Effect of container openings and airflow rate on energy required for forced-air cooling of horticultural produce

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de Castro, L.R., Vigneault, C. and Cortez, L.A.B. 2005. **Effect of container openings and airflow rate on energy required for forced-air cooling of horticultural produce**. Canadian Biosystems Engineering/Le génie des biosystèmes au Canada **47**: 3.1-3.9. A research tool previously developed to investigate air distribution in horticultural produce containers during forced-air precooling was used to determine the effect of airflow rate and opening configuration on air pressure drop and rate and uniformity of the cooling process. Further analysis performed on previously tested opening configurations determined their influence on energy efficiency. A system efficiency coefficient, consisting of the overall Energy Added Ratio (EAR) was demonstrated as a functional tool during the container design, since it considers peculiarities of the forced-air cooling system and produce physiology. The results obtained for containers with handle openings and 2, 4, 8, and 16% opening area were used to evaluate the additional energy required to remove the heat generated by the forced-air fan and produce respiration. These results were also compared to produce in bulk and to produce packed in containers having four 0.5%-holes in the corners to analyze the influence of hole positioning. The four large 0.5% opening configuration resulted in poor energy performance and cooling uniformity when compared to uniformly distributed smaller holes. Furthermore, the airflow rate could be optimized based on the respiration rate of the produce and container opening area. **Keywords**: efficiency, design, respiration rate, box, handling, packing.

Un outil de recherche développé précédemment pour quantifier la distribution de l'air dans les contenants de produits horticoles en cours de refroidissement à l'air forcé a été utilisé pour déterminer l'effet du débit d'air et de la position des ouvertures sur les pertes de pression et l'uniformité du procédé. Des analyses supplémentaires ont été réalisées sur des résultats expérimentaux sur les effets de la position des ouvertures pour déterminer leur effet sur l'efficacité énergétique. Un coefficient d'efficacité énergétique considérant l'énergie additionnelle ajoutée au cours du refroidissement des produits (EAR) s'est avéré être une outil pratique pour la conception de contenants puisqu'il prend en compte les coûts associés à l'inefficacité de systèmes à l'air forcé et des particularités physiologiques des produits. Les résultats obtenus pour des contenants ayant des poignées ouvertes et des ouvertures de 2, 4, 8 et 16% de surface ont été utilisés pour évaluer l'énergie additionnelle ajoutée par les ventilateurs et la respiration des produits pendant le refroidissement. Ces résultats ont aussi été comparés à des produits en vrac et des produits emballés dans des boîtes standards ayant quatre ouvertures de 0.5% chacune situées près de leurs coins pour déterminer l'effet de la position des ouvertures. Cette comparaison démontre la faible efficacité énergétique et la grande hétérogénéité obtenues en utilisant les contenants ayant quatre ouvertures périphériques de 0.5% par rapport à des petites ouvertures uniformément réparties sur toute la surface des boîtes. De plus, les débits d'air à utiliser lors de prérefroidissement peuvent être optimisés en fonction du taux de respiration des produis et des grandeurs des ouvertures des contenants.

Introduction

The efficiency of a forced-air cooling process for fruits and vegetables is mainly indicated by the cooling rate and cooling uniformity it produces (Vigneault et al. 2005) in contrast to the energy input required by precooling and refrigeration systems (Thompson and Chen 1988). However, a more efficient process is needed to better maintain produce quality and reduce energy input. Cooling process performance determines the amount of electrical energy inputed directly to operate the compressor and fans (ASHRAE 2000). Energy input to the forced-air circulation system depends on the airflow rate through the fans, which is related to the fan's physical characteristics, operating speed, and pressure drop (Brooker et al. 1974).

The energy effectiveness of a particular cooling method is expressed in terms of an energy coefficient, which is defined as the ratio of total thermal energy removed and the sum of the electrical energy used through the cooling process (ASHRAE 2000), that is:

$$EC = \frac{E_t}{\sum EE} \tag{1}$$

where:

EC = energy coefficient,
E_t = total thermal energy removed, and
EE = electrical energy inputed.

E_t refers to the heat load to be removed such as produce heat (field and respiration) and heat transferred to the cold chamber through doors, walls, ceiling, floor, lights, people, and machines (Thompson and Chen 1988). Produce field and respiration heat account for the majority of E_t, but the portion of each factor also varies with the system. The term, EE, includes the energy inputed to the refrigeration system and the forced-air precooling equipment.

In this case, EC is also known as COP, the Coefficient Of Performance (ASHRAE 2000) and varies between 2.5 and 3.5.

However, this value does not include the energy used by the forced-air circulation system required for the precooling process, which depends on the characteristics of the cooling system. Thompson and Chen (1988) claimed a *COP* of 0.4 for forced-air cooling, which is in the 0.25 to 0.47 range reported by Kader (2002) who recommended 5% as the maximum container opening area. This could affect the *COP* of the system (Vigneault and Goyette 2002).

The ventilation system used to force the air through the horticultural produce requires some energy to transfer the thermal energy to the refrigeration system. This energy is released to the air during the cooling process when it crosses the fan blades or circulates around the driving motor. The amount of energy is proportional to the electrical input to run the fan (ASHRAE 2001). According to ASHRAE (2001), the total heat transferred by an air circulation system to the environment is given by:

$$Q_{air} = EE_m\left[\eta_m + \left(1 - \eta_m\right) f_{m,h}\right] \tag{2}$$

where:

Q_{air} = total heat transferred by air circulation system to the environment,
$f_{m,h}$ = fraction of motor heat loss transferred to air stream,
η_m = motor efficiency, and
EE_m = electrical energy used by the motor driving the fan.

There are several factors affecting the efficiency of forced air cooling of packed horticultural crops. Some parameters may be set to decrease the process time and improve the cooling uniformity but also increase the energy consumption. For example, increasing air velocity through the produce enhances the cooling homogeneity (Castro et al. 2004a); however, it requires more energy to drive the fan and more refrigeration energy to remove all the additional heat produced by air friction and fan inefficiency (Baird et al. 1988). The additional energy resulting from faster air circulation could be compensated, up to a certain limit, by more uniform air distribution and faster cooling rate.

On the other hand, lowering the air velocity through a mass of produce could reduce the energy required per unit of time, but the operating time often increases the total energy required. Vigneault et al. (2004a) developed a dimensionless number (*Vi*) to compare the performance of different forced air cooling systems. The *Vi* number was defined as the coefficient of heterogeneity of air velocity distribution through a porous medium.

Optimal operating conditions should be determined for maximal efficiency. One way of enhancing the efficiency is by enlarging the container opening area, which reduces the pressure drop through the whole system. However, since this method decreases the structural resistance of the container and the supporting surface supplied to the produce, it should be carefully studied. The container opening area and position, rather than the shape, play an important role in cooling efficiency (Vigneault and Goyette 2002; Kader 2002; Edeogu et al. 1997; Arifin and Chau 1988; Baird et al. 1988; Haas et al. 1976). For liquid-ice processes, the vented area must be sufficiently narrow to minimize the loss of ice particles (Vigneault and Goyette 2001). In the case of forced-air cooling, however, restricting the openings to less than 25% of the container surface significantly increases the air pressure drop and consequently, the energy required to run the fan (Vigneault and Goyette 2002; Haas et al. 1976). Baird et al. (1988) also reported that energy consumption increases as the opening area decreases, suggesting 10% as the minimum opening area needed to not compromise the cooling rate.

Performance calculations of the fan and air distribution systems require a detailed pressure balance of the entire network. Air is driven by the pressure differential, so any obstruction in the air path restricts its circulation through packed produce. Therefore, cooling efficiency can be jeopardized by misalignment of the openings of palletized containers, inappropriate produce stacking arrangement, or secondary packaging (Faubion and Kader 1997; Chau et al. 1985). Since so many factors affect the cooling efficiency, a research tool was developed by Vigneault and Castro (2005) and Vigneault et al. (2005) to perform tests under controlled and stable conditions.

The aim of this research was to evaluate the effect of opening configuration, total area and position, and airflow rate on the overall cooling system efficiency. Particular objectives were to develop a coefficient to verify the influence of produce respiration rate on energy requirement and to establish additional criteria for designing containers for fruit and vegetable handling.

MATERIAL and METHODS

Produce simulator

Solid polymer spherical balls of 52.4 mm diameter and 125.6 g were used to represent horticultural produce of the same shape as described in detail by Vigneault and Castro (2005). The balls were selected for their relatively high uniformity in terms of cooling index (-0.141 ± 0.008 min^{-1}) and heat capacity (1.12 ± 0.07 kJ kg^{-1} °C^{-1}). Each ball was instrumented with a thermocouple positioned at its center.

Experimental set-up

Sixty-four instrumented balls were stacked uniformly distributed throughout a stack of other 448 balls to form a cubic matrix of 8x8x8 balls. Four acrylic plates were assembled to simulate a forced-air cooling tunnel containing the ball matrix (Fig. 1). The air-outlet of the tunnel consisted of a 610 mm long plenum enabling air pressure drop (*APD*) measurements across the ball matrix. The end of the air-outlet tunnel was tightly sealed to an aspiration chamber. The air was released to the atmosphere through an airflow measurement device. The whole experimental set-up was placed in a cold chamber maintained at 4°C to generate a precooling process for the ball matrix. A heat-exchanger was built to minimize the temperature variation at the air inlet during the experiments. The center temperature of the sixty-four balls, the air temperature in the cooling tunnel before and after the ball matrix, the temperature of the cold chamber, the pressure drop through the ball matrix and plates, and the dynamic pressure of the air circulating through the airflow measurement device were simultaneously recorded at 20-s intervals.

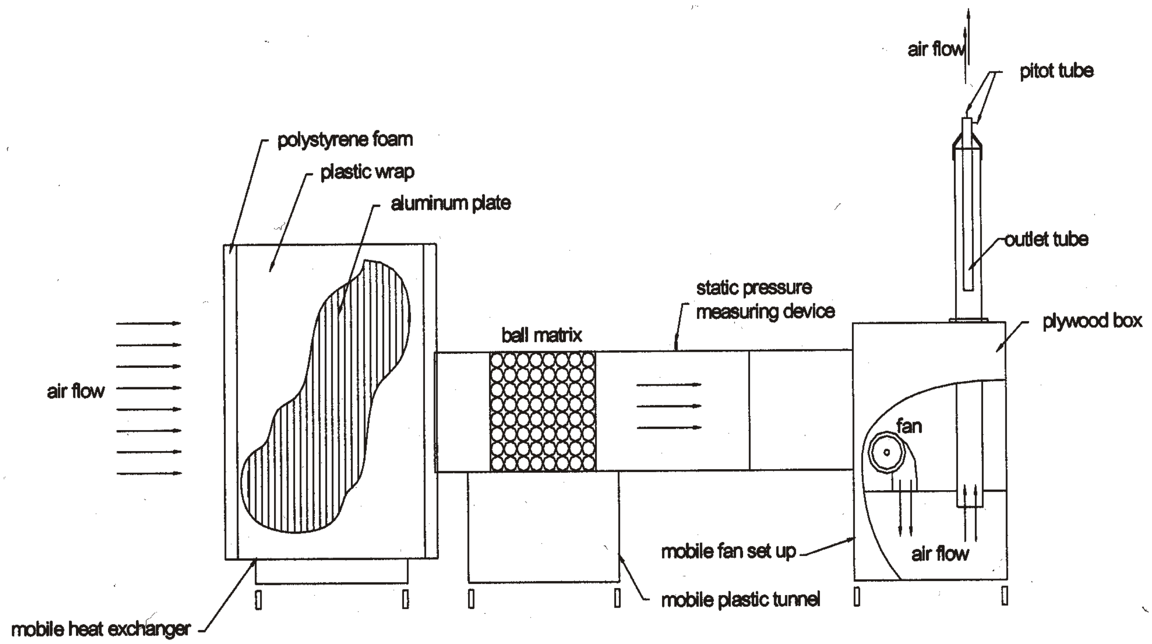

Fig. 1. Experimental set-up with forced air tunnel, ball matrix, fan, and dynamic and static pressure measuring devices.

Opening configurations

A fully open configuration was initially tested to determine the correlation between the half-cooling time (*HCT*) of the 64 produce simulators and air approach velocity. Based on the ball matrix volume, the airflow rates studied were equivalent to 0.5, 1, 2, and 3.9 L s^{-1} kg^{-1} of apple (Vigneault et al. 2004b). A group of opening configurations was investigated by placing a pair of plates on both open sides of the matrix to simulate a two-sided-perforated package. The opening configurations (Fig. 2) were selected from those studied in previous research (Castro et al. 2004b; Vigneault et al. 2004a), according to their responses to air distribution heterogeneity (*Vi*) and air pressure drop (*APD*) when compared to the fully open matrix. The first configuration with peripheral openings only, which is commonly used in the fruit and vegetable packing industry, consisted of four 0.5%-holes distributed near the corners of the package and accounting for a total open area of 2%. The four other configurations consisted of a standard handle and uniformly distributed openings (Vigneault et al. 2004a). The openings were made by 3-mm width opened slots of different lengths uniformly distributed on the plate resulting in total venting areas of 2, 4, 8, and 16%.

Experimental procedure

Prior to the beginning of each test, the forced-air tunnel containing the balls was placed in a warm chamber maintained at 28±1.0°C. The balls were warmed using a forced air system. After this conditioning period, the perforated plates were installed and the tunnel was placed in the 4°C cold room. The forced air cooling system was immediately turned on. The data were recorded until the temperature of the warmest ball had reached 6.9°C. The temperature-time data recorded were used to calculate the *HCT* and cooling rate (*CR*) of each ball for all treatments by using a dedicated Excel™ macro developed by Goyette et al. (1996).

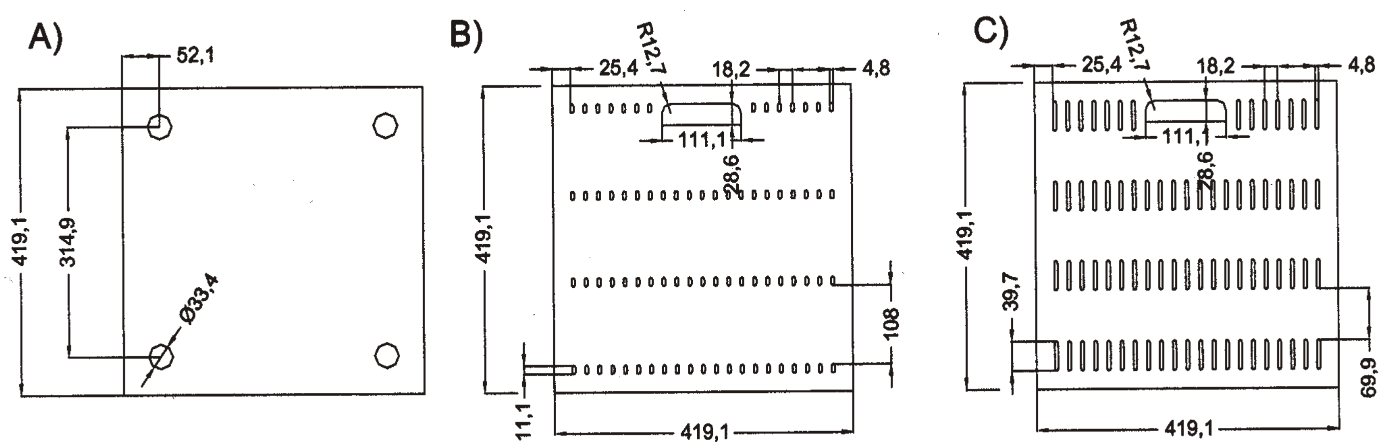

Fig. 2. Opening configurations studied: (A) four 0.5%-holes distributed in corners; (B) container with handles and 2% opening area; and (C) container with handles and 16% opening area.

Energy input

The total thermal energy, E_t, (Eq. 1) was estimated by calculating independently the different energy sources involved in the process, which included the produce field heat (E_p), respiration heat (E_r), and the ventilation energy required by the forced air system (E_v).

The field heat is the energy removed from a mass of produce and is given by:

$$E_p = mc_p\left(T_i - T_f\right) \tag{3}$$

where:

E_p = produce field heat (kJ),
m = mass of produce (kg),
c_p = specific heat of produce (kJ kg^{-1} °C^{-1}),
T_i = initial temperature of produce (°C), and
T_f = final temperature of produce (°C).

The respiration heat, E_r, generated during the cooling process is the bio-chemical energy generated by physiological activity of living vegetable or fruit. The heat released in this process depends upon the type and the mass of produce, the temperature along the cooling process, and the duration of the precooling process which corresponds to the time required for the slowest cooling ball to reach a 7/8 cooling process (Eq. 4). All the calculations were made assuming a 1 kg mass of produce.

$$E_r = 10^{-6} m \sum Q_{r,produce} \Delta t \tag{4}$$

where:

E_r = respiration heat (kJ),
$Q_{r,\,produce}$ = respiration rate for particular produce (mW/kg), and
Δt = time period under consideration.

The four horticultural produce selected for comparison covered the full range of respiration rates suggested by Kader (2002): low (apple), moderate (lettuce), high (strawberry), and very high (broccoli). These respiration rates were calculated with equations (Eq. 5a, 5b, 5c, 5d) obtained by regression analysis of data presented by ASHRAE (2002) for temperatures ranging from 0 to 25 ºC.

$$Q_{r,apple} = 8.848\exp(0.0933T) \tag{5a}$$

$$Q_{r,lettuce} = 36.54\exp(0.0807T) \tag{5b}$$

$$Q_{r,strawberry} = 48.67\exp(0.1048T) \tag{5c}$$

$$Q_{r,broccoli} = 87.01\exp(0.1197T) \tag{5d}$$

where: T = temperature (°C).

Considering a precooling system where the fan and motor assembly is mounted inside the air stream, the fraction of motor heat loss transferred to the air stream ($f_{m,h}$, Eq. 2) is equal to 1 (ASHRAE 2000). The ventilation energy, E_v, (Eq. 6), depends only on the airflow, the total pressure drop across produce and container, the fan efficiency (η_f) generally considered as 0.6 (ASHRAE 2000), and the fan operating time.

$$E_v = \frac{D_{air}\,APD}{\eta_f}\,t \tag{6}$$

where:

E_v = ventilation energy (kJ),
D_{air} = airflow rate (m^3/s),
APD = total pressure drop across produce and container (kPa), and
t = operating time (s).

The *APD* was calculated by adding the *APD* through the produce and through the container openings, which were experimentally obtained by Vigneault et al. (2004b), and Vigneault et al. (2004a) and Castro et al. (2004b), respectively.

The fan operation time was considered here as only package and airflow rate dependent. The produce cooling process was considered the same for the fruits and vegetables used as examples (apple, lettuce, strawberry, and broccoli) and was based on the results obtained using the produce simulator. This simplification does not compromise the analyses performed because the main objective was to determine the effect of the different package configurations and airflow rates on energy efficiency. Besides, the aim was to represent produce from four different categories rather than specific produce. Using the cooling rates of specific fruits and vegetables would not achieve either of these research goals. Nevertheless, respiration energy (E_r) and ventilation energy (E_v) are both dependent on the duration of the cooling process. Therefore, the absolute values of the different sources of energy would change if the actual cooling rates of those fruits and vegetables were used, but their relative values should be about the same.

Energy Added Ratio (EAR)

Since the mass of produce and the temperature differential, and thus the field heat, were the same for each produce and airflow comparison, an overall energy added ratio (*EAR*) was developed to measure the effect of container openings and airflow on the energy to be removed during the cooling process. *EAR* (Eq. 7) is the ratio of the energy added during the cooling process, which are the respiration energy (E_r) and the ventilation energy (E_v), compared to the initial energy that the produce contains at the beginning of this process, namely the field heat energy (E_p).

$$EAR = \frac{E_r + E_v}{E_p} \tag{7}$$

The advantage of using *EAR* to compare the different systems instead of the standard *COP* is that *EAR* considers E_p, E_r, and E_v which do not depend on the mechanical characteristics of the cooling system. The other sources of energy to be removed as well as the total electrical energy that would be required for the cooling process, including the cold chamber operation energy, are not considered. Therefore, a performance analysis using *EAR* could be applied as a simple and more practical method to compare the performances of any precooling system for different airflow rates and opening configurations. A value of 0 would represent a system that extracts instantaneously the field heat of the produce; thus, any higher value of *EAR* represents a decrease of energy performance.

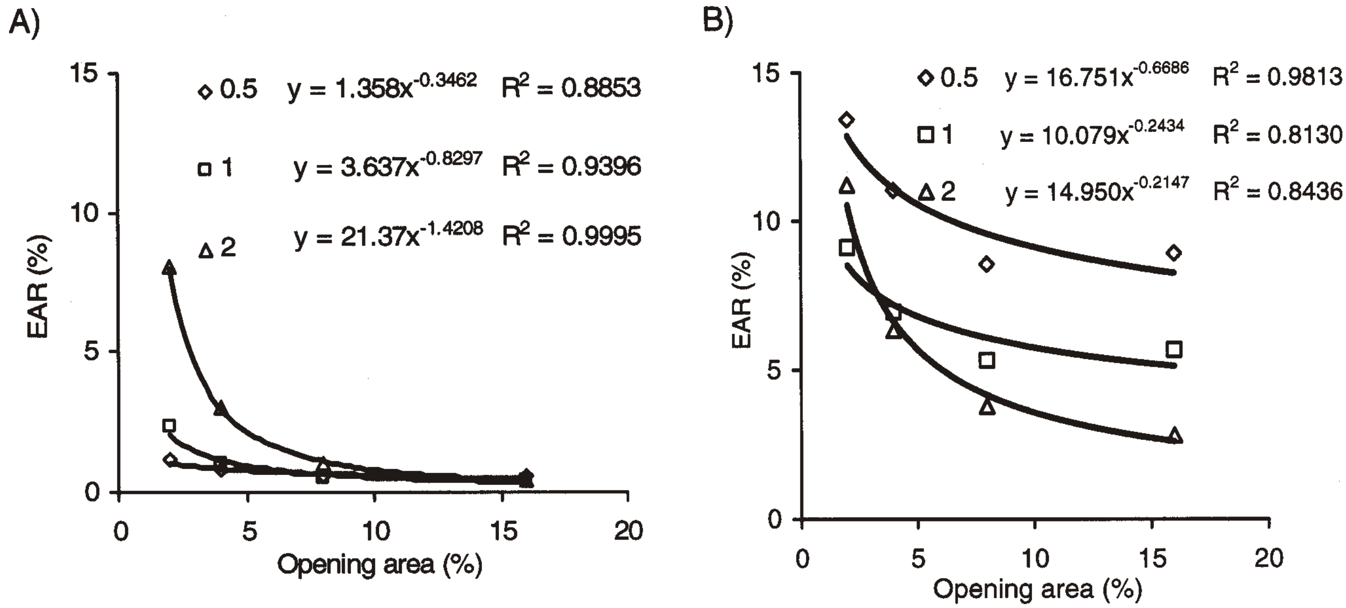

Fig. 3. Energy Added Ratio (EAR) versus opening area at each airflow rate for (A) low and (B) very high respiration produce, respectively.

Table 1. Energy Added Ratio (EAR, %) for each combination of opening area (OA), airflow rate (D_{air}), and respiration activity (L = low; M = moderate; H = high; VH = very high).

	OA (%)	D_{air} ($L\ s^{-1}\ kg^{-1}$)	Respiration activity L	M	H	VH
Container with handles	2	0.5	1.158 abcd*	2.968 lmn	5.805 uvx	13.432 ϵ
		1	2.387 ghijkl	3.582 opq	5.045 stu	9.104 βθ
		2	8.045 α	9.363 βθ	9.656 θ	11.208 δ
	4	0.5	0.794 abcd	2.351 ghijk	4.709 rst	11.033 δ
		1	1.004 abcd	2.064 fghij	3.358 nop	6.946 z
		2	3.008 mno	4.329 qrs	4.665 rst	6.331 vxz
	8	0.5	0.582 abc	1.891 efgh	3.705 pqr	8.544 αβ
		1	0.538 abc	1.447 bcd	2.476 hijkl	5.329 stuv
		2	1.069 abcd	2.234 ghijk	2.475 hijkl	3.802 pqr
	16	0.5	0.578 abc	1.921 efgh	3.824 pqr	8.906 βθ
		1	0.432 ab	1.398 bcd	2.530 ijkl	5.667 tuvx
		2	0.426 ab	1.506 cdef	1.695 defg	2.835 klmn
	100 (fully open)	0.5	0.434 ab	1.486 cdef	2.860 klm	6.512 xz
		1	0.279 a	1.026 abc	1.774 defg	3.858 pqr
		2	0.271 a	1.343 bcd	1.509 cdef	2.591 jklm

* Values followed by the same letter are not significantly different based on Tukey Test using α=0.05.

Statistical analysis

The effect of opening area and airflow rate on energy were investigated for produce of different respiration rates and porosities, resulting in various operating times, pressure drops, and energy requirements. *EAR* was calculated for each combination of opening configuration, airflow rate, and respiration activity level. The *EAR* results were then analyzed through a Multivariate Analysis of Variance followed by a Tukey test at 0.05 significance level using SPSS v. 11.5 (SPSS 2004).

RESULTS

The opening configuration and airflow rate both had a significant effect on the energy added ratio (*EAR*). Nevertheless, their effects were highly influenced by produce respiration rates (Fig. 3).

In general, *EAR* of the cooling process decreased as the opening area increased. Enlarging the opening area to 8% reduced the added energy but only a slight decrease was observed when increasing from 8 to 100% (Table 1).

For all respiration rates, the difference between the *EAR* results with 8 and 16% opening area was not significant, except with an airflow rate of 2 $L\ s^{-1}\ kg^{-1}$. No statistical difference was found between 8 and 16% for the full duration of the process and for all the pressure drops with an airflow rate up to 0.5 $L\ s^{-1}\ kg^{-1}$. Since both respiration and ventilation energies depend on the cooling time, the shorter cooling time resulting from a higher airflow rate (1 $L\ s^{-1}\ kg^{-1}$) compensated for the higher pressure drop caused by less opening area. However, at the maximum airflow rate and 8% opening area, the cooling process was not fast enough to maintain *EAR* at low levels. At this airflow rate the 16% opening showed an *EAR* as low as 0.426% demonstrating very

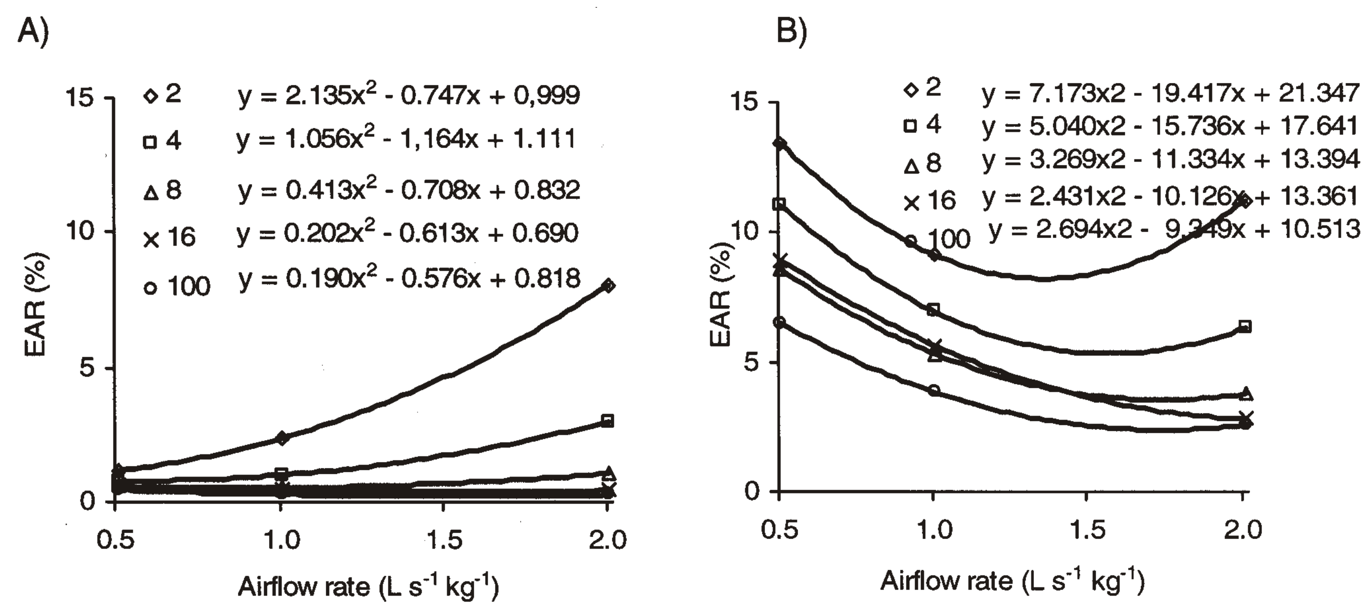

Fig. 4. Energy Added Ratio (EAR) versus airflow rate for each opening area for (A) low and (B) very high respiration produce, respectively.

little opportunity for increasing the performance of the system when the opening area is fairly large at low produce respiration rates, which explains the non-significant difference obtained between 16 and 100% openings at 2 L s^{-1} kg^{-1}.

Respiration rates

Low respiration rate Figure 4 shows a general increase of *EAR* while increasing the airflow rate for low respiration produce. According to the same figure, decreasing the opening area and increasing the airflow rate required additional energy. However, the effect of airflow rate on energy progressively declined as the opening area was increased. In fact, for the largest opening areas (16 and 100%), no significant difference was found between 1 and 2 L s^{-1} kg^{-1}. Moreover, an inversion of effect occurred since at these two opening percentages the highest airflow rate required less energy than at 0.5 L s^{-1} kg^{-1}. This was due to the fact that when the airflow rate is enhanced at these opening areas the increase in added energy (20%) is considerably less than the energy reduction (40%) caused by the decrease of the cooling process time resulting from the respiration and ventilation energies. Therefore, when the container is designed for low respiration rate produce, the pressure drop will be the limiting factor for lower opening areas but not for larger opening areas. For containers with less opening area, increasing the airflow rate to reduce cooling time and improve uniformity of air distribution does not result in energy savings because it is not enough to overcome the higher increment in pressure drop. Baird et al. (1988) also stated that this increment would result in a critical increase of cooling costs especially for areas less than 3%. On the other hand, for areas of 16% and higher the cooling time determined the energy efficiency since the container openings did not generate an important *APD*. In this case, increasing the airflow had greater effect on increasing cooling rate and therefore limiting respiration activity, than on the increasing of *APD*.

Besides reducing opening area, the stacking arrangement may also aggravate the air restriction. It is important to mention that this research considered the data produced by a columnar stack pattern, but the actual pressure drop would be even higher for a random stacking as observed by Chau et al. (1985) with oranges in cartons having 4% opening area. However, the results would be approximately the same if low respiration produce of other shapes were considered, such as carrot or celery, since the arrangement porosity was previously showed to influence *APD* only on spherical produce (Vigneault et al 2004b).

Very high respiration rate For very high respiration rate produce, the *EAR* reached minimum values as a function of the airflow rate for each opening area (Fig. 4). Beyond these points, any increase in airflow rate to reduce cooling time and consequently respiration and ventilation energy, did not compensate for the energy increase due to the *APD*. Likewise, below these optimum airflow rates, lowering the airflow, and thus the pressure drop, would not compensate for the longer processing time and higher respiration activities.

The regression equations presented for each curve (Fig. 4) were used to calculate the optimum airflow rates, which increased as the area increased. Optimum airflow rates of 1.35, 1.56, 1.73, and 2.08 L s^{-1} kg^{-1} were found for opening areas of 2, 4, 8, and 16%, respectively. This optimum airflow rate tendency was also noticed at a moderate respiration rate for opening areas equal or superior to 4%. However, this tendency for optimum airflow rates increased for produce generating more respiration heat (Fig. 5). The curve tendency could not be identified at low respiration rate likely because the inflexion point occurred at a lower value than the one studied (0.5 L s^{-1} kg^{-1}), especially for smaller opening areas.

Therefore, with higher produce respiratory activity, increased airflow is required to hasten the cooling process and compensate for the respiration energy increase. When a smaller opening area is used, a lower airflow rate is necessary to produce this equilibrium due to a larger *APD*. On the other hand, with a larger opening area, the energy efficiency must be

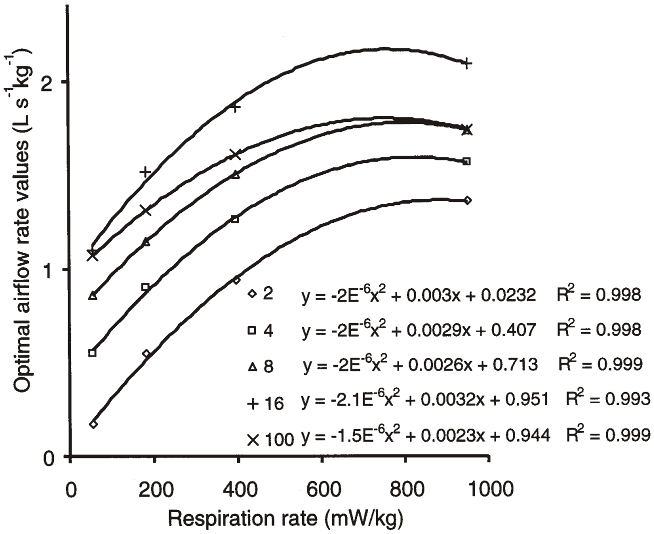

Fig. 5. Optimal airflow rate versus respiration rate for opening percentages of 2, 4, 8, 16, and 100%.

optimized by decreasing the magnitude of respiration energy through faster cooling obtained with a higher airflow rate. This outcome is confirmed with the cooling time responses found by Arifin and Chau (1988) for strawberries packed in carton with four openings and 18% total vented area. They reported a 50% reduction in cooling time (136 to 72 min) when increasing airflow from 1 to 2 L s^{-1} kg^{-1}. Similar results were found by the present authors for this high respiration product (129 to 65 min).

Opening positions

The 2% opening area formed by four 0.5%-holes distributed on the corners of the package surface presented the highest energy added during the cooling process. Increasing the airflow increased *EAR*, although no significant difference was found between the results produced with 0.5 and 1 L s^{-1} kg^{-1}. Table 2 shows the magnitude of the effect for the maximum airflow rate on energy efficiency compared to produce respiration activity. The four-hole container configuration submitted to 2 L s^{-1} kg^{-1} generated the highest energy added regardless of the respiration activity. These high values were partially due to the poor uniformity of air distribution (*Vi*=0.83) and especially to the high *APD*, 1.27 kPa (Castro et al. 2004b), compared to the 2% uniformly distributed opening area package which produced a *Vi* of 0.34 and *APD* of 0.43 kPa at the same airflow rate (Vigneault et al. 2004a).

On the other hand, the lowest *EAR* was obtained with the holes distributed uniformly on the package surface at the lowest respiration and airflow rates (Table 2). At high and very high respiration activity, *EAR* tended to first decrease but then to increase as more air was circulated through the system. Lowering the produce respiration activity decreased the airflow rate value correspondening to the minimum *EAR* result.

The main result obtained for the comparison of the two types of 2% opening configurations was the sharp difference between their additional energy required at the maximum airflow supplied. This large difference was due to the contrary effect of airflow rate on cooling uniformity between the two opening positions. When airflow rose, heterogeneity of air distribution (*Vi*) was reduced in containers with evenly distributed openings, but enhanced in peripherally positioned openings. Furthermore, this increase of *Vi* limited the improvement of the cooling rate at higher air velocities. Thus, for the corner opening configuration, the reduction of cooling time at the maximum airflow level is not enough to offset the greater increase of *Vi* and *APD*.

For instance, at 0.5 and 2 L s^{-1} kg^{-1} the maximum half-cooling times (HCT_{max}) obtained with the peripheral openings are 12 and 42% higher than for uniformly distributed holes, respectively. The difference between the *Vi* values for the two opening configurations was even larger than for HCT_{max}, going from 10 to 60%. Yet for the same airflow rate levels, the pressure drop difference showed a lower increase, from 42% at 0.5 L s^{-1} kg^{-1} and 66% at 2 L s^{-1} kg^{-1} (Castro et al. 2004b; Vigneault et al. 2004b), but in the latter case, this is equivalent to a considerable *APD* value of 0.85 kPa. Therefore, container design involving many small openings uniformly distributed on the package surface would be preferable for a more efficient cooling process. This is contrary to Kader's (2002) recommendation of few larger holes.

The optimum airflow value was not verified for corner holes at very high respiration likely because it did not occur within the range of the airflow rates studied. Table 2 shows that with the peripheral configuration, no significant effect has been identified for 0.5 and 1 L s^{-1} kg^{-1}airflow rate on the *EAR*, however the difference between these values decreased as the

Table 2. Energy Added Ratio (EAR, %) of containers with 2% opening area as a function of airflow rate (D_{air}) and respiration activity (L = low; M = moderate; H = high; VH = very high).

Position of opening	D_{air} (L s^{-1} kg^{-1})	Respiration activity			
		L	M	H	VH
On corners	0.5	2.347 ab*	4.299 abc	7.203 bcdef	14.967 h
	1	7.354 bcde	8.486 cdef	9.727 efg	11.208 gh
	2	36.940 i	38.873 i	39.324 i	41.650 i
Uniformly distributed	0.5	1.158 a	2.968 ab	5.805 abcde	13.432 gh
	1	2.387 ab	3.582 abc	5.045 abcde	9.104 defg
	2	8.045 cdef	9.363 efg	9.656 efg	11.208 fg

* Values followed by the same letter are not significantly different based on Tukey Test using α=0.05.

respiration activity increased. Thus this outcome could suggest that *EAR* tended to decline beyond 0.5 L s^{-1} kg^{-1}, reaching a minimum point at some value around 1 L s^{-1} kg^{-1} and then rising until reaching 2 L s^{-1} kg^{-1}.

As aforementioned, decreasing the opening area and the produce respiration activity lowered the optimum airflow rate required to balance respiration and ventilation heat. Thus, the inflexion point likely occurs at a lower airflow rate than the minimum value studied (0.5 L s^{-1} kg^{-1}) for low respiration activity produce. The smallest opened package (2%) had pressure drop as the limiting factor in the cooling process and, although four-0.5% holes form the same total vented area, the effect of the holes positioning on cooling heterogeneity likely added a further restriction, reducing even more the optimum airflow rate. In this case, the selection of airflow to maximize the energy efficiency should be carefully analyzed since long cooling time can also compromise the produce quality by modifying the gas atmosphere. Even low respiration produce, such as pear, can be harmed if the package opening and airflow rate are not sufficiently high to dissipate the gases released during respiration (Faubion and Kader 1997). These authors claimed that although accumulation of carbon dioxide can reduce the sensitivity to ethylene, concentrations of more than 10% could cause critical internal carbon dioxide injuries.

CONCLUSION

The system efficiency coefficient *EAR* was demonstrated as a functional tool during container design, since it considers the peculiarities of the forced air cooling system and produce physiology. The coefficient rapidly decreased as the opening area was gradually increased from 2 to 16% and continued decreasing, although only slightly until reaching a fully open condition. Therefore, the results pointed to an opening area between 8 and 16% for energy use optimization. Further investigation would be necessary to identify a specific value within this range, however container design, manufacturing and raw material costs, and many other parameters should be considered in determining the best opening configuration.

The optimum airflow rate for the cooling process, however, was closely dependent on the produce respiration rate and the container opening area investigated. A higher airflow rate is required for produce with high respiration activity in a container with a larger opening area to enhance the cooling time and balance the heat produced by respiration and ventilation.

Since decreasing the opening area restricts air circulation, the pressure drop becomes the limiting factor and the optimum airflow rate is reduced. In addition, if this opening area is formed by non-uniformly distributed holes, any increase in airflow will generate an even higher pressure drop and air distribution heterogeneity, which also increases the cooling time and the total energy to be removed by the refrigeration system. Therefore, for the non-uniform configuration, only a very low airflow would be able to offset ventilation and respiration energies with a slight improvement of cooling time. In this case, optimization of energy consumption should be carefully analyzed not to generate produce quality deterioration.

By comparing the results of four 0.5%-holes distributed on corners to 8% uniformly distributed openings on the package surface, it could be concluded that the 0.5%-holes distributed on corners should generally be avoided. The reason is that the four 0.5%-holes generate from 1.75 up to 34 times more energy to be removed by the cooling system when using low airflow rate (0.5 L s^{-1} kg^{-1}) with high respiration activity produce, and high airflow rate (2 L s^{-1} kg^{-1}) with low respiration activity produce, respectively.

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