

Comparison of the Effects of Air Flow and Product Arrangement on Freezing Process by Convective Heat Transfer Coefficient Measurement

Douglas Fernandes Barbin¹ and Vivaldo Silveira Junior²

¹*University College Dublin*

²*University of Campinas*

¹*Ireland*

²*Brazil*

1. Introduction

Since energy level could implement on the final cost of food products, the reduction of the freezing time is a major goal for industries and researchers. The product thermal load is generally the greatest of the demands on the refrigeration system, most of the heat removal occurring during the first hours of the process, mainly the freezing step.

Food products are predominantly frozen in air blast freezing tunnels. Air blast systems are based on fluid convection, where the cooling air flows around the product that must have the temperature reduced. Convection is the transfer of thermal energy by the movement of molecules from one part of the material to another. The rate of heat exchange between the bulk of a fluid and a known solid is given by (1):

$$\frac{dQ}{dt} = h_c A (T_b - T_\infty) \quad (1)$$

where Q is the amount of energy (J) drawn back per time t (s); h_c is the convection heat transfer coefficient ($W/m^2 \text{ } ^\circ\text{C}$); A is the heat transfer area (m^2); T_b is the product temperature ($^\circ\text{C}$) and T_∞ is the air temperature ($^\circ\text{C}$).

The convective heat flow is enhanced as the fluid motion increases. Air blast cooling processes are majorly ruled by the convective heat transfer, which relates the amount of transferred energy from the product surface to the cooling air. Wide variations in convective heat transfer coefficients may occur in different positions. Determination of this parameter can be a useful tool to measure the efficiency of the process.

In industrial situations, the freezing process of large amounts of food products may be very complex. Products to be cooled act like a barrier to air flow, and heat transfer can be compromised. A careful investigation of product characteristics and processing conditions must be conducted, as inadequate sample positioning and poor air circulation can cause longer cooling periods. If the system is not correctly designed, problems related to product or process qualities can appear, such as inadequate cooling or freezing rate, microbiological

growth, alterations in consistency and changes in its composition caused by chemical reactions during the subsequent storage. There is also need to consider proper packaging materials and the possibility of the existence of voids of air bubbles inside the package, as well as improper arrangement of products, which could lead to poor heat transfer and ineffective temperature reduction.

To produce homogeneous heat transfer it is important to have proper sample arrangement inside the system, as well as adequate air flow. Therefore, exhausting air may represent an economical choice for air blast systems, since it minimizes air short-circuiting, leading to more uniform cooling. Several implementations of forced air devices to aid cooling and freezing process have been reported. The most common method considers the use of a forced-air freezing tunnel that enhances the air flow around the products. Hence, the investigations of objective methods to determine the best design for cooling and freezing systems have been recently reported.

Frequently air velocity and air temperature are the parameters which regard most attention from designers of air blast systems. However, it is also important to consider the ways this air will flow through, as product distribution may directly affect the efficiency of cooling and freezing processes. Assessing the efficiency of freezing processes is a troublesome task. Recent researches have suggested that the convective coefficient, among other parameters, can be a useful tool for comparing different processing conditions.

Considering the complexity of the industrial process, the objective of this chapter is to discuss the influence of the main factors that affect the freezing time of food samples measuring the convective heat transfer coefficient. Results from comparative studies for different air flow and samples arrangement inside the system will be discussed, showing how slight changes to product arrangement and air flow orientation can affect the heat transfer.

2. Literature review

2.1 Cooling and freezing processes for food products

Regarding the rapid increase in frozen foods production and consumption, there is growing interest in determining food thermal properties during freezing process for the development of new systems and improvement of processes equipments (Scott et al. 1992).

Temperature reduction processes aims at reducing microbial growth and hence extending the shelf life of perishable food. Freezing and cooling processes are driven by the heat exchange between the product to be cooled and the cooling medium (Wolti-Chanes et al., 2005). Some researchers have presented models that can be used to assess the cost of food freezing by different methods (Becker & Fricke, 1999; Chourot et al., 2003).

The processes of cooling and freezing of food are complex. Freezing food basically depends on the amount of water that is present in the food and will freeze during the process. Prior to freezing, sensible heat must be removed from food until reaching the freezing temperature. Supercooling occurs when temperature reaches values below the freezing point and crystal nucleation starts. In pure water, heat is released during nucleation, causing a rise in temperature to the initial freezing point, and the temperature remains constant until all the water is converted to ice. However, it decreases slightly in foods, due to the increasing concentration of solutes in the unfrozen water portion.

After that, during the phase change of water into ice, there is the removal of latent heat from the frozen product. It starts when most freezable water has been converted to ice, and ends when the temperature is reduced to storage temperature (Zaritzky, 2000).

Among most popular frozen food products are fruits and fruit pulps, normally used as raw material for processing industries as ice cream, yogurt, jams and others. Such products can be frozen in batch or continuous processing (Salvadori & Mascheroni, 1996). In order to make cooling processes financially affordable, studies based on information from manufacturers and users are necessary to design refrigeration equipment in accordance with the demanded application.

2.2 Heat transfer for cooling process

Cooling time is directly influenced by the ratio between the resistances to heat transfer inside and outside the sample. This ratio is called the Biot number, defined by Equation 1:

$$Bi = \frac{hL_c}{k_b} \quad (2)$$

where h is the convective heat transfer coefficient ($W/m^2 \text{ } ^\circ C$), L_c is the characteristic length of the body (m), usually defined as the volume of the body divided by the surface area in contact with the cooling medium, and k_b is the thermal conductivity of the body ($W/m \text{ } ^\circ C$). Values of Biot number close to zero ($Bi \rightarrow 0$), imply that the heat conduction inside the body is much faster than the heat convection away from its surface, and temperature gradients are negligible within the body (Heldman, 1992). This condition allows for the applicability of certain methods of solving transient heat transfer problems. Within this condition, it can be assumed a lumped-capacitance model of transient heat transfer, leading to Newtonian cooling behaviour since the amount of thermal energy in the body is directly proportional to its temperature, which can be assumed uniform throughout the body. This leads to a simple first-order differential equation which describes heat transfer in these systems.

When the Biot number is very large ($Bi \rightarrow \infty$), the internal resistance to heat transfer is much larger and it only can be assumed that the surface temperature is equal to the cooling medium, but not the interior of the body. For this situation, solutions of the equation of Fourier heat transfer are useful. When the Biot number is within the range of $0.1 < Bi < 40$, both internal and external resistance should be considered.

The factors that are inherent to the products such as thermal conductivity and diffusivity cannot be changed. Hence, the reduction of cooling and freezing time must be achieved by changing system variables such as temperature and velocity of the cooling air, and product arrangement

2.3 Parameters affecting cooling time

The cooling or freezing rate is among the most important parameters in designing freezing systems, as it can directly affect the quality of products. For industrial applications, it is the most essential parameter in the process when comparing different types of systems and equipment. Besides the characteristics of the products, the temperature removal time varies according to some parameters involved in the heat transfer process, such as size and areas of openings of the packaging and characteristics of the cooling medium. The cost of the cooling process is related to cooling rate, which is directly affected by the opening area of packaging for air circulation, bed depth, temperature and speed of the cooling air (Baird et al., 1988).

The freezing systems can be described according to different methods of heat removal. Among the most used methods are forced air freezers. This technique has several

advantages, from ease of installation and operation and efficiency of the process to the variety of products that can be cooled in this type of equipments.

The selection of the best cooling method varies according to the desired application and depends on several factors, including the cooling rate required, subsequent storage conditions and costs of equipment and operation. Systems properly designed may increase efficiency and reduce the cost of operation (Talbot & Chau, 1998; ASHRAE, 2002).

Talbot & Fletcher (1996) compared the efficiency between an air blast system and a storage chamber. During the cooling process of grapes, there was a reduction of 6.7 °C in one hour and 14.6 °C after 2.5 hours, compared to a decrease of only 2 °C in one hour and 3.5 °C in 2.5 hours in the storage chamber. Experiments carried out using a prototype portable forced-air device offered promising results (Barbin et al., 2009). The system was designed to be used inside cooling and freezing chambers, aiming to improve heat flow rates.

2.4 Air flow

Cooling time by forced air systems is determined by airflow and product thermal load, which affects the amount of energy to move the air around the product and inside the system. The most common industrial applications use direct cold air insufflations inside the system. The airflow varies according to the speed and amount of air flowing through products and its variation results in longer or shorter freezing time. A correct orientation of the air flow inside the equipment and around the product can significantly reduce processing times (Cortbaoui et al., 2006). Surface area of contact between products and cooling air and products arrangement are other parameters that affect forced air cooling (Baird et al., 1988; Fraser, 1998; Laguerre et al., 2006).

In industrial plants, air flow is highly turbulent due to the fans movement and to wakes originated from upstream obstacles. Resende & Silveira Jr. (2002b) showed that the air velocity in forced air tunnels are strongly influenced by any changes in the amount of product inside the system, causing the air to flow through preferential paths, leading to increased freezing time and poor heat transfer coefficients. Results show that variations in heat transfer coefficients may occur according to the product positioning inside the equipment. Vigneault et al. (2005) studied how gravity influences air circulation in horizontal air flow, showing that low levels of air flows may be more affected, causing temperature variation of the cooling air and reducing the flow to the upper chamber. This could be an important parameter to consider when designing cooling and freezing systems.

Exhausting air is more appropriate to avoid air to flow through preferential corridors, leading to more uniform heat exchange when compared to insufflations processes (Fraser, 1998).

Conventional cooling methods by forced air are an efficient alternative to removing the heat load of fruits and vegetables during post-harvest cooling. Air exhaustion is widely used for this purpose, as it improves the air distribution in the products surrounding. This system is usually used inside cooling chambers. With the fan in operation, it creates a low pressure region surrounding the products. The cooling air flows through this region between the small opening areas, reducing the product temperature (Talbot & Fletcher, 1996; Talbot & Chau, 1998; Fraser, 1998; Abrahao, 2008). The possibility of adapting a cold room for use as a system for forced air represents an economical advantage of this process (Talbot & Fletcher, 1996).

Baird et al. (1988) showed that the velocity of cooling air influences directly the operational cost of cooling systems, as it can change with the increase of air velocity in the system. The

lowest costs were obtained with air velocities between 0.1 and 0.3 m/s. To study the cooling of plastic balls filled with a solution of carrageen, Allais et al. (2006) showed that increasing the speed of air flow, ranging from 0.25 m/s to 6 m/s, reduced the half-time cooling of samples from 800 s to 500 s. But this variation is exponential, and the reduction tends to be smaller from speed of 2 m/s. Results obtained by Vigneault et al. (2004a, b) for cooling process using forced air show that air flows above 2 l/s.kg and air velocities of insufflations higher than 0.5 m/s cause no influence on half-cooling time of the samples.

2.5 Instruments and methods for measurement of air flow velocity

There are several methods for measuring air flow velocity described in the literature, with different principles, and the accuracy of the sensors used in each of these techniques varies significantly, making them suitable for particular applications. Hot-wire anemometer is one of the most used instruments because of its wide applicability. Due to the small size and short response time, these instruments are suitable for detailed study of fluid flow, and are commonly used to measure the air flow in ventilation systems and air conditioning. The hot-wire anemometer measures the instantaneous velocity of fluids.

The core of the anemometer is an exposed very fine hot wire heated by a constant current up to some temperature above ambient. Air flowing past the wire has a cooling effect on the wire. By measuring the change in wire temperature under constant current, a relationship can be obtained between the resistance of the wire and the fluid flow velocity, as the electrical resistance of most metals is dependent upon the temperature of the metal. This kind of instrument has a fine spatial resolution compared to other measurement methods, and as such is employed for the detailed study of turbulent flows, or when rapid velocity fluctuations are of interest.

Resende & Silveira Jr. (2002b) and Nunes et al. (2003) suggest that several measurements in the cross-section of the air flow could lead to a more accurate determination of the air velocity. The average velocity is therefore used for determination of the air flow in the selected position, according to (3):

$$\dot{V} = \int_S \bar{v} dS \quad (3)$$

In this equation, \dot{V} is the flow (m^3/s), S is the total area (m^2) and v is the velocity vector (m/s).

2.6 Packaging and storage

Packaging affects the heat transfer coefficients of food items in several ways. It is a barrier to the transfer of energy from the food by acting as insulation to the food item, thus lowering the heat transfer coefficient. Packaging may also create air-filled voids and bubbles around the food item which further insulates the food and lowers the heat transfer coefficient (Becker and Fricke, 2004). Results presented by Santos et al. (2008) showed that freezing process of meat in cardboard boxes is underrated and the processing time sometimes is not enough for all the samples to reach the desired temperature. Replacing cardboard boxes by metal perforated boxes produced a reduction of up to 45% at the freezing time for this product.

Becker and Fricke (2004) developed an iterative algorithm to estimate the surface heat transfer coefficients of irregularly shaped food items based upon their cooling curves, considering the density of the food item and the packaging. This algorithm extends to

irregularly shaped food items existing techniques for the calculation of the surface heat transfer coefficient previously applicable to only regularly shaped food items, taking into account the concept of equivalent heat transfer dimensionality. In this method, the density used to calculate the heat transfer coefficient is affected by the packaging, as it is calculated from the mass of the food item plus the packaging and the outside dimensions of the package around the food item, generating results for the heat transfer coefficient for the food within its packaging.

An important parameter for improved performance of an air blast cooling system is the apertures and gaps that the packages and pallets must have to allow the circulation of the cold air through the packed product in order to achieve rapid and uniform heat transfer between the cooling air and the product (Vigneault et al. 2004a; Zou et al., 2006a, b).

Results obtained by Talbot & Fletcher (1996) and Abrahao (2008) showed the importance of proper cooling system design, proving that the larger the opening area in the packaging, the lower the requirement on refrigeration and air circulation systems to obtain a more uniform cooling rate. Meana et al. (2005) showed that the empty regions between the plastic containers that are used in the cooling of strawberries by forced air influence significantly the cooling time of the products. According to Baird et al. (1988), opening areas smaller than 10% of the total area of the box can significantly increase the cost of cooling processes. Castro et al. (2003) suggest that an opening area of 14% is appropriate for a rapid and uniform cooling process.

Large opening areas can lead to poorly designed boxes that are not suitable for industrial processing. The main goal is to get an optimal opening area of the boxes to enable a low freezing time without, however, affecting the mechanical structure of boxes.

2.7 Convective heat transfer coefficients (h_c)

Convective heat transfer is related to the amount of energy transferred from the product surface when it is in contact with the refrigerating fluid (Welty et al, 2000). Dincer (1995a) determined the experimental heat transfer coefficient with data obtained during forced air cooling of figs in air blast systems, with results varying from 21.1 to 32.1 $\text{Wm}^{-2}\text{C}^{-1}$ for air velocities of 1.1 to 2.5 ms^{-1} .

Experiments carried out in a forced air room with air velocities in the range of 1 to 2 ms^{-1} resulted in h_c values varying from 28 $\text{Wm}^{-2}\text{C}^{-1}$ up to 52 $\text{Wm}^{-2}\text{C}^{-1}$ for cylindrical products (cucumber) during cooling (Dincer and Genceli, 1994). Mohsenin (1980) obtained h_c values in the range of 20 to 35 $\text{Wm}^{-2}\text{C}^{-1}$ for forced air systems with air velocity from 1.5 to 5.0 ms^{-1} . Dussán Sarria et al. (2006) studied the influence of the air velocity in a cooling tunnel. According to the authors, air velocities greater than 2.0 ms^{-1} did not affect the convective coefficients (h_c), as results obtained were not greater than 23.8 $\text{Wm}^{-2}\text{C}^{-1}$.

Considering the complexity of freezing processes and the recent results presented, many parameters influence the experimental results for heat transfer coefficients, thus varying according to the flow characteristics of the cooling medium and the products involved.

Accurate descriptions of the boundary conditions are rather difficult for industrial air blast systems, and software solutions such as CFD will not be effective in solving the momentum and heat transport equation without precise information (Mohamed, 2008). Regarding the wide range of convective heat transfer coefficients (h_c) reported, it is important to calculate this coefficient in order to understand different operating conditions of distinct cooling systems and compare to any new systems developed. Several methods for convective heat

transfer measurements are reported. The most common are those involving temperature measurements in permanent and transient state (Cleland, 1990).

2.7.1 Temperature measurements in steady state

In this method, a constant thermal load is created in the system such as an electrical heating probe, for example. The coefficient of heat transfer can be calculated using the values of the surface area of the heating probe, the amount of energy added and the temperatures of the cooling medium and the probe. However, the temperature and velocity of the cooling medium should be kept constant, which is not an easy task in experimental conditions, limiting the use of this method.

2.7.2 Temperature measurements in transient state

Temperature measurement in transient state consists of a metallic test body with a known high thermal conductivity being used to minimize the temperature gradient during the heat exchange between the cooling medium and the product ($Bi < 0.1$), allowing the test body to have an almost uniform temperature during the cooling process. When the internal resistance of the test body to heat transfer is neglected, an energy balance conducts to the convective heat transfer coefficient. By Newton's cooling law, the rate of heat transfer in a given volume of control is given by equation 1.

The variation of energy in a metal body with constant properties is given by the equation:

$$\frac{dQ}{dt} = \rho_m V c_{pm} \frac{dT}{dt} \quad (4)$$

where ρ_m is the density, V is the volume and c_{pm} is the specific heat of the metallic body, respectively. Combining equations 1 and 4, integrating and adopting the initial boundary condition $T(t=0) = T_i$, leads to the solution for the temperature variation as a function of time:

$$\frac{T_b - T_\infty}{T_i - T_\infty} = e^{\frac{-h_c A t}{\rho_m c_{pm} V}} \quad (5)$$

Equation 5 proves that the cooling process has an exponential behaviour, as verified by several authors for horticultural products (Mohsenin, 1980; Dincer, 1995a).

In practice, this method consists of using a test body made of some material with high thermal conductivity, so that tests are carried out without phase change and assuming the constant thermal properties within temperature variation range. Le Blanc et al. (1990a, b), Resende et al. (2002), Mohamed (2008) and Barbin et al. (2010) reported experiments using the described method for obtaining convective coefficients from the cooling curves obtained for a metallic test body, indicating the capability of the present method in handling complex boundary situation such as encountered in industrial systems. Results for convective heat transfer coefficients were reported by Barbin et al. (2010), comparing two air flow direction in the same equipment, concluding that this is a useful method for studying temperature reduction processes.

According to Resende et al. (2002), some points arise when using this method. If the test body consists of a metal block, there may be heat transfer through the edges of the material, affecting the values of h_c calculated. Furthermore, condensation can occur in experiments with cooling air, causing changes to the measurements. Thus, the positioning of the test body must

be carefully chosen in order to prevent any condensation of water during the tests, and the edges of the body or other parts that may interfere with the temperature measurements during the process must be perfectly insulated to avoid heat transfer through these regions.

3. Experiments

3.1 Samples for simulation of thermal load

Food model system with 15% (weight / weight of solution) of sucrose and 0.5% (weight / weight of solution) of carboxyl-methyl cellulose (Carbocel AM, Arinos, SP, Brazil) was packed in polyethylene bags (0.1 kg) with similar dimensions (0.095 m x 0.07 m x 0.015 m) to pulp fruit products in the market. The samples were stored in 35 plastic boxes (Figure 1a), with external dimensions of 0.6 m x 0.4 m x 0.12 m, which were stacked on a commercial pallet (1.00 m x 1.20 m, Figure 1b) and kept inside the freezing room. The boxes had an opening area of 21% of the total area, accounting for more than the minimum values recommended for proper air flow (Castro et al., 2003).

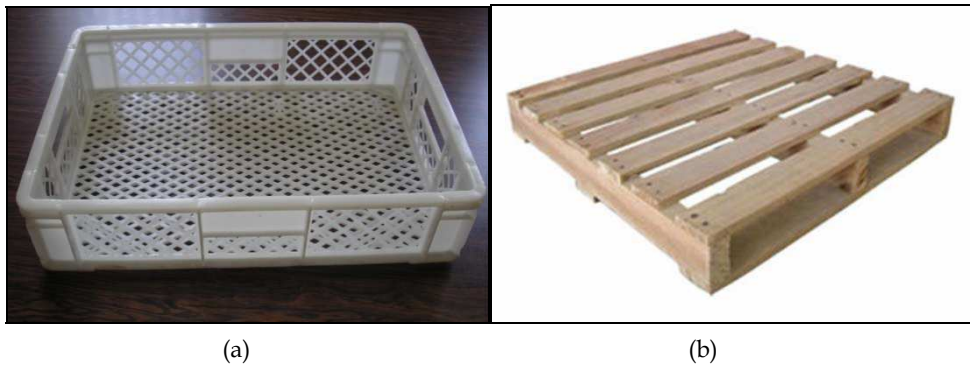


Fig. 1. (a) Plastic box for freezing products; (b) Commercial pallet

3.2 Product arrangement

Two arrangements of samples were tested to determine the influence of opening areas to the refrigeration process. Using the industrial arrangement of samples, ninety six packages of sample were allocated in each box, in three layers (top, middle and bottom), with thirty two packs in each layer, corresponding to about 9.6 kg of product, similar to the amount used in the industrial process. This assembly is shown in Figure 2 (Arrangement 1). The boxes were piled in six layers with five boxes per layer, totalling thirty boxes, simulating a commercial assembly of a pallet used regularly in the process (Figure 3, Arrangement 1).

A second distribution of packages inside the boxes was tested, with larger distance between the packages inside the boxes in order to improve the circulation of air around the samples. In this assembly, eighty four packs of sample were allocated in each box, within five layers three layers with twenty units, and two layers of twelve units, distanced from each other for air circulation, totaling 8.4 kg of product per box.

Figure 2 (Arrangement 2) shows the new distribution of packaging inside the boxes. The second arrangement had a smaller amount of samples in each box. Hence, it was added another layer of boxes in the system in order to have the same amount of product and the same thermal load for all the tests (Figure 3, Arrangement 2).

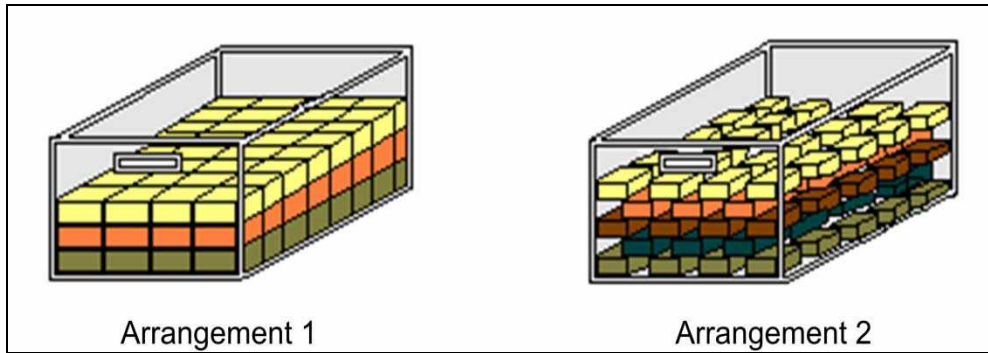


Fig. 2. Schematic diagram for sample distribution used for packaging boxes.

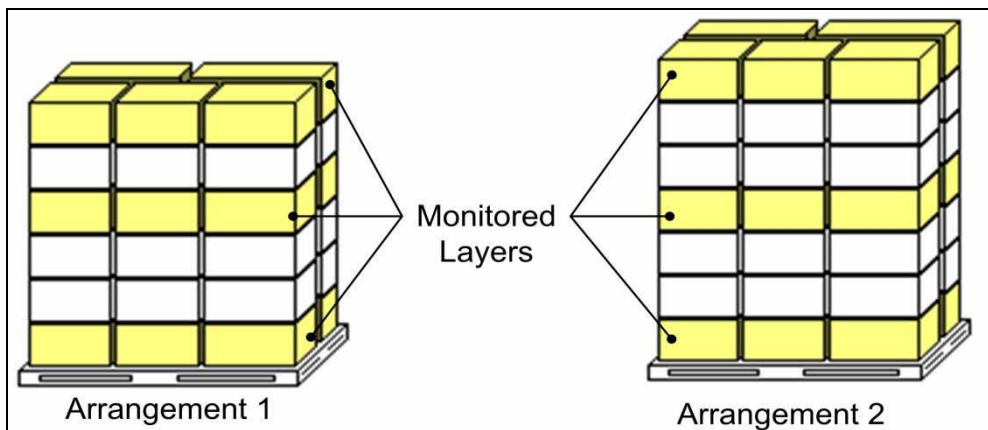


Fig. 3. Pallet with boxes and layers that were monitored

3.3 Temperature measurement

Insufflations and exhaustion air tests were run in triplicate. The velocity of the cooling air was measured for comparison with the convective coefficients. Three layers of boxes had its temperature monitored until the centre of the samples reached -18°C . Each of the monitored layers had four thermocouples in the corners of the layers and one in the middle, as shown in Figure 4. The thermocouples were inserted inside the samples in the plastic bags to measure the samples temperatures variation during the freezing process.

The monitoring system used for temperature acquisition is composed by an automatic channel selector system (Scanner 706, Keithley Instruments Inc.). Samples temperatures were monitored using T-type thermocouples (copper-constantan). The thermocouples were calibrated using a controlled temperature bath with a propylene glycol solution and a standard thermometer as reference. Five different temperature values were chosen (-19°C , -10°C , 0°C , 10°C and 20°C). The average temperatures measured in the water bath (10 measurements for each one of the five different chosen values) were plotted against the corresponding thermocouple (mV) values (ASTM, 1989). The difference for the correlation coefficient of the curve-fitted line (R^2) were not lower than 0.99.

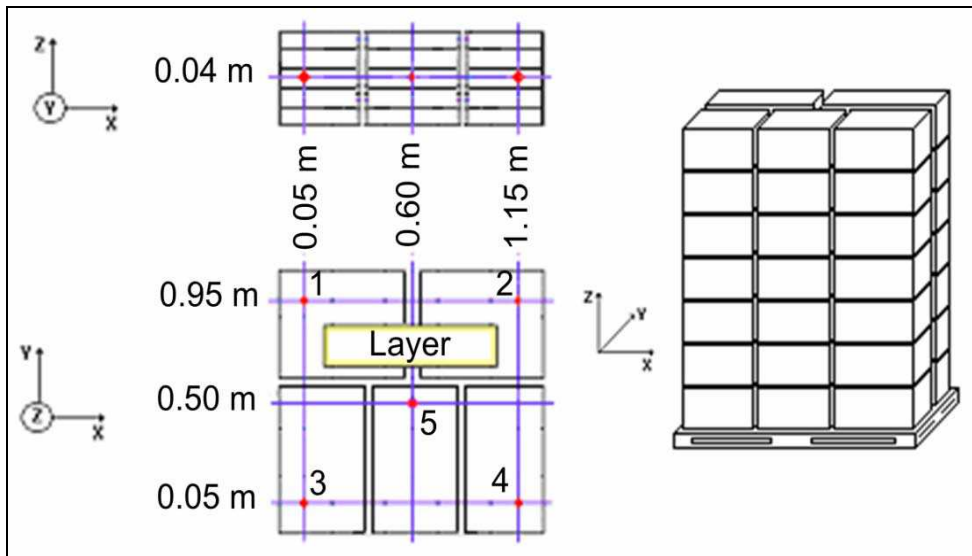


Fig. 4. Samples monitored in the layers of boxes

3.4 Portable forced air system

The forced air system was designed as described in Barbin et al. (2009), with a plastic sheet cover connected to a flexible duct and a fan that insufflates or exhausts the air inside the system. The plastic covers the boxes that contain the product, stacked on a commercial pallet. The portable tunnel fan used has axial airscrews with a tri-phase induction engine (Weg, Brazil, model 71586, 0.5 hp). The device was placed inside a freezing storage room (Recrusul, Brazil), with internal dimensions of 3 m x 3 m x 2.3 m (20.7m³) and walls made of 0.01 m aluminium panels filled with expanded polystyrene as insulation.

The cooling process consists in circulating the internal air of the storage room through the boxes open spaces and around the product samples. In the exhaustion process, the system is connected to the fan suction, and the air flows from the lower part of the system to inside the boxes and through the fan back to the room. In the insufflations processes, the airflow is changed, blowing the cooling air from the room directly to the product. The forced air circulation is vertically oriented in both the exhaustion and the insufflations process. During exhaustion, it goes from the bottom to the top of the pallet; while in the blowing process, it goes from top to bottom (Figure 5).

3.5 Air flow measurement

A hot-wire anemometer (Tri-Sense, model EW-37000-00, Cole-Parmer Instrument Company, IL, USA) was used for measurements of air velocity. The sensor was inserted through openings in the air diffuser for measuring air velocity in different positions of the area normal to the air flow. The measurement points were aligned and positioned at regular distances. The sensor was introduced for measuring the air speed with different depth of insertion, providing fixed points in the surface area perpendicular to the airflow.

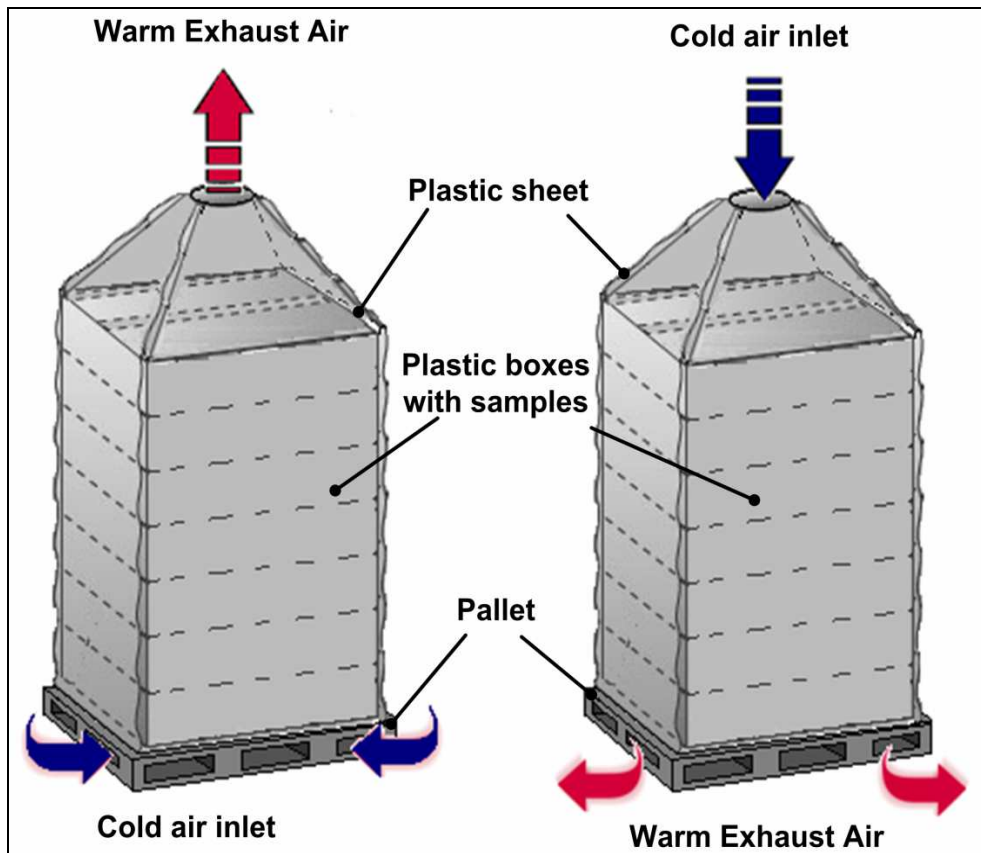


Fig. 5. Portable tunnel with boxes stacked on a commercial transport pallet covered with plastic, and air flow orientation during the exhaustion and insufflations processes.

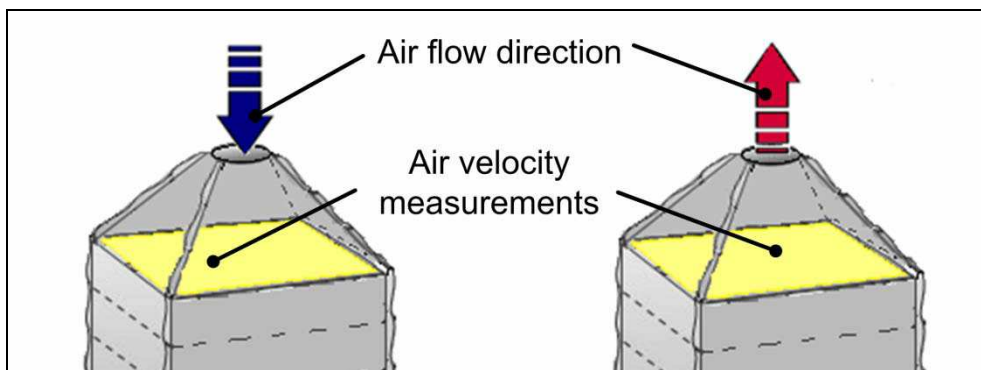


Fig. 6. Surface area for air velocity measurements during insufflations and exhaustion processes.

Velocities were measured for comparison between the exhaustion and insufflations with the fan operating at steady state and no obstructions to the air flow. In this study, the area that the air flows through is a cross-section of the pallet represented by the five boxes. The greater the number of velocity measurements, the more accurate the result of air flow. Thus the new equation for calculating the flow in the tunnel is:

$$\dot{V} = \int_{x_0}^{x_1} \int_{y_0}^{y_1} \bar{v}(x, y) dy dx \quad (6)$$

where x and y represent the coordinates of the cross-section perpendicular to the air flowing stream, comprising the dimensions of the surface formed by the boxes from the pallet. Common approach is to measure the air velocities in several points of the flow and obtain one average result, for a more consistent representation of the profile of the flow and avoid the high variability of measurements.

3.6 Convective heat transfer measurement

The experiments for determining the convective heat transfer coefficients during freezing processes were carried out according to procedures described by Le Blanc et al. (1990a, b) and Resende et al. (2002), using a specimen of high thermal conductivity metal. The method consisted of measuring the temperature variation of a test body with high thermal conductivity during cooling. The high conductivity is necessary to minimize the temperature gradient formed during the heat transfer process between the sample and the cooling medium. The test body shown in Figure 4a is an aluminium brick with known dimensions (0.10 m x 0.07 m x 0.025 m), with perforations for insertion of thermocouples for temperature measurement. Empty spaces around the thermocouples were filled with thermal paste to prevent formation of air pockets within the holes that could affect the measurements. Aluminium thermo physical properties (as a metallic test body) used for the determination of the convective heat transfer coefficients at 20°C are: density ($\rho_{Al}=2701.1 \text{ kgm}^{-3}$), specific heat ($C_{pAl}=938.3 \text{ Jkg}^{-1} \text{ }^\circ\text{C}^{-1}$), thermal conductivity ($k_{Al}=229 \text{ Wm}^{-1} \text{ }^\circ\text{C}^{-1}$) (Welty et al., 2000).

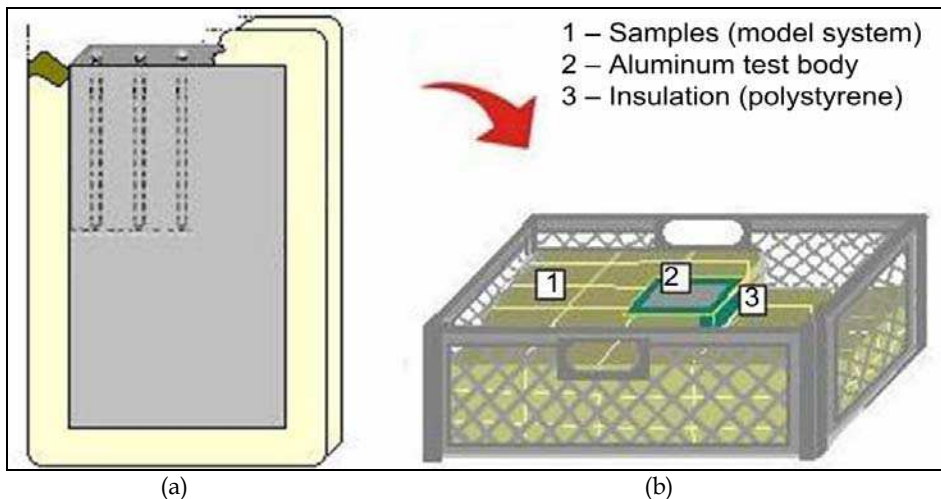


Fig. 7. (a) Aluminium test body insulation and (b) positioning inside the box.

Regarding heat flow analysis, polystyrene was used as insulation around the test body to keep only one surface exposed in contact with the cooling air. The test body had all of its edges insulated, except the upper face which was kept exposed to the cold air flow (Figure 4). This procedure was adopted to avoid edge effects and generate a one-dimensional heat flow. The test bodies were assembled inside the boxes and over the packages of samples, as shown in Figure 4b.

The fast cooling curve can be described by Equation 7, which is a simplification of Equation 6:

$$\frac{(T - T_{\infty})}{(T_i - T_{\infty})} = e^{S_2 t} \quad (7)$$

where T is the average value obtained from three thermocouple temperature measurements inside the test body during the cooling process, T_i is the initial temperature of the aluminium plate and T_{∞} is the cooling medium temperature. The S_2 parameter represents the cooling coefficient, defined as the test body temperature change per time for each temperature degree difference between the product and the cooling medium, and is expressed in s^{-1} .

The values of S_2 were used for the calculation of the heat transfer coefficients for the period of sensible heat removal of the samples during the cooling process, as Equation 8:

$$h_c = \frac{-\rho_m V c_{pm} S_2}{A} \quad (8)$$

Values for the average dimensionless temperature $[(T - T_1) / (T_i - T_1)]$ logarithm were calculated for every test body monitored during chilling period, with values obtained plotted versus cooling time. The angular coefficients (S_2) were calculated from these graphs. Convective heat transfer coefficients were obtained according to Equation 8, using temperature measurements of the 5 identified (T_1 to T_5) aluminium test bodies, distributed in layers 1, 3, 4, 5 and 6, respectively, including both the extreme layers (1 and 7) and the central layers (3, 4 and 5) (Barbin et al., 2010). With the new arrangement of samples inside the boxes, one layer of boxes was added to the pallet, namely layer number 7. In this new arrangement, the layer number 6 was not monitored. All the aluminium test bodies were positioned over the samples in the centre of the boxes along with the thermocouples identified with number 5 as last algorithm (15, 35, 45, 55 and 75, Figure 3) and in contact with the cooling air (Figure 4b). Only the second layer in the first arrangement, and both second and sixth layers in the second arrangement, did not have a test body. The main objective was to analyze the air circulation in the central layers in comparison to the top and bottom layers, and to determine the local convection coefficient of heat transfer. After obtaining S_2 values (according to equation 7), the effective heat transfer coefficients were calculated for the samples in the sensible heat loss phase, as shown in Equation 8.

4. Experimental results

4.1 Determination of air flow velocity

The air flow in the region surrounding the product was investigated and results were previously reported by Barbin et al. (2009). These results are reproduced here for comparison with the heat transfer coefficients. Flow direction and turbulence intensity are generally not accurately known in practical situations. Furthermore, impending variation of heat transfer coefficient value for a product of certain size is primarily dependent on airflow

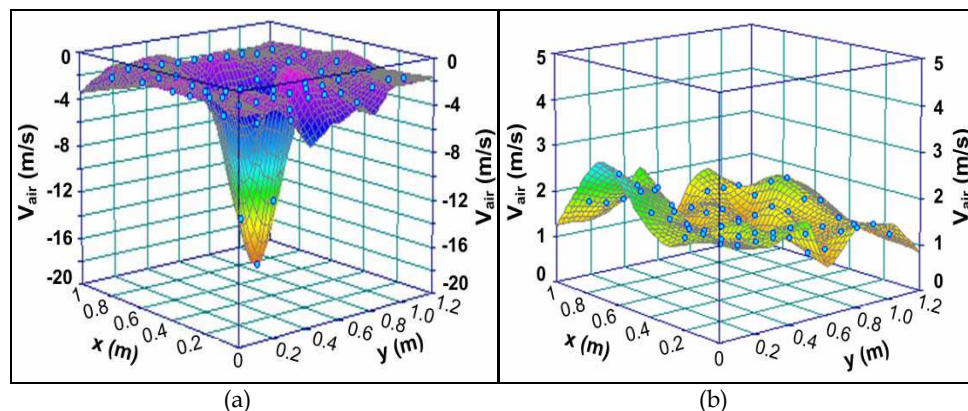


Fig. 8. Results for air flow velocity obtained in the product surface for: (a) insufflation process, and (b) exhaustion process.

properties such as velocity and turbulence, and less affected by product shape and direction into the flow. Due to high turbulence of air flow in industrial applications, wind tunnel experiments are not useful to determine heat transfer values (Kondjoyan, 2006); hence, the importance in studying the airflow behaviour for the experiments.

The graphs shown in Figure 8 illustrate the results of measurements of air velocity for each process (Barbin et al., 2009). Figure 8a shows the experimental values for the air velocities measured during the process of insufflations, while Figure 8b shows the results for the exhaustion process. The dimensions x (m) and y (m) represent the surface area perpendicular to the air flow, where velocities were measured. The vertical axis shows the velocity results (m/s), containing a response surface for visualization of the airflow.

High values for the air velocity were observed in the insufflation process, reaching above 15 m/s in the central region of measurement, while in the edge and corner positions the velocity results are lower, ranging between 1 and 2 m/s. This difference or lack of uniformity in this distribution of velocities is not observed in the exhaustion process, as it can be seen in Figure 5. This phenomenon may be consequence of the fact that the exhaustion is performed more uniformly, causing the distribution and movement of air to be more even inside the tunnel. Another reason could be that the air had not enough space to expand properly along the short region between the duct exit and the uppermost layer of samples. This makes it a difficult task to overcome because it needs more space for the system according to the appropriate technique for the design of air blast systems (ASHRAE, 1977). However, given the dimensions of the chamber and set of boxes with the product, the assembly was done with the maximum extension possible to fit inside the cold room.

Another factor that may cause interference measures in this place of assembly is that the anemometer does not indicate the direction of air flow during the measurements. Table 1 shows the values obtained and the average velocity of air flow calculated for each flow.

| Test | Average air velocity (m/s) | Air flow (m ³ /s.kg) |
|--------------|----------------------------|---------------------------------|
| Insufflation | -3.05 ± 0.2 | 3.7×10^{-3} |
| Exhaustion | 1.88 ± 0.2 | 2.3×10^{-3} |

Table 1. Average values obtained for air velocity and air flow during insufflation and exhaustion processes.

The negative sign for the value of the inflation rate represents the direction of flow opposite to the exhaustion, as the z axis of the graph shown in Figure 8, and it did not affect air flow calculations, since the absolute values of air velocity were used for both cases.

4.2 Freezing time

A freezing process without the portable tunnel was carried out as a reference test to be compared to the experiments using the tunnel device. This reference freezing process consisted in leaving the boxes inside the cold room until all of the samples reached the final freezing temperature.

The average freezing time for the pallet with the industrial sample arrangement was 47 hours, for the reference test, without the use of the portable tunnel. It was also observed the lack of uniformity of process during the freezing process, where the samples located in the upper and lower layers reached -18°C in about 42 hours, while it took more than 52 hours for all the samples in the central layer to reach the final freezing temperature.

The top layer of boxes (layer 6) was quickly frozen in the insufflation test, as all the samples reached -18°C after 35 hours. The samples in the bottom and middle layers were frozen after about 45 hours, showing that the cooling air was not flowing properly in the bottom part of the pallet.

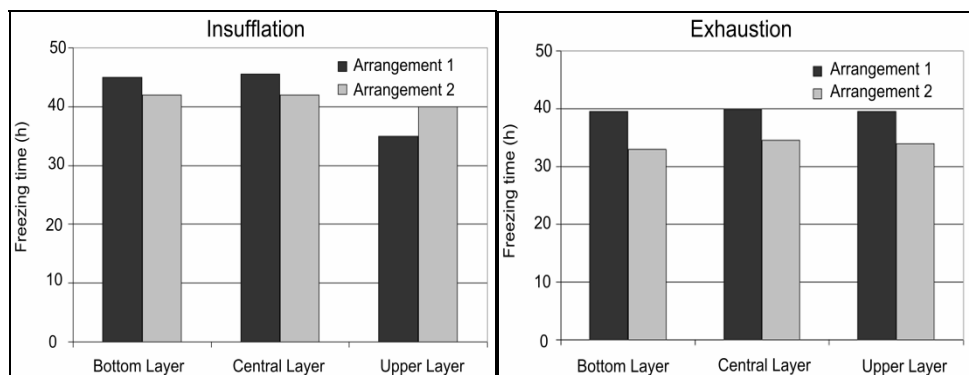


Fig. 9. Comparison of freezing time between two arrangements of samples for insufflation and exhaustion processes.

During the exhaustion test it was observed a difference for the freezing time between the central and peripheral layers of about 1 to 2 hours; however, all samples reached the temperature of -18°C after 40 hours of processing.

Regarding the new arrangement of samples, the freezing time for the exhaustion process was 38 hours. It took 43 hours for the samples to be frozen in the insufflation process. The comparison between the freezing time for industrial and new arrangement can be seen in Figure 9.

The heat transfer analysis comparing the air distribution performed shows that, although showing values around 2 m/s for air velocity results, the exhaustion process reduced up to 14% of the freezing time compared to the blowing system, where air velocity were up to 19m/s.

The top layer of the assembly showed an increase in freezing time for the new configuration of the samples for the testing of inflation (Figure 9). This may have occurred because the

industrial configuration might block air circulation for the other layers, reducing the efficiency of heat exchange between air and the other layers of product assembly.

4.3 Convective heat transfer coefficients

The lumped-capacitance method was applied to obtain the heat transfer coefficients for comparison between two different arrangements of products inside the cold chamber. Results of Table 2 show that there is an increase in the convective coefficient for the all the layers of boxes in the pallet, except the test body number 4, located on the second layer from the top.

In the insufflation process for the industrial arrangement, the results were in the range from 3 to 6 W/m²°C for the three lower layers increasing to 15 W/m²°C for the test body 4 and reaching 30 W/m²°C the top layer. The values for the new proposed arrangement reached 10 W/m²°C for the first two bodies and were greater than 30 W/m²°C the top layer.

In the exhaustion process, results were nearly twice as high as the industrial process for the first two test bodies. The top layer had also higher values compared to the industrial arrange. For the central layer, results were similar, around 10 W/m²°C. Only the test body in the second layer from the top showed higher results for the industrial arrange. Thus, results show that enlarging opening space for air flow increases the convective heat transfer, even if there is more product or layers in the pallet.

| Test body | Convective coefficient (W/m ² °C) | | | |
|-----------|--|---------------|---------------|---------------|
| | Exhaustion | | Insufflation | |
| | Arrangement 1 | Arrangement 2 | Arrangement 1 | Arrangement 2 |
| T1 lower | 5.79 | 10.74 | 3.58 | 9.90 |
| T2 | 5.25 | 12.88 | 6.13 | 9.92 |
| T3 middle | 10.41 | 10.76 | 5.77 | 6.38 |
| T4 | 14.97 | 11.06 | 15.41 | 13.04 |
| T5 upper | 5.92 | 7.73 | 29.46 | 31.72 |

Table 2. Results for convective coefficient measured between two arrangements of samples in the boxes and two different air flow orientation

The variability of the heat transfer coefficients with position inside a refrigerated room is a crucial aspect to be considered as this will directly affect cooling time. Considering that energy costs are the major expense associated with most refrigerated warehouses, this becomes critical to the managerial decisions, spanning from the initial investment to the long term running costs.

5. Conclusion

This study was initiated to resolve deficiencies in temperature reduction processes by investigating the convective heat transfer coefficient data for food cooling and freezing processes, endeavouring to assist the food refrigeration industry to improve heat transfer process efficiency with uncomplicated solutions. A literature examination was conducted to collect support information and compare with the results obtained.

Results have shown that increasing the cooling air velocity inside the system, or using more powerful equipments to introduce cooling air with lower temperatures is not the best way

to improve cooling rates. Improved results can be achieved by applying simple changes to the process, such as rearranging the product or samples to be cooled or frozen and changing the direction of cooling air to flow around the product can save some time in freezing processes. On the other hand, the determination of convective heat transfer coefficient can help to determine how the cooling air is working on some specific region of the system or samples, leading to more precise ways to improve the cooling process.

The experimental data resulting from this project will be used by designers of cooling and freezing systems for foods. This information will make possible a more accurate determination of convective heat transfer coefficients inside these equipments, leading to more efficient systems and reduction in cooling and freezing times. Such information is essential in the venture and operation of cooling and freezing facilities and will be of immediate usefulness to engineers involved in the design and manoeuvre of such systems. Taking the above into consideration and incorporating these factors to the equipment's design will have a significant impact on energy savings.

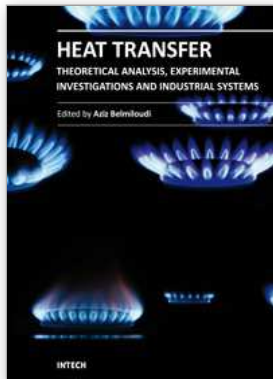
6. References

- Abrahamo, R. F.; Correia, E.; Teruel, B. J. (2008). Computational simulation for the development of packages for bananas [Simulação computacional aplicada ao desenvolvimento de embalagens para bananas]. *Revista Brasileira de Fruticultura*, 30 (1). p. 79-87.
- Allais, I.; Alvarez, G.; Flick, D. (2006). Modeling cooling kinetics of spheres during mist chilling. *Journal of Food Engineering*. V. 72, p. 197-209.
- ASHRAE Handbook. *Fundamentals*. (1977) Atlanta, Georgia: American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. 1977.
- ASHRAE. *Refrigeration*. (2002). Atlanta, Georgia: American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. 2002.
- ASTM. (1989). *Manual on the use of thermocouples in temperature measurement*. American society for testing and materials.
- Baird, C. D.; Gaffney, J. J.; Talbot, M. T. (1988). Design criteria for efficient and cost effective forced air cooling systems for fruits and vegetables. *ASHRAE Transactions*, Atlanta, v.94, p.1434-1453.
- Barbin, D. F.; Neves Filho, L. C.; Silveira Junior, V. (2009). Processo de congelamento em túnel portátil com convecção forçada por exaustão e insuflação para paletes. [Freezing process evaluation using a portable forced air system with air evacuation and air blowing in pallets]. *Ciencia e Tecnologia de Alimentos*, 29, (3), p.1-9.
- Barbin, D. F.; Neves Filho, L. C.; Silveira Junior, V. (2010). Convective heat transfer coefficients evaluation for a portable forced air tunnel. *Applied Thermal Engineering*, 30. p.229-233.
- Becker, B. R.; Fricke, B. A. (1999). Food thermophysical property models. *International Communications in Heat and Mass Transfer*, v.26, n.5, p.627-636.
- Becker, B. R.; Fricke, B. A. (2004). Heat transfer coefficients for forced-air cooling and freezing of selected foods. *International Journal of Refrigeration*, v. 27, (5). p. 540-551.
- Castro, L. R.; Vigneault, C.; Cortez, L. A. B. (2003). Container opening design for horticultural produce cooling efficiency. *International Journal of Food Agriculture and Environment*, v.2, n.1, p.135-140.

- Chourot, J. M.; Macchi, H.; Fournaison, L.; Guilpart, J. (2003). Technical and economical model for the freezing cost comparison of immersion, cryomechanical and air blast freezing processes. *Energy Conversion and Management*, n.44, p.559-571.
- Cleland, A. C. (1990). *Food Refrigeration Process. Analysis, Design and Simulation*. Elsevier Applied Science. London and New York. 1990. 284p.
- Cortbaoui, P.; Goyette, B.; Gariépy, Y.; Charles, M. T.; Raghavan, V. G. S.; Vigneault, C. (2006). Forced air cooling system for Zea mays. *Journal of Food Agriculture and Environment*. V. 4, p. 100-104.
- Dincer, I. (1995a). Thermal cooling data for figs exposed to air cooling. *International Communications Heat Mass Transfer*. v.22, n.4. p.559-566.
- Dincer, I. (1995b). Transient heat transfer analysis in air cooling of individual spherical products. *Journal of Food Engineering*, v. 26, p. 453-467.
- Dincer, I.; Genceli, F. (1994). Cooling process and heat transfer parameters of cylindrical products cooled both in water and air. *International Journal Heat Transfer*, v. 37, n. 4, p. 625-633.
- Dussán Sarria, S.; Honório, S.L.; Nogueira, D.H. (2006). Precooling parameters of 'Roxo de Valinhos' figs (*Ficus carica* L.) packed in a carton box. *Fruits*. v. 61, Issue 6, Nov. 2006, p 401-406.
- Fraser, H. (1998). Tunnel Forced-Air Coolers for Fresh Fruits & Vegetables. Ministry of Agriculture, Food and Rural Affairs, Government of Ontario, Canada, 1998. Available: <<http://www.omafra.gov.on.ca/english/engineer/facts/98-031.htm>>.
- Heldman, D.R. Food Freezing. In: Heldman, D. R.; Lund, D. B. *Handbook of Food Engineering*. New York: Dekker, 1992. 277-315.
- Kondjoyan, A. (2006). A review on surface heat and mass transfer coefficients during air chilling and storage of food products. *International Journal of Refrigeration*, v.29, p.863-875.
- Laguerre, O.; Ben Amara, S.; Flick, D. Heat transfer between wall and packed bed crossed by low velocity airflow. *Applied Thermal Engineering*, v. 26, p. 1951-1960, 2006.
- Le Blanc, D. I.; Kok, R.; Timbers, G. E. (1990a). Freezing of a parallelepiped food product. Part 1: Experimental determination. *International Journal of Refrigeration*, v.13, p.371-378.
- Le Blanc, D. I.; Kok, R.; Timbers, G. E. (1990b). Freezing of a parallelepiped food product. Part 2: Comparison of Experimental and Calculated Results. *International Journal of Refrigeration*, v.13, p.379-392.
- Meana, M. B.; Chau, K. V.; Emond, J. P.; Talbot, M. T. Forced-air cooling of strawberries in reusable plastic containers. *Proceedings of Florida State Horticultural Society*. v. 118, p. 379-382, 2005.
- Mohamed, I. O. (2008). An inverse lumped capacitance method for determination of heat transfer coefficients for industrial air blast chillers. *Food Research International*, 41. p.404-410.
- Mohsenin, N. N. (1980). *Thermal properties of foods and agricultural materials*. New York: Gordon and Breach. p. 198-224. 1980.
- Nunes, M.; Nader, G.; Jesus, F. B. G.; Cardoso, M.; Jabardo, P. J. S.; Pereira, M. T. (2003). Estudo sobre técnicas de medição de vazão em bocas de insuflamento e de exaustão de ar. Metrologia-2003 - Metrologia para a Vida Sociedade Brasileira de Metrologia (SBM) Setembro 01-05, 2003, Recife, Pernambuco - BRASIL

- Resende, J. V.; Silveira Jr, V. (2002a). Medidas da Condutividade Térmica Efetiva em Modelos de Polpas de Frutas no Estado Congelado. *Ciência e Tecnologia de Alimentos*, v.22, n.2, p.177-183.
- Resende, J. V.; Silveira Jr. V. (2002b). Escoamento de ar através de embalagens de polpa de frutas em caixas comerciais: Efeitos sobre os perfis de velocidade em túneis de congelamento. *Ciência e Tecnologia de Alimentos*, v.22, n.2, p.184-191.
- Resende, J. V.; Neves Filho, L. C.; Silveira Jr, V. (2002). Coeficientes de Transferência de Calor Efetivos no Congelamento com Ar Forçado de Modelos de Polpas de Frutas em Caixas Comerciais. *Brazilian Journal of Food Technology*, v.5, p.33-42.
- Salvadori, V. O.; Mascheroni, R. H. (1996). Freezing of strawberry pulp in large containers: experimental determination and prediction of freezing times. *International Journal of Refrigeration*, v.19, n.2, p.87-94.
- Santos, C. A.; Laurindo, J. B.; Silveira Junior, V.; Hense, H. (2008). Influence of secondary packing on the freezing time of chicken meat in air blast freezing tunnels [Influência da embalagem secundária sobre o tempo de congelamento de carne de frango em túneis de circulação de ar forçada]. *Ciencia e Tecnologia de Alimentos*, 28 (SUPPL.). p.252-258.
- Scott, E. P.; Beck, J. V.; Heldman, D. R. (1992). Estimation of Time Variable Heat Transfer Coefficients in Frozen Foods during Storage. *Journal of Food Engineering*, v.15, p.99-121, 1992.
- Talbot, M. T.; Chau, K. V. (1998). Precooling Strawberries. Agricultural and Biological Engineering Department, Florida Cooperative Extension Service, Institute of Food and Agricultural Sciences, University of Florida, pub.CIR942/AE136, 1998. Available in: <<http://edis.at.ufl.edu/AE136>>.
- Talbot, M. T.; Fletcher, J. H. A (1996). Portable Demonstration Forced-Air Cooler. Agricultural and Biological Engineering Department, Florida Cooperative Extension Service, Institute of Food and Agricultural Sciences, University of Florida, pub.CIR1166/AE096, 1996. Available in:<<http://edis.at.ufl.edu/AE096>>.
- Vigneault, C.; Goyette, B.; Markarian, N. R.; Hui, C. K. P.; Cote, S.; Charles, M. T.; Emond, J.-P. (2004a). Plastic container opening area for optimum hydrocooling. *Canadian Biosystems Engineering*, v.46, p.41-44.
- Vigneault, C.; De Castro, L. R.; Gautron, G. (2004b). Effect of open handles on packages during precooling process of horticultural produce. *ASAE Annual International Meeting*. p. 6901-6908.
- Vigneault, C.; De Castro, L. R.; Cortez, L. A. B. (2005). Effect of Gravity on Forced-air Precooling. *IASME Transactions* Vol. 2(3):459-463.
- Welty, J. R.; Wicks, C. E.; Wilson, R. E.; Rorrer, G. (2000). *Fundamentals of Momentum, Heat, and Mass Transfer*. 4th Edition, John Wiley & Sons: New York, 2000.
- Welty-Chanes, J.; Vergara-Balderas, F.; Bermudez-Aguirre, D. (2005). Transport phenomena in food engineering: Basic concepts and advances. *Journal of Food Engineering* V. 67, (1-2), March 2005, Pages 113-128.
- Zaritzky, N. E. (2000). Factors affecting the stability of frozen foods. In: KENNEDY, C. J. *Managing frozen foods*. Cambridge: Woodhead Publishing Limited, 2000.
- Zou, Q; Opara, L. U.; Mckibbin, R. (2006a). A CFD modeling system for airflow and heat transfer in ventilated packaging for fresh foods: I. Initial analysis and development of mathematical models. *Journal of Food Engineering*. v. 77, Issue 4, p. 1037-1047.

Zou, Q; Opara, L. U.; Mckibbin, R. (2006b). A CFD modeling system for airflow and heat transfer in ventilated packaging for fresh foods: II. Computational solution, software development, and model testing. *Journal of Food Engineering*. v. 77, Issue 4, p. 1048-1058.



Heat Transfer - Theoretical Analysis, Experimental Investigations and Industrial Systems

Edited by Prof. Aziz Belmiloudi

ISBN 978-953-307-226-5

Hard cover, 654 pages

Publisher InTech

Published online 28, January, 2011

Published in print edition January, 2011

Over the past few decades there has been a prolific increase in research and development in area of heat transfer, heat exchangers and their associated technologies. This book is a collection of current research in the above mentioned areas and discusses experimental, theoretical and calculation approaches and industrial utilizations with modern ideas and methods to study heat transfer for single and multiphase systems. The topics considered include various basic concepts of heat transfer, the fundamental modes of heat transfer (namely conduction, convection and radiation), thermophysical properties, condensation, boiling, freezing, innovative experiments, measurement analysis, theoretical models and simulations, with many real-world problems and important modern applications. The book is divided in four sections : "Heat Transfer in Micro Systems", "Boiling, Freezing and Condensation Heat Transfer", "Heat Transfer and its Assessment", "Heat Transfer Calculations", and each section discusses a wide variety of techniques, methods and applications in accordance with the subjects. The combination of theoretical and experimental investigations with many important practical applications of current interest will make this book of interest to researchers, scientists, engineers and graduate students, who make use of experimental and theoretical investigations, assessment and enhancement techniques in this multidisciplinary field as well as to researchers in mathematical modelling, computer simulations and information sciences, who make use of experimental and theoretical investigations as a means of critical assessment of models and results derived from advanced numerical simulations and improvement of the developed models and numerical methods.

How to reference

In order to correctly reference this scholarly work, feel free to copy and paste the following:

Douglas Fernandes Barbin and Vivaldo Silveira Junior (2011). Comparison of the Effects of Air Flow and Product Arrangement on Freezing Process by Convective Heat Transfer Coefficient Measurement, Heat Transfer - Theoretical Analysis, Experimental Investigations and Industrial Systems, Prof. Aziz Belmiloudi (Ed.), ISBN: 978-953-307-226-5, InTech, Available from: <http://www.intechopen.com/books/heat-transfer-theoretical-analysis-experimental-investigations-and-industrial-systems/comparison-of-the-effects-of-air-flow-and-product-arrangement-on-freezing-process-by-convective-heat>

INTECH
open science | open minds

InTech Europe

University Campus STeP Ri

InTech China

Unit 405, Office Block, Hotel Equatorial Shanghai

Slavka Krautzeka 83/A
51000 Rijeka, Croatia
Phone: +385 (51) 770 447
Fax: +385 (51) 686 166
www.intechopen.com

No.65, Yan An Road (West), Shanghai, 200040, China
中国上海市延安西路65号上海国际贵都大饭店办公楼405单元
Phone: +86-21-62489820
Fax: +86-21-62489821

© 2011 The Author(s). Licensee IntechOpen. This chapter is distributed under the terms of the [Creative Commons Attribution-NonCommercial-ShareAlike-3.0 License](#), which permits use, distribution and reproduction for non-commercial purposes, provided the original is properly cited and derivative works building on this content are distributed under the same license.