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Experimental study of heat transfer by natural convection in a closed cavity: application in a domestic refrigerator

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Abstract

An experiment was carried out using a refrigerator model in which heat is transferred by natural convection between a cold vertical wall and the other walls, which are exposed to heat losses. The air temperature profile in the boundary layers and in the central zone of the empty refrigerator model was investigated. Temperature stratification in the vertical direction was observed with the cold zone at the bottom and warm zone at the top of the cavity. The effects of temperature and the surface area of the cold wall were studied.

In order to study the effect of obstacles on temperature profiles, the refrigerator model was filled with 4 blocks of hollow spheres. Temperature profiles in this case were compared with the results with no blocks. The air temperature is lower almost everywhere in the model containing blocks. The presence of the blocks seems to enhance heat transfer particularly near the cold wall.

Keywords: Natural convection; Closed cavity; Temperature profile; Domestic refrigerator

1. Introduction

Natural convection is a phenomenon frequently encountered during the shelf life of foodstuff, for example, during transport in a container, during storage in an unventilated cold room or in a domestic refrigerator. In practice, the high temperature is frequently observed in the cold chain, particularly in domestic refrigerators where foods spend the longest part of their shelf life. Two types of domestic refrigerators are available on the market: static and ventilated. In Europe, the static system (without ventilation) is widely used. For this refrigerator type, heat is transferred principally by natural convection and airflow is due to variations in air density. These variations are related principally to the temperature and humidity gradients. The vertical force, which results from air weight and buoyancy, is ascendant if air is locally lighter than the average and descendant where the opposite is true (hot/humid air is lighter than cold/dry air). Surveys carried out in the United Kingdom, Netherlands, Greece, Northern Ireland, and New Zealand show that many refrigerators operate at a high temperature; for example, in France, 26% of refrigerators run at an average temperature above 8 °C, whereas the temperature specified in standards is 4 °C (Laguerre, Derens, & Palagos, 2002). Product temperature is a quality and safety-determining factor, and it is therefore necessary to fully understand the mechanism of heat transfer.

Knowledge of air temperature and velocity profiles in a refrigerator is important for food quality control. In fact, if the consumer knows the position of warm and cold

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Nomenclature

g	acceleration due to gravity (9.81 m/s^2)
H	height (m)
K	permeability of the porous media (m^2)
L	width (m)
N_{RC}	radiation–convection interaction parameter
Ra	Rayleigh number
Ra_p	Rayleigh number of porous media
Ra_c	critical Rayleigh number
T	temperature (°C or K)
T_c	cold wall temperature (°C or K)
T_h	hot wall temperature (°C or K)
T_{amb}	ambient temperature (°C or K)
T^*	dimensionless temperature
ΔT	temperature difference between the cold and warm walls

Greek symbols

λ	thermal conductivity of air (W/m/K)
α	thermal diffusivity of air (m^2/s)
α_p	thermal diffusivity of the porous media (m^2/s)
β	thermal expansion coefficient (K^{-1})
v	kinematic viscosity (m^2/s)
ε	surface emissivity of wall
σ	Stefan–Boltzmann constant (=5.667×10^{-8} W/m^2/K^4)

zones in the refrigerator, the product can be placed correctly. Knowledge of the thickness of thermal and hydrodynamic boundary layers near the evaporator and the other walls is also important. If the product is too close to the evaporator wall, freezing can occur, and if it is too close to the other walls, there may be health risks.

This study was carried out in order to develop experimental data on temperature distribution in a static domestic refrigerator (without a fan). The objective was to gain a better insight into the mechanism of heat transfer by natural convection in the refrigerator and to use these data to validate a modelling. To achieve this objective, an experiment was carried out using a refrigerator model, which makes it possible to observe the same phenomena as those in a domestic refrigerator but with better-controlled boundary conditions and simpler geometry.

In practice, food products stored in a domestic refrigerator have different forms, dimensions (generally form 5 to 20 cm) and occupied volumes (from nearly empty to nearly full). In order to better understand the phenomena, firstly, a simple configuration such as an aligned stack of spheres was studied.

Temperature cartography was established in an empty refrigerator model and a second refrigerator model loaded with hollow spheres. Comparison of the results obtained for these two cases makes it possible to determine the influence of the presence of obstacles on heat transfer in the refrigerator. Moreover, the influence of two parameters was studied: the surface area of the cold wall (parameter related to design) and its temperature (parameter related to thermostat setting by consumer).

2. Literature review

Firstly, a literature review on natural convection in empty closed cavities and in cavities filled with porous media will be presented. Some limits of the application of these studies to our case (refrigerator loaded with a food product) will also be given.

2.1. Empty closed cavity

Several experimental studies were carried out to measure air temperature and/or velocity in closed cavities (Tian & Karayiannis, 2000; Ampofo & Karayiannis, 2003; Betts & Bokhari, 2000; Mergui & Penot, 1996; Armaly, Li, & Nie, 2003).

Ostrach (1988), Catton (1978) and Yang (1987) carried out a literature review on this subject, which presents the experimental and modelling results (2D and 3D). These authors emphasize the importance of the aspect ratio of the cavity and the temperature difference between walls on the flow regimes. The flow regime in natural convection is characterised by the Rayleigh number (Ra) defined as:

$$Ra = \frac{g\beta\Delta T L^3}{\alpha v}$$

In general, the critical Rayleigh number, which distinguishes the transition from laminar to turbulent flows, is approximately 10^9 (depending on the geometry and boundary conditions, Incropera & Dewitt, 1996).

When the bottom horizontal wall is cold, stable temperature stratification is observed in the cavity (cold zone at the bottom and warm zone at the top), and there is no airflow. When the upper horizontal wall is cold, unstable flow is observed (Ostrach, 1988) since heavier air is above the lighter one. The state of unstable equilibrium occurs until a critical density gradient is exceeded. A spontaneous flow then results that eventually becomes steady and cellular-like. When a vertical wall is cold, circular flow is observed along walls and the air is almost stagnant at the centre of the cavity; thermal stratification is also observed. This case is similar to that of a domestic refrigerator, since the evaporator is often inserted in the vertical back wall.

There are fewer experimental studies on natural convection than on forced convection due to experimental difficulties in terms of metrology for low velocity and design of experimental devices maintaining given wall conditions. In fact, measurement is very sensitive to experimental and boundary conditions. In spite of that, at Eurotherm Seminar (Henkes & Hoogendoorn, 1993), some experimental results obtained with a standard case were compared ($Ra = 5 \times 10^{10}$, cavity aspect ration $H/L = 1$ in 3-dimension, adiabatic horizontal walls). Good agreement between results was found, particularly about temperature and velocity in the boundary layers.

Ramesh and Venkateshan (2001) used a differential interferometer to visualize conditions in the boundary layer along the wall ($10^5 < Ra < 10^6$). They found that it is generally stable except in the corner. Mergui and Penot (1996) carried out a visualization of flow in an empty cavity using a laser tomography ($Ra = 1.7 \times 10^9$); they observed the same phenomena as Ramesh and Venkateshan (2001).

Deschamps, Prata, Lopes, and Schmid (1999) reported that in a domestic refrigerator, the Rayleigh number varies from 10^8 and 10^9, and that flow is therefore in the transition regime between laminar and turbulent flow.

Heat exchange by radiation between internal walls of the cavity is as important as that achieved with natural convection and this should be taken into account. Several authors (Balaji & Venkateshan, 1994; Ramesh & Venkateshan, 1999; Velusamy, Sundarajan, & Seetharamu, 2001; Li & Li, 2002) showed by experimental and numerical approaches that these two heat transfer modes occur simultaneously. Ramesh and Venkateshan (1999) showed experimentally that for a square enclosure (vertical walls maintained at 35 and 65 °C, adiabatic horizontal walls, $Ra = 5 \times 10^5$), the heat transfer by convection and radiation between high emissive vertical walls ($\varepsilon = 0.85$) is twice that between polished ones ($\varepsilon = 0.05$). Balaji and Venkateshan (1994) proposed correlations (established from numerical simulations) to express the convection and radiation in a square cavity in function of $\varepsilon, Ra, T_c/T_h$ and a radiation–convection interaction parameter ($N_{RC} = \frac{\sigma T_h^4 H}{\lambda(T_h - T_c)}$).

These correlations show that the radiation effect increases when the wall emisivity and/or wall temperatures increase. Moreover, Li and Li (2002) reported that the radiation increases in comparison with convection as the size of the enclosure increases. An estimation of convection and radiation heat transfer in a refrigerator was carried out in our previous study (Laguerre & Flick, 2004), which confirms the importance of radiation.

2.2. Cavity completely or partially filled with porous media

Several reviews on heat transfer by natural convection in a cavity filled with porous media have been carried out: Nield and Bejan (1992), Cheng (1979), Kaviany (1991) and Oosthuizen (2000).

In the case of porous media, the Rayleigh number is defined as:

$$Ra_p = \frac{g\beta\Delta T \cdot H \cdot K}{\alpha_p \nu}$$

When the Rayleigh number is less than a critical value (Ra_c), the heat transfer is principally by conduction. When $Ra_p > Ra_c$, airflow is observed and leads to heat transfer by convection. Oosthuizen (2000) reported a value of $Ra_c = 40$ in a rectangular cavity heated from below.

Airflow in the cavity filled with porous media is generally laminar. Circular flow, similar to that of an empty cavity, is observed principally in the boundary layer along the walls (Fig. 1). Velocity is much smaller at the centre of the cavity.

Literature concerning heat transfers in porous media and in packed beds (Padet, 1997; Rohensenow, Hartnett, & Cho, 1998, Chapter 3 & 4; Wakao & Kaguei, 1992) presents several approaches taking into account conduction, convection and radiation. Moreover, these studies distinguish the one-temperature models, in which local equilibrium between product and air is supposed, from the two-temperature models, in which different temperatures represent product and air.

2.3. Food porous media

Typically, porous media models can be applied to study the airflow and heat transfer in the case of small products, for example, grain storage in a silo; these models can improve understanding of water condensation phenomena which contribute to mould growth in certain

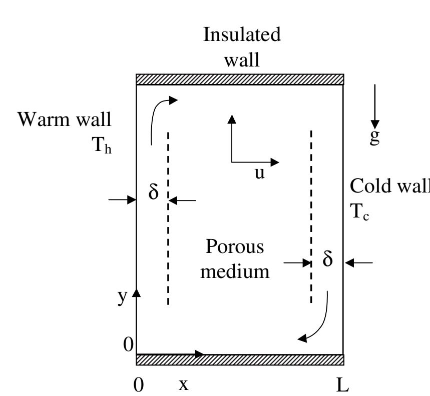

Fig. 1. Two-dimensional rectangular porous layer held between differentially heated side walls (Source: Bejan, 1984).

zones (Cruz & Akins, 1983; Nguyen, 1987; Thibaud, 1988; Dona & Stewart, 1988;, Fohr & Moussa, 1994).

Some particular precautions must be taken to study airflow and heat transfer in stacks of food such as fruit and vegetables, which are larger than grain. In the entrance zone, the air passage is abruptly reduced; this contributes to the increase in heat exchange (Ben Amara, Laguerre, & Flick, 2004). Taking into consideration the product size, this zone occupies an appreciable volume in the stack, whereas it is generally small and neglected in porous media. These phenomena contribute to product temperature heterogeneity observed in food stacks (Alvarez & Flick, 1999).

Moreover, a temperature gradient between the surface and the center of the product that depends on its shape is often observed (Tang & Johnson, 1992). Therefore, the one-temperature and two-temperature models cannot be applied in such cases.

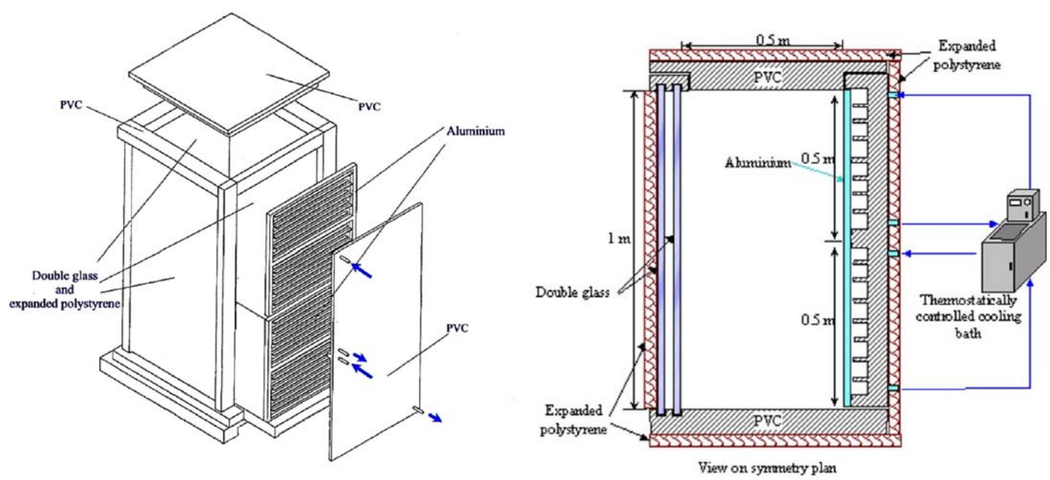

Fig. 2. Refrigerator model.

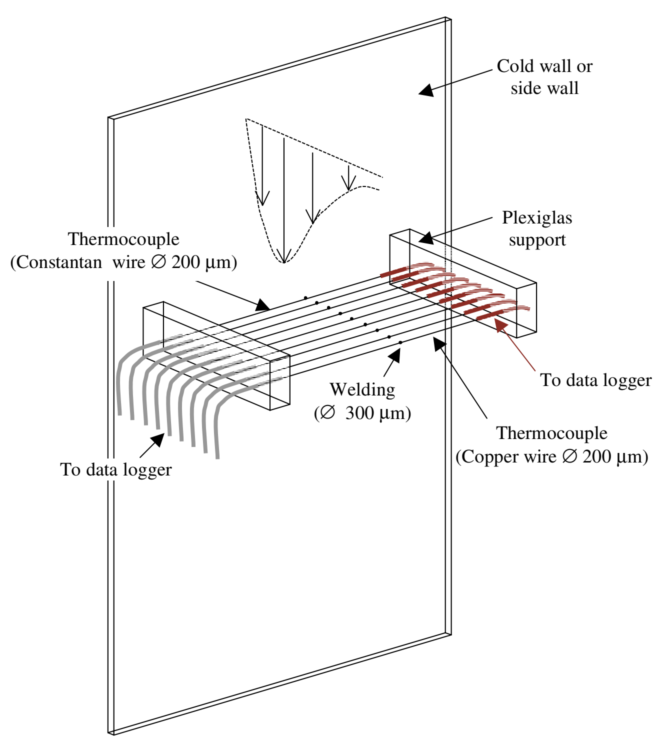

Fig. 3. Device used for temperature profile measurement in the boundary layer (measurement carried out in an empty refrigerator model).

2.4. The domestic refrigerator

Concerning domestic refrigerators, several studies have been carried out on the cold production system (Chen, Wu, & Sun, 1996; Radermacher & Kim, 1996; Graviss & Zurada, 1998; Alsaad & Hammad, 1998; Bansal, Wich, & Browne, 2001; Grazzini & Rinaldi, 2001). The principal objective of these studies is to optimize energy consumption. Some numerical studies have been carried out on heat transfer in empty domestic refrigerators (Pereira & Nieckele, 1997; Silva & Melo, 1998; Deschamps et al., 1999). However, few studies have been carried out on loaded refrigerators.

Masjuki et al. (2001) and James and Evans (1992) carried out experimental studies on empty and loaded refrigerators. The objective was to analyze the effects of several parameters on the temperature in the refrigerating compartment (thermostat setting, frequency of door openings, filled volume, temperature and humidity of ambient air). However, it is difficult to understand the mechanism of heat transfer by natural convection from the results obtained, due to the complexity of the refrigerator operation (compressor "on" and "off" cycles, different degrees of insulation in walls, heat loss through gaps etc.).

This study proposes to establish experimental data using a device in which the same phenomena as those encountered in a domestic refrigerator can be observed, but simple and well-controlled conditions are used. The dimensions of this refrigerator model are of the same order of magnitude as those of a real one. The results obtained from our study can be used to validate a natural convection model in a food stack and this model can then be applied to a domestic refrigerator.

3. Material and methods

3.1. Refrigerator model

The internal dimensions of the refrigerator model are: $0.5 \times 0.5 \times 1$ m (length × width × height). It is composed of 3 vertical double glass walls (glass thickness: 6 mm and air thickness between glass walls: 10 mm) and one vertical aluminum wall (thickness 2 cm, containing a coil) see Fig. 2. A low-temperature water–glycol mixture was prepared in a thermostatically controlled cooling bath to maintain a constant temperature in this aluminum wall. The top and bottom horizontal walls are

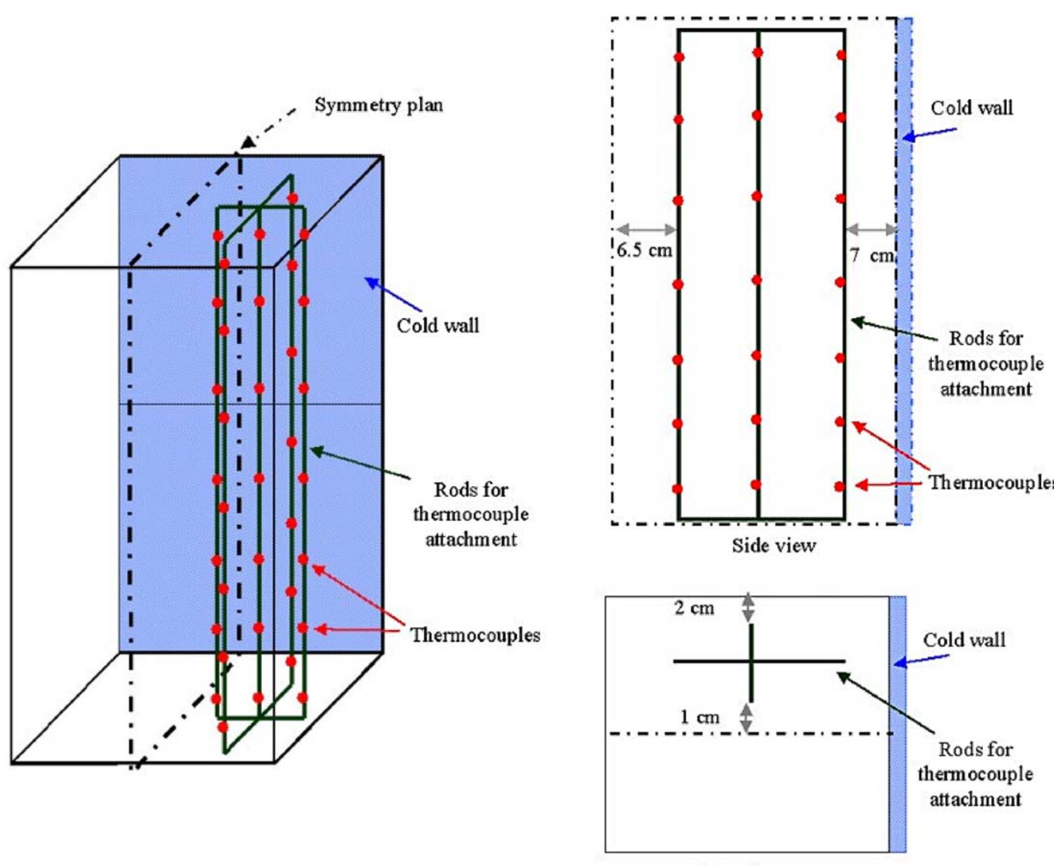

Fig. 4. Diagram showing thermocouple positions on vertical rods (measurements carried out in an empty refrigerator model).

made of PVC (thickness: 2 cm). All external walls are insulated using expanded polystyrene plates (thickness: 4 cm). These plates can be taken off to visualize airflow and allow air velocity measurement by optical methods through the double glass walls.

The flow circuit of the water–glycol mixture is ensured in such a manner that either the total surface of aluminum plate or only the upper half is cooled.

3.2. Experimental assembly

3.2.1. Measurement of temperature profiles in the boundary layers (empty refrigerator model)

Air temperature was measured using calibrated T type thermocouples (200 μm diameter, precision ± 0.2 °C), with the wires tightened near the wall as shown in Fig. 3. In this manner, airflow was very slightly dis-

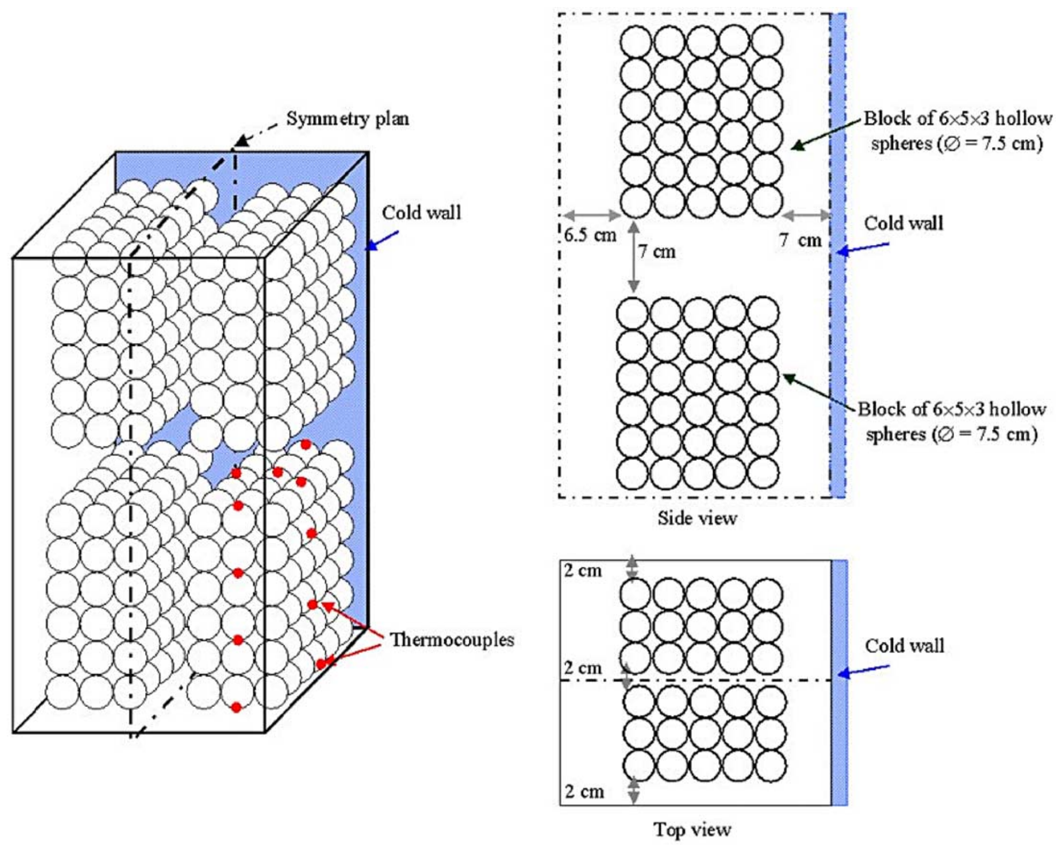

Fig. 5. Diagram showing the positions of 4 blocks of hollow spheres.

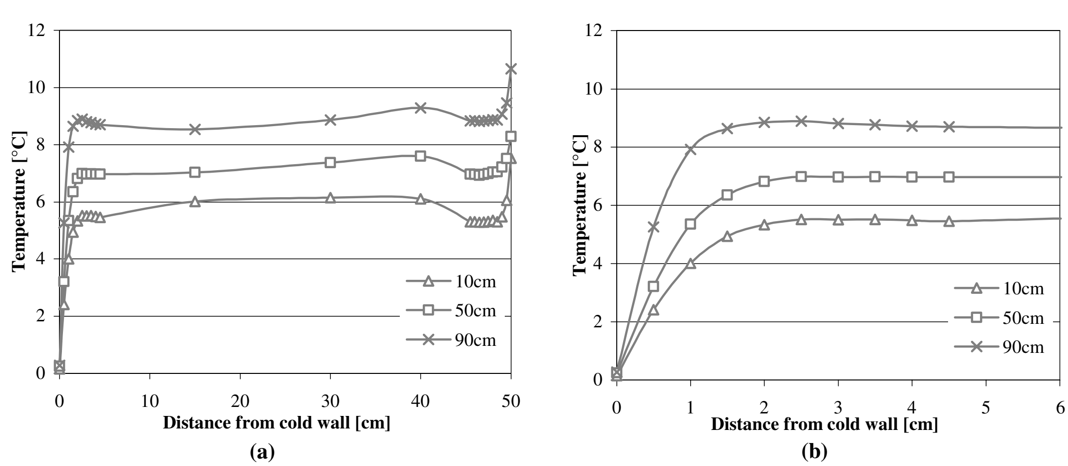

Fig. 6. Horizontal temperature profile over 3 heights on the symmetry plane of an empty refrigerator model, with the entire cold wall maintained at 0 °C (a) overall view and (b) thermal boundary layers near the cold wall.

turbed by the presence of the thermocouples (the thermocouples occupy only 4% of the flow cross-section). Nine thermocouples were attached to a Plexiglas support on each side (5 ± 0.1 mm space between thermocouples) and the height level of thermocouples can be adjusted. In our experiment, the temperature profiles at 3 heights were measured: 10 cm, 50 cm and 90 cm (precision ± 0.2 cm).

3.2.2. *Measurement of vertical temperature profiles far from walls: empty refrigerator model*

Seven previously calibrated thermocouples were attached to a vertical rod (diameter: 4 mm, length: 1 m, precision ± 0.2°C) and five rods were used to measure air temperature at different locations (Fig. 4). The tips of these thermocouples projected 1 cm from the rod.

In order to verify the disturbance due to the presence of rods, the temperature measured by a thermocouple fixed on a rod was compared with that measured by a thermocouple fixed on a plastic thread at the same location. The same temperature values were observed (temperature variation in the two cases <0.2 °C).

3.2.3. *Measurement of the vertical temperature profile far from walls: refrigerator model filled with hollow spheres*

The refrigerator model was filled with 4 blocks of hollow PVC spheres. The diameter of each sphere was 7.5 cm and each block was composed of $6 \times 5 \times 3$ spheres (Fig. 5). In steady state, the heat transfer between air and the spheres can be supposed to be negligible. These spheres are only airflow obstacles and are inert to heat exchange; they modify nevertheless the air temperature field.

Only one block was instrumented by thermocouples to minimize flow disturbance due to the presence of many thermocouples in the refrigerator model. The position of these thermocouples was the same as that of the

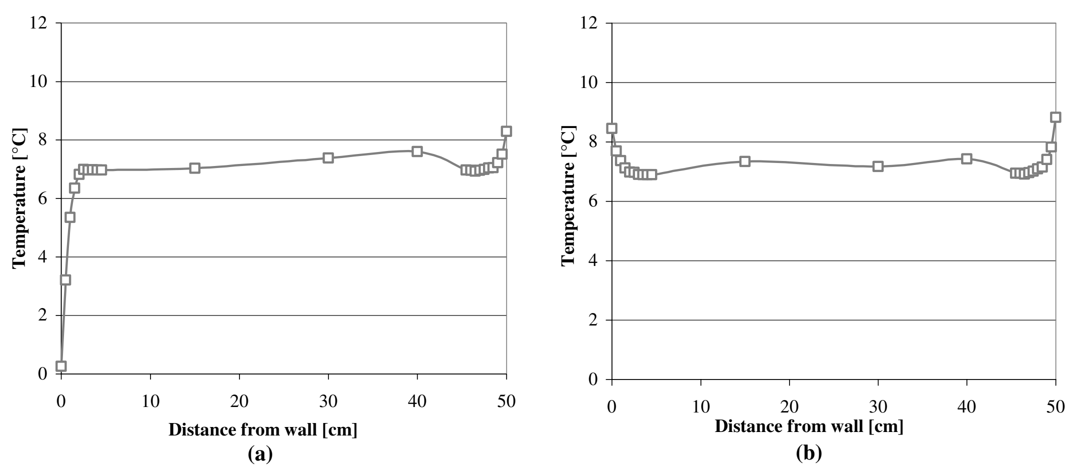

Fig. 7. Comparison of horizontal temperature profiles at mid-height (50 cm) of an empty refrigerator model (with the entire cold wall maintained at 0 °C) (a) between the cold wall and the glass wall opposite and (b) between the two sidewalls.

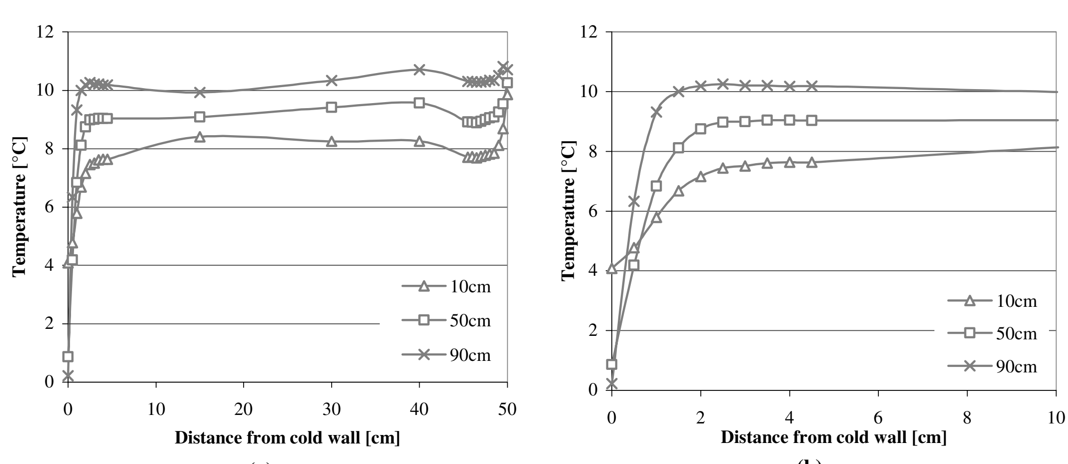

Fig. 8. Horizontal temperature profile over 3 heights on the symmetry plane of an empty refrigerator model; only the top half of the cold wall was maintained at 0 °C (a) overall view and (b) thermal boundary layers near the cold wall.

empty case. The position of this instrumented block was changed in order to establish the temperature cartography over the entire height of the refrigerator model.

3.3. Experimental conditions

The experiment was carried out in a controlled temperature room. Ambient temperature (T_{amb}) was set at 20 °C for all experiments. Two cold wall temperatures were studied: −10 °C (corresponding in general to the evaporator temperature at the end of the compressor work cycle) and 0 °C (corresponding to an average evaporator temperature during an ''on'' and ''off'' cycle).

For domestic refrigerators, the surface area of the cold wall can vary from half to the entire back vertical wall. For our study, the aluminum wall was cooled either totally (100%), or only the top half (50%) was cooled.

It is to be emphasized that steady state regime was reached for all experiments. Therefore, the refrigerator

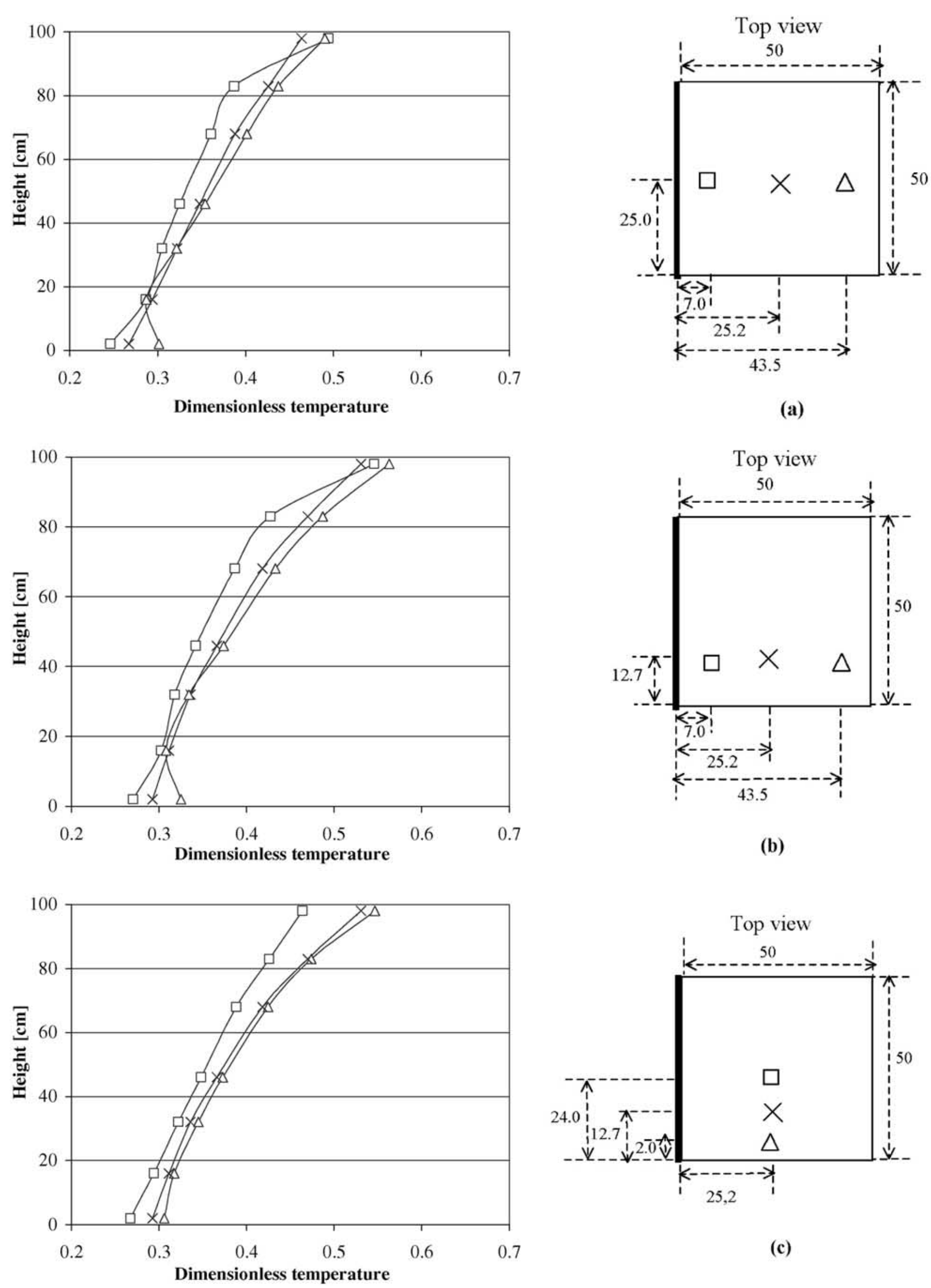

Fig. 9. Vertical air temperature profile in an empty refrigerator in various positions, with the entire cold wall maintained at −10 °C (dimension in cm).

model was cooled until that the internal air temperature became constant. Then the average air temperature at each measured point was calculated over a stable period (average of 100 measurements over 1.5 h).

The Rayleigh number in the refrigerator model varied between 1.92×10^8 and 3.04×10^8. These values were calculated using the width of refrigerator model (0.5 m) and the temperature difference between the cold wall (0 or −10 °C) and the centre of the glass wall situated oppositely (varying from 4 to 11 °C).

4. Results and discussion

In order to compare the results obtained for different cold wall temperatures (T_c), the dimensionless value $\left(T^* = \frac{T-T_c}{T_{amb}-T_c}\right)$ is used in the curves shown hereafter.

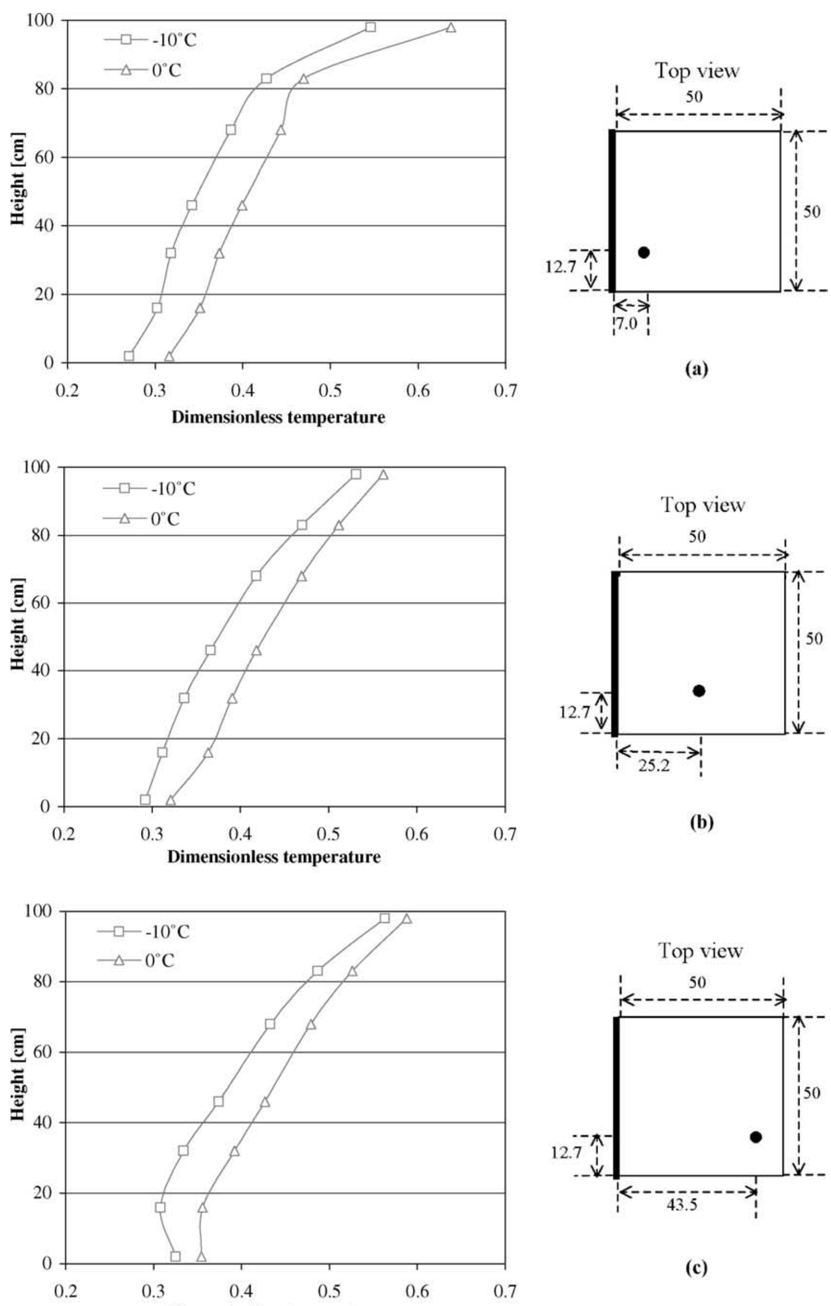

Fig. 10. Influence of cold wall temperature (T_f) (a) near the cold wall, (b) in the middle between the cold wall and the glass wall opposite and (c) near the glass wall, with the entire cold wall maintained at −10 or 0 °C (dimension in cm).

The inaccuracy of temperature measurement leads to $\leqslant$4% error on T^* and Ra.

4.1. Empty cavity

4.1.1. Temperature profile in the boundary layers: empty refrigerator model

The air temperature profile in the boundary layer was measured near the cold wall and near the glass wall situated opposite; this was carried our over 3 heights: 10, 50 and 90 cm.

When the entire cold wall was used (Fig. 6), the temperature at the centre of the refrigerator model was constant at a given height and increases with the height. The thickness of the boundary layer near the cold wall was around 2 cm. This layer was slightly thinner (around 1.5 cm) near the glass wall. This can be explained by the fact that there is less temperature variation in the

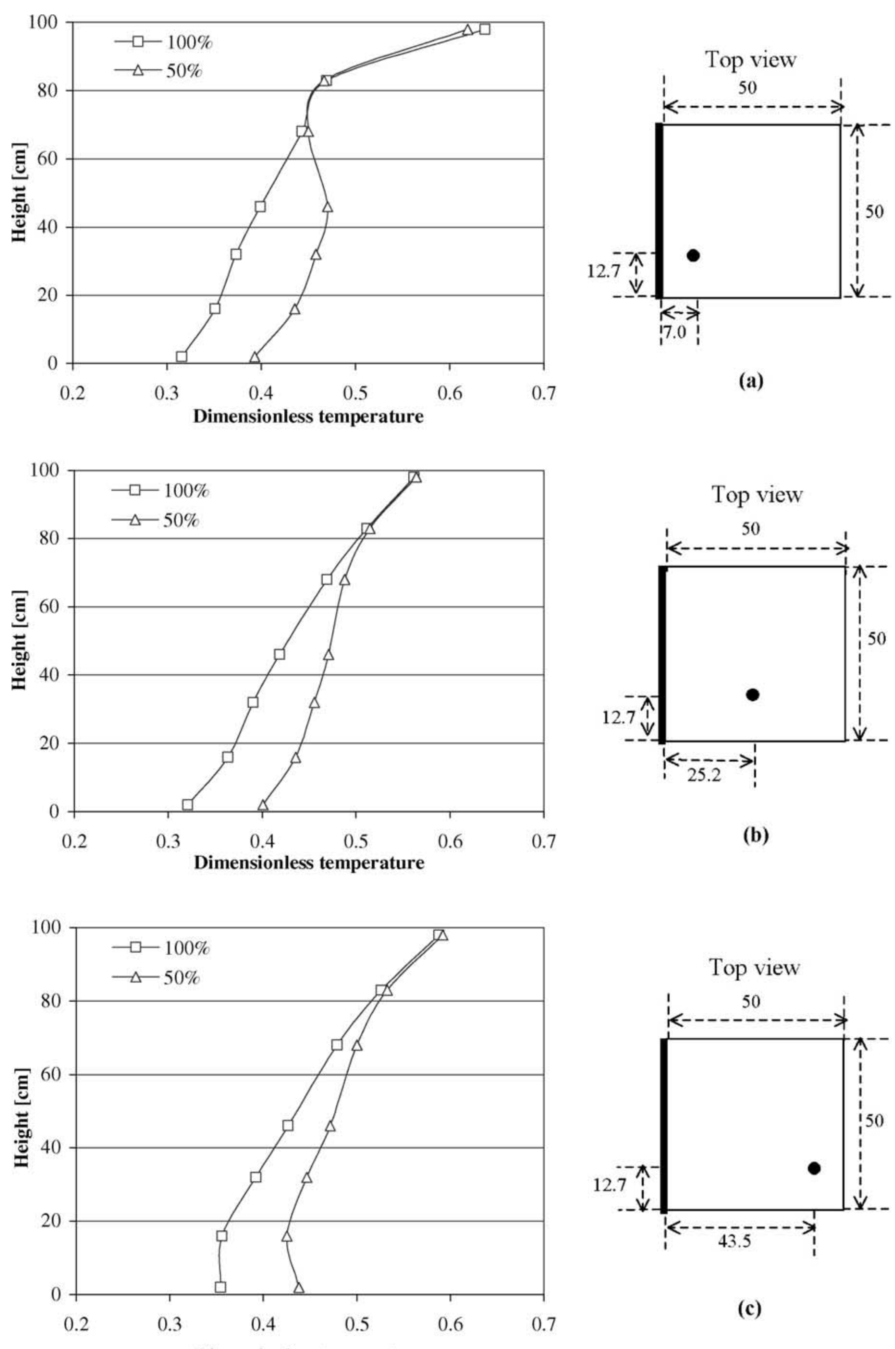

Fig. 11. Influence of the surface area of the cold wall (a) near the cold wall, (b) in the middle between the cold wall and the glass wall opposite and (c) near the glass wall, with the cold wall maintained at 0 °C (dimension in cm).

boundary layer near the glass wall (around 2 °C) than that near the cold wall (6 to 9 °C according to the height). According to the theory, the thickness of boundary layer approaches zero at the top of the cold wall and increases progressively towards the bottom. In our study, there was a 5 mm space between thermocouples; thus the thickness of the boundary layer is difficult to estimate accurately using our device.

The temperature profile at mid-height (50 cm) measured on one hand between the cold wall and the opposite glass wall and on the other hand between the two lateral glass walls are compared (Fig. 7). It is observed that the temperature profile in the boundary layer near the glass walls is similar whatever its position, and this profile is different from that observed near the cold wall.

When only the top half of the cold wall is cooled (50%) (Fig. 8), compared with the first case (100% of the cold surface), the temperature at the centre of the refrigerator model increases by an average of 2 °C. But the same temperature profile in the boundary layer and the same order of magnitude of thickness are observed except at a height of 10 cm, since at this height the cold plate is not supplied by the water–glycol mixture; the surrounding air temperature is therefore, higher (around 4 °C).

4.1.2. Temperature distribution in the central zone: empty refrigerator model

In order to obtain overall information on air temperature distribution in the refrigerator model, the detailed results of an experiment carried out when the entire cold wall is maintained at −10 °C are presented as an example in Fig. 9.

It can be observed that on the symmetry plan (Fig. 9a), the temperature increases with the distance from the cold wall. The vertical temperature profile shape near the cold wall is different from that near the glass wall opposite, particularly at the bottom of the refrigerator cavity. This may indicate recirculation at this location. This phenomenon was also observed in a numerical study performed by Sun and Emery (1997). The air temperature at the top of the refrigerator cavity is relatively homogeneous, whatever the distance from the cold wall, which may indicate air stagnation at this location. Moreover, the vertical profile at the centre of the cavity is closer to that near the opposite glass wall than that near the cold wall, except at the bottom. The vertical temperature profile on the middle plan between the symmetry plan and the side glass wall is shown in Fig. 9b. This temperature is higher than that on the symmetry plan but the profile shape is similar. The vertical temperature profile over the width of the refrigerator model is shown in Fig. 9c. It can be seen that the temperatures at 2 cm and at 12.7 cm from the side walls are practically identical; this observation confirms that the thickness of boundary layer (near the side walls) is less than 2 cm.

By measuring the temperature profile on the left and on the right of the cavity, temperature symmetry was verified.

4.1.3. Influence of the cold wall temperature

The air temperature stratification in the refrigerator cavity during two experiments carried out while the entire cold wall was maintained at −10 °C and 0 °C is presented in Fig. 10. The dimensionless temperature is appreciably lower for −10 °C than for 0 °C, whatever the height and the distance from the cold wall. This can be explained by the fact that the temperature variation between the cold wall and ambient air is higher in the case of −10 °C. Thus, there is more air circulation by natural convection ($Ra = 3.04 \times 10^8$ for −10 °C and $Ra = 1.92 \times 10^8$ for 0 °C) near the cold wall. For the same reason, the convective exchange near the glass walls increases also. But this has a minor effect because heat transfer here is limited by the conductive resistance of the double glass and polystyrene insulation.

The curve "break" is still observed on the temperature profile at the bottom of the refrigerator model near the glass wall, whatever the cold wall temperature (Fig. 10c); this may be related to air recirculation. A high temperature is always observed at the top of the refrigerator model due to air stagnation.

4.1.4. Influence of the dimension of the cold surface

Two experiments using different cold wall surface areas were carried out: the entire aluminum plate was cooled (100%), or only the top half of it (50%); the temperature of this wall was maintained at 0 °C.

The vertical air temperature profiles are shown in Fig. 11. It can be observed that the difference between the two cases (50% and 100%) is significant from the bottom up to a height of 80 cm. But the temperature profiles are quite close from 80 cm to 100 cm (warm air stagnation zone). It can be seen that the "break" in the vertical

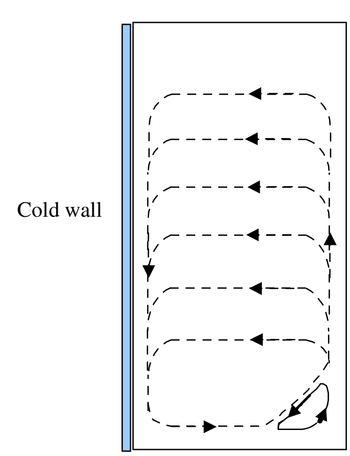

Fig. 12. Theoretical airflow pattern in the refrigerator model.

temperature profile near the glass wall is observed in both cases.

This comparison provides an insight into the influence of the position and the surface area of the evaporator on the temperature distribution in a domestic refrigerator i.e. there is no effect of surface area of the cold wall on the temperature profile at the top of the refrigerator model, but this effect is noticeable at the bottom.

The experimental data on air temperature makes it possible to propose a possible flow scheme for the refrigerator model. This flow is overall similar, whatever the temperature and the surface area of the cold wall (Fig. 12). It is composed of principal air circulation along the walls, transversal flow from the glass walls to the cold wall, at least recirculation in one corner and an air stagnation zone at the top. Of course, flow is 3

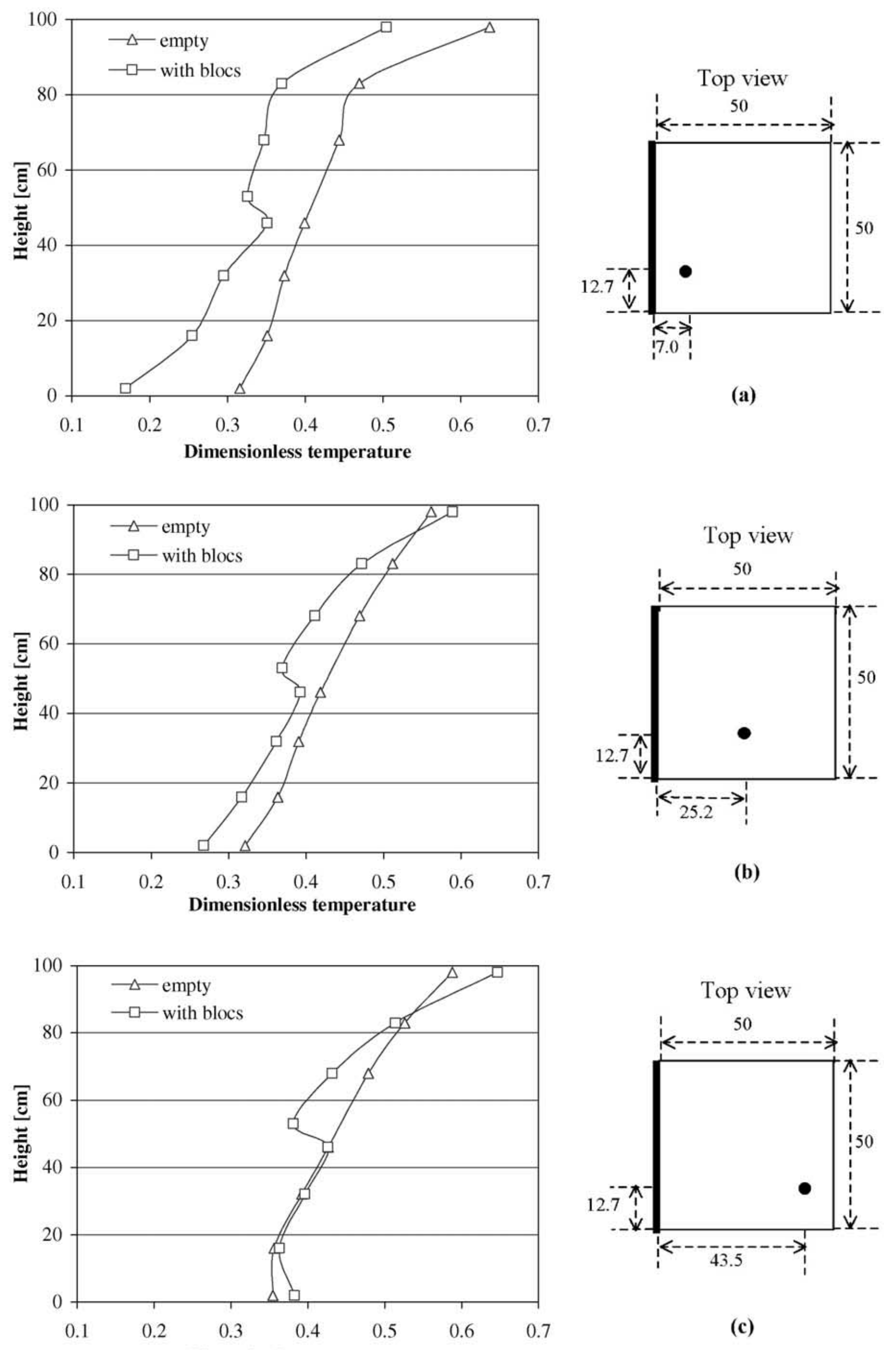

Fig. 13. Influence of obstacles in the refrigerator model, comparison of the temperature profile between the empty model and that filled with 4 blocks of hollow spheres (a) near the cold wall, (b) in the middle between the cold wall and the glass wall opposite and (c) near the glass wall, with the entire cold wall maintained at 0 °C (dimension in cm).

dimensional and this flow hypothesis should be confirmed by visualization and air velocity measurement.

4.2. Refrigerator model filled with hollow spheres

The air temperatures measured when the refrigerator model was empty or filled with 4 blocks of hollow spheres are compared in Fig. 13. It can be observed that near the cold wall (Fig. 13a), the air temperature in the presence of the blocks is lower than that without them. This can be explained by the fact that the blocks reduce the section available for airflow near the wall and this contributes to the velocity increase. Consequently, the convective exchanges with air, which flows in the space between the walls and the blocks, increase. This influence is more visible near the cold wall than near the glass wall situated opposite because the thermal resistance between air and the glass wall is low compared with that

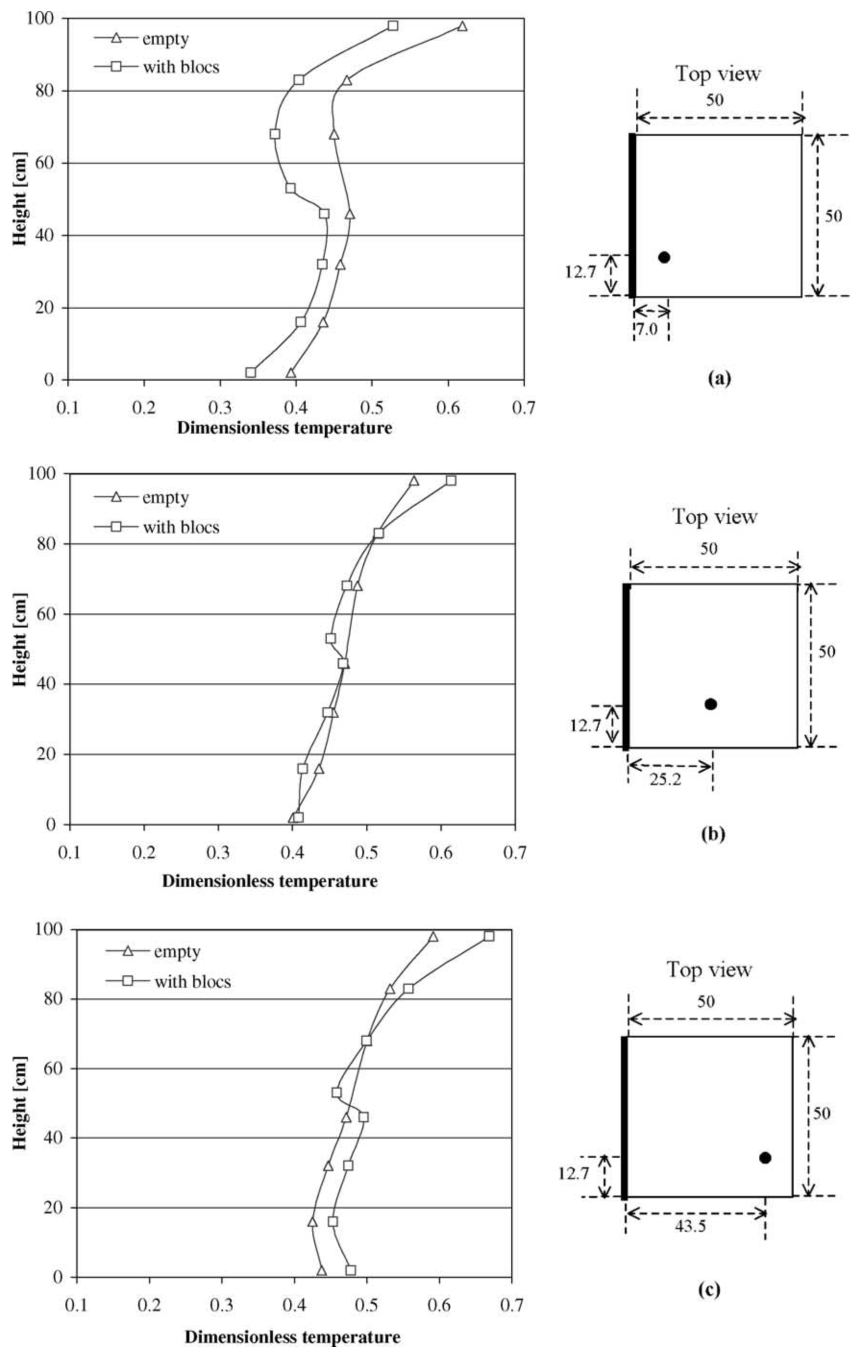

Fig. 14. Influence of obstacles in the refrigerator model, comparison of temperature profiles in the empty refrigerator and that filled with 4 blocks of hollow spheres (a) near the cold wall, (b) in the middle between the cold wall and the glass wall opposite and (c) near the glass wall, with the top half of the cold wall maintained at 0 °C (dimension in cm).

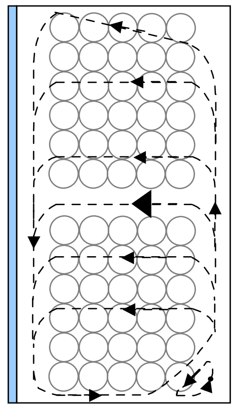

Fig. 15. Theoretical airflow pattern in the refrigerator model filled with 4 blocks of hollow spheres.

of the composite wall insulation (double glass and polystyrene plate).

The temperature at the centre of the blocks is lower than that measured in the empty refrigerator model at the same location (Fig. 13b), but this difference is less noticeable than near the cold wall (Fig. 13a). Near the glass wall, there is practically no difference in the bottom of the two cases (empty and with blocks) (Fig. 13c). A curve "break" is always observed at mid-height between the two blocks (Fig. 13a–c); this may be due to transversal airflow channeling (from the glass wall to the cold wall).

It can be seen that at the top of the refrigerator model, the air is warmer in the presence of the blocks than that in the absence of blocks (except near the cold wall). This may be due to the fact that the blocks prevent air circulation; thus, the stagnation zone is more developed. Similar observations are obtained in the case where only the top half of the aluminum wall is cooled (Fig. 14): near the cold wall, the temperature is lower in presence of blocks.

Finally, the experimental data on the vertical air temperature profile in the refrigerator model filled with 4 blocks of hollow spheres make it possible to propose a possible flow scheme (Fig. 15). This scheme should be confirmed by flow visualization and air velocity measurements.

5. Conclusions

The experimental results of temperature distribution in the refrigerator model confirm the theory that there is stratification: a warm zone at the top and a cold zone at the bottom. Therefore, food that presents microbiological risks should be placed in the bottom zone.

The temperature measurements inside the boundary layer make it possible to conclude that its thickness is approximately 2 cm. Therefore, the consumer should place food at least 2 cm from the refrigerator walls (evaporator or side walls) in order to avoid health risks and freezing.

Investigation of the cold wall temperature shows that the lower this temperature (compared with the ambient temperature), the higher the heat exchange intensity with the cold wall. This explains the reduction in the air temperature in the cavity, even in dimensionless terms, whereas in forced convection (with constant heat transfer coefficient), the dimensionless temperature would be independent of wall temperature.

The temperature is higher at the bottom of the refrigerator model when only the top half of the aluminum wall is cooled, compared with the totally cooled wall. However, the temperature difference in these two cases is not significant above a height of 80 cm.

The presence of obstacles significantly modifies the heat transfer in the refrigerator model. The obstacles enhance channeling near the wall and the convective heat exchange is thus improved. It was observed that for almost all measured points, the air temperature is lower in presence of blocks than without them. This makes it possible to conclude that the temperature of the filled refrigerator (containing cold food products) is not always higher than that of the empty one (for a given cold wall temperature). However, the presence of blocks increases the maximum air temperature observed at the top level; this may due to the fact that air stagnation is more marked in this case.

An attempt was carried out to compare our results with those obtained by other studies (Mergui & Penot, 1996; Tian & Karayiannis, 2000; Ramesh & Venkateshan, 2001). But, this comparison was delicate due to the differences of experimental conditions (dimension of cavity, wall insulation, wall and ambient temperatures).

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