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Modelling the Environment Within a Wet Air-cooled Vegetable Store

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ABSTRACT

This paper presents a model for a wet air-cooled store based on computational fluid dynamics. Unlike previous works on cold room modelling published in the literature, the model considers the effects of buoyancy by coupling the fundamental energy and diffusion equations with the momentum equation. It has been found that the consideration of buoyancy is important in regions of low air velocity. The model which can be applied to other types of cold store has been validated using experimental results from a wet air-cooled store. A good agreement was found between the experimental and simulation results. The modelling results identify a number of design problem areas and illustrate the usefulness of the model in investigating the effects of design parameters on the performance of the cold store. © 1998 Elsevier Science Limited. All rights reserved.

NOMENCLATURE

A	Surface area (m^2)
A_o	Surface area per unit volume (m^{-1})
c	Specific heat at constant pressure (J $kg^{-}1$ $k^{-}1$)
C	Inertial resistance ($m^{-}1$)
d_p	Produce diameter (m)
D	Diffusion coefficient (m^2 s^{-1})
D_e	Equivalent diameter (m)
g	Gravitational acceleration (m s^{-2})

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h	Specific enthalpy (J kg^{-1})
J	Diffusion flux of each species (kg m^2 s^{-1})
K	Turbulent kinetic energy (J kg^{-1})
m	Mass fraction
p	Pressure (ρ_a)
S	Source term
t	Time (s)
T	Temperature (K)
u	Velocity (m s^{-1})
V	Time average velocity (m s^{-1})
x,y,z	Coordinate (m)

Greek symbols

α	Permeability of the material (m^2)
Γ	Diffusion coefficient (N s m^{-2})
ε	Dissipation rate (J kg^{-1} s^{-1})
μ	Dynamic viscosity (N s m^{-2})
ρ	Density (kg m^{-3})
σ_k	Constant in the turbulent model
σ_ε	Constant in the turbulent model
ϕ	Void fraction

Superscripts

$\rightarrow$	Vector

Subscript

b	Bulk
eff	Effective
f	Fluid
h	Heat
i,j	Coordinate number
m	Mass
s	Solid
t	Turbulence
w	Wall

INTRODUCTION

Worldwide competition in the food industry is forcing manufacturers and retailers to seek improvements in food processing and storage methods to improve quality and length of storage. For many vegetables, short-term storage may be required to accommodate fluctuations in supply and demand, both of which may be affected by weather. Demand is also subject to large regular fluctuations on a weekly cycle, with almost 75% of all produce sales taking place between Thursday and Saturday. Fluctuations in weather, such as very hot or dry periods that may accelerate growth, or cold and wet periods that may hinder harvesting, may necessitate holding buffer stocks for longer periods of time. Longer period of storage may also be required if vegetables are to be exported to European or other markets further away.

The relative humidity and temperature of the air are important parameters in cold store design and operation. Relative humidity is the ratio of the partial pressure of water vapour in the air to the partial pressure at saturation at the same temperature. Since it is the water vapour pressure difference between the store air and the surface of the produce that drives the transpiration process, to minimise weight loss it is important to maintain the store air at as high a value of relative humidity as possible, typically 90–97% RH (Dennis, 1984). Excessive RH levels in certain cases, however, may encourage microbial spoilage and it is thus necessary to ensure that the humidity in the store remains within acceptable limits (Geeson, 1989). The humidity in the store also exerts an influence on the performance of the cooling coils. The higher the humidity for a given temperature, the greater will be the rate of frost formation on the cooling coils and the higher will be the energy losses caused by frosting and defrosting of the coils.

Although certain bacteria and fungi are capable of causing decay of stored fruits and vegetables even at low temperatures, as low as 0°C, the spoilage losses are considerably retarded and some post-harvest diseases can be completely eliminated by using low storage temperatures. Low temperatures and high humidities can be achieved in a cold store by using either forced air cooling with conventional coils or, forced air cooling with wet cooling systems (ice banks). The latter system provides much higher cooling rates than the conventional system and is thus widely used for rapid cooling and short-term holding of fruits, vegetables and salads. Research at the Institute of Food Research in the UK has also shown that these systems have the ability to maintain low temperatures and high relative humidities with lower running costs than conventional systems, making them suitable for long- and medium-term storage of a number of vegetable crops (Farrimond *et al.*, 1979).

The maintenance of low temperatures and high relative humidities in cold stores requires accurate control and uniform distribution of air flow. In positively ventilated cold stores, the spacial distribution of both temperature and relative humidity is a function of a number of interrelated variables which include the stacking arrangement, air circulation arrangement, and air flow conditions. It is difficult to determine the combined effects of all these variables experimentally. Traditionally, engineers considered the effects of certain variables, such as temperature and velocity variations, in isolation.

Theoretical research into cold stores began in the late 1960s. Meffert *et al.* (1971) analysed a cold store and derived correlations for the temperature distribution inside product stacks, by considering the store to be a non-ventilated room with heat transfer taking place by conduction only. van der Ree *et al.* (1974) built a finite element model to predict the temperature distribution in refrigerated cargoes. They assumed that the cold store space was almost completely occupied by produce. The only empty spaces were the crevices between the boxes and passages at the top and bottom of the boxes to allow for air circulation. Heat transfer was considered to be non-steady and spatially distributed. The model did not account for momentum and vapour transport. Wang *et al.* (Wang & Touber, 1988, 1990; Wang & Visser, 1991) used a commercial computational fluid dynamics (CFD) package to model a refrigerated store. The effect of buoyancy was neglected, and the energy and diffusion equations were decoupled from the momentum equations. The produce was modelled as porous media using a two step approach. First, the boxes were assumed to be impenetrable blocks to determine the macro velocity and pressure distributions around them and then these parameters were used to calculate the micro velocity through the product.

In this paper a modelling methodology for cold stores is presented, based on CFD. Unlike previous cold store models reported in the literature, the analysis considers buoyancy effects and the coupling of the energy and diffusion equations with the momentum equation. The methodology was tested against experimental results from a wet air-cooled store. Good agreement was found between the experimental and modelling results.

THE WET AIR-COOLED (ICE BANK) STORE

In wet air cooling systems, a small refrigeration plant is used to make a 'bank' of ice on extended surface plates which are suspended in a tank of water. The water is prevented from freezing completely through mechanical agitation which also maintains good heat transfer rates between the refrigerated plates and the water. The ice is usually formed during the night, employing off-peak tariff electricity. The chilled water from the tank which is at 0°C, or very close to 0°C, is pumped to a cooling tower which is filled with packing to increase the contact area between the chilled water and air drawn from the cold store in the opposite flow direction to that of the water. The water chills the air to approximately 0·5°C and 97–98% RH (Geeson, 1989). This air is distributed to the store either directly or more commonly through a plenum chamber which has controllable louvres. The flow area through the opened louvres coincides with the space between the stacked bins as shown in Fig. 1, allowing air to flow freely through the louvres and up through the storage bins. The bins employed have perforated or slated bases and solid sides to allow the flow of air up through them. The warm air is then recirculated by a fan from the top of the store through the cooling tower.

MODELLING METHODOLOGY

Modelling of buoyancy effects

Although the tendency in past investigations has been to neglect the effects of buoyancy due to the relatively high off-coil air velocities in the store, in certain cases such as the downwind side of stacked boxes or between rows of boxes, the air velocity can be so small that heat transfer by natural convection becomes much more significant than heat transfer by forced convection.

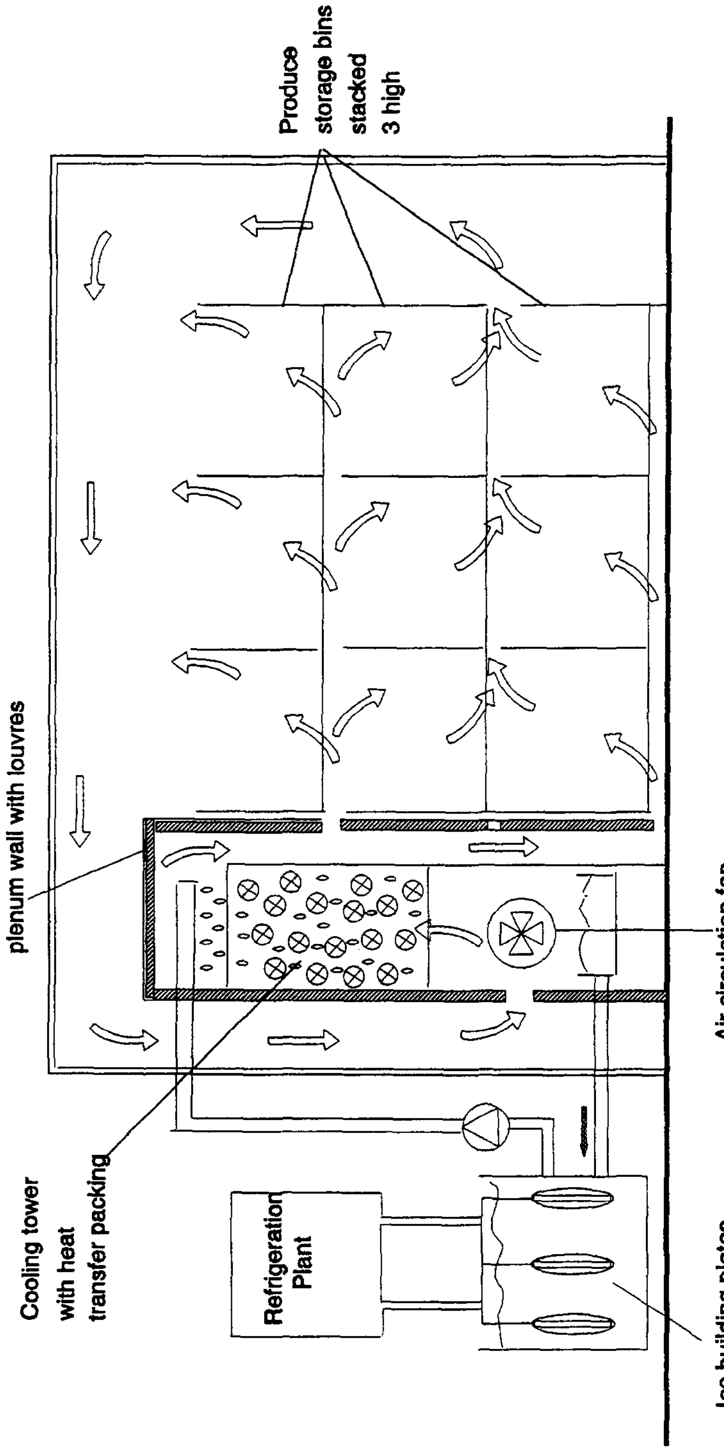


Fig. 1. Schematic diagram of a wet air-cooled store showing air circulation pattern.

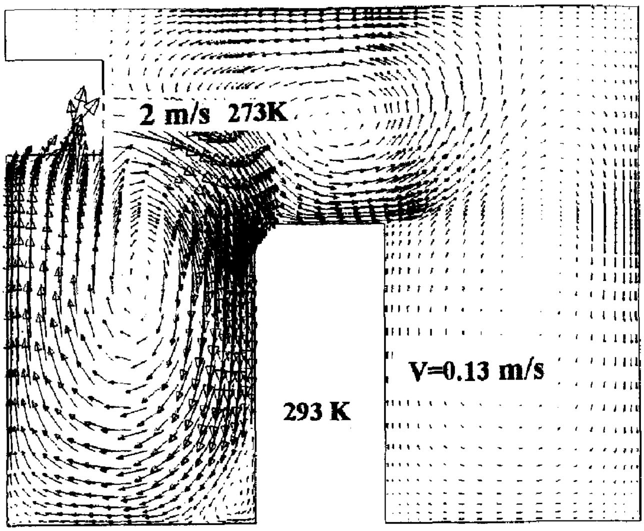
To demonstrate the effects of buoyancy, calculations were carried out for different air circulation rates through a model cold store containing a single stack of boxes as illustrated in Fig. 2. Figure 2a shows the resulting velocity vectors for off-coil air temperature of 273 K, product temperature of 293 K and off-coil velocity of $2{\cdot}0\ \mathrm{ms}^{-1}$ when the effect of buoancy was neglected. Figure 2b shows the velocity vectors for the same conditions with buoancy taken into consideration in the calculations. Comparing the two figures, it can be seen that the upward velocity on the downwind side of the box is much higher when buoncy is included, giving a higher overall heat transfer coefficient compared to the case when the effect of buoancy was neglected. The overall heat transfer coefficient was taken as the mean between the average heat transfer coefficients on the upwind and downwind sides of the box with respect to the direction of air flow from the cooling coil. The resulting overall heat transfer coefficients between the box and the air for different air circulation rates (off-coil air velocities) and different temperature differences between the air and the surface of the box are plotted in Fig. 3.

It can be seen that neglecting the effect of buoyancy underestimates considerably the overall heat transfer coefficient due to underestimation of the coefficients in areas of low air velocity, such as the downwind side of the box. When the temperature difference between the surface of the box and the air is small, the effect of buoyancy is to disturb the boundary layer at the surface, leading to an increase in the heat transfer coefficient. At higher temperature differences, however, natural convection becomes so strong that it counteracts the effect of forced convection on the upwind side of the box, causing a reduction in the relative effect of buoancy on the overall heat transfer coefficient. It can be seen from Fig. 3 that increasing the temperature difference from 5 K to 20 K causes only a small increase in the heat transfer coefficient. Figure 3 also indicates that the effect of buoancy is more pronounced at low air circulation rates. As circulation rate is increased, forced convection becomes the dominant factor and the effect of buoancy is reduced.

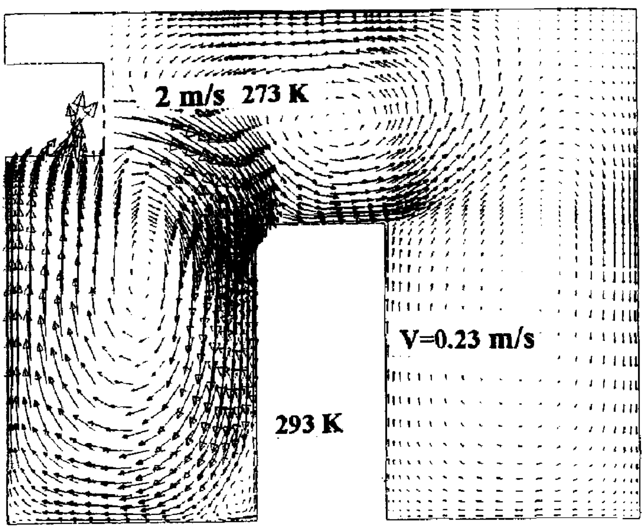
In order to incorporate the effect of buoyancy in the calculations it is necessary to couple the energy and diffusion equations with the momentum equation. The solution of the coupled fundamental equations takes more computing time than the solution of the uncoupled equations but, with modern computer technology, it is possible to solve the fully coupled equations for a 3-D time dependent refrigerated store model on a PC within a few hours.

Modelling of produce

An added difficulty in the modelling of cold stores is the representation of the vegetables and fruits which are usually stored in boxes or bins. Most of the boxes have openings or holes to let the air through the produce. If the void fraction and the average diameter of the produce can be defined, then the produce can be modelled as porous media (Becker *et al.*, 1996a,b). This approach was followed in this study. For simplicity, the temperature variation within individual items of produce was not considered, but instead, it was assumed that each item was at a uniform temperature which varied across the store depending on flow and heat transfer conditions. The modelling approach followed, which was implemented within a commercially available CFD package, is illustrated below.



a. Without buoyancy



b. With buoyancy

Fig. 2. Velocity distribution in the cold store with and without buoyancy; V is the upward average velocity on the downwind side of the box.

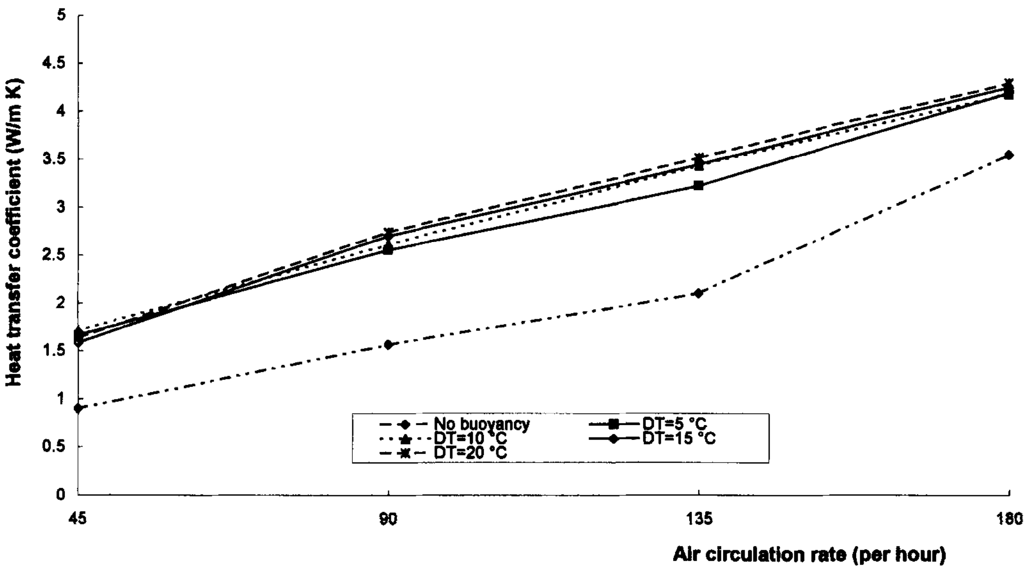


Fig. 3. Effect of buoyancy on the overall heat transfer coefficient in cold store.

Fluid flow and pressure drop through a packed fruit or vegetable bed

When solving the flow through a packed vegetable and fruit bed, the general momentum equation (Hinze, 1975) can be augmented by the introduction of a momentum sink:

$$\frac{\partial}{\partial t}(\rho\mu_i)+\frac{\partial}{\partial x_j}(\rho\mu_i\mu_j)=\frac{\partial}{\partial x_j}\mu\left(\frac{\partial\mu_i}{\partial x_j}+\frac{\partial\mu_j}{\partial x_i}\right)-\frac{\partial\rho}{\partial x_i}+\rho g_i+\frac{\partial}{\partial x_j}(\rho\overline{\mu_i\mu_j})+S_{\text{bed}} \quad (1)$$

The sink is a function of Reynolds number and porosity of the bed. In a packed bed of unit volume, the volume occupied by the void and the solid particles can be denoted as ϕ and $(1-\phi)$ respectively. ϕ is the void fraction or porosity of the bed and can be calculated from (Dullien, 1979):

$$\phi=1-\frac{\rho_b}{\rho_s} \quad (2)$$

where ρ_b is the bulk density of the sample, and ρ_s is the density of the solid in the sample. The total surface area in a bed of unit volume is given by:

$$A=(1-\phi)A_o \quad (3)$$

where A_o is the surface area per unit volume of the solid material.

The Reynolds number of the fluid flow through a packed bed is given by (Holland, 1973):

$$Re = \frac{\rho \mu D_e}{6\mu(1-\phi)} \tag{4}$$

where the equivalent diameter D_e can be defined as:

$$D_e = \frac{4\phi}{(1-\phi)A_o} \tag{5}$$

For laminar flow where $Re \leq 2000$ the momentum sink is given by (Ergun, 1952):

$$S_{\text{bed}} = \frac{\mu}{\alpha} V \tag{6}$$

and for turbulent flow

$$S_{\text{bed}} = \frac{\mu}{\alpha} V + C_2\left(\frac{1}{2} \rho V|V|\right) \tag{7}$$

where α is the permeability of the porous media calculated from:

$$\alpha = \frac{d^2}{150} \frac{\phi^3}{(1-\phi)^2} \tag{8}$$

and C_2 is the inertial loss coefficient calculated from:

$$C_2 = \frac{3.5}{d} \frac{(1-\phi)}{\phi^3} \tag{9}$$

Heat and mass transfer in a packed bed

Heat transfer in the fruit or vegetable regions of a cold store can be determined by modifying the standard enthalpy transport equation (Hinze, 1975) to include a conduction flux and a transient term.

$$\frac{\partial}{\partial t}[TR] + \frac{\partial}{\partial x_i}(\rho_f h_f \mu_f) = \frac{\partial}{\partial x_i}(k_{\text{eff}})\frac{\partial T}{\partial x_i} - \frac{\partial}{\partial x_i}\sum_{i=1}^{i} h'_j J_i + S_h \tag{10}$$

The conduction flux uses an effective conductivity, k_{eff}, and the transient term, *TR*, includes the thermal inertia of the solid region of the media.

The transient term is given by:

$$TR = (\phi \rho_f h_f + (1-\phi)\rho_s h_s) \tag{11}$$

and the effective conductivity by:

$$k_{\text{eff}} = \phi k_f + (1-\phi)k_s \tag{12}$$

where k_s is the thermal conductivity of the produce. It can be calculated from (Becker *et al.*, 1996a):

$$k_s = 0{\cdot}148 + 0{\cdot}493\left(\frac{W_{H_2O}}{100}\right) \tag{13}$$

where W_{H_2O} is the water content in percentage terms. The water content ranges from 80 to 90%.

Another feature of fruit and vegetable heat transfer is that they generate heat during respiration. The generated heat becomes a source term in the enthalpy equation. It can be calculated from (Becker *et al.*, 1996a):

$$S_h = 10{\cdot}7 m_{CO_2} \tag{14}$$

where m_{CO_2} is the carbon dioxide production per unit mass of commodity ($mg\,kg^{-1}\,h$) given by:

$$m_{CO_2} = f\left(\frac{9T_m}{5} + 32\right)^g \tag{15}$$

where T_m is the mass average temperature and f and g are the coefficients of respiration.

Mass transfer in porous media can be dealt with in a similar way to heat transfer. The general mass transfer equation can be modified to include a transient and a diffusion term with all other terms remaining unchanged (Hinze, 1975).

$$\frac{\partial}{\partial t}(Tr_i) + \frac{\partial}{\partial x}(\rho \mu_i m_i) = -\frac{\partial}{\partial x_i} J_{i',j} + S_v \tag{16}$$

The mass species in the bed are air, vapour, and solid. The transient term in eqn (16) can be calculated from:

$$TR_i = \phi m_{vapour} \tag{17}$$

The diffusion term in eqn (16) can be calculated from: (1) for laminar flow:

$$J_{i,i'} = -\rho D_{i',m} \frac{\partial m_{i'}}{\partial x_i} \tag{18}$$

where $D_{i',m}$ is the diffusion coefficient for species i' in the mixture and is a function of temperature and pressure; and (2) for turbulent flow:

$$J_{i,i'} = -\left(\rho D_{i',m} + \frac{\mu_i}{Sc_t}\right)\frac{\partial m_t}{\partial x_i} \tag{19}$$

where Sc_t is the effective Schmitt number and μ_t is the effective viscosity. To model the transpiration from fruits and vegetables an extra term, S_v, is added to the diffusion equation. This is given by (Becker *et al.*, 1996a):

$$S_v = E_o(\rho_s - \rho_a) \tag{20}$$

where E_o is the mass transfer (transpiration) coefficient based on vapour pressure and can be calculated from:

$$E_o = \frac{1}{\dfrac{1}{E_a} + \dfrac{1}{E_s}} \tag{21}$$

where E_a is the air film mass transfer coefficient which describes the convective mass transfer which occurs at the surface of the commodity and is a function of air flow rate. E_s is the skin mass transfer coefficient which describes the skin's diffusional resistance to moisture migration.

Model validation

The model was validated using cooling down curves for beetroot obtained in an experimental wet air-cooled store (Geeson, 1989). The geometric characteristics and design parameters of the store as well as the thermophysical properties of beetroot are given in Table 1.

The store had a maximum capacity of 15 tons, and held bulk bins in three rows, as shown in Fig. 1. Each bin had a holding capacity of 0·8 m^3. The store was served by a vapour compression refrigeration system and eight ice bank plates positioned in water channels below floor level.

TABLE 1
Model Inputs

Air flow rate	0·11 $m^3\ s^{-1}$ per 1000 kg of beetroot
Ambient temperature	293·15 K
Supply air temperature	273·65 K
Bin depth	0·7 m
The height of the passage of the refrigerated air	0·1 m
Bulk density of beetroot in bin[15]	$\rho_{bin} = 700$ kg m^3
Density of the beetroot (Neale & Messer, 1976)	$\rho_{beet} = 1000$ kg m^3
Porosity of the packed bed (beetroot in bins)	$\varepsilon = 0{\cdot}3$
Average diameter of the beetroot	$d = 100$ mm
Permeability of the bed (beetroot in bins)	$\alpha = 3{\cdot}67$E-6
The inertial loss coefficient	$C_2 = 3640$
The specific heat of beetroot	$c = 3770$ J kg^{-1} K
Thermal conductivity of beetroot	$k_b = 0{\cdot}571$ kJ m^{-1} K
Thermal conductivity of store wall	$k_w = 0{\cdot}33$ W m^2 K
Wall thickness (polystyrene with aluzink cladding)	$x = 0{\cdot}1$ m
Heat generation of the beetroot during storage	60 W per 1000 kg

Since the height of the air passage between the rows of bins (see Fig. 1) is only 100 mm and much smaller than the width of the air flow passage (the width of the passage is 1 m), a 2-D model could be justified for the simulation. The grid density used was 62×56 in the i,j directions. The grid was irregular, with a smaller grid size near the air passages. Runs with higher density grids did not produce any appreciable difference in the velocity and temperature distributions. The time step employed was 1 min at the initial stages of simulation which increased to 3 min as simulation progressed towards steady state.

Cooling down characteristics

Figure 4 compares the cooling down curves for the top, middle, and bottom of bins obtained experimentally and from simulation, over a cooling period of 15 h starting with an initial beetroot temperature of 8°C. The temperature at the top of the bins represents the average temperature of the produce at the top surface of all the bins in the store, whereas the temperatures at the middle and bottom of the bins represent the average temperature of the produce at the middle and bottom surfaces of all the bins in the store respectively. Apart from a slight underprediction of the cooling rate during the first 3 h of operation, there is a very good agreement between the experimental and modelling results for the top of the bins. For the bottom of the bins, the model overpredicts the cooling rate during the first 3 h of operation and underpredicts the cooling rate after that. The maximum difference

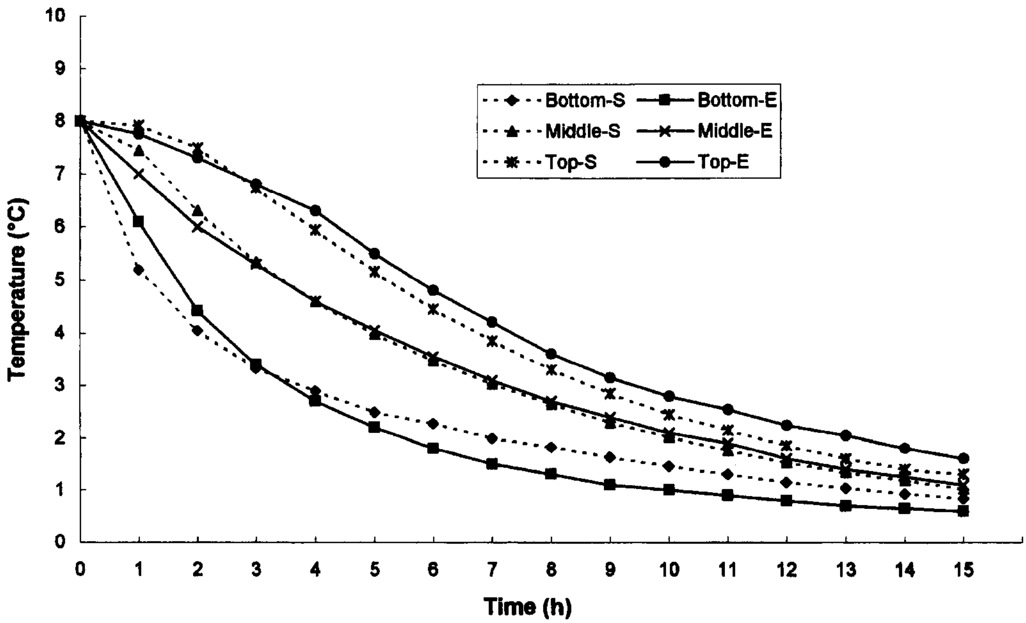


Fig. 4. Cooling curves of beetroot at top, middle and bottom of the bins in the wet air-cooled store (E = experiment, S = simulation).

between the measured and predicted temperature after 15 h of operation, however, is only 0·5°C. The tendency to underpredict the temperature after a long period of operation is also evident at the other two positions in the store but to a much lower extent. There are various reasons for the small discrepancies between the modelling and experimental results. Such reasons include: (1) the actual position of the thermocouples in the experimental tests, which was not provided by the investigators, (Farrimond *et al.*, 1979) which may not coincide exactly with the positions for which the simulation results have been plotted; and (2) possible errors in experimental measurement of temperatures and air flows in each channel. In the model, a constant heat generation rate for the beetroot equivalent to 60 W ton^{-1} was used. In practice, the heat generation rate will reduce slightly with the reduction in temperature but this is not thought to have influenced significantly the accuracy of the results due to the small value of heat generation rates involved.

Flow characteristics in the cold store

Figure 5 shows the resulting velocity vectors in the store. It can be seen that the initial velocities at the top and bottom passages are quite high, 2.8 and 1.4 ms^{-1},

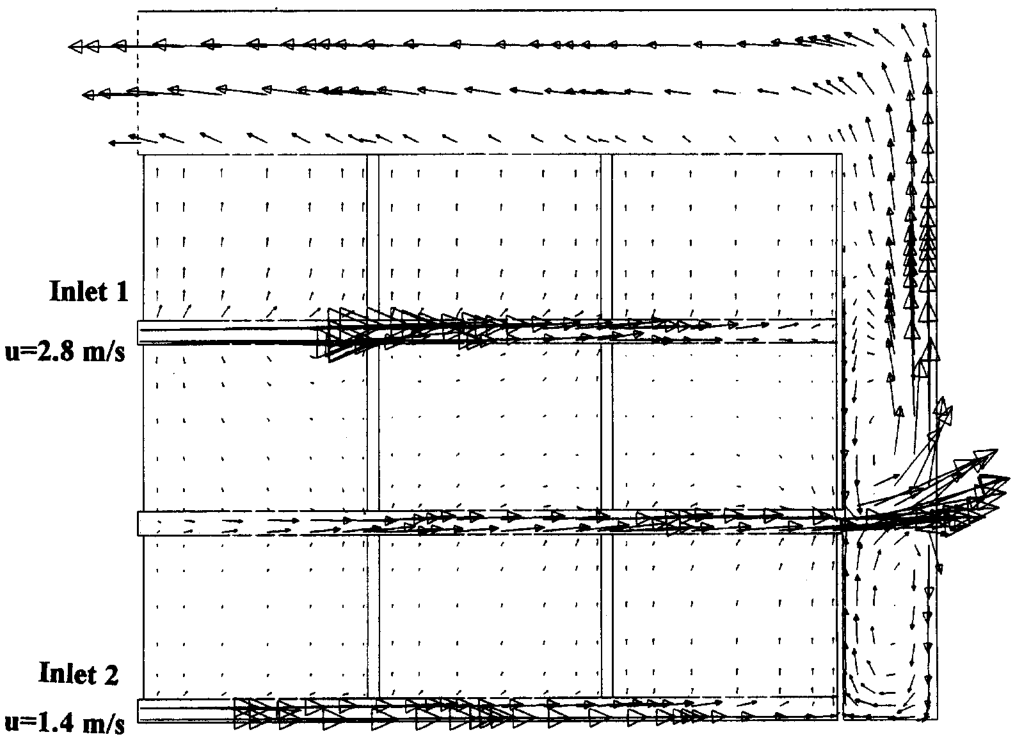


Fig. 5. Velocity distribution in the wet air-cooled store.

respectively. As the air travels along the passages its velocity reduces due to the fact that some of it flows through the produce in the bins. The middle passage acts as the return air passage. It can be seen that in the bottom row of bins the air flows through the produce from bottom to top and the velocity distribution is quite uniform. The middle row utilises the same air supply passage as the top row and the same return passage as the bottom row. This makes the air flow through the middle row complex and it can be seen from Fig. 5 that the flow which should be from top to bottom reverses in some areas.

Since less cold air flows through this row, a worse cooling effect is expected. Moreover, since there is some air that flows reversely and goes back into the air supply passage between the bottom and middle rows, the air temperature in the middle row particularly towards the end of the passage is expected to be higher than the rest of the row. It can also be seen from the velocity vectors that the velocity in the first stack of bins in the top row is higher than the rest of the bins. A greater cooling effect is expecetd in this area and this can be confirmed by looking at the temperature distribution curves in Figs 6–8 which indicate consistently lower temperatures in this part of the store.

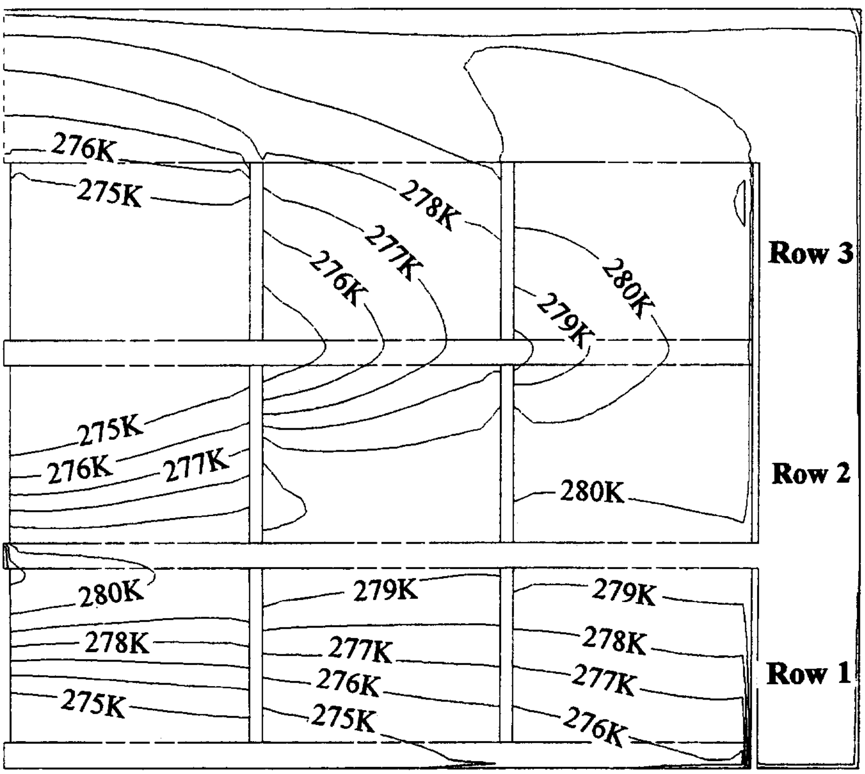


Fig. 6. Average produce temperature distribution in the wet air-cooled store after 5 h operation.

Temperature distribution in the store

Figues 6–8 show the temperature distributions in the store after 5, 10 and 15 h, respectively. It can be seen that cooling progresses slowly with time. Since the air flows through the bottom row passage uniformly and unidirectionally, the temperature distribution in this row is quite uniform.

The cooling does not progress so well in the middle row because less cold air flows from the top passage down through the produce and the air rising through the bottom row has slightly warmed up by the heat released by the produce in the bottom row.

Cooling in the top row progresses faster than cooling in the middle row, because as mentioned earlier, most of the air from the top supply passage tends to rise through the top row rather than fall through the middle row.

After 15 h of operation, the temperature in the bulk of the produce is fairly uniform, between 1°C and 2°C, with a supply air temperature of 0·5°C. However, the temperature of the produce on the right-hand top bin remains fairly high at between 2°C and 4°C. This region is affected by the return air flow from the bottom rows of

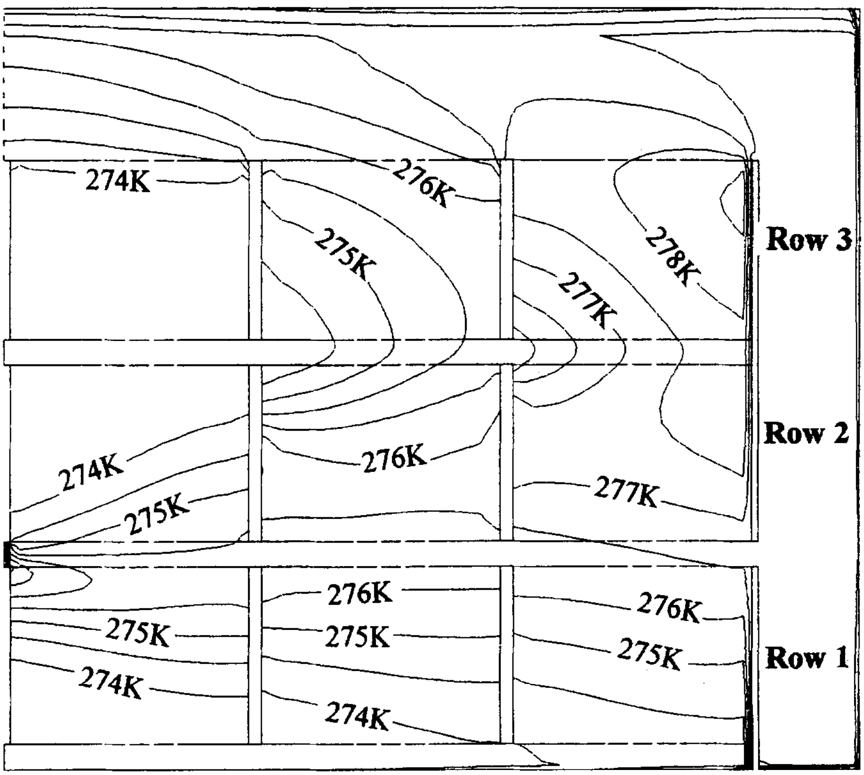


Fig. 7. Average produce temperature distribution in the wet air-cooled store after 10 h operation.

bins and the higher temperatures may lead to faster deterioration of the stored commodity. This problem, once identified, can be addressed by modifying the air passage through the produce. The effect of such modification is illustrated in Fig. 9. Here, the height of the stored produce in the last two bins in the middle row has been reduced, increasing the flow area and hence the static pressure of the air above these two bins. The increased static pressure forces more air to travel upwards through the produce in the last two bins on the top row, reducing the temperature in these bins by about 2°C and improving the uniformity of temperature in the store at no extra cost.

The model once validated can be used to investigate the effect of various design parameters on the performance of the cold store. One such parameter is the air circulation rate through the store. Figure 10 shows the influence of increasing the design flow rate by 20%. It can be seen that increasing the air flow rate will increase the rate of cooling particularly during the initial stages. The higher air flow rate will also result in a much more uniform temperature in the produce with only a very small temperature difference of about 0·2°C between the three rows. Figure 11 illustrates the effect of reducing the air flow rate by 20% below the design flow. It

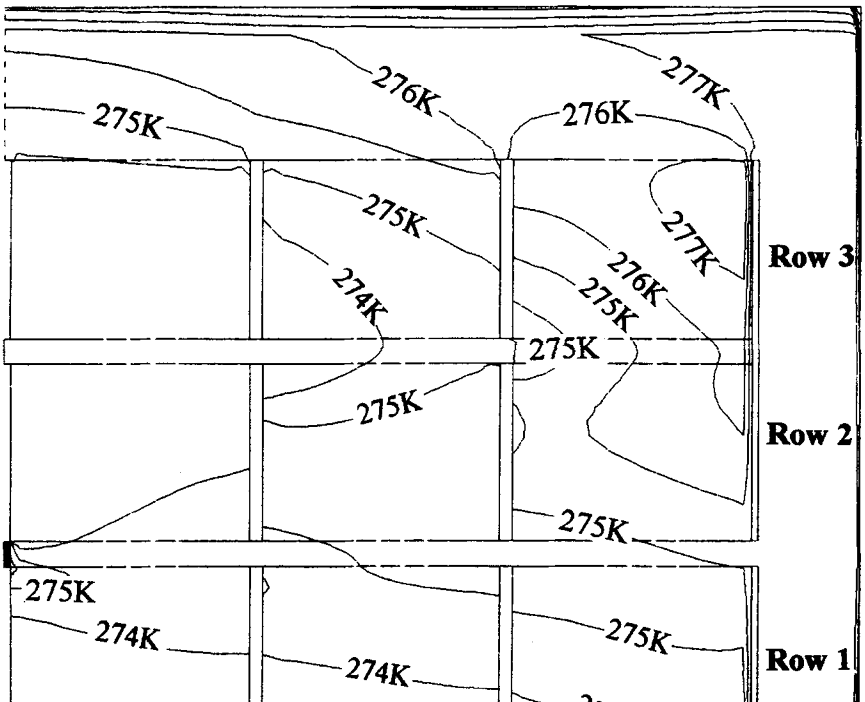


Fig. 8. Average produce temperature distribution in the wet air-cooled store after 15 h operation.

can be seen that at the lower flow rate the cooling rate at the top and middle rows is much slower. After 15 h of operation the average temperature in the three rows is about 1°C higher than in the case of the design flow rate. There is also a wider variation in temperature between the three rows, around 1°C compared to 0·5°C, for the design flow rate.

CONCLUSIONS

The results of this study indicate that wet air-cooled stores can be effectively modelled using computational fluid dynamics. Commercial CFD packages, with certain modifications, have the ability to model the heat and mass transfer processes and provide temperature, velocity and water vapour distributions in the store. Although the modelling results are in good agreement with experimental results published in the literature, the accuracy of the results can be improved further by introducing greater sophistications into the model such as variable heat generation rates from the stored commodity.

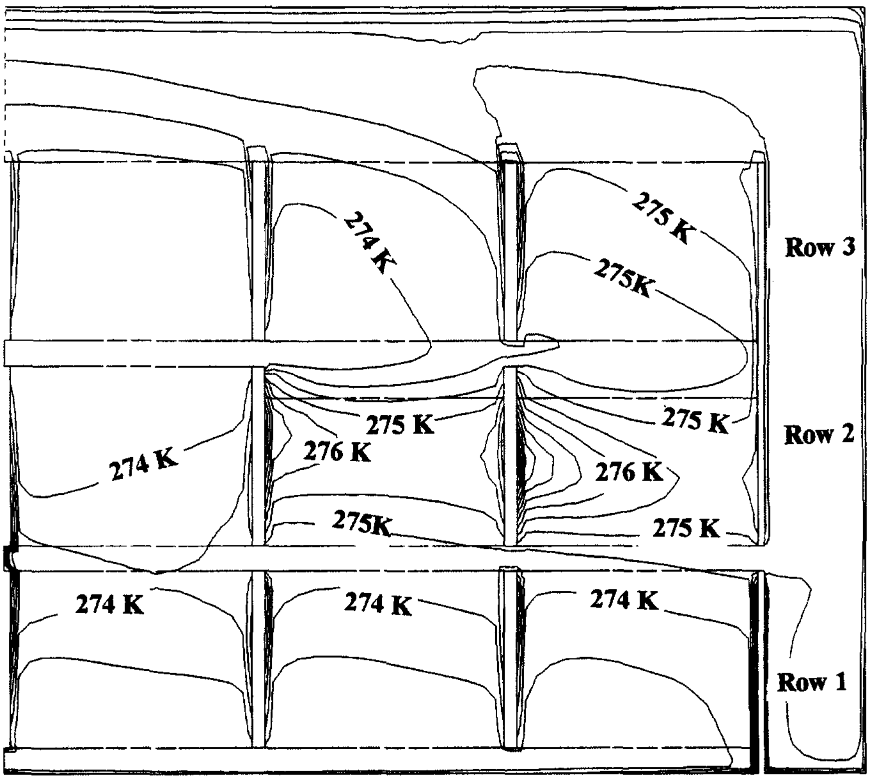


Fig. 9. Average produce temperature distribution in the modified cold store after 15 h operation (the height of the produce in the middle and right bins of the middle row has been changed).

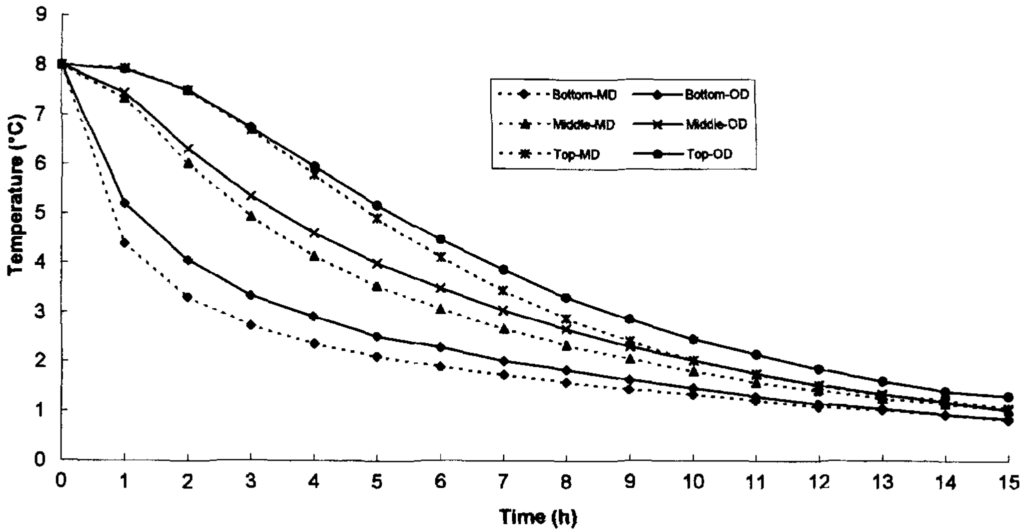


Fig. 10. Cooling curves for beetroot at top, middle and bottom of bins in wet air-cooled store (OD = original design, MD = air circulation rate increased by 20%).

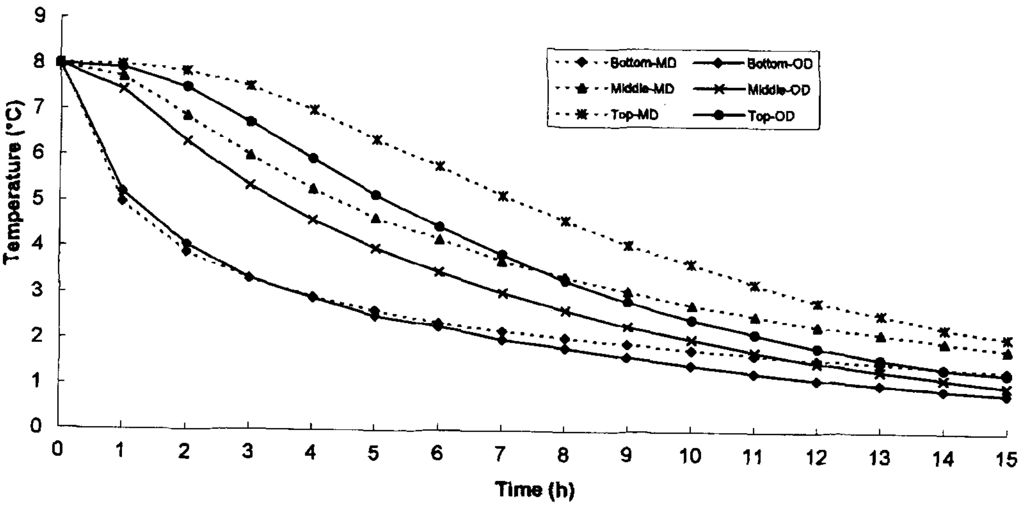


Fig. 11. Cooling curves for beetroot at top, middle and bottom of bins in wet air-cooled store (OD = original design, MD = air circulation rate decreased by 20%).

The study also shows that the porous media model can be used to model the flow and heat transfer through the stored commodity if bins and boxes which are penetrable to the air passage are employed in the store.

The results of the simulations show that there are stagnant regions in the store which may lead to higher weight loss and faster deterioration of the stored commodity. This problem can be overcome by carrying out modifications to the design of the flow passages.

Parametric analysis of the air flow rate through the cold store indicates how the model can be used to optimise store design parameters.

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