AN ANALYTICAL TECHNIQUE OF TRANSIENT PCM MELTING CALCULATION FOR DIFFERENT PCM CONTAINERS CASES

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ABSTRACT

An analytical model earlier developed for calculation of transient phase-change material (PCM) melting (Dubovsky et al. [1]) proved to be a useful tool for use in the analysis of various structures. As shown in our subsequent studies, despite the serious assumptions in its development, the model is effective in optimization of the design and parameters of systems, including the case of transition from analytical methods to numerical calculations. The present work shows that the use of the analytical technique and its modifications for the analyzed PCM arrangements is beneficial. Proper application of the technique makes it possible to obtain the parameters of the real PCM melting process in the form of algebraic formulas, both for the transient values of variables over time, and for the overall process characteristics. Our analysis is performed for the commonly used PCM containers where we use classifications in terms of the geometry and configuration as defined in the literature. A comparison with the results of numerical calculations of transient melting by CFD FLUENT, confirms the validity of algebraic formulas and allows to assess the nature and value of the introduced error in the results of our analytical method, for each analyzed case.

INTRODUCTION

In contrast to ordinary heat exchangers operating in a steady-state mode, heat exchangers with PCM, or storage units, operate during a limited period of time in a transient regime. Correspondingly, all the parameters (heat transfer rate, heat transfer fluid (HTF) temperature, melt fraction) are variables in time. This refers both to local values of these parameters and to overall heat exchanger characteristics like HTF temperature at the outlet and average melt fraction.

The available literature includes studies of various storage unit configurations. It appears that among them, shell-and-tube systems are the most popular ones. According to Agyenim et al. [2], this geometry accounts for more than 70% of articles in the literature. This is related to the fact that tubular heat exchangers are widely used in various engineering applications, with the data on their design and performance readily available.

When a tubular unit is used for heat storage, a PCM may be stored in the tubes whereas HTF passes through the shell, or vice versa. Accordingly, the analyses presented in the literature reflect the specific configurations used. For instance, Lacroix [3] analyzes a unit with forced laminar convection of water as the heat-transfer fluid inside the tube, whereas the PCM melts in a hollow cylinder around it. Trp [4] and Trp et al. [5] investigate the effect of various operating conditions and geometric parameters on a PCM-water shell-and-tube unit.

Multi-tube heat transfer arrays are considered by Agyenim et al. [6]. More complex configurations may include a triplex concentric tube where the PCM fills the middle channel, a hot heat transfer fluid flows in the outer channel, and a cold heat transfer fluid flows in the inner channel (Long [7]), and a double pipe with the PCM embedded in a graphite matrix (Medrano et al. [8]).

The literature reports also configurations with a PCM placed inside the tube and a longitudinal fluid flows outside it (Bansal and Buddhi [9], Esen et al. [10]). Effect of various PCMs, tube radii, total PCM volume, heat transfer fluid mass flow rate and inlet temperature of fluid are analyzed. The units designed specifically for solar energy applications have been analyzed by Esen and Ayhan [11] and Esen [12]. Some work in the field includes analyses of a latent-heat storage system for direct steam generation in solar thermal power plants (Bayón et al. [13]) and of a PCM canister suggested for a heat pipe solar receiver (Xiaohong et al. [14]).

In the paper by Dubovsky et al. [1] we firstly introduced a basic mathematical model, consisting of a system of equations describing the heat transfer and PCM melting process, and the analytical solution of this system. In this solution, both integral and local process parameters, including heat transfer rate, HTF temperature, and PCM melt fraction were obtained as explicit functions of time and, for local parameters, of coordinate. The basic model has been presented for the case where the PCM melts inside tubes while HTF (air) flows across the tube banks.

The main assumptions of the model are:

1. Melting is concentric – hence, one-dimensional PCM melting in the radial direction is assumed.
2. Temperature gradient in cross-section is neglected both in HTF and in tube material and in PCM.
3. Sensible heat of both the tubes and the PCM is neglected – hence, all of the heat transferred from the HTF is absorbed by the latent heat of the PCM.
4. Other properties are taken constant.
5. Heat transfer coefficient from HTF to tube is known and constant, regardless of time and coordinates.
6. Heat transfer in the axial direction in the PCM and tube is neglected.
7. The partial time derivative of the HTF temperature is neglected in the HTF energy equation – hence, the model is valid for low HTF density, that is, air or other gas stream.

Further, in the paper by Dubovsky et al. [15], the analytic solution has been used to investigate the performance of a cross-flow PCM-air heat exchanger of in-line tube banks. Full variety of the standard tube pitches was considered. Calculations have been performed for two basic cases: a constant overall PCM mass, and a constant overall volume of the exchanger.

In our recent study (Ezra et al., [16]), the Latent Heat Thermal Energy Storage (LHTES) units, which contain multiple phase-change materials with different melting temperatures, have been analyzed. The materials were arranged in a cascade, where the melting temperature decreased from the entrance to the exit of the unit. The corresponding modification of the basic mathematical model was solved numerically using an explicit numerical scheme, implemented in MATLAB computer software. Nevertheless, for the minimum melting time calculations, the analytical solution by [1] has been used, as well as for the maximum relative performance improvement calculation.

Note that in papers [1], [15] and [16] the PCM melting inside tubes was studied, with external cross-flow of hot air over in-line arranged tube banks. This configuration with a sufficiently large number of tube rows (90 rows in [1]), practically eliminates the use of computer programs for direct processes calculation (as CFD FLUENT). That is posing a direct numerical experiment to compare and verify the applicability of analytical solutions to the real process.

The recent completion of a mathematical model for the case of shell-and-tube systems is represented by Dubovsky et al. [17]. Development of a technique in this regard is primarily due to the above-mentioned general prevalence of such systems. At the same time, an important aspect is the possibility of setting a numerical experiment by direct calculation of these cases with the use of the CFD FLUENT and comparing and testing so the proposed analytical methodology. But in [17] is presented only a mathematical justification of the analytical model both for the case when the PCM is stored in the shell whereas HTF passes through the tubes, and vice versa.

In this paper, we compare the results of numerical experiments using FLUENT, with the calculations of the analytical model. We consider both cases of shell-and-tube configuration and some other cases in which we can directly estimate the CFD FLUENT program.

NOMENCLATURE

A	[m^2]	Heat transfer area, related to diameter D
c_p	[J/kg^{o}C]	HTF specific heat
c_{pp}	[J/kg^{o}C]	PCM specific heat
c_{pt}	[J/kg^{o}C]	Tube material specific heat
D	[m]	Tube diameter related to PCM
D_p	[m]	Inner diameter of outer shell-tube
f_m	[-]	PCM melt fraction
h	[W/m^{2o}C]	Heat transfer coefficient from HTF to PCM
k	[W/m ^{o}C]	PCM thermal conductivity
L	[J/kg]	PCM latent heat
M	[kg]	PCM mass
M_t	[kg]	Mass of tube material
$\dot{m}$	[kg/s]	HTF flow rate
Q	[J]	Absorbed heat
Q_0	[J]	Overall absorbed heat
q'	[W/m]	Heat transfer rate per unit length
q_0	[W]	Overall heat transfer rate
$q_{\max}$	[W]	Maximum overall heat transfer rate
T	[^{o}C]	HTF temperature
T_{in}	[^{o}C]	Inlet HTF temperature
T_m	[^{o}C]	PCM melting temperature
t	[s]	Time
t_i	[s]	Time of complete melting of initial cross-section
v	[m/s]	HTF velocity
X	[m]	Length of PCM unit
x	[m]	Current coordinate

Greek letters

ρ	[kg/m^3]	PCM density
τ	[-]	Dimensionless time
τ_{pc}	[-]	Time of complete melting of particular cross-section
τ_0	[-]	Overall time of complete melting

Auxiliary values

θ, ϕ	[-]	Auxiliary dimensionless functions of time
h_0, h_f	[W/m^{2o}C]	Constants of heat transfer coefficient dimension
w, b, b_1, b_2	[-]	Dimensionless constants

PHYSICAL MODEL AND ANALYTICAL SOLUTION

We analyze shell-and-tube systems with PCM stored in the shell whereas HTF passes through the tube, or vice versa. We follow the classification in terms of geometry and configuration as defined by Agyenim et al. [2], ***pipe*** and ***cylinder*** models (Figure 1). The system is placed initially at the melting temperature $\boldsymbol{T_m}$ with PCM in solid state. HTF is supplied to the tube or to the shell. Along the entire process of melting, HTF is at the known and constant inlet temperature $\boldsymbol{T_{in} > T_m}$ and flow rate $\dot{m}$.

The mathematical justification of the analytical model is presented in [17]. Here we present the initial equations and the final formulas of the analytical calculation of the process parameters.

The HTF energy equation:

$$\left(\dot{m}c_p \frac{\partial T}{\partial x}\right)_{HTF} = -q' \qquad (1)$$

Relation between heat transfer rate and local PCM melt fraction:

$$\frac{\partial f_m}{\partial t} = \frac{q'X}{Q_0} \qquad (2)$$

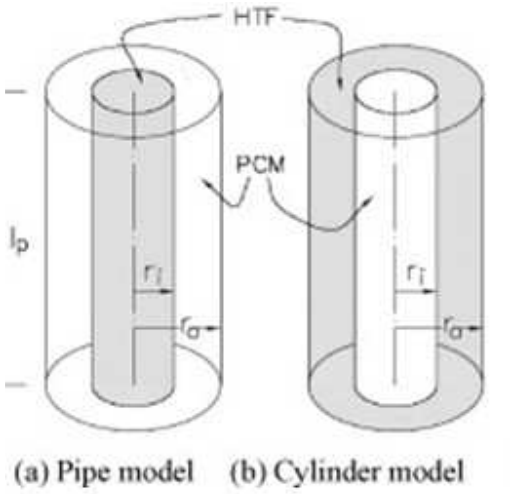


Figure 1. Classification of used PCM containers [2]

The instantaneous balance is given by:

$$q' = \frac{A/X}{1/h + D/4k \log(1+wf_m)}(T - T_m) \tag{3}$$

for pipe model, and

$$q' = \frac{A/X}{1/h - D/4k \log(1-f_m)}(T - T_m) \tag{3'}$$

for cylinder model. Here, h is an average heat transfer coefficient from HTF to PCM, related to the surface A, and taking into account the thermal resistance of the tube wall.

This system of two partial differential and one algebraic equations is subject to the following conditions:

$$f_m(x, t=0) = 0$$
$$T(x=0, t) = T_{in} \tag{4}$$

The following structural and modal constants are used:

$$w = \left(\frac{D_p}{D}\right)^2 - 1; \quad \textit{for pipe systems only}$$

$$h_0 = \left\{\frac{1}{h} + \frac{D}{4k}\left[\left(1+\frac{1}{w}\right)\log(1+w) - 1\right]\right\}^{-1}; \ \textit{for pipe}$$

$$h_0 = \left[\frac{1}{h} + \frac{D}{4k}\right]^{-1}; \quad \textit{for cylinder} \tag{5}$$

$$h_f = \frac{\dot{m}c_p}{A};$$

$$t_i = \frac{Q_0}{A(T_{in} - T_m)h_0};$$

$$b = \log\left[1 + h\frac{D}{4k}\log(1+w)\right]; \ \textit{for pipe systems}$$

$$b = \log\left[1 + h\frac{D}{k}\right]; \ \textit{for cylinder systems} \tag{6}$$

$$b_1 = \frac{b}{1-\exp(-b)}; \quad b_2 = \exp\left(b_1\frac{h_0}{h_f}\right) - 1.$$

Note that t_i is the time of complete melting of initial cross-section.

Solution of the system of equations is presented in non-dimensional form with respect to the non-dimensional independent variables x/X and $\tau = t/t_i$.

PCM melt fraction:

$$f_m(x,\tau) = \begin{cases} \dfrac{b_1[1-\exp(-b\tau)]}{b\left\{1+\exp(-b\tau)\left(\exp\left[b_1\dfrac{h_0}{h_f}\dfrac{x}{X}\right]-1\right)\right\}}, & \tau \le 1 \\ \dfrac{1}{1+\exp(-b)\left\{\exp\left(b_1\left[\dfrac{h_0}{h_f}\dfrac{x}{X} - (\tau-1)\right]\right)-1\right\}}, & 1 \le \tau \le \tau_{pc}(x) \\ 1, & \tau_{pc}(x) \le \tau \le \tau_0 \end{cases} \tag{7}$$

where time of complete melting for the x cross-section $\tau_{pc}(x)$, as well as time of complete unit melting τ_0 are the following:

$$\tau_{pc}(x) = 1 + \frac{h_0}{h_f}\frac{x}{X} \qquad \tau_0 = 1 + \frac{h_0}{h_f} \tag{8}$$

The overall PCM unit parameters are as follows.

Non-dimensional heat transfer rate:

$$q_0(\tau) = \frac{1}{q_{\max} t_i}\frac{dQ}{d\tau} = \begin{cases} \dfrac{b_2\exp(-b\tau)}{1+b_2\exp(-b\tau)}, & \tau \le 1 \\ \dfrac{\theta - 1}{\theta}, & 1 \le \tau \le \tau_0 \end{cases} \tag{9}$$

Correspondingly, heat transfer rate q_{max} is determined as the maximum possible HTF cooling:

$$q_{\max} = \dot{m}c_p(T_{in} - T_m) = \frac{Q_0}{t_i}\frac{h_f}{h_0}; \tag{10}$$

Heat absorbed by the entire unit from the beginning of the heat transfer process up to time τ in non-dimensional form:

$$\frac{Q(\tau)}{Q_0} = \begin{cases} \dfrac{1}{b}\left[b_1 - \dfrac{h_f}{h_0}\log\{1+b_2\exp(-b\tau)\}\right], & \tau \le 1 \\ \dfrac{1}{b}\left[b_1\phi - \dfrac{h_f}{h_0}\log(\theta)\right], & 1 \le \tau \le \tau_0 \end{cases} \tag{11}$$

In relations (9) and (11) we denote:

$$\theta(\tau) = 1 + \exp(-b)\left\{\exp\left(b_1\left[\frac{h_0}{h_f} - (\tau-1)\right]\right) - 1\right\} \tag{12}$$

$$\phi(\tau) = 1 - \exp(-b)\frac{h_f}{h_0}(\tau - 1)$$

NUMERICAL SIMULATION

The following shell-and-tube structure was used for the calculation: X = 0.5 m and 1m, aluminium tube with outer diameter of 0.012m and 1mm thick, where shell diameter is D_p=0.016m. For the two-dimensional CFD FLUENT calculations the 40000 Gambit-built Cartesian axisymmeric elements were adopted with linear size of 0.2-0.5 mm of each.

Three-dimensional case of a block with PCM continuously placed around the same tubes of length 1m, where HTF flows inside, was analyzed too. The tubes were placed in a square arrangement with the distance between axes of 25 mm. For the FLUENT calculations, 1/8 (triangular) part of a unit cross-section has been selected, with a total amount of about 350000 elements of mixed cells.

Air was used as HTF, with T_{in}=35°C and velocity 1, 3.5 and 7 m/s both for pipe and cylinder cases. PCM with melting temperature T_m=23°C, density 760 kg/m^3, k=0.2W/m°C and latent heat L=206 (104 for 3-D case) kJ/kg has been used. In order to describe the PCM melting, the so-called "volume-of-fluid" (VOF) method has been used. For the phase-change region, enthalpy-porosity approach is used, by which the porosity in each cell is set equal to the liquid fraction in that cell. Accordingly, the porosity is zero in fully solid regions.

In contrast to the analytical model, in the development of which the serious simplifications listed in the Introduction have been made, CFD FLUENT calculations take into account a three-dimensional non-concentric PCM melting process. Further, the sensible heat is included. Thus, a real transitional three-dimensional heat transfer process is analyzed. Figure 2 illustrates an example of such melting. Calculation of real temperature fields in all materials is made, too. A typical example is shown in Figure 3. Thus, the CFD FLUENT calculations are fully consistent with the heat transfer and melting processes, and can be used for the validation of analytical relationships.

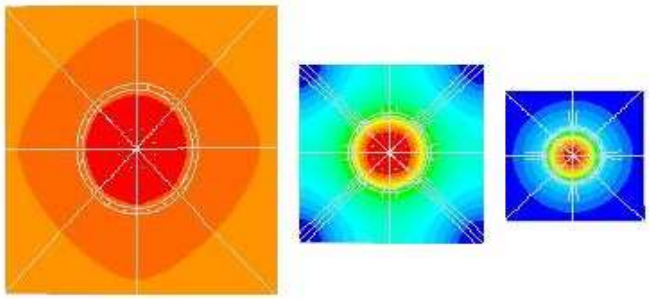


Figure 2. 3-D cross-sections temperature field for x=10, 50, 90 cm (left to right). Time 3500s for air velocity 3.5 m/s. Note that size distortion is associated with isometric view.

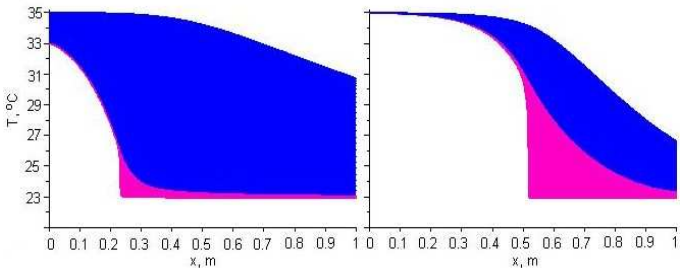


Figure 3. 2-D temperature field for pipe model (left), and cylinder model (right). Time 3000s for air velocity 3.5 m/s in both cases. Air coloured blue, PCM coloured red.

COMPARISON WITH ANALYTICAL RESULTS

It was clear that the greatest distortions in the analytical results compared with the actual melting process relates to assumption 3 listed in the Introduction (neglecting of sensible heat). This is evident from the fact that this assumption violates the overall energy balance of the process. Accordingly, a method of saving energy balance of the process has been proposed even in [1] through the use of efficient latent heat (Eq. (13)). This method has been used in analytical calculations of real melting processes.

$$L_{eff} = L + \left(c_{pp} + c_{pt} \frac{M_t}{M} \right) (T_{in} - T_m) \qquad (13)$$

Heat transfer coefficient between air and PCM, h, has been calculated in accordance with table 6.1 and relation 6.6 by Holman [18], also taking into account the tube thermal resistance.

Aluminum as tubes material with high thermal conductivity was chosen to maximize detection of the error introduced in the analytical results by assumption 6. Naturally, the error caused in the analytical result due to neglecting axial heat transfer for materials with lower thermal conductivity is even smaller.

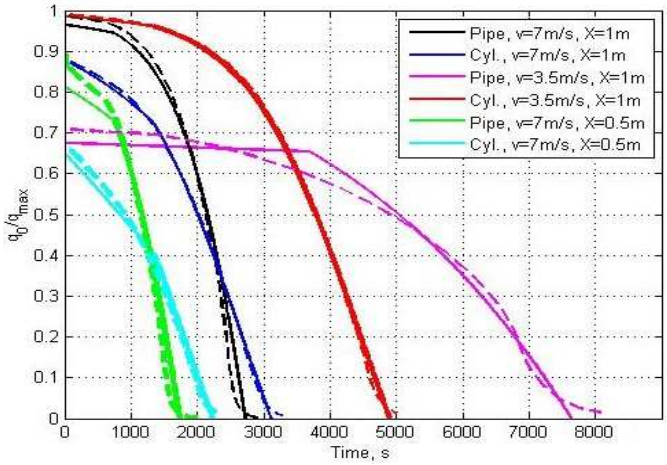
Figure 4 presents comparison of analytical and FLUENT overall heat transfer results for 2-D shell-and tube systems.

Figure 5 presents comparison of analytical and FLUENT results for 3-D system of PCM block around square arrangement of tubes. For analytic calculation, size D_p has been selected for preserving condition of cross-section area.

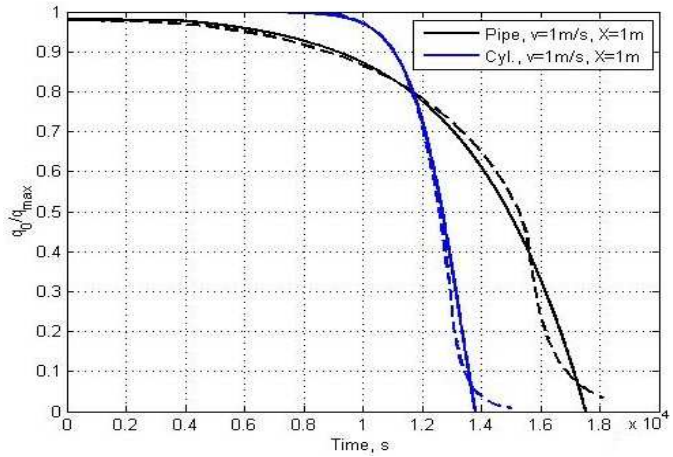
A very good agreement is observed between the analytical and numerical predictions in spite of serious analytical assumptions. The best agreement is obtained for shell-and-tube cylinder model, and very good agreement is obtained for the shell-and-tube pipe model. Somewhat worse correspondence is shown for 3-D PCM-block system, however, it is sufficient for engineering calculations.

A characteristic feature of the analytical results, that is clearly visible in all the Figures, is a broken relationship (an abrupt change in the derivative) at time τ = 1. This is obviously due to the specific analytical model, which at the end of melting in any cross-section, the local heat flux value vanishes abruptly. Of course, such abrupt change in local heat flux in the real process is not observed, which clearly shows a comparison of the local results of analytical model with CFD FLUENT, where aliasing occurs due PCM and tube sensible heat (Figure 6). Using in the analytical calculation of effective latent heat value (Eq. (13)) ensures the total energy balance, but can not eliminate this fundamental feature of the analytical model.

Comparison of analytic and FLUENT calculation for water flow is shown in Figure 7. Water flows at velocities of 1 m/s and 0.1 m/s were analyzed for the same shell-and-tube. Assumption 7 listed in Introduction is fundamentally incorrect for water flow, contrary to airflow case. Nevertheless, except for the initial part of process, heat transfer rate is described satisfactorily by the analytical prediction. Most important is that the final time of the process is predicted exactly. So, the simplest relations, Eq. (8) may be used in water case, too.



a) Air velocities 7 and 3.5 m/s



b) Air velocity 1 m/s

Figure 4. Comparison of analytical results (solid curves) and FLUENT results (dashed curves). Non-dimensional heat transfer rate of whole heat unit is presented versus time of process.

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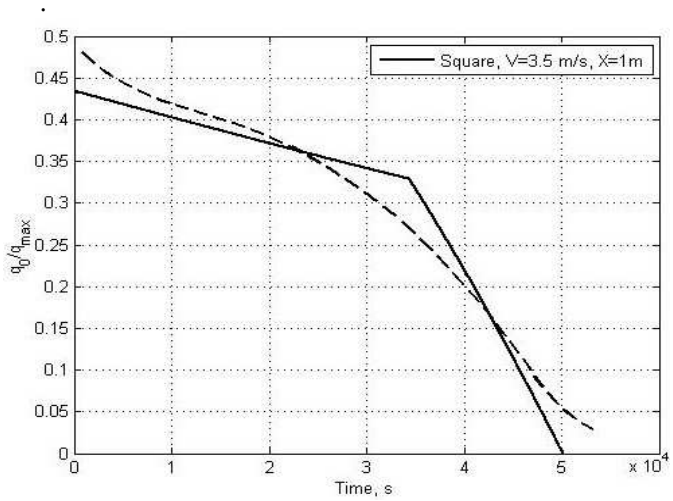


Figure 5. Comparison of analytical results (solid curves) and FLUENT results (dashed curves). Non-dimensional heat transfer rate of whole 3-D heat unit is presented vs. time of the process.

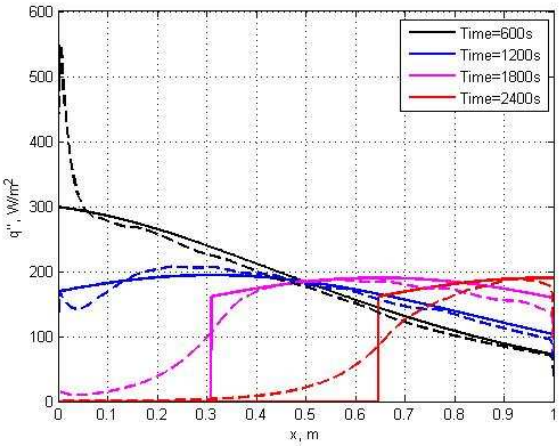
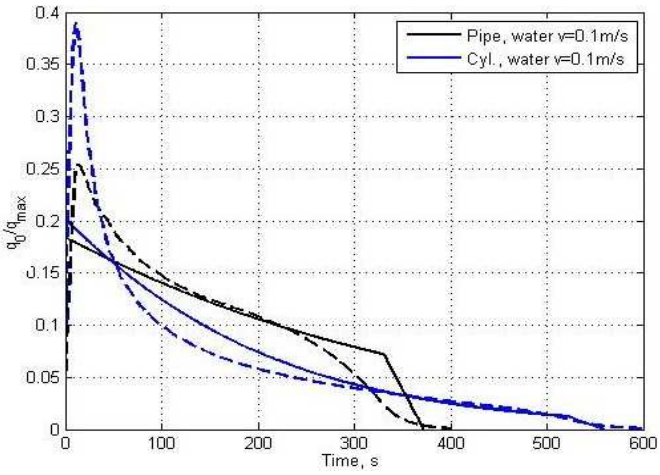
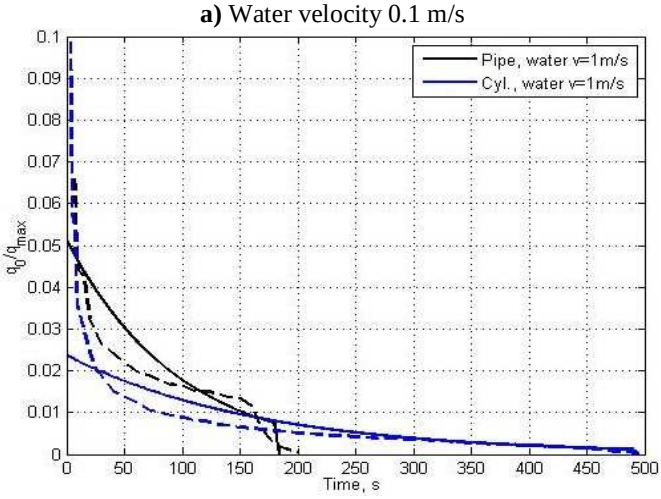


Figure 6 Comparison of analytical results (solid curves) and FLUENT results (dashed curves). Local heat flux is presented versus coordinate *x* for different times. Cylinder model, air flows with v=7 m/s.



a) Water velocity 0.1 m/s



b) Water velocity 1 m/s

Figure 7. Comparison of analytical results (solid curves) and FLUENT results (dashed curves). Water flow. Non-dimensional heat transfer rate of the entire shall-and-tube unit is presented versus time of process.

CONCLUSIONS

An analytical analysis of a shell-and-tube heat transfer unit which utilizes the latent heat of a phase-change material (PCM) has been presented. In the unit, the PCM melts inside the tube, while air flows in the shell, or PCM melts outside the tube, while air flows in the tube. Our method is applicable to both these cases. A system of partial differential equations, which describes heat transfer and melting of the PCM and heat transfer in the air, has been derived with some assumptions, and solved analytically. Simple formulas are obtained for the overall heat exchange parameters, like heat transfer rate, stored energy and overall heat transfer time.

An analytical solution of the system of equations has been compared with the results of a melting numerical calculation by the CFD FLUENT, where assumptions of analytical model are absent. A very good agreement has been demonstrated.

Thus, this analytical model can be used not only for optimization of the design and parameters of systems, but for a concrete calculation of real PCM melting process, and obtain heat transfer and melting parameters in the form of algebraic formulas.

The same conclusion can be drawn with respect to the analytical calculations of the tubular heat exchangers with air cross-flow across the tube banks where PCM melts inside tubes, Dubovsky et al., [1]. This obvious conclusion follows from the fact that there has been used completely analogous mathematical model with exactly the same set of assumptions.

In addition, the three-dimensional case with the non-concentric PCM melting has been considered. PCM was placed around the tubes and air was flowing inside. Results of analytical calculations of PCM melting were compared with exact FLUENT numerical simulation. It has demonstrated a satisfactory agreement, although the accuracy of the analytical solutions is much lower than for systems with two-dimensional PCM concentric melting.

For the case with water flow instead of air flow, the exactness of analytical model obviously shows low precision. Nevertheless, except for the initial part of the process, heat transfer rates are described satisfactorily by the analytical prediction. Most important is that the final time of the process is predicted exactly. Thus, the general simplest relations may be in use in this case, too.

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