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Numerical Heat Transfer Modelling of a Cold Thermal Energy Storage System for a Refrigerated Vehicle

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Abstract:

In this paper an enthalpy formation model with fully implicit finite difference method has been used for one dimensional analysis to study the dynamic behavior of heat absorbed and melt fraction of PCM in the designed model. The developed model can guide to design a refrigerated truck for PCM storage quantity and time required to maintain the desired temperature.

Keywords: *Moving boundary, Cold storage chain, Phase change material, Enthalpy formation, Horticulture*

1. INTRODUCTION

India is the second largest producer of vegetables and fruits in the world. The production of fruits and vegetables in recent years was more than 300 Million Tonnes from an area of 25 Million Hectares. Over the last decade, annual production increased by 4.8%. The percentage share of horticulture output in Agriculture has become 33%. There are up-to 15% wastage of fruits and vegetables every year in India due to lack of harvesting and insufficient cold chain establishments. About 2-3% of the total vegetables and fruits are transported in refrigerated trucks in India [1].The economic growth, and population growth has continuously increased the demand of quality products and is also increasing the demand for refrigerated fruits and vegetables.

In refrigerated transportation of fruits and vegetables desired temperature and air movement is required to maintain the quality and extending the shelf-life span of perishable products. The temperature of the fruits and vegetables during storage and transportation needs to be kept at which the metabolic and microbial deterioration is minimized. These trucks can be integrated with cold thermal storage system having phase change material (PCM) as thermal storage material to maintain the desired thermal conditions. On average, foodstuff is transported over a distance of hundreds of k before arriving to the consumer[1]. The drive distance, use of fossil energy and greenhouse gas emissions co-related with this transport system has received substantial attention. In United States, more than 75% of food produced is transported and stored at the desired temperature using refrigeration. More than 1.5 million refrigerated trucks are used worldwide for this service [2-4].
Phase change material based systems are preferred over other thermal energy storage system due to high energy density and isothermal phase change process to regulate the temperature [6-11].This paper deals with heat transfer modelling of

May-June 2020
ISSN: 0193-4120 Page No. 10739 – 10744

refrigerated truck incorporated phase change material. Heat transfer equation in one dimensional geometry has been solved numerically by introducing enthalpy method. The purpose of this study to calculate the effect of heat source on solid – liquid interface motion with time and quantity of heat stored, hence, the time lag between input heat and heat entering into the refrigerated truck. Effect of variation of absorber plate temperature on heat storage and PCM melt fraction with time has been predicted.

Specification of the refrigeration truck and phase change material

In this study, a stationary prototype refrigeration system has been considered. The dimensions of the refrigerated vehicle were 4.22 m x 2.1 m x 1.8 m.with ten cm insulation. The PCM panels of -15.2^{o}C freezing temperature with 10 cm width were installed at the walls of the truck. To freeze the PCM panels, a refrigeration unit was used. Heat was introduced through the wall of refrigerated truck which equivalent to the heat loss from the walls [5]. The specifications of refrigerated truck and used PCM are given in table 1 and 2 respectively.

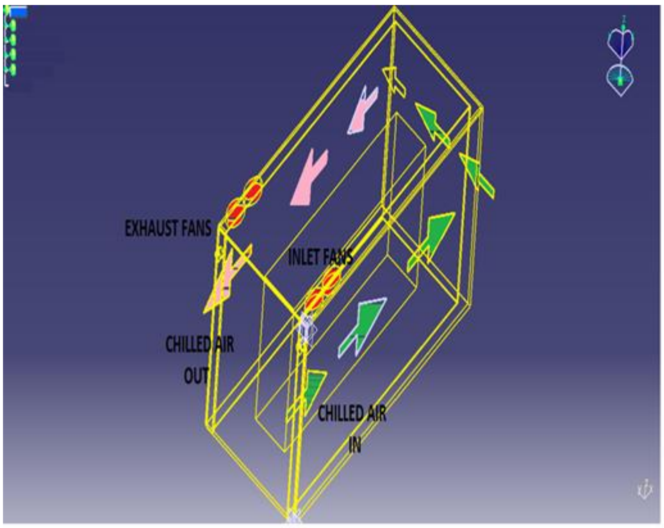

Figure 1: Chilled air circulation

Table 1: Specification of the refrigerating truck

Characteristics	Units	Value
Length	m	8.22
Width	m	2.1
High	m	1.8
Capacity	m 3	31.07
Surface area	m 2	71.6
Conductivity of polystyrene	W/m ^{o}C	0.038
Thickness of insulation	m	0.1
Maximum ambient temperature	^{o}C	40
Store temperature	^{o}C	0
Overall heat transfer coefficient	W/m^{2o}C	0.38

Table 2: Thermo-physical properties of PCM

Characteristics	Units	Value
Phase change temperature	^{o}C	-15.2
Latent heat of fusion	kJ/ kg	284
Approx. specific heat in PCM	kJ/ kg K	4.21
Density (liquid)	Kg/ m^{3}	1013
Thermal conductivity	W/ m ^{o}C	0.5

Numerical Simulation of the System

The system considered for simulation is shown in figure 1 and the specifications are given in table 1.

In this simulation models, enthalpy based formulation for the study of refrigerating truck having phase change material as thermal management unit has been used. Using enthalpy method the phase change problem converts into a simpler formi.e. heat transfer equation becomes similar to the single phase equation heat transfer and mushy zone can exitbetween the solid-liquid phases [8,12].

May-June 2020
ISSN: 0193-4120 Page No. 10739 – 10744

For the heat transfer analysis of the system, the assumptions have been made are (i) thermo-physical properties of PCM are temperature independent both the phases and (ii) the PCM is initially in the solid phase.

For a PCMchanging a phase transformation (solid to liquid), one-dimensional heat transfer equation in the PCM can be written in terms of temperature and total volumetric enthalpy as

$$\frac{\partial h}{\partial t} = \frac{\partial}{\partial x}\left(\alpha \frac{\partial h}{\partial x}\right) - \rho_l \lambda \frac{\partial f}{\partial t} \tag{1}$$

The total volumetric enthalpy H is the sum of the sensible and latent heats of the PCM, i.e.

$$H(T) = h(T) + \rho_l f(T)\lambda \ , \tag{2}$$

In equation (2) $h(T) = \int_{T_m}^{T} \rho_k C_k dT$.

In case of isothermal phase change, the liquid fraction of melt "f" is given in table 4 for different conditions.

Table 3: Melt fraction conditions

Melt fraction	Temperature condition	phase
f = 0	if T < Tm	Solid
f=1	if T > Tm	Liquid
0 < f < 1	-	Mushy zone

Temperature- enthalpy relation [xx]for the PCM, is given in table 4.

Table 4: Temperature-enthalpy relation

Temperature	Enthalpy	Phase
$T = T_m + \frac{H}{\rho_s C_s}$	H < 0	solid
$T = T_m$	0 ≤ H ≤ $\rho_L \lambda$	interface
$T = T_m + \frac{H - \rho_l \lambda}{\rho_l C_l}$	H> $\rho_L \lambda$	liquid

Finite difference method has been used to solve Equation (1). The finite difference equation for the PCM is obtained on integrating equation (1) over each control volume [12-13].

Boundary and Initial Conditions

Following initial and boundary conditions have been used to solve equation (1) and PCM temperature is below the melting point, viz.

$$h_{init} = \rho_s C_s (T_m - T_{init}) . \tag{4}$$

The boundary condition at x = 0 can be written as

$$h(0,t) = \rho_k C_k \left(T_{Abs} - T_m\right) , \tag{5}$$

i.e. at x = 0 the volumetric enthalpy is known at each time and, the face x = L is adiabatic, i.e.

$$\frac{dh}{dx} = 0 \ , \tag{6}$$

One-dimensional discretization of equation (1) is shown in figure 2.

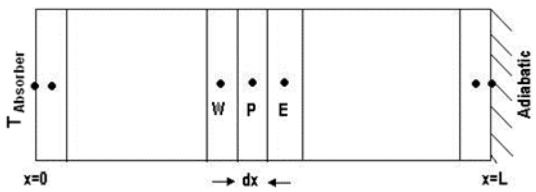

Figure 2; One-dimensional domain

Results and Discussion

Thirty two space steps of 3.1 mm and the time step of 2 minutes has been used for one dimensional analysis. These steps were obtained by optimizing

May-June 2020
ISSN: 0193-4120 Page No. 10739 – 10744

for space and time domain being selected. The thermo-physical properties of PCM used in the analysis are given in table 2. The PCM was in solid phase and consideredat -5°C below the melting temperature of the PCM. To see the effect of heat absorbing surface temperature, the surface temperature was maintained 1°C and 2°C, higher temperature than the melting temperature for melt fraction, heat absorbed and PCM temperature. This absorbing surface temperature is indicated as Tm+ 1°C and Tm+2 °C. The latent heat energy absorbed by the PCM system for different input temperature of absorber surface is depicted in figures 3. From the figure, it can also be seen that as the temperature difference between the absorber and the melting point of the PCM increases the higher amount of heat is absorbed in less time and the same is also tabulated in table 7.

Variation of melt fraction of PCM with time is plotted in figure 4 for different surface temperature. It can be seen from the figure that the time taken for melting of PCM decreases as the exposed surface temperature increase from -15.2 °C to -13.2 °C higher than the melting point. The time taken for 100% melting decreases from 600 minutes to 240 minutes as the surface temperature increases from 1°C to 2°C higher than the melting point. Time taken for complete melting for various temperature differences is shown in table 6. To see the variation of temperature at different nodes inside the PCM for temperature difference of 1°C and 2°C at different given times are shown in figure 5. Obviously, the temperature as the melt fraction of PCM increases with time the temperature of the last node PCM shown in Table 6.

Table 5: Time taken, heat stored and melt fraction with absorber surface temperature

Absorber surface temperature (°C)	Time (minutes)	Heat absorbed (kJ/m^2)	Fraction of melt (%)
Tm+2	240	7.19x10^6	100
Tm+1	600	7.19x10^6	100

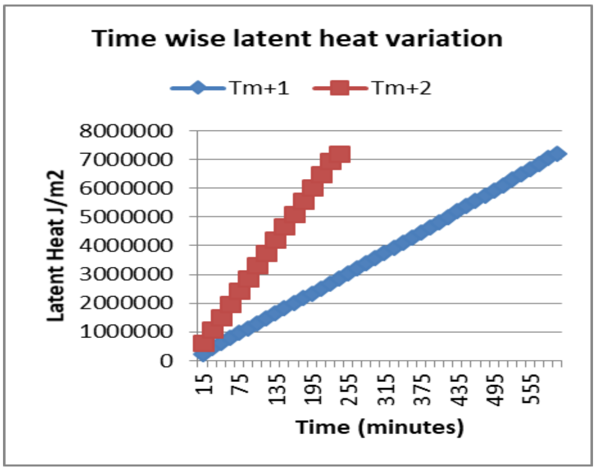

Figure 3: Time wise variation of heat absorbed in thermal management system at different surface temperature

Table 6: Melt fraction of PCM with time

Time (minutes)	Melt fraction (%)	
	Tm+1	Tm+2
0	0	0
30	6	15
105	18	46
210	35	90
240	40	100
315	52	-
420	70	-
525	87	-
600	100	

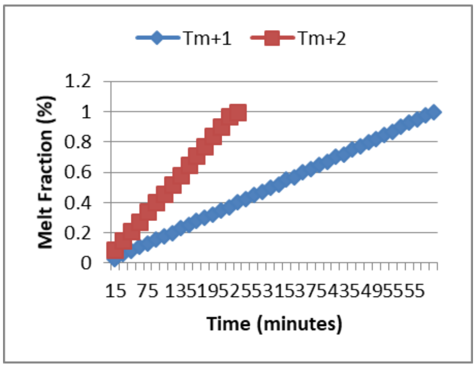

Figure 4: Variation of melt fraction of PCM with time at different absorbing surface temperatures

May-June 2020
ISSN: 0193-4120 Page No. 10739 – 10744

Table 7: Variation of nodes temperature, with time for different absorbing surface temperature

Time (Minutes)	Node No. 1 Temperature (°C)	Node No. 16 Temperature (°C)	Node No. 32 Temperature (°C))			
	Tm+1	Tm+2	Tm+1	Tm+2	Tm+1	Tm+2
30	-14.2	-13.2	-15.7	-16.06	-15.87	-16.38
105	-14.2	-13.2	-15.21	-15.2	-15.21	-15.2
105	-14.2	-13.2	-15.2	-14.4	-15.2	-15.2
210	-14.2	-	-15.2	37.4	-15.2	-
315	-14.2	-	-14.93	-	-15.1	-
525	-14.2	-	-14.77	-	-	-

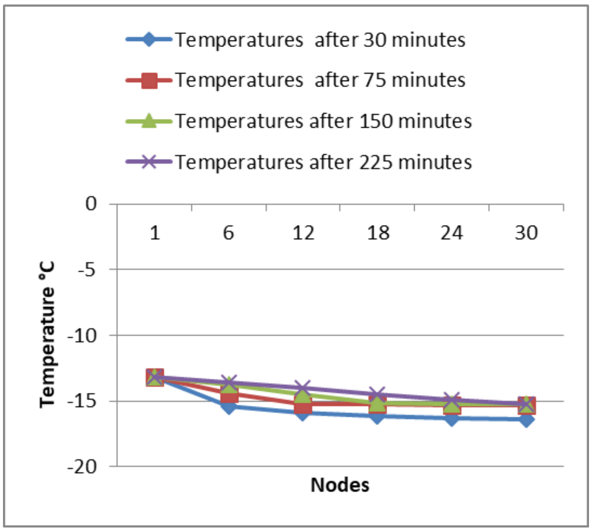

Figure 5: Time wise variation in temperatures at nodes at Tm+2 (°C) temperature of absorbing surface

5. CONCLUSION

The one dimensional enthalpy based numerical model has been developed to predict the time-wise variation of heat absorbed and melt percentage of PCM in the designed system. Variation of temperature with time for each node in one-dimension as model was also obtained. The developed model can be used for designing cold thermal storage system in terms of quantity of PCM, temperature, heat transfer area and travel distance to maintained the quality of vegetables and fruits. As it may eliminate the active refrigeration machine during transport, therefore energy & environmental benefits can be estimated separately.

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