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Analysis of the spatiotemporal temperature fluctuations inside an apple cool store in response to energy use concerns

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Highlights

- ✓ Air circulation fans are the major source of heat load during apple cold storage.
- ✓ Attempt to reduce energy cost should deliberate on reducing fan operation time.
- ✓ The temperature of the apple can be made to fluctuate within limited range.
- ✓ It has been demonstrated that fan and refrigeration cycling works to save energy.
- ✓ Load shifting of controlled atmosphere stores for saving energy costs seems worthwhile.

Abstract

Energy cost concerns have led to question the temperature and air flow regimes in cool store facilities. In this paper, a CFD model of the air flow and temperature dynamics inside an apple fruit

cool store was developed, validated and used to analyse energy cost saving alternatives. Load shifting was attempted by cycling the temperature set point between 1.2 °C and 0.6 °C following a day/night regime. Discontinuous use of the cooling operation, including 12 h on / 12 h off; 10 h on / 14 h off; and 8 h on / 16 h off were also investigated. The study showed that the air circulation fan is the major source of heat load. Hence, an attempt to reduce energy cost should first deliberate on reducing the fan operation time. The study developed a generic tool to analyse temperature control options inside an apple cool store in response to energy use concerns.

Keywords: [Postharvest](#), Fruit cold storage, computational fluid dynamics, Energy conservation, cooling cycle, Load shifting

1 Introduction

Temperature management is one of the most important factor affecting quality during postharvest handling of fresh commodities. For apple fruit, temperature control starts immediately after harvest by [precooling](#) the produce from field temperature to optimal storage temperature. During the storage period, temperature should be kept low and within a narrow range of the optimum value (Kupferman, 2003). Temperature, if not controlled, rises due to heat production by the commodity and other heat sources such as fan motors, heat infiltration from outside, solar radiation, etc. Hence, strict control of constant low temperature is a standard practice to avoid losses during postharvest storage of fresh commodities. However, constant low temperature cooling contributes significantly to the energy use and also to the carbon footprint (East et al., 2013, Liu et al., 2010, Stoessel et al., 2012).

Sufficiently cooling the produce to the lowest possible temperature during period of cheap electricity price (off-peak hours) and reducing the cooling operation during high price period (peak-hours) can reduce energy cost (East et al., 2013; Koca and Hellickson, 1993; Waelti 1983; Yost, 1984). Such strategies must be applied carefully in such a way that [produce](#) shelf life is unaffected while energy efficiency is enhanced. To this end, it is crucial to know the temperature, air flow, moisture and other gases distributions inside the cool store room for appraising effects on the shelf life of [produce](#).

This depends upon many factors including the thermal insulation of the cool room, the product's rate of respiration and the air circulation which depends on the physical layout of the cool room and the bins (Amos, 2004; East et al., 2013).

Understanding the influence of the aforementioned factors experimentally alone is time consuming and inadequate. Numerical models, on the other hand, can predict the air flow and temperature patterns at a high spatial and temporal resolution (Dehghannya et al., 2010; Smale et al., 2006). Due to its powerful visualization capabilities and acceptable accuracy of the predictions, computational fluid dynamics (CFD) is the primary method of choice (Ambaw et al., 2013a, Norton and Sun Da-Wen, 2006; Smale et al., 2006; Verboven et al., 2006). However, CFD modeling of a cool store room invokes the use of several simplifying assumptions. Hence, a priori validation with experimental data is necessary to assess accuracies (Norton and Sun Da-Wen, 2006; Smale et al., 2006).

In a cool store room there are large number of bins, each holding randomly stacked fruits of different shapes and sizes forming a complex system with spatially non uniform thermodynamic and aerodynamic properties. The porous medium approach is normally used for simplification. In this approach the complex geometry of the stacked fruit is transformed in to a spatially uniform porous medium in which volume-averaged equations are used to model the heat and mass transfer processes (Hassanizadeh and Gray, 1979; Liu and Masliyah, 2005; Quintard and Whitaker, 2005; Whitaker, 1999). Often local equilibrium between the fluid and solid phases is assumed to further simplify the problem. However, such assumptions may lead to errors in situations where the transport process is transient (Lerew, 1978; Verboven et al., 2006).

The aim of this work was to investigate the effect of temperature set point cycling and fan cycling during cold storage of apple fruit to address energy use concerns. A two phase porous medium CFD model of a typical apple cool store was developed, validated and used for the analysis. Air flow and temperature distributions in the cool store under different temperature control scenarios were quantified and visualized for the analysis.

2 Materials and methods

2.1 Fruit

Freshly harvested commercial grade ‘Jonagold’ apple (*Malus × domestica* Borkh, cv. ‘Jonagold’) from four different growers were collected immediately after picking at optimal harvest time in September, 2012 and transported to the cooling facilities (BelOrta, Borgloon, Belgium). Fruits were stored at 1 °C in normal air before the start of the experiment that took place over a period of three days.

2.2 Fruit bins

Fruit were placed in standard wooden bin with external Length / Width / Depth of 1.20 m × 1.20 m × 0.90 m (see Fig. 1(a)). The air gap (vent-area) between the planks of a typical bin was 5% and 6% on the vertical and the bottom faces, respectively. Each bin contained 380 ± 20 kg apple fruit. Apple fruit were randomly packed in the bin.

2.3 Cool store facility

Two cool stores with dimensions 12.0 m × 5.6 m × 8.0 m were used for experiments (see Fig. 1(c)). Fig. 1(b) shows the porous medium illustration of fruit loaded individual bin used to create the stacking in Fig. 1(c). The loading and storage conditions of the two cool store rooms were made identical so that two repetitions could be performed. Fruit bins were stacked in four columns 15 cm apart. The load consisted of six rows with eight layers of bins and two rows with six layers. Altogether, there were 240 bins, thus a load of 91 tons of ‘Jonagold’ apple fruit inside each cold store room. Each cool room was equipped with a cooling unit (evaporator-coils and fan assembly) located near the ceiling at one end of the cool store, blowing air out over the bins (see Fig. 1(c)). Six direct driven fans each with 550 W motor power blow the air across the evaporator-coils. The motors of the fans were inside the cool store, positioned upstream of the cooling coil. The air flow capacity of the cooling unit was $41\,500\text{ m}^3\text{ h}^{-1}$, with a refrigeration capacity of 70 kW. Each room was fitted with a

temperature controller which uses the returning air temperature measured at the back of the cooling unit as a monitor point. The rooms were kept under optimal controlled atmosphere (CA) conditions for ‘Jonagold’ apple ($[O_2] = 1 \%$ and $[CO_2] = 2.5 \%$).

2.4 Measurements

Six thermocouples (TESTO Saveris system, using PT100 probes) placed at six positions (positions indicated by spheres in Fig. 2) measured the air temperature inside the stack. The temperature probes had a measurement range of $-50 \text{ }^{\circ}\text{C}$ to $150 \text{ }^{\circ}\text{C}$ and an accuracy of $\pm 0.15 \text{ }^{\circ}\text{C}$. The thermocouples were wirelessly connected to extenders placed on top of the stack in the cool room. The extenders transmitted the measurement signal to a base station which was placed outside the cool room. The system saved the temperature information of the thermocouples every hour for three days.

The temperature of the cold air leaving the cooling unit was measured by a separate thermocouple placed 10 mm downstream of the cooling unit, giving the actual temperature-time profile of the air leaving the cooling unit.

The air velocity as it left the cooling unit was measured on a vertical grid of measurement points 10 mm in front of the cooling unit using air velocity transducers (TSI 8465, TSI, Shoreview, MN) that were based on thermal anemometry with an accuracy of $\pm 2 \%$ of reading.

2.5 Model formulation

2.5.1 Assumptions

The volume averaging technique (Hassanizadeh and Gray, 1979; Liu and Masliyah, 2005; Quintard and Whitaker, 2005; Whitaker, 1999) was used to develop a two-phase porous medium CFD model of the air flow and heat transfer. The model assumptions are discussed below.

- Fruit loaded bins were simplified to a porous medium consisting of a fluid phase (air) and a solid phase (apple fruit) with constant volume fraction in space and time.

- The flow resistance characteristics of individual bins as loaded with apple fruit were modeled using the Darcy-Forchheimer law with coefficients obtained from simulated pressure drop vs. velocity data using a detailed CFD model of the air flow and pressure drop across a single bin loaded with fruit (Ambaw et al., 2013b; Delele et al., 2009).
- Turbulence was modelled using the SST $k-\omega$ model, not only at the clear fluid region (outside the porous domain) but also inside the porous medium. Turbulence in the porous domain was treated as though the solid medium has no effect on the turbulence generation or dissipation rates.
- The physical properties of air and apple fruit were assumed constant within the temperature range of the application.
- Evaporation/condensation of moisture from the fruit was not included in the model.
- The heat of respiration was calculated as a function of temperature and gas (O_2 and CO_2) concentrations in the cool store room.
- Heat conduction through the walls, floor, door and ceiling was calculated based on an average outside air temperature of 10 °C. The insulation on all boundaries was assumed uniform.
- Since the air circulation fans were positioned upstream of the evaporator coil (a blow-through configuration), the fan heat was assumed totally removed directly by the evaporator coil.
- Heat loads due to lighting and other activities were not considered in the model since in practical CA storage of apples, the rooms are not opened, and there are no lights, and activities within the cool rooms are minimal.

2.5.2 Governing equations

Eqn. (1) and Eqn. (2) give the mass and momentum conservation, respectively.

$$\nabla \cdot \mathbf{u} = 0 \quad (1)$$

$$\frac{\partial(\mathbf{u})}{\partial t} + \nabla \cdot ((\mathbf{u} \otimes \mathbf{u})) - \nabla \cdot \left(\left(\frac{\mu + \mu_t}{\rho} \right) \nabla \mathbf{u} \right) = \mathbf{S}_U - \frac{1}{\rho} \nabla p \quad (2)$$

where $\mathbf{u}$ is the vector of the velocity (m s^{-1}), t is time(s), μ is the dynamic viscosity of air ($\text{kg m}^{-1}\text{s}^{-1}$), μ_t is the turbulent eddy viscosity ($\text{kg m}^{-1}\text{s}^{-1}$), p is pressure (Pa) and $\mathbf{S}_U$ (m/s^2) is the momentum source term. Inside the porous domain, $\mathbf{S}_U$, accounts for the pressure loss as given by Eqn. (3).

$$S_{u,i} = C_{R1i} u_i + C_{R2i} |\mathbf{u}| u_i \quad (3)$$

where i stands for direction in the Cartesian coordinate system (x, y or z). C_{R1i} , C_{R2i} are the viscous and inertial resistance coefficients, respectively. These coefficients were determined using a detailed CFD model of airflow across a fully loaded bin (Ambaw et al., 2013b). The calculated viscous and inertial resistance coefficients are summarized in Table 1.

The fluid (air) and solid (fruit) phase volume-averaged heat transport equations are given by Eq. (4) and Eq. (5), respectively:

$$\varepsilon \left(\rho_a C_{pa} \right) \left(\frac{\partial T_a}{\partial t} + \mathbf{U} \cdot \nabla T_a \right) = \varepsilon \nabla \cdot ((k_a + k_t) \nabla T_a) + a_{sa} h_{sa} (T_s - T_a) \quad (4)$$

$$(1 - \varepsilon) \left(\rho_s C_{ps} \right) \frac{\partial T_s}{\partial t} = (1 - \varepsilon) \nabla \cdot (k_s \nabla T_s) + (1 - \varepsilon) Q_s + a_{sa} h_{sa} (T_a - T_s) \quad (5)$$

where ε (-) is the porosity of the stack, C_{pa} ($\text{J kg}^{-1}\text{K}^{-1}$) is the heat capacity of air, C_{ps} ($\text{J kg}^{-1}\text{K}^{-1}$) is the heat capacity of apple fruit, ρ_a (kg m^{-3}) is the density of air, ρ_s (kg m^{-3}) is the density of apple fruit, T_a (K) is the air temperature, T_s (K) is the produce temperature, k_a ($\text{Wm}^{-1}\text{K}^{-1}$) is the thermal conductivity of air, k_t ($\text{Wm}^{-1}\text{K}^{-1}$) is the turbulent thermal conductivity. a_{sa} (m^{-1}) and h_{sa} ($\text{W m}^{-2} \text{K}^{-1}$) are the specific surface area and the interfacial heat transfer coefficient, given by Eq. (6) and Eq. (7), respectively (Nield and Bejan, 2013),

$$a_{sa} = \frac{6(1-\varepsilon)}{d_p} \quad (6)$$

$$\frac{1}{h_{sa}} = \frac{d_p}{Nu_{sa} k_a} + \frac{d_p}{\beta k_s} \quad (7)$$

where d_p (m) is the mean fruit diameter, ε is the porosity, β is a constant equal to 10 for a porous bed of spherical particles. The fluid-to-solid Nusselt number Nu_{sa} is, for Reynolds numbers (based on d_p) $Re_p > 100$, given by Eq. (8) (Nield and Bejan, 2013).

$$Nu_{sa} = 2.0 + 3.223 Pr^{1/3} Re_p^{0.6} (1-\varepsilon)^{0.6} \quad (8)$$

where Pr (-) is the dimensionless Prandtl number, equal to 0.72 for air. The heat of respiration Q_s (W m⁻³) was calculated from the rate of respiration of the fruit as a function of temperature and environmental gas conditions (O₂ and CO₂ concentration) (Hertog et al. 1998, Ho et al. 2013). Table 2 summarizes the model parameters and their values.

Since the fan was upstream of the evaporator coil in a blow-through configuration, the fan heat was assumed absorbed directly by the coil. The fan heat adds to the cooling coil load (ASHRAE, 2002).

2.5.3 Model geometry and boundary conditions

Fig. 1 (c) depicts the cool store room in full to be modelled. Fig. 2 (a) depicts half of the cool store room assumed sufficient based on symmetry principle. This imposes a zero flux of all quantities across the symmetry boundary. There are no convective or diffusive fluxes across the symmetry plane. This approach assumes the physical geometry of interest and the pattern of the airflow and heat transfer to be symmetrical.

The model contains a fluid domain and porous domain. The fluid domain is bounded by the room walls, door, ceiling, floor and the symmetry plane. The air gaps between the boxes were explicitly defined and incorporated into the fluid domain. Fruit loaded boxes were individually considered and modelled as porous domains as shown in Fig. 1 (b). The exit of the cooling unit (which is also the inlet

boundary to the free air domain) was defined by velocity and temperature boundary conditions. The inlet to the cooling unit, which corresponds to the outlet from the free air domain, was specified by velocity boundary condition in such a way that the mass flow in and the mass flow out from the air domain was balanced. The walls, ceiling, floor, door and surfaces of the cooling unit were considered as no slip boundaries.

The heat gain from the outside environment was included by incorporating heat fluxes on the walls, ceiling, floor and the door as given by Eqn. (10).

$$q_{wall} = \frac{k_w}{t_w} (T_\infty - T_w) \quad (10)$$

where q_{wall} is the heat flux (W m^{-2}), k_w ($= 0.022 \text{ W m}^{-1} \text{ }^\circ\text{C}^{-1}$) is the thermal conductivity and t_w ($= 0.12 \text{ m}$) is the thickness of the insulated walls and floor, T_∞ is outside temperature assumed to be $10 \text{ }^\circ\text{C}$, and T_w is wall temperature.

2.5.4 Solving method

The mode of air flow and heat transfer in the cool store cycles in time, requiring a transient and fully coupled air flow and heat transfer computation. Such simulation requires high CPU usage and a large simulation time than available. To this end, the analysis compared average temperature histories of cycling schemes based on a steady state air flow profile in the room, corresponding to an air exchange rate of 75 h^{-1} . Separately, a transient and fully coupled air flow and heat transfer model was used to simulate the spatiotemporal air flow and temperature dynamics under natural convection condition.

The problem was solved using ANSYS-CFX-15 software running on a Windows 7 PC with an Intel Xeon processor (2.66 GHz) with 48 Gb RAM. Hexahedral (for the porous domain) and tetrahedral (for the air domain) mesh elements were used for discretization of the computational domains. Level of grid independence was evaluated using the method of Richardson extrapolation (Franke et al., 2007; Roache, 1994). A grid of 7 million cells, which corresponds to discretization

errors of 2% and 5 % in estimating mass and heat fluxes through boundaries of the porous medium, respectively, was used. The advection scheme of the numerical solution uses a blend of central differencing and upwind differencing, locally. The transient scheme uses a second order backward Euler method. A time step of 360 s with 10 iterations per time step was used.

2.6 Validation

The airflow profile predictions of the porous CFD model were previously validated using a dedicated procedure (Ambaw et al. 2013b). In the current study the prediction of the transient temperature profiles was validated using experimental data from two operating commercial apple cool store rooms under cyclic temperature conditions (explained below).

Data taken from two identically operating commercial cool stores were used. Inside each cool store room, six thermocouples (TESTO Saveris system, using PT100 probes) were placed at six positions (balls in Fig. 2) to collect the air temperature inside the stack every hour for three days.

3 Cases studied

3.1 Cycling the set-point temperature of the cooling controller

Here, the temperature set point was made to cycle in a day/night regime. During the day (from 6.00 h to 22.00 h) the set point was 1.2 °C, at night (from 22.00 h to 6.00 h) it was 0.6 °C. Hence, the produce would be cooled slightly more during the night and the temperature is allowed to rise above the normal set point of 1 °C during the day while maintaining on average the optimal storage temperature of 1 °C. The style of the cycling was intended to exploit the night time cheap electricity price. To assess the premise of energy cost minimization, the day/night cycling scheme was compared with a continuous cooling scheme at 1 °C.

3.2 On-off cycling of the cooling unit

Here, the premise of adequately cooling the produce to the lowest possible temperature during period of cheap electricity price (off-peak hours) and turning the cooling system “off” during high price period was investigated. “On” implies cooling is undertaking at a constant cooling air temperature of 0.5 °C and at an air exchange rate of 75 h⁻¹, and “off” implies both cooling and fan operations were stopped. Cycling schemes including 16 h off / 8 h on, 14 h off /10 h on and 12 h off / 12 h on were examined. The pattern of cycling in the 16 h off / 8 h on scenario is similar to the set-point cycling scenario. However, while the air circulation fan is continuously running under the set-point cycling scenario, it is discontinuously running under the 16 h off /8 h on scenario.

3.3 Cool store room under cooling-off and fan-off condition

The natural convection flow inside the cool store room was simulated using a transient and fully coupled air flow and heat transfer model. Hence this case provided a detailed visualization and quantification of the spatial and temporal distribution of air flow and temperatures inside the cool store room under cooling off and fan-off condition.

4 Results and discussion

4.1 Air flow profiles

The forced convection air flow profile inside the cool store room is shown in Fig. 3. Air leaves the cooling unit at an average velocity of 3.2 m s⁻¹ and flows over the bins directly to the opposite corner (see Fig. 3(a) and (b)). The air speed decelerates to 1 m s⁻¹ before reaching the opposite wall. It then flows through the stack while returning back to the cooling unit, taking heat from the produce. Fig. 3(b) shows stream lines on vertical plane bisecting column1 (C1) and the volume average superficial air velocity per bin. The air velocity increases from bottom to top (along -Y-axis) and from the back wall towards the cooling unit (along - Z-axis). Inside the stack, air velocity was below 0.6m s⁻¹

¹ and in average it was 0.3 m s^{-1} . Fig. 3 (c) show the stream lines on horizontal plane bisecting layer 4 (L4).

Notice the recirculation zone below the evaporator. This is due to the uneven stacking in the cool store. The load in the cool store room consisted of six rows with eight layers of bins and two rows with six layers. The front rows (door side rows) were made shorter due to space limitation to manoeuvre a forklift during loading and unloading of the boxes. This creates unfilled space that affects the airflow distribution in the cool store room, and created the recirculation zone under evaporator units. Similar findings have been reported by Hellickson and Baskins (2003).

4.2 Temperature profiles

4.2.1 Day/night cycling of the temperature set point

Fig. 4 shows the measured time-temperature data of the cold air leaving the cooling unit as monitored by a thermocouple placed 10 mm in front of the cooling unit (full curve). The broken curve shows the set-point temperature to control the return air. The temperature set-point was 1.2°C during day time (from 6.00 h to 22.00 h) and 0.6°C during night time (from 22.00 h to 6.00 h). An overshoot/undershoot of about 0.6°C from the set-point was observed while the air was leaving the cooling unit. This was a characteristic feature of the controller and configuration of this particular cool store room. Fig. 5 depicts the time-temperature history at the six locations inside the cool store room. The measurements from the two cool store rooms were comparable, indicating the repeatability of the experiment. Temperatures vary both spatially and temporarily inside the cool store. TC1, TC2 and TC3, which were directly downstream of the cooling unit, were under high temperature fluctuation. TC4, which was at the centre, has a more stable temperature profile. TC5 and TC6, being at the far end of the returning air, receives warmer air. The model properly captured the temperature-time dynamics; but overestimated values by up to 0.8°C . The error was highest at TC3 (see Fig. 4 (c)). The observed overestimation may be caused by estimation errors of the actual thermal mass of the stack that was unknown. At TC3, the pronounced error was probably due to poor calibration of this particular

thermocouple. As a result, the model generally provided a worst case scenario of the actual temperature fluctuations.

The maximum, the average and the minimum simulated produce time-temperature profiles are given in Fig. 6 starting from a uniform temperature of 0 °C in the room. Correspondingly, Fig. 7 show the spatiotemporal distribution of produce temperature. The time-averaged produce temperature was 0.94 °C. The midnight temperature of fruits in the top bins was higher than 1.5 °C and early in the morning the same fruits were subjected to temperatures below 0.5 °C. The effect of such temperature swing on the quality of the commodity is to be investigated.

The fans were continuously operating in the aforementioned day/night cycling of the set-point temperature. However, air circulation fans can contribute considerably to the heat load inside the cool store. After sufficient cooling, the air circulation fans can be operated in cycles to possibly reduce the fan heat load inside the cool store room. Such schemes are analysed below.

4.2.2 On-off cycling of the cooler and fan

In these scenarios, “off” implies the cooler and the air circulation fan were turned off and “on” implies the cooling was undertaking at a simulated constant cooling air temperature of 0.5 °C and the fan was operating at an air exchange rate of 75 h⁻¹. With respect to observations of temperature at the outlet of the evaporator such as in Fig. 4 it will be probably difficult to achieve constant blower temperatures in practice, so the simulations below should be considered as idealized cases. The effect of the three different cycling schemes on the simulated volume average, minimum and maximum produce temperature is shown in Fig. 8. For the 12 h off / 12 h on (Figure 8a), the average produce temperature ranged between 0.51 and 1.22 °C and over time the average was 0.88 °C. For the 14 h off / 10 h on (Figure 8b), the time averaged produce temperature was 0.94 °C with minimum and maximum equal to 0.51 and 1.32 °C, respectively. For scenario three, 16 h off / 8 h on (Figure 8c), the time averaged produce temperature was 1.01 °C and fluctuating between 0.52 and 1.50 °C. Thus show that cooling system can cycle up to 16 h off / 8 h on with a safe produce temperature fluctuation.

The simulated results also suggested that the applied regimes result in temperatures slightly higher than the starting cold store room temperature (Fig. 8). This effect is pronounced at 16 h off / 8 h on scenario. This may suggest unsustainability of the temperature set-point as over long periods of time (months of storage) temperature would continue to rise beyond the optimal point. Hence, on-off cycling of the cooler of the kind discussed in this simulation results should be thoroughly evaluated before implementation.

4.2.3 *Natural convection*

This assesses the temperature-time profile inside the cool store under cooling-off and fan-off condition. A fully coupled, buoyancy driven air flow and heat transfer model was used to realistically capture the spatiotemporal temperature distribution. Fig. 9 shows the instantaneous natural convection air flow patterns inside the cool room at 16 h after fan-off and cooling off. The air flow inside the cool store was because of air density differences occurring due to temperature gradients (natural convection). Heat conducted through the side walls from the outside environment creates higher temperature air in the regions between the stack and the side walls. Hence, air rises in these regions (Fig. 9 (a)), and to replace it, air descends down through the stack (Fig. 9 (b) and (c)) creating a distinctive flow path in the room as summarized in Fig. 9(d). In average the air velocity in the free air (outside the porous domain) was 0.1 m s^{-1} and in the porous domain (in the stack) the average superficial air velocity was 0.01 m s^{-1} .

Fig.10 shows contours of produce temperatures in time. As time goes temperature in the room increases from the initial 0.5°C . After about 16 h, portion of the produce in the cool store start experiencing temperatures above 1.5°C . The proportion of fruit at a certain temperature is shown by temperature frequency curves in time (see Fig. 11). Unambiguously, the top bins were prone to experience higher temperatures. Waelti (1983) also suggested this region to be critical for the same reason.

4.3 Refrigeration load

The components and total refrigeration loads of the different scenarios are discussed in this section by calculating the loads for the whole cold store room.

Under cooling a constant blow temperature of 1 °C (Fig. 12(a), full curve), the simulated room is at steady state. As a consequence, the total respiration heat was constant. We obtained a value of 147 W in the filled CA room. Under the 1.2 °C / 0.6 °C day/night cycling of the temperature set-point (Fig. 12(a), dot curve) the respiration heat fluctuated between 140 W and 155 W and for the 16 h off / 8 h on cycling of the cooling process (Fig. 12(a), dash curve) the respiration heat fluctuated between 140 W and 145 W. The on-off scenario is thus less severe for fluctuations in product conditions as the blower temperature (0.5 °C or not blowing) is less extreme than in the scenario with 1.2 °C / 0.6 °C day/night cycling, where blower temperatures fluctuated between 0 °C and values as high as 1.7 °C (Fig. 4).

Fig. 12(b) shows the heat loads due to conduction through room boundaries. Under the constant cooling regime this load was a constant of 675 W. Under the 1.2 °C / 0.6 °C day/night cycling of the temperature set-point, the heat load due to conduction from the outside varied between 625 W and 740 W. For the 16 h off / 8 h on cooling cycle, this load varied between 690 W and 710 W. Notice that the heat load due to conduction is five to six times larger than the load due to produce respiration.

At motor drive efficiency of 80 %, and for a cooling unit operating at an air exchange rate of 75 h⁻¹ the heat gain from the electric motor can reach up to 4 kW, making the fan heat the major load during CA storage (Fig. 12(c)).

The calculated cumulated refrigeration loads are summarized in Fig. 12(d). The cumulated refrigeration load under day/night cycling of the temperature set-point is equal to the load under constant cooling. This indicates that the anticipated electricity cost saving through day/night cycling of the temperature set-point between 1.2 °C and 0.6 °C is trivial. On the other hand, the on/off cycling of the air circulation fans clearly shows benefits in terms of reducing the heat load. There is a significant

reduction of the heat load by reducing the working hours of the air circulation fan. For instance, under the 16 h off / 8 h on cycle scenario, refrigeration load can be reduced by 55 % compared to constant cooling at 1 °C. Also, fans would operate only 33 % of the time, leading to an extra saving of fan motor power.

The result of our study goes in accordance to what other researchers demonstrated experimentally. Koca and Hellickson (1993) showed discontinuous operation of the air circulating fans offering energy savings in excess of 50 % whilst resulting in only 0.1 °C temperature oscillations during storage. East et al. (2013) similarly showed that temperatures during a 9 h on and 15 h off cycle remained within a range of ± 1 °C with an approximate 40 % of electricity cost savings. It should be noted that such small fluctuations in produce temperature during apple storage has minimal impact in quality (Brown, 2011; East et al., 2013; Gwanpua et al., 2013; 2014). Consequently, there is enough window for energy reduction in cold storage of apples, without negatively impacting on quality. Our study provides some approaches to do so.

5 Conclusions

This paper analysed energy saving options during cold storage of apple fruit. A two-phase porous medium CFD model was developed. Data taken from commercial CA store rooms were used for model validation. The model properly captured the spatiotemporal temperature distribution with slightly overestimated values. As a result the model thus provided the worst case scenario.

The study demonstrated that the air circulation fans are the major source of heat load. Hence, an attempt to reduce energy cost should deliberate on reducing fan operation time. It has been demonstrated that fan and refrigeration cycling works, save energy while the temperature of the apple fluctuates within limited range. While it has been shown that small fluctuations in temperatures have minimal impact on quality, the impact of temperature perturbations on several quality indicators should be thoroughly evaluated before implementing such changes.

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7 References

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Figures captions

Fig. 1: Schematic diagrams of the cold store room. (a) wooden bin loaded with randomly packed apple fruit (≈ 380 kg), (b) the porous medium illustration of the fruit loaded bin, and (c) the cool store (BelOrta, Borgloon, Belgium) loaded with 240 bins each holding ≈ 380 kg apple fruit.

Fig. 2: (a) Schematics of half of the cool store room used as the model geometry based on symmetry principle, (b) location of the six thermocouples inside the cool store. (1) thermocouples, (2) cooling unit outlet, (3) cooling unit inlets, (4) symmetry plane, (5) apple loaded bins constituting the porous medium, C1 and C2 are column indexes and L1 to L8 are layer indexes. The schematics corresponds to half of the cool store (BelOrta, Borgloon, Belgium) loaded with 240 bins each holding ≈ 380 kg apple fruit.

Fig. 3: Velocity profiles inside the cool room. (a) velocity stream line showing the air flow path inside the cool room, (b) surface stream line on vertical plane bisecting the 1st column of stack from the left wall, the numerical values show the volume average superficial air velocity per bin, (c) surface stream line on horizontal plane bisecting the fourth layer of stacks from the floor. The

simulation corresponds to an air exchange rate of 75 h^{-1} . The color bars indicate local velocity magnitudes and arrows indicate velocity directions.

Fig. 4: The temperature set point of the cooling unit (dashed curve) and the measured cooling air temperature (full curve) inside the cool store room under the $1.2 \text{ }^{\circ}\text{C} / 0.6 \text{ }^{\circ}\text{C}$ day/night cycling of the temperature set-point. The temperature of the cold air leaving the cooling unit was measured by a thermocouple placed 10 mm in front (downstream) of the cooling unit.

Fig. 5: Temperature measurements in the two cool store rooms (full curves) and simulated (dash curves) temperature histories from six spots inside a cold storage room. Figure (a) to (f) correspond, respectively, to thermocouples TC1 to TC6 shown in Figure 2. Due to malfunctioning of TC5 and TC6 in Room 2, only measurements from Room 1 were shown in (e) and (f). The two cool stores were loaded each with a total of 240 wooden bins each holding $380 \pm 20 \text{ kg}$ Jonagold apple fruit and were kept under controlled atmosphere (CA) conditions ($[\text{O}_2] = 1\%$ and $[\text{CO}_2] = 2.5\%$). Simulations were performed with continuous forced air flow.

Fig. 6: Simulated minimum, average and maximum produce temperature histories under the $1.2 \text{ }^{\circ}\text{C} / 0.6 \text{ }^{\circ}\text{C}$ day/night cycling of the temperature set-point inside the cool store (BelOrta, Borgloon, Belgium) loaded with 240 bins each holding $\approx 380 \text{ kg}$ Jonagold apple fruit and kept under controlled atmosphere (CA) conditions ($[\text{O}_2] = \text{ kPa}$ and $[\text{CO}_2] = 2. \text{ kPa}$). Continuous forced air flow was applied.

Fig. 7: Contours of produce temperature in a typical day inside the cool store room under the $1.2 \text{ }^{\circ}\text{C} / 0.6 \text{ }^{\circ}\text{C}$ day/night cycling of the temperature set-point inside the cool store (BelOrta, Borgloon, Belgium) loaded with 240 bins each holding $\approx 380 \text{ kg}$ Jonagold apple fruit and kept under controlled atmosphere (CA) conditions ($[\text{O}_2] = 1\%$ and $[\text{CO}_2] = 2.5\%$). Continuous forced air flow was applied.

Fig. 8: Time profile of the minimum, average and maximum produce temperature inside the cool store under various fan-cycling schemes. 12 h off / 12 h on (a), 14 h off / 10 h on (b) and 16 h off / 8 h on (c). The cool store (BelOrta, Borgloon, Belgium) loaded with 240 bins each holding ≈ 380 kg Jonagold apple fruit and kept under controlled atmosphere (CA) conditions ($[O_2] = 1\%$ and $[CO_2] = 2.5\%$).

Fig. 9: Simulated instantaneous natural convection air flow pattern inside the cool room at 16 h after fan off and cooling off. (a) surface stream line on vertical plane (Y-Z plane) bisecting the region between the 1st column of stack and the left wall, (b) surface stream line on vertical plane bisecting the 1st column of stack (c), surface stream line on symmetry plane and (d) surface stream line on vertical plane (Y-X plane) bisecting the cool store. The color bars indicate local velocity magnitudes and arrows indicate velocity directions.

Fig. 10: Simulated produce temperature contours in time after fan off and cooling off inside the cool store (BelOrta, Borgloon, Belgium) loaded with 240 bins each holding ≈ 380 kg Jonagold apple fruit and kept under controlled atmosphere (CA) conditions ($[O_2] = 1\%$ and $[CO_2] = 2.5\%$).

Fig. 11: Simulated frequency curves of produce temperatures in time after fan off and cooling off inside the cool store (BelOrta, Borgloon, Belgium) loaded with 240 bins each holding ≈ 380 kg Jonagold apple fruit and kept under controlled atmosphere (CA) conditions ($[O_2] = 1\%$ and $[CO_2] = 2.5\%$).

Fig. 12: Components of the refrigeration loads of cool room with a constant set point temperature (full curve) and with cyclic set point temperature (dot curve) and with on-off cycling (16 h off / 8 h on) of the cooling unit (dash curve). Heat loads: due to respiration of the produce (a), due to conduction through boundaries (b), fan heat (c) and cumulated refrigeration load (d). The cool store was loaded with a total of 240 wooden bins each holding 380 ± 20 kg Jonagold apple fruit and were kept under controlled atmosphere (CA) conditions ($[O_2] = 1\%$ and $[CO_2] =$

2.5%). The dot curve corresponding to the cyclic temperature set point case is not visible in (c) and (d) because it overlaps with the full curve of the constant set point temperature case.

Figures

Fig. 1:

Fig. 2:

Fig. 3:

Fig. 4:

Fig. 5:

Fig. 6:

Fig. 7:

Fig. 8:

Fig. 9:

Fig. 10:

Fig. 11:

Fig. 12:

Tables

Table 1: Viscous and inertial coefficients of the air flow resistance model of the porous medium, obtained from detailed CFD model (Ambaw et al., 2013b).

Air flow direction	viscous loss coefficient [kg m ⁻³ s ⁻¹]	Inertial loss coefficient [kg m ⁻⁴]
X or Z	68	276
Y	2	269

Table 2: Model parameters and their values

Parameters	Value	Source
Bulk porosity (r_a)	0.43	Estimated ^(a)
Density of air (ρ_a)	1.28 kg/m ³	Keenan et al. (1984)
Density of apple (ρ_p)	870 kg/m ³	Measured on samples
Heat capacity of air (c_{pa})	1005 J/kg.K	Keenan et al. (1984)
Heat capacity of apple (c_{ps})	3640 J/kg.K	Becker et al. (1996)
Mean fruit diameter (d_p)	0.075 m	Measured samples ^(b)
Specific surface area (A_{spec})	47.0 m ² /m ³	Nield and Bejan (2013) ^(c)
Thermal conductivity of apple (k_p)	0.50 W/mK	Becker et al. (1996)
Viscosity of air (μ)	1.78×10 ⁻⁵ kg/ms	Keenan et al. (1984)

^(a) Estimated from the volume of a stack (1.12 m × 1.12 m × 0.8 m), the volume of apples in a stack (a total of 2000 apple fruit with diameter of 0.075m are assumed in a bin) and the volume of wood of the bin (≈ 0.13 m³).

^(b) Calculated from measured mass and volume of sample apple fruits. The mean $\pm$ standard deviation (n=10) of the mass and volume were 192g $\pm$ 13g and 221 mL $\pm$ 24 mL, respectively.

^(c) Calculated using: $A_{spec} = 6(1 - r_a)/d_p$.

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