A simplified heat transfer model for predicting temperature change inside food package kept in cold room

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Abstract A simple analytical heat flow model for a closed rectangular food package containing fruits or vegetables is proposed for predicting time temperature distribution during transient cooling in a controlled environment cold room. It is based on the assumption of only conductive heat transfer inside a closed food package with effective thermal properties, and convective and radiative heat transfer at the outside of the package. The effective thermal conductivity of the food package is determined by evaluating its effective thermal resistance to heat conduction in the packages. Food packages both as an infinite slab and a finite slab have been investigated. The finite slab solution has been obtained as the product of three infinite slab solutions describe in ASHRAE guide and data book. Time temperature variation has been determined and is presented graphically. The cooling rate and the half cooling time were also obtained. These predicted values, are compared with the experimentally measured values for both the finite and infinite closed packages containing oranges. An excellent agreement between them validated the simple proposed model.

Keywords Room cooling · Precooling · Food packages · Heat transfer through fruits & vegetables

Introduction

Precooling of fruits & vegetables prior to shipment to a cold storage warehouse or before long-term storage helps to reduce deterioration during the transportation and storage. Since most of the food packages are of rectangular shape, prediction of cooling time for these configurations is of great importance to food industry. In the present paper, the heat transfer through fruits and vegetables in a rectangular package, both as infinite slab (i.e. one dimensional heat flow) and as finite slab (i.e. three dimensional heat flow), have been investigated with a view (a) to obtain a simple prediction method for the time temperature variation, cooling rate and the half cooling time.(b) To measure these quantities experimentally and collect data useful to a designer, and (c) to compare the accuracy of the prediction method with the experimentally obtained results. These rectangular packages are cooled in a cold room in still air, thus the cooling occurs by free or natural convection. Though the mechanism of heat transfer in a package containing fruits and vegetables is very complex involving processes discussed later in the paper, it has been observed that considering only conduction inside the package with the convection and radiation at the outside surface of the package yields a predication method of fairly good accuracy and shall prove useful to a practical engineer.

The cooling rate denotes the change in the product temperature per unit change of cooling time for each degree temperature difference between the product and its surrounding, and the ‘half cooling’ time is the time at which the temperature difference between the product and its surrounding become one-half of the initial temperature difference.

A survey of literature reveals that the removal of field heat from fruits and vegetables prior to transportation and

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marketing has been recognized (Dincer 1993) as important in increasing their storage life. These investigations include (a) air cooling (Gaffney et al. 1985; Pan and Bhowmic 1993; NarasimhaRao et al. 1993) and (b) hydro cooling (Dincer 1993) (c) experimental studies (Abdul 1988; Raval et al. 1994; Ansari et al. 1984) and (d) analytical studies (Abdul 1988; Mincer and Shove 1988; Bennet 1964) encompassing computer simulation (Bennet 1964; Ansari et al. 1983) and numerical procedure (Abdul 1988; Pan and Bhowmic 1993; NarasimhaRao et al. 1993; Raval et al. 1994). As the heat and mass transfer analysis requires the knowledge of the thermo physical properties of fruits and vegetables, a large number of researchers (Mattea 1986; Hayakawa 1973; Bhowmik 1979), devoted their efforts in obtaining these values experimentally and some of them published the models for predicting these properties (Bhowmik 1979; Mannapperuma and Singh 1989; Nas-vadha 1989; Yagi and Kunij 1957; Mohsenin 1980; Hayakawa and Bakal 1983; Progethof et al. 1976), have made a comprehensive review of different predictive methods. The effective medium theory (EMT) has been used successfully by Mattea (1986) for obtaining effective thermal conductivity of apples, pears and potatoes.

Analytical model

The mechanism of heat transfer in food packages containing fruits and vegetables is very complex. Figure 1 represents the various modes and processes of heat transfer in bulk of fruits or vegetables in a package, as discussed by Yagi and Kunij (1957) and by Mohsenin (1980).

Fig. 1 A sectional view of a food package showing various modes of heat transfer

It involves (Mohsenin 1980) heat exchange 1) through the solids (fruits or vegetables) 2) solid to solid through the contact surfaces. 3) Radiation between solid surfaces 4) and 5) radiation within voids (air) and solid to voids, and 6) the air film near point of contact between the solids. The heat transfer by lateral mixing of fruits 3) does not occur in the still cold air. From the outside surface of the package to the surrounding cold air the heat exchange may occur 7) by free convection (in the still air) and 8) by radiation.

The air in the void of a closed package is still, which implies negligible convective heat transfer. Moreover, the temperature of the air surrounding the food produce inside the package and the food commodity itself is experimentally found to be approximately the same. Thus it is reasonable to assume that the major mode of heat transfer inside the composite package including product, air and packaging material, is conduction (Hicks 1955). Therefore, in the present analysis it is assumed that the heat transfer occurs by conduction through the solid and the void inside the package; and by natural convection and by radiation from the outside surface of the package. The present analysis is based on designing the effective resistance to heat conduction through two parallel paths (viz. through fruits & through air). The fruits are assumed to be arranged regularly one over the other inside the package as shown in the Fig. 1. Though there are various methods proposed (Castillo et al. 1983), for obtaining effective thermal properties, the present proposed method of getting these properties is very simple. In this work, it has been found that even this very simple model which neglects all other mechanisms of heat transfer except conduction inside the package and convection and radiation outside of the package gives results which are in excellent agreement with the experimentally determined time temperature variation and half cooling time.

Evaluation of effective thermo-physical ($K_e, \rho_e, \alpha_e, c_{pe}$) properties

The procedure for obtaining the effective thermal conductivity of the package by determining its effective thermal resistance is explained with the help of Figs. 2 and 3. The package as shown in Fig. 4(a) is presently considered as an infinite slab (i.e. heat conduction in only one-direction from two larger parallel faces, $H \times L$, and other faces being insulated).

Thermal resistance

The thermal resistance of package box material (cardboard, for example), resistance of food commodity and thermal

Fig. 2 Proposed heat transfer model

resistance of air inside the package were determined by Eqs. 1, 2, and 4, respectively.

$$R_c = \frac{W_c}{A_c \times k_c} \quad (1)$$

Where $A_c = H \times L$

W_c is thickness of card board, A_c is cross section area of cardboard through which heat transfer takes place and k_c is the thermal conductivity of cardboard.

Thermal resistance of air Column is

$$R_{air} = \frac{(W/2)}{A_{air} \times k_{air}} \quad (2)$$

Where, k_{air} is the thermal conductivity of air at mean temperature, W the distance between the two parallel conducting faces of the package box, and A_{air} the area occupied by air columns calculated by Eq. 3.

$$A_{air} = \frac{\text{Volume of entrapped air}}{W} \quad (3)$$

Fig. 3 Thermal resistance circuit for heat transfer

The resistance of food produce column is

$$R_f = \frac{(W/2)}{A_f \times k_f} \quad (4)$$

$$A_f = \frac{\text{Volume of fruits}}{W} \quad (5)$$

Where A_f is area occupied by fruits, and k_f the thermal conductivity of food commodity.

Effective thermal resistance

The thermal resistance circuit for the package is shown in Fig. 3, which yields the effective thermal resistance as

$$R_e = R_c + \frac{R_{air} \times R_f}{R_{air} + R_f} \quad (6)$$

Effective thermal conductivity

The effective thermal conductivity of whole package is found out as

$$K_e = \frac{[(W/2) + W_c]}{A_c \times R_e} \quad (7)$$

The effective specific heat

The effective specific heat of the whole food package is obtained by considering it as a composite material with three components viz. Air, food produce & cardboard. The total enthalpy change is the sum of the enthalpy change of the three components for the same temperature change. This therefore, yields the effective specific heat as

$$C_{pe} = \frac{m_1 \cdot cp_1 + m_2 \cdot cp_2 + m_3 \cdot cp_3}{m_1 + m_2 + m_3} \quad (8)$$

Where m_1 , m_2 and m_3 are mass of cardboard, food commodity and the air; cp_1 , cp_2 and cp_3 are specific heat of cardboard, food commodity and air respectively. Since mass of air inside the package box is negligible compared with the mass of mass of food material and cardboard, the amount of energy stored in air is not significant.

Mass density

The value of overall mass density (ρ_e), for each food package was determined from the calculated total volume and the measured total mass of the package including fruits contained inside the package.

Fig. 4 Conversion of finite slab into three infinite slabs

Thermal diffusivity: (α)

The thermal diffusivity of the package is calculated as the ratio of the effective thermal conductivity to the product of the effective specific heat and the mass density of the package.

$$\alpha_e = \frac{k_e}{\rho_e \times c_{p_e}} \quad (9)$$

The thermophysical properties for oranges, air, and cardboard required in determining the above effective properties are taken from the literature (Ansari et al. 1984), and are listed in Table 4.

Calculation of time temperature variation during cooling of the packages

Infinite slab

The governing equation for transient conduction heat transfer from an infinite slab (one dimensional heat flow) initially at a uniform temperature (t_i) and surrounded by a fluid of constant temperature (t_a), (McAdams 1954) are:

Table 1 Size of the food packages

Package no.	Size of food package boxes. (LxWxH)	Characteristic ^a dimension 2rm = w
1	0.14 m×0.14 m×0.14 m	0.14 m
2	0.28 m×0.21 m×0.21 m	0.21 m
3	0.36 m×0.36 m×0.36 m	0.38 m

^a In case of boxes with two faces exposed

$$\frac{\partial T}{\partial \theta} = \alpha_e \frac{\partial^2 T}{\partial X^2} \quad (10)$$

$$T = T_0 \text{ at } \theta = 0 \quad (11)$$

$$\frac{\partial T}{\partial x} = 0 \text{ at } x = 0 \quad (12)$$

$$\frac{\partial T}{\partial X} = -\frac{h_e}{k_e} (T - T_a) \text{ at } x = rm \quad (13)$$

Fig. 5 Room cooling of oranges in package no. 1 (infinite slab)

Fig. 6 Room cooling of oranges in package no. 2 (infinite slab)

here $r_m = W/2$ or $L/2$ or $H/2$ for infinite slab 1,2 or 3 respectively as shown in Fig. 4(e). r_m is used as a characteristic length in the definition of biot and fourier numbers and in the dimensionless distance n defined as x/r_m . Fig. 4(e).

h_e , the effective heat transfer coefficient takes into account the convective as well as the radiative heat transfer from the outside surface of the package as given by Eq. 14.

$$h_e = h_c + h_r \quad (14)$$

h_c represents the convective heat transfer coefficient for natural convection and is calculated for the case of an infinite slab with two long vertical faces exposed to cold air and other faces insulated from the correlation Eq. 15 given by McAdams (1954).

$$\frac{h_c L_c}{k_a} = 0.59 \{ (Gr) \times (Pr) \}^{0.25} \quad (15)$$

Fig. 7 Room cooling of oranges in package no. 3 (infinite slab)

Fig. 8 Room cooling of oranges in package no. 1 (infinite slab)

Where $Gr = \frac{L_c^3 \rho_a^2 g \beta_a \Delta T}{\mu_a^2}$ for $(Gr)(Pr) = 10^4$ to 10^8 where L_c is the characteristic dimension (height of the vertical face of the package), and the subscript a designates air.

h_r , which represents the radiative heat transfer coefficient is given by the Eq. 16 for the case of radiative heat exchange between a hotter small body (food package) kept in a large enclosure like cold room.

$$h_r = \sigma \epsilon (T^2 + T_a^2)(T + T_a) \quad (16)$$

where T_a is the absolute temperature of the surrounding and T is the absolute temperature of the package surface. Here T varies with time. Hence, h_c and h_r and therefore the value of h_e decrease with time. The instantaneous value of h_e corresponding to existing surface temperature is assumed constant over the whole surface of the package.

The solution of the above Eq. 10 with the associate initial and boundary conditions given by Eqs. 11 to 13 is given by the rapidly converging infinite series as mentioned below.

Fig. 9 Room cooling of oranges in package no. 2 (infinite slab)

Fig. 10 Room cooling of oranges in package no. 3 (infinite slab)

$$\frac{T - T_a}{T_i - T_{aj}} = \sum \frac{j = \alpha}{\lambda_j + \sin \lambda_j \cos \lambda_j} \cos(\lambda_j \cdot n) \exp(-\lambda_j^2 \cdot Fo) \quad (17)$$

where λ 's are eigen values obtained with the help of following equation

$$Bi = \lambda_j / \cot \lambda_j \quad (18)$$

A computer programmed was developed to solve the Eq. 17 and to determine the temperature t , at any instant, at location and thence to obtain the corresponding Biot number (Bi) at different Fourier numbers, (i.e. at different time intervals).

Finite slab

The procedure to calculate temperature distribution for a finite slab is describe in the ASHRAE guide and data book (1972). A rectangular package of finite dimension $L \times W \times H$ as shown in Fig. 4(a) is obtained by intersection of three infinite slab viz. the first with a distance of w between two vertical heat conducting surfaces (infinite slab 1), the second with a distance L between the other two vertical heat conducting surfaces (finite slab-2) and third with

vertical heat conducting surface H between the two horizontal surfaces (infinite slab-3) which are shown respectively in Fig. 4(a–d). Each of above is assumed to have all surfaces, other than heat conducting surface as mentioned above, perfectly insulated. the convective heat transfer co-efficient h_c for the two infinite slabs (1&2) is the same as each heat transferring surfaces is a vertical surface of characteristic length $L_c = H$ and is obtained from Eq. 15. The convective heat transfer co-efficient for infinite slab-3 are those for a horizontal hot surface facing up and down and are different from that of infinite slab 1 & 2.

The following empirical correlations for horizontal plates suggested by McAdams (1954) were used.

- (a) Empirical correlation for heat transfer from horizontal hot plate facing upward,

$$\frac{h_{c1} L_c}{k_a} = 0.54 \{ (Gr) \cdot (Pr) \}^{0.25} \quad (19)$$

for $(Gr) \times (Pr) = 3 \times 10^5$ to 10^{10}

- (b) Empirical correlation for heat transfer from horizontal hot plat facing down.

$$\frac{h_{c1} L_c}{k_a} = 0.27 \{ (Gr) \cdot (Pr) \}^{0.25} \quad (20)$$

for $(Gr) \times (Pr) = 3 \times 10^5$ to 10^{10}

In the case of a horizontal surface Grasshof and Nusselt number are computed by using the characteristic dimension L_c which is defined (Incropera 1981) as follows.

$$L_c = \frac{A_s}{p} \quad (21)$$

Where A_s and P are the surface area and perimeter respectively.

Average of convective heat transfer coefficient h_{c1} and h_{c2} is taken as h_c for the infinite slab-3. Here again, the instantaneous value of the effective heat transfer coefficient h_e is calculated by adding h_r to h_c from the above equation corresponding to the surface temperature at the moment.

The computer programmed developed to evaluate the Eq. 17 as mentioned above, is used to obtain the non-dimensional temperature ratio at the mid-plane (i.e. $n=x/L$)

Table 2 Cooling rate and half cooling time at the center of the food package

Package no.	Infinite slab				Finite slab			
	Cooling rate °C/°C/hr		Half cooling time (hr)		Cooling rate °C/°C/hr		Half cooling time (hr)	
	Predict.	Experi.	Predict.	Experi.	Predict.	Experi.	Predict.	Experi.
1	0.078	0.075	10.01	9.86	0.240	0.203	3.98	4.01
2	0.038	0.038	19.11	18.55	0.094	0.097	8.29	7.97
3	0.016	0.017	41.59	39.62	0.039	0.035	19.86	22.39

Table 3 Percentage deviation and absolute deviation of predicted time temperature

Package no	Location	Infinite slab		Finite slab	
		% absolute deviation	Absolute deviation	% absolute deviation	Absolute deviation
1	Center $n=0.4$	3.3	0.40	17.8	1.00
		3.2	0.06		
2	Center $n=0.4$	2.6	0.22	0.04	0.65
		4.9	0.19		
3	Center $n=0.4$	2.6	0.37	2.8	0.38
		1.9	0.18		

$r_m=0$), for the three infinite slabs. The values of three Biot number (Bi) are obtained from effective heat transfer coefficients values of the three infinite slabs. The Fourier number $Fo = \alpha \cdot \theta / r_m^2$ used is also calculated at any instant. Three values of non-dimensional temperature ratio of the finite rectangular solid at its center are as follows.

$$Y_{fr} = (Y_1) \times (Y_2) \times (Y_3) \quad (22)$$

Where Y_{fr} is the temperature ratio of the finite rectangular solid and subscript 1,2, and 3 designate the three infinite slabs as describe above.

In order to validate the prediction method of the analytical model discussed above, an experimental investigation was planned as mentioned below.

Experimental investigation

Three package of different size (given in Table 1) of cardboard were prepared. Oranges were placed inside them one over another in line as shown in Fig. 1. The package boxes were tested as an infinite slab insulating all the faces excepting two heat conducting faces (specified by characteristic length r_m which is the half the thickness of the slab between the heat transferring surface given in column 3 of Table 1). Later these boxes were tested as a finite slab i.e. with all the faces conducting heat all the package were closed and tightly seal with the help of the tape all around it. The test is carried in a controlled environment cold room, of $3.65 \text{ m} \times 3.65 \text{ m} \times 2.44 \text{ m}$ with a 5 ton refrigeration plant and a constant temperature of 1 ± 0.5 was maintained in the cold room which is in the recommended temperature range for storage of orange (Ansari et al. 1984). A high relative humidity of $94 \pm 1\%$ was maintained in the room in order to reduce the moisture loss from the orange. Temperature probe were prepared by fixing 28 SWG copper constantan thermocouples on pointed plastic strip, which can be easily insert in the orange to measure the temperature of the oranges at desired location of the package. The temperature was

measured at two location viz. at the center of the package (i.e. $n=0$) and at $n=0.4$; these the location are shown in the Fig. 4(e). The temperature thus recorded is the temperature of an orange located at $n=0$ and $n=0.4$ in the package. For package tested as finite slab, the temperature measured only at the center ($n=0$) of the package. Air velocity was checked with help of pitot static tube at several points in the neighbourhood of a package and it was observed that there was no significant bulk air motion and the condition of the natural convection, as assumed in the analytical model, existed practically. Mass density and porosity were determined after completion of each test run.

Result and discussion

The temperature were recorded regularly an interval of 30 min. The experimental data so obtained are presented as time–temperature variation for the infinite slabs in Figs. 5, 6, and 7 and for the infinite slabs in Figs. 8, 9 and 10 along with the corresponding results predicated by the analytical model. In both the cases, different physical properties (K_e , ρ_e , α_e , c_{pe}) are given the on the figure themselves.

The quantities of practical interest to an engineer viz. the cooling rates and half cooling time were determined by the

Table 4 Thermal properties of food products and cardboard

Thermal conductivity of orange (Ansari et al. 1984)	0.553 W/m.K
Specific heat of orange (Ansari et al. 1984)	4.066 kJ/kg.K
Thermal conductivity of peas (Ansari et al. 1984)	0.255 W/m.K
Specific heat of peas (Ansari et al. 1984)	4.174 kJ/kg.K
Thermal conductivity of grapes (ASHRAE 1978)	0.567 W/m.K
Specific heat of grapes (ASHRAE 1978)	3.629 kJ/kg.K
Specific heat of cardboard (Isachenko 1977)	1.51 kJ/kg.K
Emissivity of cardboard (Isachenko 1977)	0.96

method (Mohsenin 1980) based on the slop of the straight portion of the cooling curve, both from the analytical model and the experimental data and are given in Table 2. Here, the cooling rate denotes the change in the product temperature per unit change of cooling time for each degree temperature difference between the product and its surrounding, and the ‘half cooling’ time is the time at which the temperature difference between the product and its surrounding become one-half of the initial temperature difference.

The predicated and experimental values of time temperature distribution have been compared on the basis of absolute percentage deviation (pd) for each data point as under.

$$Pd = \frac{|T(\text{pre}) - T(\text{exp})|}{T(\text{pre})} \times 100 \quad (23)$$

Table 3 represented percentage deviation of predicated and experimental time temperature variation for infinite and finite slabs and Table 4, represents thermal properties of food products and cardboard.

Comparison of analytical prediction with experimental results

As mentioned earlier, the time temperature variation determine both analytical and experimentally for all the packages are presented in Figs. 5, 6 and 7 for infinite slab and Figs. 8, 9 and 10 for finite slabs. The figure shows that the predicted values are very close to the experimental values. The analytical and experimental time temperature curves for most of the packages are quite similar in qualitative variation. Quantitatively 90% of the data have absolute average percentage deviation less than 5%. This excellent agreement validate the proposed simple method based on only conductive heat transfer inside a closed food package with effective thermal properties and convective and radiative heat transfer at the out of the package.

Table 2, gives the overall quantities of engineering interest viz. cooling rates and half cooling time. Both predicted and measured values are given. As seen from the table that the half cooling time as predicted by the proposed analytical model is very closed to the experimental values for the entire test run. For example, it is found that in the finite package no. 1 containing 2 kg of orange has been found to be 4.01 h experimentally and 3.98 h by the analytical model. Similarly for package no. 3 the experimental and analytical half cooling times are 19.8 h and 20.3 h, respectively. There is again an excellent agreement between the experimental measurement and analysis.

Conclusions

The predicated temperature distribution agrees well with experimentally measured temperature distribution. Hence the proposed simplified heat flow model for a closed rectangular food package is validated, both for an infinite slab and finite slab heat flow cases. The proposed model is very simple, efficient and of acceptable accuracy and can be easily used by a practical engineer and designer working in the low temperature preservation food industry.

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