EXPERIMENTAL EVALUATION AND QUALITATIVE INCREASE OF THERMAL LOAD IN REFRIGERATED DISPLAY CABINETS DUE TO BREAKAGE OF THE AIR CURTAIN

S. M. NASCIMENTO[(a)], G. G. HEIDINGER[(b)], P. D. GASPAR[(c)] , P. D. SILVA[(d)]
[(a)] [(b)] Eletrofrio Refrigeração Ltda, r João Chede, 1599, Cidade Industrial,
Curitiba - PR, 81170-220, Brazil
+55 (41) 2105-6097, [(a)]samuel@eletrofrio.com.br; [(b)]gustavo@eletrofrio.com.br
[(c)] University of Beira Interior, Engineering Faculty, Dept. of Electromechanical Engineering,
Covilhã, 6201-001, Portugal
+351 275 329 972, [(c)]dinis@ubi.pt; [(d)]dinho@ubi.pt

ABSTRACT

Open refrigerated display cabinets 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 cabinets for products sale. The vertical open refrigerated display cabinet is the model that consumes more energy. About 81% 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. Experimental tests were conducted in open and closed hours of a food store. The TEF increases by 22% and cooling load by 18% when the shop is open. The results comparison allows evaluating the thermal performance of the equipment among test cases for a real scenario.

1. INTRODUCTION

Since ancient times, man has the need and the will of obtaining cooling ways that make the temperature of foods products to reach a value below the environmental temperature in order to preserve them for longer periods of time. The perishable food products, from production to the final consumer, are preserved and channelled through the system named as cold chain. According to Rigot (1991), the cold chain can be described by five main links: Cold in the production stage; Cold during storage, Refrigerated transportation; Cold in the distribution stage; Home cooling. The fourth link in the cold chain, which is the subject of this paper, is commonly referred as commercial refrigeration by being placed at the trade level. ASHRAE (2010) indicates that the percentage of energy consumed in a typical supermarket due to the refrigeration systems reaches 50%. This energy is consumed by compressors, refrigerated display cabinets, walk-ins and condensers. Among the refrigerated display cabinets installed in a supermarket, which consumes more energy are of the vertical and open type. According to Faramarzi (1999), ASHRAE (2010) and Gaspar *et al.* (2011), the thermal load due to ambient air infiltration to a vertical open refrigerated display cabinet (ORDC) corresponds respectively to 67% - 77%, 73.5% and 78% - 81% of the total thermal load. This condition results from the low efficiency of the curtain air which forms a physical barrier between the internal and external environments of the equipment.

The work developed by various researchers has focused, for this type of equipment, in qualifying and quantifying the perceptible thermo-physical properties of the jet that provides a cold air curtain. 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. Faramarzi (1999) determined the relative weight of the total cooling load components for ORDC, composed by the loads from infiltration, radiation, conduction, product pull-down cooling, devices (lights and fans), defrost and anti-sweat heaters, and product respiration. According to EN-ISO 23953 (2005), the total thermal cooling load can be determined by eq. (1).

$$\dot{Q}_{tot} = \dot{m}_{ref} \cdot \Delta i \tag{1}$$

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 eq. (2) to eq. (5) respectively, for different aspect ratios (height/width) of the air curtain.

$$Re = \left(\frac{u \cdot b}{\nu}\right)_{DAG} \tag{2}$$

$$Gr = \frac{g\xi(T_{amb} - T_{DAG})H^3}{\nu_{DAG}^2} \tag{3}$$

$$Ri = \frac{Gr}{Re^2} \tag{4}$$

$$X_j = \frac{T_{RAG} - T_{DAG}}{T_{Amb} - T_{DAG}} \tag{5}$$

The results provided the following conclusions: (1) for a given Grashof number, there is a critical value of the Richardson number for optimal thermal sealing ability of the air curtain, which means that the value of the Reynolds number decreases by reducing the height/width ratio; (2) the range of values for the Richardson number increases reducing the height/width ratio; (3) for a minimum infiltration rate, the Richardson number decreases reducing the height/width ratio; (4) the volumetric infiltration rate decreases reducing the height/width ratio. 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 in studying the effectiveness of the curtain and maintaining the temperature of food products to a predetermined value. The results evaluation indicates 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 the discharge air grille (DAG) is about 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, varying $0 < TEF < 1$. The analysis to the correlation 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 air flow through the perforated back panel (PBP). Yu *et al.* (2009) developed the *TEF* equation considering this air flow. The new correlation is given by eq. (6) to eq. (8) including the dimensionless temperature given by eq. (5).

$$TEF = (1 - \beta)X_j - \beta X_j X_{PBP} \tag{6}$$

$$\beta = \frac{\dot{m}_{PBP}}{\dot{m}_{PBP} + \dot{m}_{DAG}} \tag{7}$$

$$X_{PBP} = \frac{T_{PBP} - T_{DAG}}{T_{Amb} - T_{DAG}} \tag{8}$$

The results obtained by Yu *et al.* (2009) show a good approximation for *TEF* and temperature value at the return air grille (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 vertical ORDC. Gaspar *et al.* (2009, 2010, 2011) 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 ORDC performance strongly depends on the ambient air conditions such as temperature, humidity, velocity and direction of ambient air flow in relation to the ORDC's 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 ORDC than the airflow direction change from parallel to perpendicular in relation the frontal opening of the ORDC, (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 is range closer to the value obtained by Faramarzi (1999) which is of 73.5%, and (5) *TEF* is not constant along

the equipment length for parallel air flow. Furthermore, the TEF value increases when the ambient air flow goes from parallel to perpendicular, being the worst case for θ_{amb} = 45°. In the case study, *TEF* = 0.25, 0.32, 0.3 for θ_{amb} = 0°, 45°, 90° 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 vertical ORDC. 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. Mousset and Libsig (2011) developed the correlation described by eq. (9) that quantifies, for any ambient air condition, the cooling load increment relatively to the cooling load in the climate class n.º 3 (25°C/60%RH) of ISO23953 (2005).

$$\Phi_{24\,(CLASS_x)} = \Phi_{24\,(CLASS_3)} \frac{Enthalpy_{(CLASS_x)}}{Enthalpy_{(CLASS_3)}} \tag{9}$$

Where, $\Phi_{24\,(CLASS_x)}$ [W] is the heat extraction rate in the ISO climate class *x* while the $\Phi_{24\,(CLASS_3)}$ [W] is the heat Extraction Rate in the ISO climate class n.º 3. Similarly, $Enthalpy_{(CLASS\ x)}$ [kJ/kg] is the enthalpy of the humid air calculated with the temperature and humidity of the class *x* and $Enthalpy_{(CLASS\ 3)}$ [kJ/kg] is the enthalpy of the humid air calculated with the temperature (25°C) and humidity (60%RH) of the climate class n.º 3.
This paper presents the field data collected in real operation of the ORDC in a commercial establishment without air conditioning. The experimental results describe the performance of the ORDC under real ambient air conditions and aim to provide information that can be use in the performance improvement of the ORDC since the design climate class condition cannot be always achieved in real operation.

2. CHARACTERISTICS OF THE OPEN REFRIGERATED DISPLAY CABINET

The vertical ORDC provided by Eletrofrio Refrigeration LTDA - Brazil has 2.5x1.1x2.1 m^3 (see Figure 1). 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 air flow exiting DAG forms an air curtain which protects the inner refrigerated compartment. The device has four fans with 53 W each to supply a flow rate of 0.4 $m^3 \cdot s^{-1}$ to DAG and PBP. The air, before reaching the DAG, passes through an evaporator with dimensions 2.20x0,13x0,35 m^3 constituted by 222 fins and three rows of tubes in the air flow direction and 8 rows of tubes perpendicular to it. The DAG has a total width, *b*, of 140 mm, which is equally distributed to form the PAC (b_{PAC} = 70 mm) and SAC (b_{SAC} = 70 mm).

3. EXPERIMENTAL STUDY

The performance benchmark of the vertical ORDC was conducted with air temperature and humidity data collected during six months. The ORDC was installed in a store with 532 m² of sales area without air conditioning. The supermarket is located in São Paulo - SP, Brazil.

2.1. Experimental testing procedure

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 ORDC 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. Table 1 shows the experimental techniques and probes/experimental measuring devices used to collect the relevant physical properties.

Table 1. Experimental techniques and probes/experimental measuring devices.

Experimental technique	Model	Measuring range	Accuracy
Thermometry	MT 530 Super	-10°C to 70°C	± 1.5 °C
Hygrometry	MT 530 Super	20% to 85%	± 5%

The probes were located in the ORDC as shown in Figure 1.

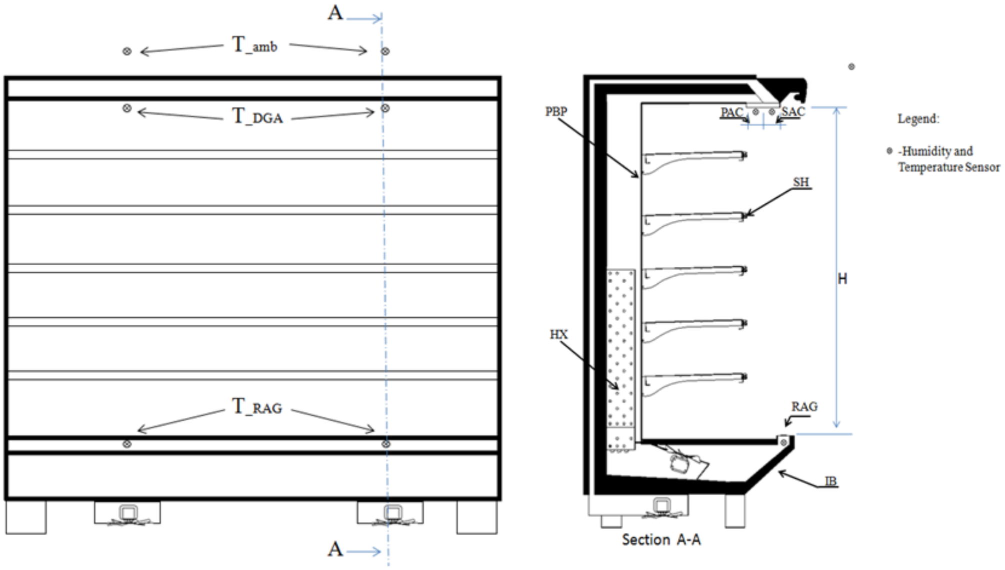

Figure 1. Air temperature and humidity probes location in the ORDC.

4. RESULTS ANALYSIS AND DISCUSSION

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 condition 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 eq. (5) to eq. (8).

4.1. Results

The annual average variation of the air temperature and relative humidity in DAG, RAG and store environment are shown in Figure 2 and Figure 3. 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 4 shows the average variation of the air temperature and humidity values collected in laboratory tests (LAB) according to EN ISO 23953 (2005). In Figure 5 are shown the TEF values for the different scenarios as well as their linear trend. Finally, Figure 6 shows the percentage increase of the cooling load calculated according to eq. (9) and its linear trend.

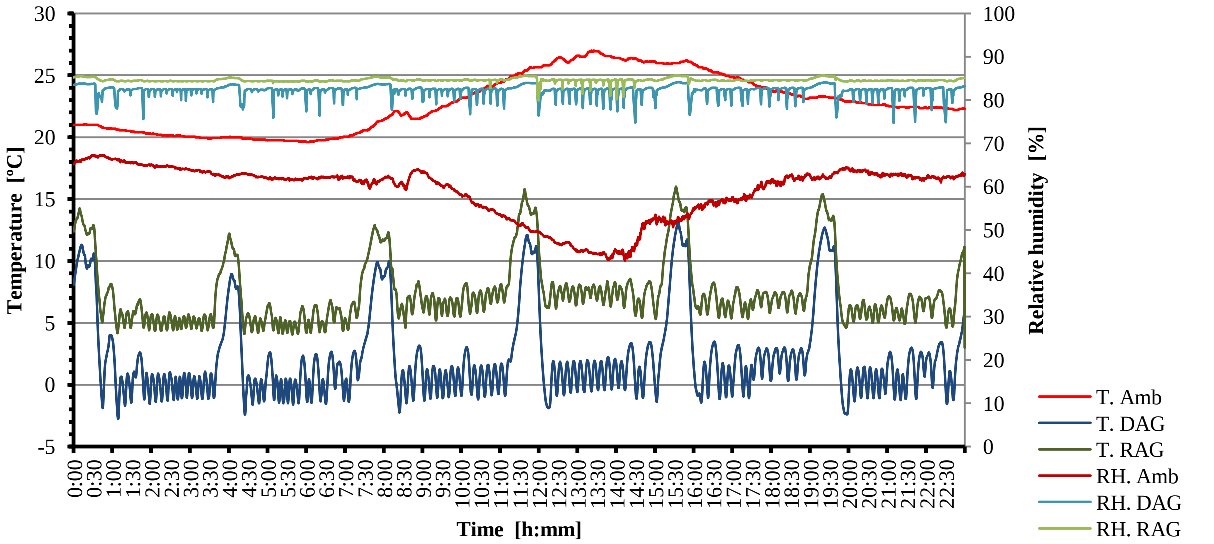

Figure 2. Average experimental results of air temperature and relative humidity - Case study: Store closed.

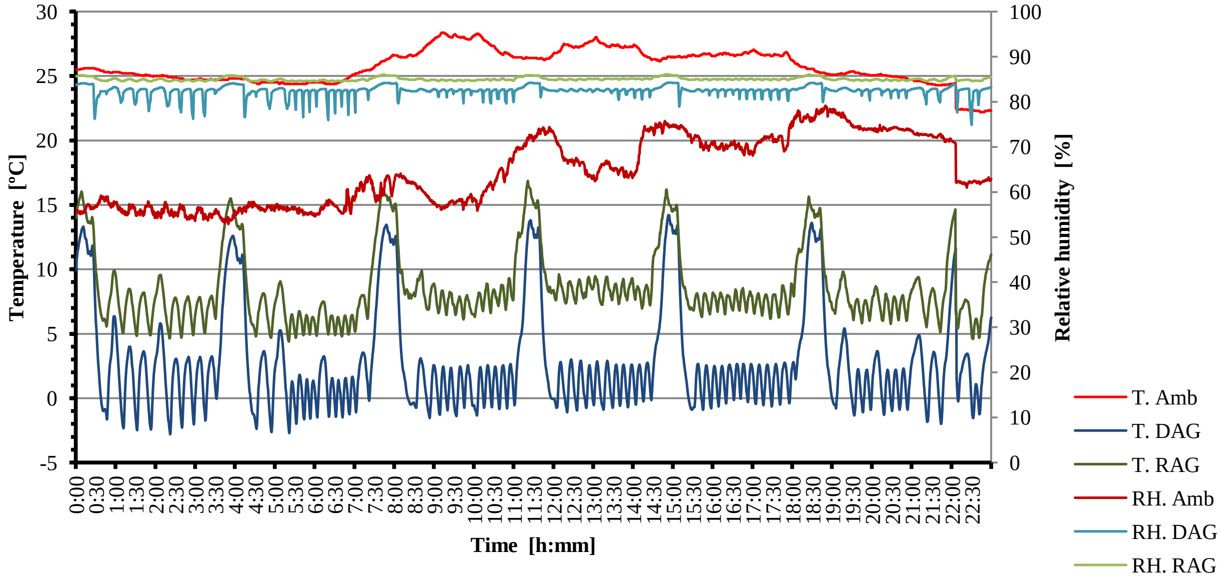

Figure 3. Average experimental results of air temperature and relative humidity - Case study: Store open.

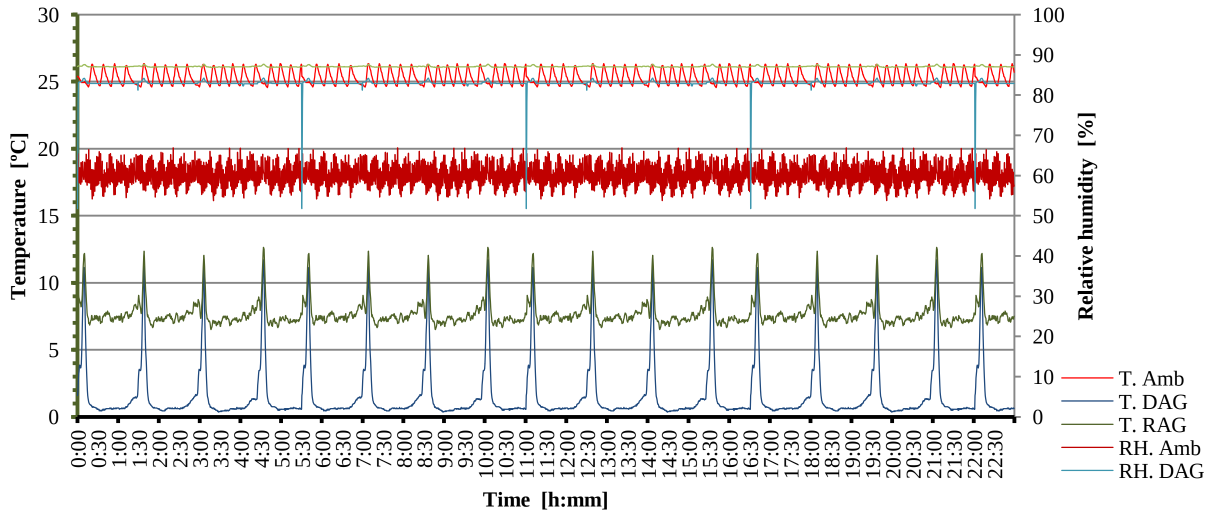

Figure 4. Daily average experimental results of air temperature and relative humidity - Case study: Climate class n.º 3 (25°C; 60%RH) (EN ISO 23953, 2005).

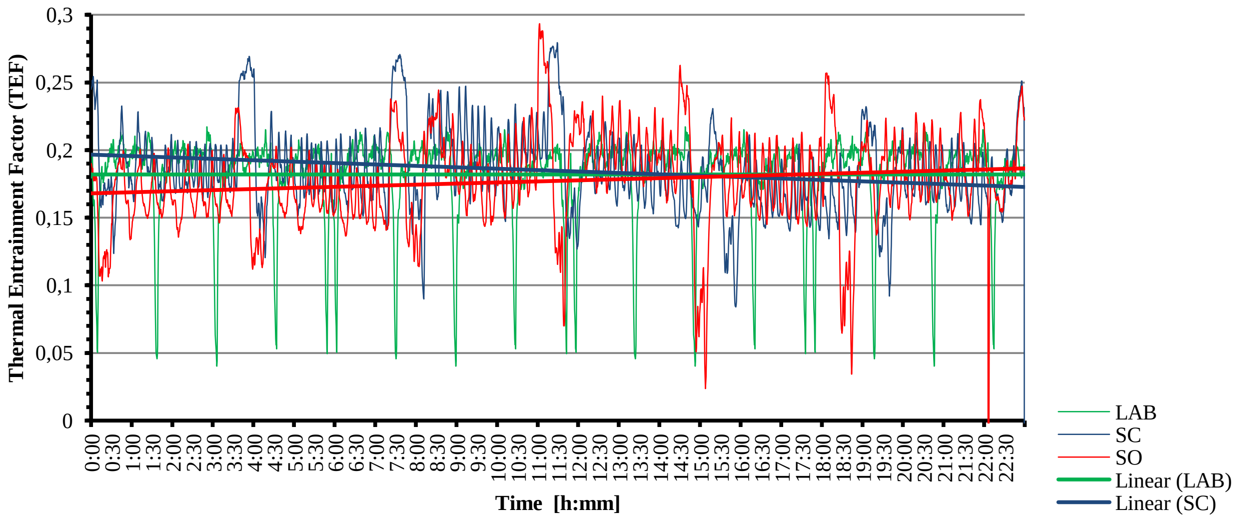

Figure 5. TEF values and its linear trend for the different case studies: Store closed (SC), Store open (SO) and laboratory (LAB).

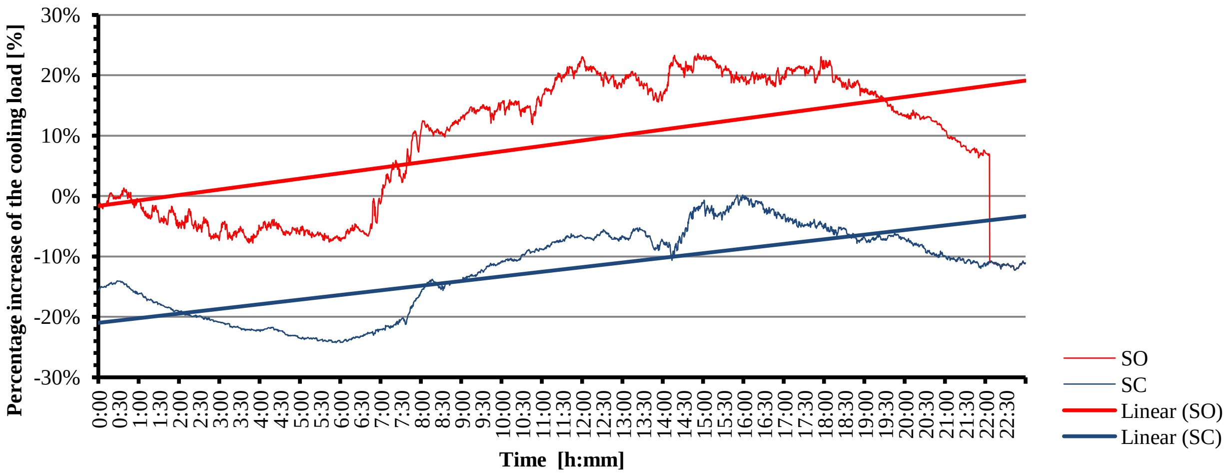

Figure 6. Percentage increase in the cooling load of case study: Store closed (SC) and Store open (SO) in relation to case study: Laboratory (LAB).

4.2. 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 Figures 3 and 4, the time period in when the temperature difference between DAG and RAG is greater coincides with the time period when the ambient air temperature increases. Figure 5 shows the air temperature variation of test conducted according to EN ISO 23953 (2005). In Figure 5, it is shown a cyclic operation of the ORDC, 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 ORDC, it is necessary to develop standard tests in a controlled environment.

In Figure 6 it is possible to evaluate the air curtain performance by the TEF values. There are fluctuations in the TEF values when the ORDC is exposed to real conditions in a store without air conditioning, both closed or open. Additionally, the linear trend of TEF increases when the store is open to public. The time period when the TEF value is greater also coincides with the time period when the temperature value is higher.

Figure 7 shows the percentage increase of cooling load when the store is open and closed to public. The two curves show oscillations throughout the time 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 time period. Thus, the power consumption of the ORDC 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.

5. CONCLUSION

In Brazil, due to its territorial extension, there are very significant different climate regions. Even with this climate difference, small supermarkets or store, mostly don't 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 cabinet, 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 cabinets 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.

NOMENCLATURE

b	air curtain width	(m)	Subscripts	
g	Gravitational accelerations	($m \cdot s^{-1}$)	amb	ambient
Gr	Grashof number	(-)	DAG	Discharge air grille
H	Air curtain height	(m)	ET	Experimental test
i	enthalpy	($J \cdot kg^{-1}$)	HX	Heat exchanger
$\dot{m}$	Mass flow rate	($kg \cdot h^{-1}$)	IB	Insulating body
$\dot{Q}$	Thermal power	(W)	PAC	Primary air grille
Re	Reynolds number	(-)	RAG	Return air grille
Ri	Richardson number	(-)	SAC	Secondary Air Curtain
RH	Relative humidity	(%)	SH	Shelve
T	Temperature	(K)	sim	simulator
u	Velocity	($m \cdot s^{-1}$)	tot	total
V	Airflow rate	(m^3/h)		
X	Dimensionless temperature	(-)		

Abbreviation

CFD	Computational fluid Dynamics
DAG	Discharge Air Grille
RAG	Return Air Grille
PBP	Perforated Back Panel
TEF	Thermal Entrainment Factor
PAC	Primary Air Curtain
SAC	Secondary Air Curtain
HVAC	heating, ventilation, and air conditioning

Greek symbols

θ	Airflow direction	(º)
ν	Kinematic viscosity	($m^2 \cdot s^{-1}$)
ξ	Thermal expansion coefficient	(K^{-1})
β	Back panel airflow ratio	(-)
Φ	Heat Extraction Rate	(W)

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