DISTRIBUTED THERMAL STORAGE IN A PLASTIC APPARATUS

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Abstract. *The purpose of this paper is to evaluate the capacity of a small-scale thermal storage apparatus constructed of plastic.The recovery fluid is air. The analysis is limited to the melting stage of the storage cycle, as this is recognized as the controlling process. For the thermal rating computations a simplified model, utilizing the overall heat transfer coefficient is used. The assumptions required for establishing the model are discussed. The results are presented in the form of a thermal recovery versus elapsed time curve. The results indicate that an apparatus of a size comparable to that of the remote unit of a split type air conditioner, could counter a thermal load of 6,5 kWh. Over a recovery period of three hours, this would be equivalent to the capacity of a window type air conditionoing unit.*

Keywords*: air conditioning, ice-storage tanks, phase-change modeling, thermal energy storage.*

1. Introduction

Over the years, development of engineered products has seen a trend towards higher efficiencies and cost effectiveness.

This has been achieved often, through progressively larger sizes of equipment and systems, taking advantage of the larger efficiencies usually inherent in larger apparatus.

Starting from the mid-twentieth century, new considerations have been brought to relevance in engineering practice; namely the concern for sustainable economic development and environmental responsibility.

This, in turn, has made feasible the development of smaller systems and units, to fulfill local requirements distributed in space.

Thus, we have witnessed the evolvement of distributed generation, small hydroelectric plants (PCH) etc.

Thermal storage equipment is another development that resulted from these new concerns. An air conditioning system provided with this feature is usually more costly and less efficient, and is nonetheless installed.

Thermal storage systems usually employ water, either with or without phase change, as the storage media. One of the greatest drawbacks in this case, is the large space used up by the water or ice tanks. Very frequently the required space is simply not available at the site.

If the total storage mass could be dispersed in the building, close to the thermal load sources, thermal storage could be provided with systems that otherwise would be deprived of this feature.

In this paper, a small unit, similar in size to the evaporator of a split type air conditioner, is analyzed to quantify the amount of energy that could be stored, and more importantly, recovered, within the space and time constraints imposed by the application.

In an effort to reduce the eventual manufacturing costs of such an apparatus, the analysis is based on the use of plastic as material of construction.

2. Modeling

The computation model used in this paper is a plastic tube embedded in ice and progress of the ice-water interface, within the allotted recovery stage, is determined.

Air flowing inside the tube, is the recovery fluid.

The variables involved are defined in Fig. (1). For this geometry, the heat conduction equation is:

$$\frac{\partial T}{\partial t} = \alpha\left[\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r}\right] = \frac{\alpha}{r}\frac{\partial T}{\partial r}\left[r\frac{\partial T}{\partial r}\right] \tag{1}$$

Where:
- r = radial coordinate [m]
- r_f = radius at the ice-water interface [m]
- T = temperature [K]
- α = thermal diffusivity [m^2/s]

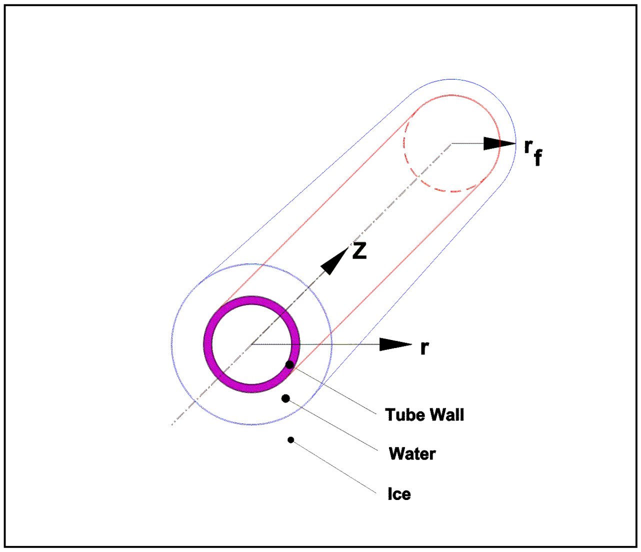

Figure 1. Geometric variables

It can be shown that Eq. (1) has no analytical solution (Mello 1992), therefore a numerical solution will be derived from the boundary conditions.

If the ice mass is considered to be at the melting temperature, the progress of the ice-water interface is conditioned by the heat supplied by conduction, thus:

$$K\frac{\partial T}{\partial r} = \rho\lambda\frac{\partial r_f}{\partial t} \tag{2}$$

Where:

K = thermal conductivity [W/m K]

ρ = density [kg/m^3]

λ = phase change enthalpy [kJ/kg]

The heat supplied by the air at the tube wall can be calculated by the following equation:

$$K\frac{\partial T}{\partial r} = U\left(T_b - T_f\right) \tag{3}$$

Where:

T_b = bulk air temperature [K]

T_f = melting temperature [K]

U = overall heat transfer coefficient, including air film and tube wall resistance [W/m^2 K]

However, if the thermal resistance of the water layer is included in the overall coefficient, equations 2 and 3 can be equated, and if finite differentials are taken, result in:

$$U\left(T_b - T_f\right) = \rho\lambda\frac{\Delta r_f}{\Delta t} \tag{4}$$

This equation can now be applied in finite increments to numerically determine the ice-water interface progress, provided the following conditions are fulfilled:

(a) there is no thermal conduction in the axial direction
(b) the heat stored in the molten water is neglected
(c) the heat conductivity of water is considered constant
(d) the convective air film inside the tube is constant
(e) there is no convective heat transfer in the molten layer

Abugederah and Ismail (1998) have compared the results obtained with the use of an overall coefficient with those of more rigorous modeling and have found no substantial discrepancies. Freezing with a liquid phase working fluid inside the tube was analyzed.

2.1. Discussion of the conditions

Conditions (a) and (b) above can be shown to have minimum effect in the results.

The effect of conditions (c) and (d) can be reduced by dividing the tube axially into a certain number of sections in which air and water properties can be considered constant.

Many authors have addressed the problem of natural convection in the melting around an horizontal pipe (condition e).

Ho and Chen (1986) have determined that, even when convection is considered, the melted volume has an approximately linear dependence with time.

Figure 2. shows a comparison of the results related in this paper to those presented in figure 7 of the cited Ho and Chen work.

In Fig. (2) :

V^*= melted volume divided by the volume of the tube
Fo = Fourier dimensionless group [$\alpha t/r^2$]
Ste = Stefan dimensionless group [$C_p\ \Delta T/\lambda$]

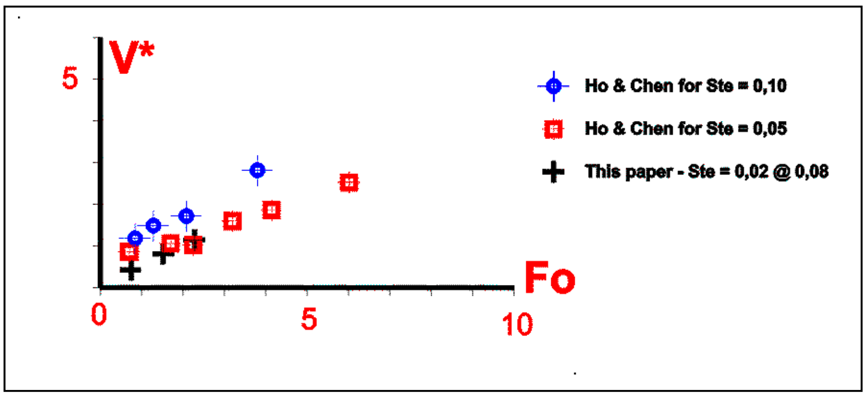

Figure 2 . Comparison with the literature

It can be seen that there is reasonable consistency. It can also be noted that the melting process develops over a much smaller range of **Fo** than that in the cited work.

When a tube bundle, rather than a single tube is studied, Bathelt et all (1979) have shown that other factors such as displacement of the ice mass and overlapping of the ice-water interface of adjoining tubes are equally important factors and are very difficult to account for analytically.

It seems, however, that heat transfer is enhanced in many instances (Jekel et all, 1993).

3. Computation model

Figure (3) shows the geometry used for the computations.

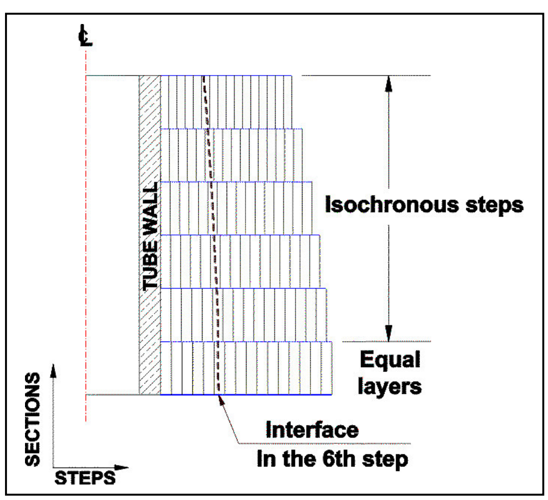

Figure 3. Geometry used in the computation

The tube is divided axially into six sections in which physical properties are considered constant.

In the first section, the ice-water interface is considered to proceed in equal thickness layers and the corresponding duration of each step is determined with the use of Eq. (4).

For the remaining sections, durations determined for the first section are used to calculate the melted thickness of the various steps.

The exit air temperature determined in each section is used as input for the next.

The number of steps considered is that which add up to the allotted time for the recovery phase.

For most applications, the recovery phase is considered to be the, so-called, peak period of energy demand which is usually limited to 3 hours.

Figure (4) is a logic diagram of the computation process.

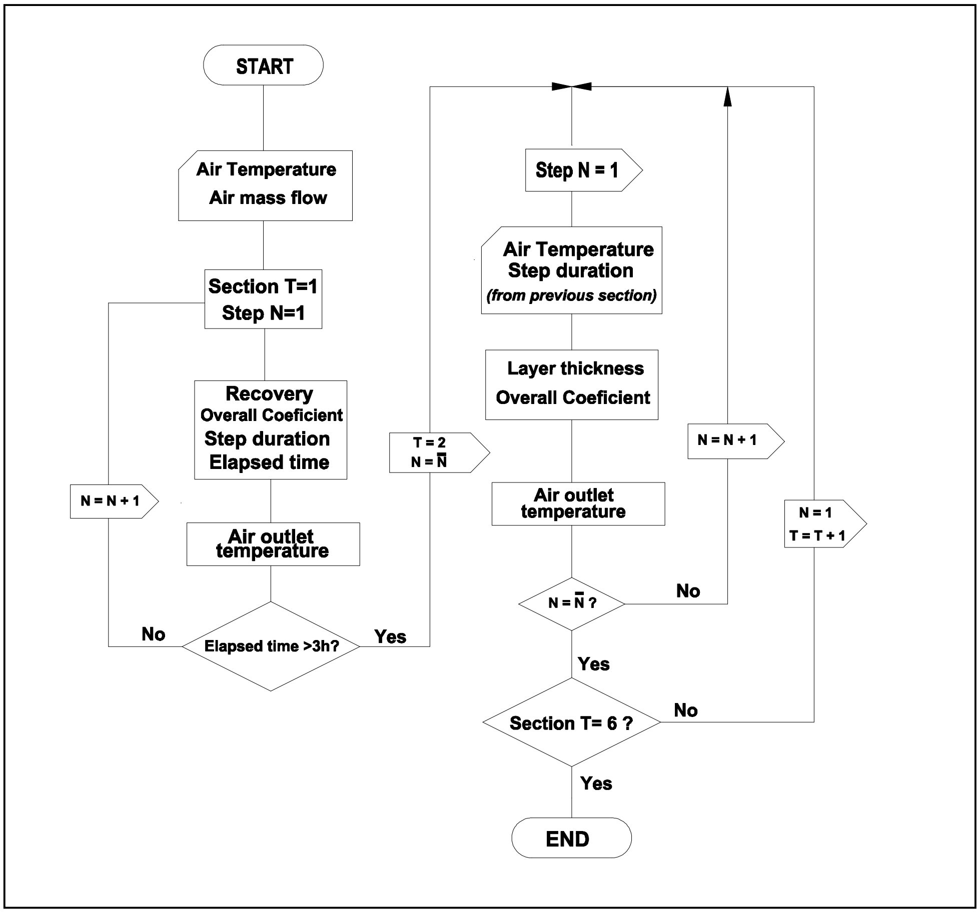

Figure 4. Logic diagram for the computation process

The physical properties and dimensions used in the computations were:

Density of ice [kg/m^3] = 916,9
Ice-water latent heat [kJ/kg] = 333,4
Thermal conductivity of water [W/m K] = 0,57
Specific heat of water [kJ/kg K] = 4,2
Specific heat of air [kJ/kg K] = 1,0
Density of air [kg/m^3] = 1,293 (@ 273 K)
Viscosity of air [cP] = 0,0018 (@ 293 K)
Entering air temperature [K] = 298
Tube length [mm] = 600
Tube outside diameter [mm] = 16,4
Tube material Thermal conductivity [W/m K] = 0,06

4. Results

Figure (5) shows the progress of the ice-water interface with time.

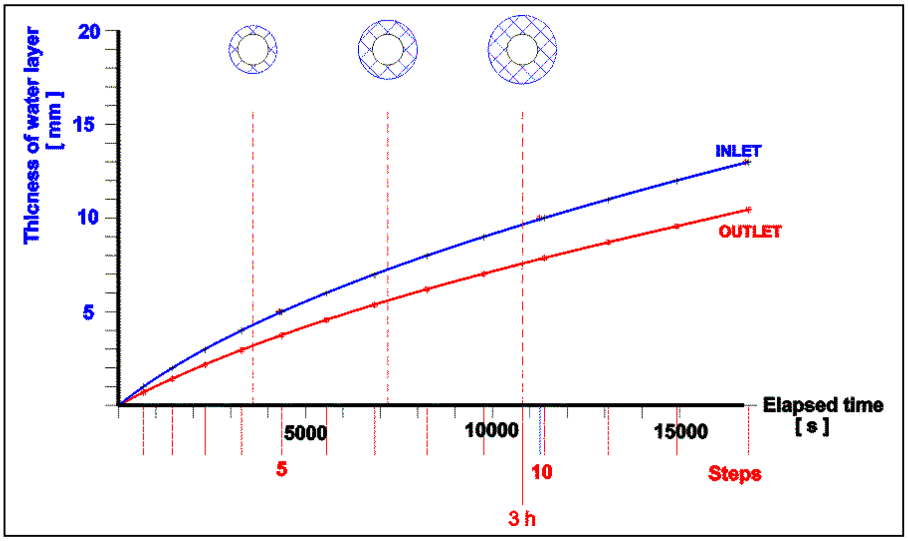

Figure 5. Progress of the molten layer thickness

Sketches showing the thickness of the molten layer relative to the tube diameter are also superimposed on Fig. (5). It can be seen that, due to the high thermal impedance of the tube material, the molten thickness is small, of the order of the tube radius. This contributes to minimize the intensity of natural convection, when compared with results of the literature cited.

Figure (6) is the performance curve for a single tube.

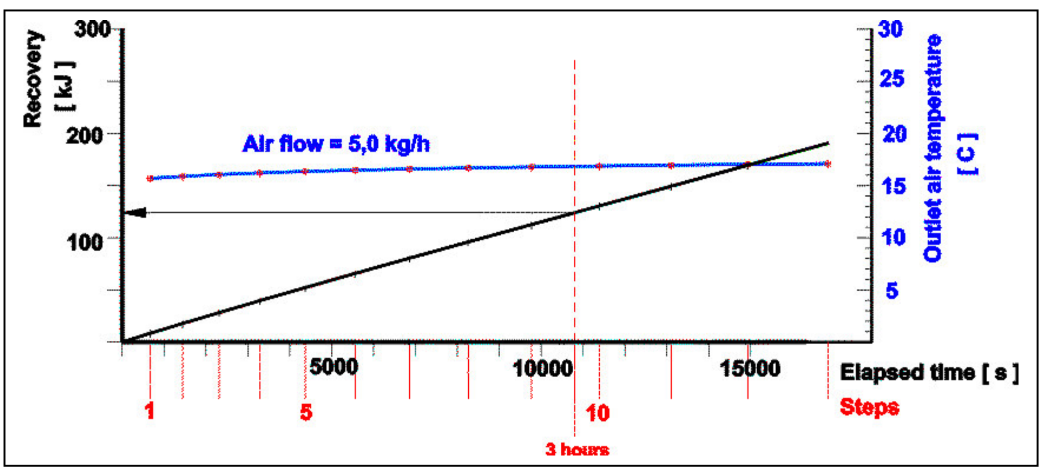

Figure 6. Performance curves

On the left side is the cumulative heat transferred to the ice by the air. It can be seen that at the end of the three-hour period, a total of 135 kJ is exchanged.

The scale on the right shows the outlet air temperature. Since the tube wall thermal impedance is comparable to that of the water layer, the outlet temperature varies very little during the run. This is a desirable characteristic since it would simplify the control of the apparatus in operation.

With the geometric constrains given by Fig. (5) and the performance given by Fig. (6) it is germane to arrive at the capacity of an apparatus of given dimensions.

Figure (7) depicts a possible arrangement for an apparatus of 170 tubes. Total capacity is 23 000 kJ. It would be approximately equivalent to a window type air conditioner of 7500 BTU.

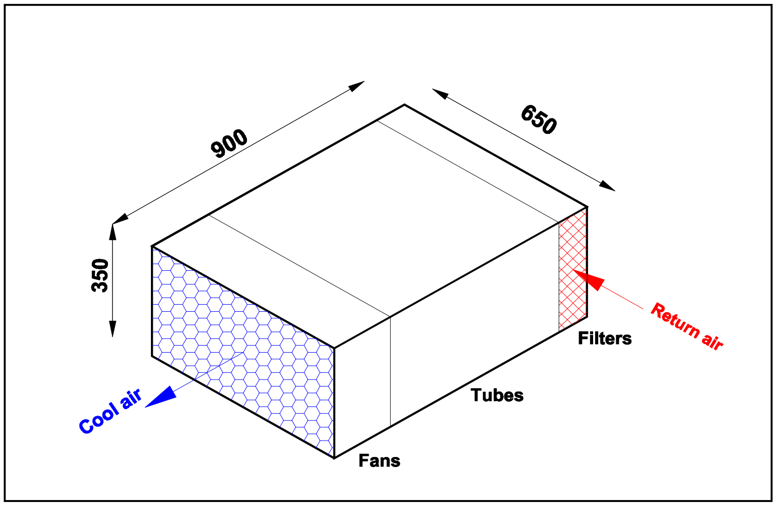

Figure 7. General arrangement of the apparatus

4. Acknowledgement

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