table 2 shows the thermal cooling load values and M-packages temperature at different heights (shelves) for the two case studies. The temperature of M-packages (product simulators) in EE1 stabilized in higher values than in EE2 (plus 56%). In EE2 is determined a reduction of the thermal cooling load of approximately 17% in relation to EE1.

The thermal cooling load and storage temperatures of products obtained in EE2 are the values reported by the manufacturer, although the test does not follow completely the standard as the calibration of air velocity in the climatic chamber is performed with the VORDC switched on. The thermal cooling load of 5.3 kW matches with the value published by the manufacturer and it is used by him in all installations, since the ventilation conditions inside the stores are less demanding than defined in EN ISO 23953 (2005).

Another comparison that is worth to be mentioned is the ambient air velocity defined in ASHRAE standard 72-2005. This standard suggests an air velocity parallel to the frontal opening of the VORDC between $0.15 \text{ m s}^{-1}$ and $0.25 \text{ m s}^{-1}$. As this standard allows testing the VORDC with lower air velocity, the performance results for the same VORDC will be better. Future work will be developed with the aim of comparing the performance of the VORDC when subjected to the test conditions of ASHRAE and ISO. The results should contribute to the development of new equipment with reduced energy consumption, heat exchangers size and production cost.

Conclusive Remarks

The numerical results show the airflow pattern inside the climate chamber. These predictions allow adjusting the airflow to perform the case studies EE1 and EE2. The case studies provided different results of thermal cooling load and M-packages temperature. These values were compared with the data reported by the manufacturer. The data analysis allows to conclude that the environmental conditions of the climatic chamber imposed in EE2 (VORDC switched on) are less demanding to the operation of the VORDC. Performing tests with this condition allows marketing a VORDC with lower energy consumption without showing operation problems. Another possibility is to perform tests according to ASHRAE standard 72-2005, as the procedure to adjust the air velocity in the climatic chamber can be carried out with the VORDC switched on and using a lower air velocity ($0.15 \text{ m s}^{-1}$). It is possible to obtain better performance results for the same VORDC performing the tests with this standard.

AIR CURTAINS INTERFERENCE DUE TO CONSUMERS MOVEMENT

VORDC used to expose perishable food for sale in convenience stores and supermarkets are subject to human interference. Clients and repositories transit in front of VORDC and frequently remove or place food products on the shelves depending on sales volume. This movement is part of the trade, although it has consequences on the VORDC performance. Each interference drags or breaks the air curtain resulting in the modification of air flow and promoting the ambient air thermal entrainment that consequently change the equipment’s working conditions and increase 2% to 5% its energy consumption. This experimental study quantifies the increase of air temperature and energy consumption when there is an interference drag due to people inside the store passing parallel to the frontal opening of the VORDC. The tests were
Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers

performed using a robotic mannequin that systematically transfers around the VORDC (5 minutes lap during 24 hours) and parallel to the frontal opening of the VORDC with a translation velocity of 0.6 m s\(^{-1}\). The results show increase of the products temperature and energy consumption respectively in 4.6\% and 16\% due to the air movement generated by the robotic mannequin transfer. These results are part of a more complex evaluation of the air curtain interference by humans to be used in the development of new products on an industrial scale. Additional details can be found in Nascimento et al. (2014b).

Materials and Methods

Robotic Mannequin

A robot: MARIA (Mannequin for Automatic Replication of the Interference in the Air curtain) was designed and constructed for the experimental study. The robot was programmed to move in front of VORDC with a period of 5 min and a translation velocity of 0.6 m s\(^{-1}\) during 24 h. This velocity value was obtained with field analysis and corresponds to the average velocity of people transferring inside a supermarket when they are in front of VORDC in the butchery sector.

Experimental Testing Procedure

Experimental tests were performed with and without the translation movement of MARIA. The experimental results allow the comparison of the VORDC performance with and without interference of the air curtain due to the systematic transfer of MARIA. Figure 23 shows the layout of the experimental testing procedure. Tests were performed under the same climate condition and air velocity in the test room.

Figure 23. MARIA passing in front of the VORDC
Figure 24. Air temperature at DAG and RAG for the case study without the interfering translational movement of MARIA

Figure 25. Air humidity at DAG and RAG for the case study without the interfering translational movement of MARIA
**Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers**

Figure 26. Air temperature at DAG and RAG for the case study with the interfering translational movement of MARI.

![Graph of Air Temperature at DAG and RAG](image)

Figure 27. Air humidity at DAG and RAG for the case study with the interfering translational movement of MARI.

![Graph of Air Humidity at DAG and RAG](image)
Analysis and Discussion of Results

The experimental results of both case studies (with and without interference) were analysed and compared based on 24h data of air temperature and relative humidity in the DAG and RAG. Additionally, the thermal load obtained in both case studies was also compared.

Results

Figure 24 and Figure 25 shown respectively the 24h data of air temperature and (absolute) humidity in the DAG and RAG for the experimental testing without movement. Figure 26 and Figure 27 shows respectively the evolution of air temperature and humidity considering the movement of MARIA parallel to the frontal opening of the VORDC, but against the direction of airflow from the test room.

Discussion

In Figure 24 is shown the effect of the period of defrost on the air temperature and humidity. Since the evaporator defrost system is a heating coil placed inside the evaporator fins, when the system operates (set to 2h30m intervals), the air temperature and humidity at DAG increase in large amount. After the defrost system stop, the refrigeration system starts up and takes approximately 30 minute to return air temperature and humidity to the proper values for the products conservation.

MARIA transfers around the VORDC during the all-testing period (refrigeration and defrost cycles). The objective of the experimental testing is to evaluate the increase of air temperature and humidity at DAG, RAG and conservation zone after 24h. During this period, the refrigerant flow and the enthalpies difference were measured/calculated. Thus, the influence on thermal performance of the parallel transfer in front of the VORDC and consequent interference of the air curtain can be quantified.

The values of air temperature and humidity at RAG show a small increase at end of the test period due to the translational movement of MARIA (see Figure 26 and Figure 27). The air temperature and humidity at RAG increased approximately 0.2 [°C] and 0.4 [gwater/kgdry air] respectively. The values of air temperature and humidity at DAG maintained approximately constant during the test period because the heat exchanger absorbs the heat gain of the air curtain due to external interferences.

The thermal load was measured based on the refrigerant flow and enthalpies difference. The latter energy parameter had a slight increase, from 5.32 [kW] to 5.67 [kW], i.e. a 4.6% increase that now can be quantified as due to perturbations, disturbances or interference of the air curtain of VORDC.

Conclusive Remarks

This section described the performance results of a VORDC when subjected to external disturbances on the air curtain. With the robotic mannequin developed for this purpose, MARIA (Mannequin for Automatic Replication of the Interference in the Air), was possible to simulate systematically the movement of customers inside the store and quantify the increase of air temperature and thermal load due to this disturbance. It has been observed experimentally that the air temperature and humidity at RAG have a slight increase, approximately 0.2 [°C] and 0.4 [gwater/kgdry air]. Thus, the air enthalpy is higher than in the case study without MARIA moving around the VORDC.
At supermarkets, there are several models of display cases, and all are subject to external interference due to customers transfer. This study allow to quantify for this condition, a thermal load increase of approximately 4.6%, which is for the model in study represents an increase of approximately 250 W.

HEAT EXCHANGERS

Introduction

The process of heat exchange between fluids at different temperatures and separated by a solid wall occurs in many engineering applications and for each type there is a most suitable heat exchanger. The finned-tube heat exchangers has been extensively used in various industrial applications, including...
refrigeration. They are very compact, lightweight, and characterized by a relative low manufacturing cost (Wang et al., 2002).

The tubes are not necessarily of circular cross-section and can also be elliptical. Matos (2003) used elliptical tubes to reduce the vortices formation in the posterior region of the tubes in the direction of flow, resulting in lower air pressure drop and increase of the coefficient of heat transfer. However, the industry generally not use this type of geometrical arrangement mainly due to the constructive difficulties.

Figure 28 shows two of the most important construction of these types of heat exchangers: (a) Plate fin-and-tube or (b) individually finned tube.

Figure 29 shows fins with the most common arrangement of the tubes that can be (a) aligned and (b) staggered, the latter option gives better performance in heat transfer due to the promotion of greater mixing in the flow. The staggered tubes result in a more efficient thermal interaction between air and the surfaces of the fins and tubes (Webb, 1994).

The coefficient of heat transfer at the airside is much smaller than the coefficient of heat transfer from the inner side of the tubes, so it is important to optimize the external heat transfer flow (Webb, 1994). The fins are responsible for a large increase in the area of external heat transfer, and still, the airside is responsible for at least 85% of the total resistance to heat transfer in this type of equipment (Wang et al., 1997, 1999). Thus, different types of fins have been studied in order to maximize the heat transfer coefficient between the air and the evaporator surfaces. The fins can be: flat, wave or with different types of surface corrugation. The fin detailed in Figure 29 (a) has a surface corrugation in the fin and the detailed Figure 29 (b) has smooth waves.

The heat exchange fluids can flow in the same direction (parallel currents) in opposing currents (counter-currents) or perpendicular to one another (crosscurrents).

In order to evaluate the performance of heat exchangers is necessary to know its geometric and operating variables.

- Geometrical variables of the tubes side: outside diameter, wall thickness, length, transverse tube pitch, row pitch, arrangement (staggered or in line), number of tubes in the transverse direction, the number of rows, and number of circuits.
- Geometrical variables on the fins side: plate or individual fins, height, length, thickness, spacing between fins. For wavy fins should note pitch, shape and angle of the corrugations.
- Operating variables of the tubes side: tube material, type of fluid and its physical properties, flow rate, operating temperatures, pressure drop.
- Operating variables of the fins side: fins material, fluid type and its physical properties, flow rate, operating temperatures, pressure drop.

**Problem**

Knowing that 28% of the energy consumed in large supermarkets is attributed to compressors of refrigeration systems (ASHRAE, 2010), any study aiming to minimize this cost is of great value and importance. Concerning the refrigerated display cases, the energy consumption is strongly connected to the environmental conditions of the store in which they are placed. The higher the temperature and absolute humidity of the ambient air, the greater the sensible and latent heat loads which the cooling system must remove. The increase of air humidity in the refrigerated space increases the condensation rate and the frost formation on the heat exchanger, which in turn requires a greater frequency of daily defrosting. This
condition increases the heat load due the need to recover the preservation temperature of food products after defrosting. Additionally, it promotes the temperature instability of the exposed products.

In general terms, VORDC are designed to operate in air-conditioned supermarkets. The international industry considers summer climatic conditions to be air temperature at 25 °C with 60% of relative humidity (ISO 23953:2005 climate class n.° 3). This climate condition is considered standard for homologation of this type of equipment (critical summer conditions). In Brazil, not all stores have air conditioning, and more adverse air temperature and humidity conditions frequently occur. Thus, in these situations, an air temperature of 27 °C with 70% of relative humidity (ISO 23953:2005 climate class n.° 6) is usually considered.

The heat exchangers used in refrigerated display cases are compact with finned tubes. The heat transfer occurs between the air flowing through the fins and the refrigerant flowing inside the tubes in cross flow. It is usual: cooper circular tubes with or without inner corrugation and aluminium-corrugated fins.

During the operation, the surfaces of the heat exchangers are at temperatures below the dew point and lower than the freezing point of water, therefore, condensation of water vapour present on the airflow with subsequent freezing on the surfaces of the fins and tubes is unavoidable.

The frost formation between the fins reduces the air passage area, thus increasing the pressure drop and reducing the airflow. The significant reduction in airflow during the frost formation deteriorates the air curtain that protects the refrigerated space in case of ORDC. Thus, more and more ambient air enters the refrigerated space, which affects the optimal storage temperature of products. This can lead to large losses due to changes in organoleptic properties of food. Hence, ORDC needs regular and scheduled stops of operation to defrost the ice from the evaporator in order to recover the initial conditions of performance without affecting the condition of the products. Regarding the process of frost formation and defrost, the optimal ORDC is the one which has the lowest frequency and time of defrosting. This way ensure a longer refrigeration time with consequent lower oscillation of displayed products temperature.

The type, frequency and duration of defrosting are determined through experimental homologation tests conducted by the manufacturers of ORDC that are guided by current technical standards and are directly related to the design of the heat exchanger. Among the design factors that are related to the formation of frost and defrosting on display case heat exchangers, the most important is the spacing between the fins. In the same operating condition, as smaller the spacing between the fins as greater will be the necessary defrost frequency. A large defrost frequency implies on a shorter time of continuous cooling and a higher pull-down load. As greater the spacing between the fins as smaller will be the defrost frequency, however a larger number of tubes is required to keep the heat exchange area and, in consequence, the size of the heat exchanger will be greater.

Therefore, this section describes the experimental procedures for tests conducted on this climate class, and presents data suited for the design of heat exchangers, due to their relevant role in improving the performance of VORDC operating in adverse climate conditions while always ensuring safe temperature levels of shelf products, as required by food safety standards.

**State of the Art**

To evaluate the performance of finned heat exchangers under frost formation conditions, Aljuwalyhel *et al.* (2008) performed experimental studies with an evaporator of an industrial cold room. The chamber operates with the room temperature of -29 °C and it is used to store ice cream. Data acquisition was performed with air velocity, temperature and humidity sensors positioned at the airflow entrance and
The experimental tests results show: (1) a reduction in the cooling capacity of the evaporator due to frost formation between the fins is responsible for the decrease of air free flow area with consequent increase of air pressure drop and air flow reduction when are used fixed speed fans; (2) The air temperature difference between inlet and outlet of the evaporator tends to increase with the reduction of air flow over time.

In industrial refrigeration applications, Jekel & Reindl (2009) studied the characteristics of frost formation in evaporators and considered spacing between fins, the location of the coil, the air humidity and type of frost as being the determining factors of the reduction capacity rate of the coil during frost formation: (1) Decreasing the pitch of the fins may be desirable to increase the heat transfer area, but in frosting conditions, the circulating air flow blockage occur faster due to less available space for growth of the frost layer. Thus, a higher frequency of defrost cycles will be required. The spacing between fins must be carefully set in the evaporator design to provide the area required for heat transfer and the correct operating conditions for the coil; (2) evaporators should not be installed directly above the doors of refrigerated chambers because this is the region where the air saturated with moisture concentrates; (3) There are unfavourable conditions for frost formation that leads to the freezing of ice crystals directly in airflow. These crystals adhere to the fins due to the impact creating a lower density frost that will block the airflow so much faster. The unfavourable condition of frost formation occurs when the humidity of the air inlet of the evaporator is very high and the dew point temperature of equipment surfaces is very

**Figure 30. Comparison of the total mass of condensate collected in different operating conditions over a period of 24 hours**

*(Adapted from Gas Research Institute, 2000)*
Thus, the air becomes saturated with moisture during the cooling process that leads to the formation of ice crystals directly in the airflow.

Tassou & Datta (1999) studied the impact of environmental conditions on frost formation and defrosting on heat exchangers of ORDC. The tests performed in the field, in winter and summer, show the drastic influence of climate on the volume of water collected by defrost period (each 6 hours in this experimental test). In summer conditions, the moisture inside the store ranged between 45% and 55% and in winter between 22% and 25%. The average volume of water collected in summer was 5.1 liters/defrost and winter of 2.5 liters/defrost. It follows that for an ORDC, the amount of daily required defrosting varies according to weather conditions.

Additional experimental tests were carried out in a test room with respect to the variation of the amount of defrosting necessary due to oscillation of the air relative humidity in an environment where dry-bulb air temperature was fixed at 26 °C. The operation of the ORDC was followed from the end of a reference defrost until an excessive formation of ice blocked the airflow through the coil. The excessive frost formation occurs in 5 hours at a relative humidity of 50%, at 6.5 hours to relative humidity of 40% and at 9 hours and at a relative humidity of 30%, respectively.

The Gas Research Institute (2000) studied the impact of ambient conditions in laboratory conditions keeping constant the ambient air dry-bulb temperature at 24°C and ranging the relative humidity between 35% and 55%.
The influence of frost formation on the performance of the ORDC was evaluated by total mass of water collected after 24 hours of operation (see Figure 30) and the increase of the latent load relative to the total cooling load (Figure 31).

Increasing relative humidity causes a significant increase for water and in the latent cooling load. Increased moisture in ambient air has direct implications on the load removed by evaporator and in the energy consumption of the compressors to keep the products in optimal conditions of temperature regardless of the external climate.

Usually, the number of defrosting is fixed according to the manufacturer’s specification. Thus, this regulations cause unnecessarily defrost in lower humidities, resulting in increased power consumption. On the other hand, for high values of humidity, the programmed defrosts are insufficient, which causes the excessive consumption of energy and commits the quality of product preservation.

It follows that for an ORDC, the amount of daily defrost required varies according to the weather condition. Therefore, the cooling period can be longer (defrosting fewer) when the value of humidity is smaller, and may have their period of cooling reduced (highest number of defrost) when the amount of moisture is higher.

Gaspar et al. (2007, 2008, 2009, 2010a, 2010b, 2011a, 2012a, 2012b) and Heidinger et al. (2014) develop investigations that have taken account not only the impact of varying the temperature and relative humidity environment but also the airflow magnitude and direction relative to the plane of the display opening of VORDC. Changing the direction of ambient airflow from 0° (parallel) to 45° (oblique) and subsequently to 90° (perpendicular), keeping the dry-bulb temperature at 25 °C, the air relative humidity at 60% and the ambient airflow at 0.2 m s⁻¹, the total heat load of the equipment increases 4.6% and 5.7% respectively. Changing the airflow direction causes a drastic drop in air curtain that affect the safe storage temperature of the product that increases, respectively, from 2.9 ºC to 8.4 ºC and 6.7 ºC. Keeping the above conditions with the parallel direction of airflow and changing the ambient air velocity from 0.2 m s⁻¹ to 0.4 m s⁻¹, there is an increase of 54% in the total load of the equipment and 77% in the latent load. Again, the product temperature reaches values that affect the ideal conservation of products. The product temperatures increased from 2.9 ºC to 11 ºC. Based on the results, it stands out the importance of avoiding unwanted air currents near ORDC. This can be caused by outflows of air conditioning, fans and doors systems that cause airflows by differences of pressure.

Mousset & Libsig (2011) conducted a comparative study between field and lab situations to evaluate the differences and impacts in the performance of ORDC. The studies were performed in medium temperature VORDC. In the laboratory, VORDC were tested in three climatic conditions referenced in ISO 23953 standard and defined as Classes n.° 1 ($T_{\text{amb}} = 16 ^\circ C; \phi_{\text{amb}} = 80\%$), n.° 2 ($T_{\text{amb}} = 22 ^\circ C; \phi_{\text{amb}} = 65\%$) and n.° 3 ($T_{\text{amb}} = 25 ^\circ C; \phi_{\text{amb}} = 60\%$). In the field, it is not possible to keep the air temperature and humidity fixed according to the climatic classes and measures fluctuated during the year. To correlate the data from the field with the reference laboratory tests the value of ambient air enthalpy was compared.

It was possible to identify that the climate condition inside the store was more than 98% of the time below the Class n.° 2. However, it is necessary that the food safety be maintained at 100% of the time. Thus, the equipment homologation tests need to be performed in climate class n.° 3. Analysing the tests data, the heat load from the equipment in the laboratory was 11% to 17% higher than that recorded in the field to climate class n.° 3. This is because the normative parallel air velocity (between 0.1 and 0.2 m s⁻¹, as imposed in ISO 2395 (2005), in the laboratory is not be repeated in the field. In the store, there is air stratification that prevents the homogeneity of the ambient air temperature.
INFLUENCE OF ENVIRONMENTAL CONDITIONS ON THE PERFORMANCE OF THE EVAPORATOR

Heidinger et al. (2013) study the influence of the environmental conditions on the performance of the evaporator through experimental tests performed according to ISO 23953 in a VORDC with dual air curtain.

Experimental Procedure

Tests on the VORDC shown in Figure 32a were conducted inside the environmental test chamber shown in Figure 4. The experimental techniques and probes/experimental measuring devices shown in Table 1 were used to measure the air temperature, relative humidity, velocity and pressure drop during 24 hours at 1-minute intervals.

To measure physical quantities on the airside, sensors were positioned at 5 equidistant points from the air inlet and outlet along the length ($y$) of the evaporator (see Figure 32b). Thus, the evaporator was divided into 5 virtual control volumes (CV), so that air temperature, humidity and velocity in each CV are considered constants (Figure 32b). With this procedure was possible to determine the air distribution in the evaporator and the zones are more or less likely to form frost.

The mass of water condensed during defrosting was manually collected and measured during each test period. The outlet mass flow of the liquid coolant, which passed through the evaporator during each operating period, was monitored by a Coriolis flow monitor.

The manufacturer of the VORDC recommends for ideal operation an evaporation temperature of -10 ºC, ensuring products temperature within the prescribed limited range between -1 ºC and 5 ºC. Initially, the VORDC defrosts 10 times per day, and each defrost period lasts 12 minutes. The defrosting control system consists of stopping the cooling system while maintaining the fans in operation in order to defrost the ice on the evaporator surface by the passage of air at the ambient temperature. For testing purposes, there is no controller limiting the compressor work, whether a thermostat that ceases its operation depending on temperature, or some other mechanism that turns off the compressor during the operating period. The compressor is only turned off when the defrosting process begins.

Laboratory tests were performed on two climate classes usually found in Brazilian food stores (n.º 3: $T_{\text{amb}} = 25$ ºC; $\phi_{\text{amb}} = 60\%$ and n.º 6: $T_{\text{amb}} = 27$ ºC; $\phi_{\text{amb}} = 70\%$). In either test, the maximum product
temperature was maintained between 4 ºC and 5 ºC. The climatic conditions of the room vary according to the climatic class desired for the test. In more pleasant climates, the product temperatures are lower. In warmer and more humid climates, the product temperatures increase. In order to make a fair comparison, the maximum temperature of the warmest product was maintained between 4 ºC and 5 ºC regardless the climatic condition. It was possible to make this adjustment by simply increasing or decreasing the usable overheating of the refrigerant fluid. In this way, there was a reduction or increase in the required refrigerant flow, and a consequent reduction or increase in the heat capacity of the evaporator according to the needs of each climate class.

**Mathematical Model**

The total heat load of the cooling system on the refrigerant liquid side is determined by Equation 3.

In order to determine the total heat load and its components on the airside, the airflow rates and the condensation and freezing rates of water for each cooling cycle were determined. Considering uniform rates of outflow for each CV, and using the average values for velocity, $V_a$, and density, $\rho_a$, of air in the inlet and outlet areas of the evaporator of cross section, $A$, obtained by experimental sampling based on the principle of conservation of mass, the mass flow of air, $\dot{m}_a$, was obtained by Equation 26.

$$\dot{m}_a = \rho_a \cdot V_a \cdot A$$  \hspace{1cm} (26)

Using the average values of air temperature and relative humidity at the inlet and outlet areas of the evaporator, absolute humidity of the air at inlet and outlet was measured using psychrometric concepts, $\omega_{a,i}$ and $\omega_{a,o}$, respectively. Thus, using Equation 27, the mass flow of condensed water, $\dot{m}_w$, is determined for the period in which the refrigeration was turned on, $\dot{m}_{w,OP}$, and for the period in which the refrigeration was turned off, $\dot{m}_{w,DEF}$. The mass flow of water collected in experimental sampling during the defrosting period, $\dot{m}_{w,EXP}$, given by Equation 28 is the sum of the mass flow of ice formation on the evaporator during the operation period, $\dot{m}_I$, and $\dot{m}_{w,DEF}$. A small part of the $\dot{m}_{w,OP}$ was drained without freezing. This mass flow, $\dot{m}_{w,DRAIN}$, is determined by Equation 29.

$$\dot{m}_w = \dot{m} \cdot (\omega_{a,i} - \omega_{a,o})$$  \hspace{1cm} (27)

$$\dot{m}_{w,EXP} = \dot{m}_I + \dot{m}_{w,DEF}$$  \hspace{1cm} (28)

$$\dot{m}_{w,DRAIN} = \dot{m}_{w,OP} - \dot{m}_I$$  \hspace{1cm} (29)

The components of heat load can be determined by Equation 30 to Equation 34, according to GAS Research Institute (2000).
Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers

\[ Q_{a,\text{COLD}} + Q_{w,\text{COND}} = \dot{m}_a \cdot (h_{a,i} - h_{a,o}) \quad (30) \]

\[ Q_{w,\text{COND}} = \dot{m}_{w,\text{OP}} \cdot h_w \quad (31) \]

\[ Q_{w,\text{COLD}} = \dot{m}_1 \cdot c_{p,w} \cdot (T_{\text{DEW}} - T_{\text{FREEZE}}) \quad (32) \]

\[ Q_{w,\text{FREEZE}} = \dot{m}_1 \cdot h_j \quad (33) \]

\[ Q_{1,\text{COLD}} = \dot{m}_1 \cdot c_{p,l} \cdot (T_{\text{FREEZE}} - T_l) \quad (34) \]

The total, sensible and latent heat loads can be determined by Equation 35, Equation 36 and Equation 37, respectively.

\[ Q_{\text{TOT}} = Q_{a,\text{COLD}} + Q_{w,\text{COND}} + Q_{w,\text{COLD}} + Q_{w,\text{FREEZE}} + Q_{1,\text{COLD}} \quad (35) \]

\[ Q_{\text{LAT}} = Q_{w,\text{COND}} + Q_{w,\text{FREEZE}} \quad (36) \]

\[ Q_{\text{SENS}} = Q_{a,\text{COLD}} + Q_{w,\text{COLD}} + Q_{1,\text{COLD}} \quad (37) \]

Correspondingly, two methods of determining the refrigeration heat load are obtained. The results of these two calculation methods are compared and discussed.

**Table 3. Absolute values for condensation and freezing of water in experimental tests on climate classes n.º 3 and n.º 6**

<table>
<thead>
<tr>
<th>Climate Class</th>
<th>( m_{w,\text{OP}} ) [kg 24h(^{-1})]</th>
<th>( m_{w,\text{DEF}} ) [kg 24h(^{-1})]</th>
<th>( m_{w,\text{DRAIN}} ) [kg 24h(^{-1})]</th>
<th>( m_j ) [kg 24h(^{-1})]</th>
<th>( m_{w,TOT} ) [kg 24h(^{-1})]</th>
</tr>
</thead>
<tbody>
<tr>
<td>n.º 3</td>
<td>45.7</td>
<td>3.3</td>
<td>2.8</td>
<td>42.9</td>
<td>49.0</td>
</tr>
<tr>
<td>n.º 6</td>
<td>66.4</td>
<td>5.5</td>
<td>7.5</td>
<td>58.9</td>
<td>71.9</td>
</tr>
<tr>
<td>Increase [%]</td>
<td>45</td>
<td>67</td>
<td>168</td>
<td>37</td>
<td>47</td>
</tr>
</tbody>
</table>
Analysis and Discussion of Results

Air Humidity, Condensation, and Freezing of Water

The rates of condensation and freezing of water between the evaporator fins increase with the humidity of the ambient air where the VORDC is operating. The aim of the experimental study was to measure how representative is the increase of ambient air humidity and temperature on the rates of condensation and freezing during the entire operation and defrosting period. The experimental results obtained for the test conducted for climate classes n.º 3 and n.º 6 are shown in Table 3.

Comparing the experimental results of climate class n.º 3 and class n.º 6 shown in Table 3, it can be concluded that: (1) the total quantity of condensed water in the evaporator is 47% greater; (2) the quantity of water which solidifies between fins is 37% greater; and (3) the quantity of water which drains during the operating period is 168% greater. Such numbers prove the significant influence of climate conditions on the operation of evaporators regarding their frost formation. The following sections detail other aspects related to the evaporator performance.

Air Distribution and Pressure Loss in the Evaporator

The airflow distribution was evaluated at two separate times: when the evaporator was completely clean, i.e. after defrosting the coil and immediately before defrosting it. The percent distribution of airflow in the five virtual CVs of the evaporator is shown in Figure 33.

From the analysis of Figure 33, it was determined that after defrosting, airflow is equally distributed in each CV for both climate classes (approximately 20% in each CV). CV 2 and CV 4 are those possessing the greatest airflow circulating, and moreover, the greater velocity of air passing between fins. The
lower quantity of airflow circulating in CV 1 and CV 5 can be attributed to extremities effects (friction and consequently pressure drop) on the evaporator. The lower proportion of airflow circulating in CV 3 is attributed to the structural column that restricts the passage of air and restrains ideal formation of air plenum in the centre of the evaporator.

When the evaporator is ready to be defrosted, i.e., before defrost, the percentage distribution of airflow is significantly different from the initial measurement. This fact is attributed to the irregular formation of frost. The airflow through CV 3 is less than other CV. It can be concluded that the frost is most likely to form in this region. The airflow reduction is more significant for climate class n.º 6, as before defrosting only 7% of the airflow circulating in the evaporator goes through this region (a quantity much lower than the initial measurement of 19%). This drastic reduction suggests that the frost formation grows significantly as a function of ambient air conditions. The increase in the percent airflow mainly in CV 2 and CV 4 is due to these regions are less likely to form frost and therefore, as the resistance to outflow is lower they become the preferred paths for air. During the cooling period, the total absolute airflow decreased 24% for climate class n.º 3 and 40% for climate class n.º 6.

Figure 34 shows the variation of the air pressure drop before and after the fan bank. The air pressure drop is more significant for climate class n.º 6 and increases approximately 33% during the cooling period, while for class n.º 3 the increase is approximately 23%. These results prove that greater reduction of airflow in hot and humid environmental conditions due to the greater frost formation, which is responsible for an increase in the resistance to air circulation between fins due to the decrease in free outflow area. The reduction in air velocity is the main factor in cooling capacity loss during the cooling period (Aljuwayhel et al., 2008).

Influence of Frost Formation on Air Temperatures

Figure 35 shows the variation of the temperature of air entering and leaving the evaporator for climate classes n.º 3 and n.º 6. The air temperature at the evaporator outlet is lower for class n.º 6. This is due to the different parametrizations needed to maintain the maximum temperature of test packages (product

Figure 34. Air pressure drop variation before and after fans
Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers

Figure 35. Air temperatures in the inlet and outlet of the evaporator

Table 4. Air temperatures in the evaporator

<table>
<thead>
<tr>
<th>Period</th>
<th>Climate Class n.º 3</th>
<th>Climate Class n.º 6</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( T_{a,i} )</td>
<td>( T_{a,o} )</td>
</tr>
<tr>
<td>After defrosting</td>
<td>4.53</td>
<td>-2.17</td>
</tr>
<tr>
<td>Before defrosting</td>
<td>5.50</td>
<td>-1.89</td>
</tr>
<tr>
<td>Increase [%]</td>
<td>21</td>
<td>15</td>
</tr>
</tbody>
</table>

Simulators) below the established 5 °C limit. Greater capacity to protect the refrigerated space is necessary when the climate is more adverse in order to maintain the product temperatures within the food safety limits.

The circulating air temperature and the difference between the temperatures at the evaporator inlet and outlet increase as the defrost period approaches. After defrosting, the air is colder, and the difference between the air at the evaporator inlet and outlet is less than before defrost. Therefore, with the evaporator free from frost, the heat transfer efficiency is at its maximum level. When the surface of fins is covered with frost, the efficiency lowers. As frost forms over time, there is an increase in the difference between the temperature of air entering and leaving the evaporator due to the decreased airflow on the evaporator. This situation means a cooling capacity loss in the heat exchange due to reduced efficiency of the air curtain, which allows a greater quantity of external air to be taken in by the fans. As a result, the temperature of the air entering the evaporator increases and consequently increases the rate of frost formation between fins. A substantial increase in the difference between air entering and leaving the evaporator is verified when the air temperature and humidity of the surrounding environment are greater (climate class n.º 6). As the layer of frost formed is thicker, the resistance to heat transfer is increased, making the conduction/convection of heat between the fluids more difficult. Table 4 shows air temperatures values at two different times: before and after defrosting the coil.

Table 4. Air temperatures in the evaporator
Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers

Figure 36. Heat load and its components
For climate class n.º 3, the difference in the temperature of air at the inlet and outlet of the evaporator increases 10% while for climate class n.º 6 the increase is 19%. The data in Figure 35 and Table 4 shows that for climate class n.º 6 the temperature of air leaving the evaporator, after reaching its highest value, starts reducing until the end of the operating period, again proving the larger reduction in the circulating airflow.

**Heat Load**

The total heat load and its components are shown in Figure 36.

The heat load of the equipment increased 25% from climate class n.º 3 to class n.º 6. In climate class n.º 3, the latent heat load (condensation and freezing of water) represents 34.6% of the total heat load, with the remaining 65.4% due to sensible heat of cooling the air, the water and frost. The latent heat due to freezing of ice represents only 3.9% of the total load and the sensible heat load due to cooling the water and ice represents only 0.3% of the total load. For climate class n.º 6, the latent heat load represents 40.9% of the total heat load, with the remaining 59.1% due to sensible heat of cooling the air, water and ice. The latent heat due to freezing the ice represents only 4.3% of the total load and the sensible heat load due to cooling water and ice represents only 0.5% of the total load.

**Conclusive Remarks**

This experimental study analysed the conditions of ambient air in the formation of frost on the surface of evaporators of VORDC. Experimental tests are designed for climate classes n.º 3 and n.º 6. The main conclusions of this study are: (1) the frost formation significantly affects the distribution of air in the evaporator due to its irregular formation in fins; (2) the quantity of air circulated in the VORDC is decreased over time due to the reduction of free area for air outflow and consequent increase of pressure drop; (3) the reduction in the evaporator’s heat capacity during the operating period is mainly due to the reduction of airflow; (4) to maintain the same conditions for preserving products under different climatic conditions, it is necessary to change the temperature at which the air leaves the evaporator in such a way that the degree of protection offered by the air curtain can be controlled; (5) climatic conditions significantly influence the energy consumption of refrigeration equipments, since the total heat load increases as climatic conditions become more adverse, and the increase in latent heat load is greater than the increase in sensible load; (6) the heat loads due to the cooling of water and ice are not significant.

**CONCLUSION**

This chapter contains an analysis of VORDC performance focused at the two areas of greatest impact on the efficiency of this type of equipment: the air curtain and the heat exchanger.

The air curtain is very important to the correct performance of VORDC. The results of specific experimental tests describe the behaviour of the VORDC focused on the thermal barrier formed by the air curtain between DAG and RAG. The results can be described as: (1) it is possible to balance the mass flow rates between PBP and DAG to reach the equipment optimal performance, (2) the use of DAG with two independent and parallel jets improves the overall performance of VORDC. It is possible to reduce 10% of the energy consumption, (3) the movement of air due to people movement inside the stores, can have a significant impact on the overall performance of the VORDC.
A careful analysis of the design of heat exchangers is important because there are not only the sensible heat transfer but also latent heat transfer. The knowledge of the environmental conditions in which the heat exchanger will operate is very important to make a correct project.

If the values of ambient air temperature and humidity are large, the heat load and the frost formation rate at the heat exchangers fins will be higher. Thus, the need of regular stops for defrost the coil will be greater. The significant airflow reduction during the frost formation deteriorates the air curtain that protects the refrigerated environment (especially in VORDC), allowing warm and humid air to infiltrate into the refrigerated space. This phenomenon affects directly the maintenance of optimal conservation temperatures of the products. This can lead to large losses due to changes in the organoleptic properties of food. It was concluded that the optimal VORDC is one that has the lowest and shorter frequency of defrosting cycles. Thus, a longer cooling with consequent lower products temperature oscillation can be achieved. The frequency of defrosting and its duration are directly related to the design of finned heat exchangers and the type of defrost to be done. The frequency of defrosting depends directly on the air curtain efficiency, which forms a thermo-physical barrier between the internal and external environment.

REFERENCES


Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers


Experimental Analysis to Optimize the Performance of Air Curtains and Heat Exchangers


**KEY TERMS AND DEFINITIONS**

**Air Curtain:** The Air Movement and Control Association International Inc. (AMCA International) defines air curtain as a controlled stream of air moving across the height and width of an opening with sufficient velocity and volume to reduce the infiltration or transfer of air from one side of the opening to the other and/or to inhibit insects, dust or debris from passing through.

**Energy Consumption:** The energy consumption of open refrigerated display cases is dependent on ambient air conditions (temperature, humidity and direction and magnitude of velocity), which will influence the heat transfer rate of the evaporator, air curtain efficiency and defrost system operation.

**Food Safety:** Food safety refers to the conditions and practices that preserve the quality of food to prevent contamination and foodborne illnesses. Ensuring the food temperature below the prescribed limits will promote their safety.

**Frost Formation:** Frost formation occurs when humid air encounters a surface whose temperature is less than the freezing temperature of water (273 K), and is less than the dew point temperature, so that water vapour goes from a gaseous to a solid state. As the frost layer increases in the evaporator surface, the cooling capacity of refrigeration is depleted due to the extra thermal resistance to the heat transfer process and also because it increases the air pressure drop, thereby substantially reducing the fan-supplied airflow rate.

**Heat Exchanger:** Piece of equipment built for efficient heat transfer from one medium to another.

**Open Refrigerated Display Case:** Refrigerated display cases are classified by standard ISO 23953-1 (2005) into vertical, semi-vertical, horizontal and combined; which may be open or closed to the store environment; for self-service or not. The specifications of each display case type are: (1) Vertical refrigerated display cases with multiple shelves have height exceeding 1.5 m; (2) Semi-vertical refrigerated display cases are whose overall height does not exceed 1.5 m and the display opening may be vertical or inclined; (3) Horizontal refrigerated display cases have horizontal display opening on its top. Open refrigerated display cases (ORDC) are those that access to the products takes place in a direct way without the need of opening doors or lids. Closed refrigerated display cases (CRDC) has the access to the products by opening of a door or lid; (4) Combined refrigerated display cases are those that combine at least two of the above characteristic. Each type of display case has its characteristic operating temperature, from preserving ice cream, frozen foods, fresh meat, dairy/deli to fruits/vegetables.

**Testing Standard:** The testing standard of performance characteristics of refrigerated display cases is performed according to ISO standard ISO 23953:2005.

**Thermal Entrainment Factor:** Dimensionless temperatures or enthalpies difference that quantifies the aero-thermodynamics blockage provided by an air curtain. This parameter varies from 0, which corresponds to no entrainment (unreachable condition) to 1, which corresponds to unblocked passage and entrainment of air between the two contiguous environments.

**Thermal Load:** Amount of heat (sensible and latent) energy to be removed from an inner environment by the refrigeration equipment to maintain that environment at the design temperature when worst case external temperature is being experienced.