A NUMERICAL AND EXPERIMENTAL STUDY OF TRANSIENT HEAT TRANSFER IN A SHELL-AND-TUBE LATENT HEAT STORAGE UNIT WITH PARAFFIN AS A PHASE CHANGE MATERIAL

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Abstract: *A physical process of transient heat transfer in the shell-and-tube latent heat storage unit has been analysed numerically and experimentally in this paper. Heat transfer considered in the paper is the conjugate problem of transient forced convection between heat transfer fluid and the wall, heat conduction through the wall and heat exchange of the phase change material. Governing dimensionless differential equations for heat transfer fluid, wall and phase change material with initial and boundary conditions have been discretised by a control volume approach and then solved by Fortran software using an iterative procedure. The numerical model is validated with experimental data. Commercial paraffin RUBITHERM RT 30 has been used in experimental investigations as a phase change material. Mutual agreement has been established between numerically and experimentally obtained timewise temperature variations. Unsteady temperature distributions of heat transfer fluid, wall and phase change material have been obtained by numerical calculations and thermal behaviour of latent heat storage unit during charging and discharging has been simulated.*

Key words: transient heat transfer, latent heat storage unit, numerical analysis, experiment

1. INTRODUCTION

Interest in utilizing renewable energy sources is growing due to efforts of rational and effective energy management and because of environmental considerations. In this field, solar energy has special significance, but due to its periodic nature, a thermal energy storage device has to be used. A latent heat storage system with solid-liquid phase change has some advantages like high storage density and short temperature interval of heat transfer.
The latent heat storage unit analysed in this paper is a shell-and-tube type of heat exchanger with the phase change material filling the shell side. Heat transfer in the thermal energy storage system of this type is a time dependent conjugate phase change – convection problem. A one-dimensional phase change problem was first described by Stefan in 1889, and because of that, it was named the Stefan problem. A class of Stefan problems for which analytic solutions exist is small. Numerical schemes require that moving phase change boundary be accurately treated. One method for a solution of this problem has been described in paper [1] introducing the enthalpy method to describe heat diffusion inside the phase change material. In this method, the energy equation for the phase change material was written in terms of the enthalpy. The standard enthalpy method has been used in mathematical approaches by many authors [2] - [6]. Belleci and Conti in papers [7], [8] and [9], using the enthalpy method,

numerically studied a solar receiving shell-and tube heat exchanger. They treated the fluid flow inside the tube as steady fully developed and employed standard correlations to calculate the convective heat transfer coefficient. The same type of latent heat exchanger was studied by Lacroix in papers [10], [11] and [12]. Cao and Faghri in [13] and [14] numerically simulated the thermal behaviour of shell-and-tube heat exchanger employing a low Prandtl number heat transfer fluid for space application. The transient fluid flow momentum and energy equations were solved simultaneously with the tube wall and phase change material energy equations. For phase change material, the temperature transforming model was used. They concluded that using a steady fully developed heat transfer correlation to calculate the convective heat transfer coefficient inside the tube would introduce significant errors in the results. A temperature transforming model for describing melting of the phase change material was also used by Zhang and Faghri in [15]. In paper [16] Zhang and Faghri described a semi-analytical model for a shell-and-tube latent heat storage unit. The flow inside the tube was treated as steady thermally developing with constant velocity profile and a one-dimensional integral method was used for solving the phase change material energy equation. They concluded that the laminar forced flow inside the tube never reached a thermally developed state. In paper [17] a transient phase change heat transfer with conjugate forced convection in the shell-and-tube latent heat storage unit, with water as a heat transfer fluid and calcium chloride hexahydrate as the phase change material was analysed numerically.
In this paper, a mathematical model of transient heat transfer in shell-and-tube latent heat storage unit has been formulated. The dimensionless transient fluid flow continuity, momentum and energy equations were solved simultaneously with the tube wall and phase change material energy equations. The enthalpy method has been used for describing heat transfer inside the phase change material. Differential equations, with initial and boundary conditions, have been discretised by control volume approach and then solved by Fortran software using an iterative procedure. The numerical model has been validated with experimental data obtained by experimental investigations performed in the Laboratory for thermal measures at the Faculty of Engineering University of Rijeka.

2. MATHEMATICAL FORMULATION OF TRANSIENT HEAT TRANSFER IN A STORAGE UNIT

The shell-and tube latent heat storage unit considered in this paper is shown in Figure 1.

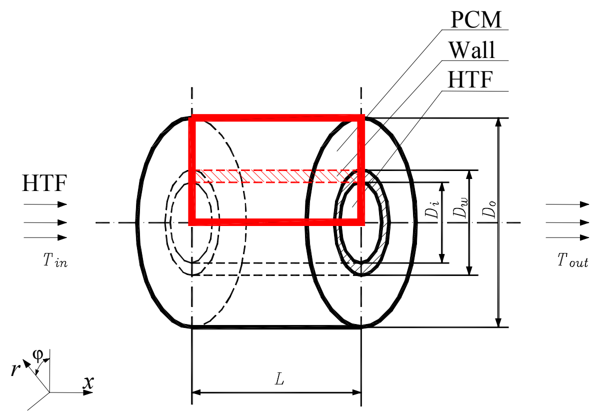

Figure 1. Latent heat storage unit

A heat transfer fluid (HTF) flows through the inner tube and exchanges heat with the phase change material (PCM) on the shell side. During sunlight, i.e. active phase, hot fluid heats the

PCM, the PCM melts and the heat is stored. During the eclipse phase, the PCM solidifies and the stored heat is delivered to the cold fluid.
To establish a convenient mathematical model of transient heat transfer, the following assumptions have been introduced:

- the heat transfer fluid is incompressible and it can be considered as a Newtonian fluid,
- flow of the HTF is laminar,
- the initial temperature of the latent heat storage unit is uniform and the PCM is in the solid phase,
- inlet velocity and inlet temperature of the HTF are constant,
- thermal losses and conduction through the outer wall of the storage unit have been ignored, i.e. adiabatic outer wall is assumed,
- fluid – wall convective heat transfer, heat conduction through the wall and phase change heat transfer can be considered as an unsteady two-dimensional problem,
- the problem is axisymmetric,
- physical properties of the HTF, wall and PCM are constant,
- natural convection in the liquid phase of the PCM has been ignored.

According to these assumptions, it follows that the domain for the mathematical description of the transient heat transfer in considered latent heat storage unit is a bolded, framed part in Figure 1.

The dimensionless continuity, momentum and energy equations, governing a transient two-dimensional problem of flow and heat transfer in a latent heat storage unit, for heat transfer fluid, wall and phase change material are as follows

- HTF

$$\frac{\partial W_X}{\partial X}+\frac{1}{R}\cdot\frac{\partial\left(R\cdot W_R\right)}{\partial R}=0 \tag{1}$$

$$\frac{\partial W_X}{\partial \tau}+W_X\cdot\frac{\partial W_X}{\partial X}+W_R\cdot\frac{\partial W_X}{\partial R}=-\frac{\partial P}{\partial X}+\frac{1}{\mathrm{Re}}\cdot\left[\frac{\partial^2 W_X}{\partial X^2}+\frac{1}{R}\cdot\frac{\partial}{\partial R}\left(R\cdot\frac{\partial W_X}{\partial R}\right)\right] \tag{2}$$

$$\frac{\partial W_R}{\partial \tau}+W_X\cdot\frac{\partial W_R}{\partial X}+W_R\cdot\frac{\partial W_R}{\partial R}=-\frac{\partial P}{\partial R}+\frac{1}{\mathrm{Re}}\cdot\left[\frac{\partial^2 W_R}{\partial X^2}+\frac{1}{R}\cdot\frac{\partial}{\partial R}\left(R\cdot\frac{\partial W_R}{\partial R}\right)-\frac{W_R}{R^2}\right] \tag{3}$$

$$\frac{\partial \Theta_\mathrm{f}}{\partial \tau}+W_X\cdot\frac{\partial \Theta_\mathrm{f}}{\partial X}+W_R\cdot\frac{\partial \Theta_\mathrm{f}}{\partial R}=\frac{1}{\mathrm{Re}\cdot\mathrm{Pr}}\cdot\left[\frac{\partial^2 \Theta_\mathrm{f}}{\partial X^2}+\frac{1}{R}\cdot\frac{\partial}{\partial R}\left(R\cdot\frac{\partial \Theta_\mathrm{f}}{\partial R}\right)\right] \tag{4}$$

- Wall

$$\frac{\partial \Theta_\mathrm{w}}{\partial \tau}=\frac{1}{\mathrm{Re}\cdot\mathrm{Pr}}\cdot\frac{a_\mathrm{w}}{a_\mathrm{f}}\cdot\left[\frac{1}{R}\cdot\frac{\partial}{\partial R}\cdot\left(R\cdot\frac{\partial \Theta_\mathrm{w}}{\partial R}\right)+\frac{\partial^2 \Theta_\mathrm{w}}{\partial X^2}\right] \tag{5}$$

- PCM

$$\frac{\partial \chi_\mathrm{p}}{\partial \tau}=\frac{1}{\mathrm{Re}\cdot\mathrm{Pr}}\cdot\frac{a_\mathrm{p}}{a_\mathrm{f}}\cdot\left[\frac{1}{R}\cdot\frac{\partial}{\partial R}\left(R\cdot\frac{\partial \chi_\mathrm{p}}{\partial R}\right)+\frac{\partial^2 \chi_\mathrm{p}}{\partial X^2}\right] \tag{6}$$

where χ is the dimensionless enthalpy related to the temperature with equation $\Theta_{\mathrm{p}} = A_{\mathrm{b}} \cdot \chi_{\mathrm{p}} + B_{\mathrm{b}}$ where factors A_{b} and B_{b} are

$$A_{\mathrm{b}} = \frac{1}{\mathrm{St}} \cdot \frac{\rho_{\mathrm{L}} \cdot c_{\mathrm{L}}}{\rho_{\mathrm{s}} \cdot c_{\mathrm{s}}}, \qquad B_{\mathrm{b}} = 0 \qquad \text{for} \quad \chi_{\mathrm{p}} < 0,$$

$$A_{\mathrm{b}} = 0, \qquad B_{\mathrm{b}} = 0 \qquad \text{for} \quad 0 \le \chi_{\mathrm{p}} \le 1,$$

$$A_{\mathrm{b}} = \frac{1}{\mathrm{St}}, \qquad B_{\mathrm{b}} = -\frac{1}{\mathrm{St}} \qquad \text{for} \quad \chi_{\mathrm{p}} > 1.$$

The equations are obtained using dimensionless variables defined as

- dimensionless coordinates $R = \dfrac{r}{D_{\mathrm{i}}}$ and $X = \dfrac{x}{D_{\mathrm{i}}}$,
- dimensionless velocities $W_X = \dfrac{w_x}{w_{\mathrm{in}}}$ and $W_R = \dfrac{w_r}{w_{\mathrm{in}}}$,
- dimensionless pressure $P = \dfrac{p - p_0}{\rho_{\mathrm{f}} \cdot w_{\mathrm{in}}^2}$,
- dimensionless time $\tau = \dfrac{w_{\mathrm{in}}}{D_{\mathrm{i}}} \cdot t$,
- dimensionless temperature $\Theta = \dfrac{T - T_{\mathrm{m}}}{T_{\mathrm{in}} - T_{\mathrm{m}}}$,
- dimensionless enthalpy $\chi = \dfrac{H - \rho_{\mathrm{s}} \cdot c_{\mathrm{s}} \cdot T_{\mathrm{m}}}{\rho_{\mathrm{L}} \cdot q}$,
- Reynolds, Prandtl, Stefan and Nusselt number

$$\mathrm{Re} = \frac{w_{\mathrm{in}} \cdot D_{\mathrm{i}}}{\upsilon_{\mathrm{f}}}, \ \mathrm{Pr} = \frac{\upsilon_{\mathrm{f}}}{a_{\mathrm{f}}}, \ \mathrm{St} = \frac{c_{\mathrm{L}} \cdot (T_{\mathrm{in}} - T_{\mathrm{m}})}{q}, \ \mathrm{Nu} = \frac{\alpha \cdot D_{\mathrm{i}}}{\lambda_{\mathrm{f}}}.$$

Initial and boundary conditions are as follows:

- initial condition $\tau = 0$

$$0 < R \le 0{,}5, \quad 0 \le X \le L / D_{\mathrm{i}} \quad \Rightarrow \quad W_X = W_R = 0$$

$$0 < R < R_{\mathrm{o}}, \quad 0 \le X \le L / D_{\mathrm{i}} \quad \Rightarrow \quad \Theta_{\mathrm{f}} = \Theta_{\mathrm{w}} = \Theta_{\mathrm{p}} = \Theta_{\mathrm{init}}$$

- boundary conditions $\tau > 0$

inlet plane $X = 0$

$$0 < R < 0{,}5 \quad \Rightarrow \quad W_X = 1, \ W_R = 0, \ \Theta_{\mathrm{f}} = 1$$

$$0{,}5 \le R \le R_{\mathrm{w}} \quad \Rightarrow \quad \frac{\partial \Theta_{\mathrm{w}}}{\partial X} = 0$$

$$R_{\mathrm{w}} < R < R_{\mathrm{o}} \quad \Rightarrow \quad \frac{\partial \Theta_{\mathrm{p}}}{\partial X} = 0$$

outlet plane $X = L / D_{\mathrm{i}}$

$$0 < R < 0{,}5 \quad \Rightarrow \quad \frac{\partial W_X}{\partial X} = 0, \ \frac{\partial W_R}{\partial X} = 0, \ \frac{\partial \Theta_{\mathrm{f}}}{\partial X} = 0$$

$$0{,}5 \le R \le R_{\mathrm{w}} \quad \Rightarrow \quad \frac{\partial \Theta_{\mathrm{w}}}{\partial X} = 0$$

$$R_{\mathrm{w}} < R < R_{\mathrm{o}} \quad \Rightarrow \quad \frac{\partial \Theta_{\mathrm{p}}}{\partial X} = 0$$

axe of symmetry $R = 0$

$$0 < X < L / D_{\mathrm{i}} \quad \Rightarrow W_R = 0 \, , \ \frac{\partial W_X}{\partial R} = 0 \, , \ \frac{\partial \Theta_{\mathrm{f}}}{\partial R} = 0$$

fluid – wall interface $R = 0{,}5$

$$0 < X < L / D_{\mathrm{i}} \quad \Rightarrow \quad W_X = W_R = 0, \quad \left(\frac{\partial \Theta_{\mathrm{f}}}{\partial R} \right)_{R=0,5} = \frac{\lambda_{\mathrm{w}}}{\lambda_{\mathrm{f}}} \cdot \left(\frac{\partial \Theta_{w}}{\partial R} \right)_{R=0,5}$$

wall – PCM interface $R = R_{\mathrm{w}}$

$$0 < X < L / D_{\mathrm{i}} \quad \Rightarrow \quad \left(\frac{\partial \Theta_{\mathrm{w}}}{\partial R} \right)_{R=R_{\mathrm{w}}} = \frac{\lambda_{\mathrm{p}}}{\lambda_{\mathrm{w}}} \cdot \left(\frac{\partial \Theta_{\mathrm{p}}}{\partial R} \right)_{R=R_{\mathrm{w}}}$$

outer wall $R = R_{\mathrm{o}}$

$$0 < X < L / D_{\mathrm{i}} \quad \Rightarrow \quad \frac{\partial \Theta_{\mathrm{p}}}{\partial R} = 0 \, .$$

3. NUMERICAL PROCEDURE

A mathematical model, i.e. differential equations, with initial and boundary conditions, has been discretised using the control volume method described by Patankar [18]. The velocities and pressure of heat transfer fluid are solved by using the SIMPLER scheme [18]. The resulting discretisation equations have been solved simultaneously using Gauss-Seidel iterative procedure. Due to nonlinearity of the problem, iterations are needed during each time step. A convergence criterion is set at 0.01% for all variables of the system. A system of algebraic equations has been solved using *Fortran* software. Unsteady temperature distributions of heat transfer fluid, wall and phase change material have been obtained by numerical calculations and thermal behaviour of latent heat storage unit during charging and discharging has been simulated.

4. EXPERIMENTAL SETUP AND VALIDATION

The computational model has been validated with experimental data. An experimental latent heat storage unit was constructed and a series of experiments were performed. An experimental test unit and thermocouple position inside the unit are shown in Figures 2 and 3. The experimental test unit consists of two concentric tubes. The inside tube (0.033 m i.d., 0.035 m o.d. and 1 m long) is made of copper while the outside tube (0.128 m i.d., 0.133 m o.d. and 1 m long) is made of brass. The outside tube is well insulated with commercial pipe insulation *K-FLEX ST H*. The HTF is water and it circulates through the inside tube. The space between both tubes is filled with phase change material. Since paraffins have been widely used in low temperature thermal storage systems, the PCM used in experiments is commercial paraffin *RUBITHERM RT 30.* Thermophysical properties of the paraffin *RUBITHERM RT 30* are as follows:

- melting temperature $T_m = 300.7$ K,
- latent heat $q = 206$ kJ/kg,
- thermal conductivity
  - solid phase $\lambda_s = 0.18$ W/(mK),
  - liquid phase $\lambda_L = 0.19$ W/(mK),
- specific heat
  - solid phase $c_s = 1.8$ kJ/(kgK),
  - liquid phase $c_L = 2.4$ kJ/(kgK),
- density
  - solid phase $\rho_s = 789$ kg/m^3,
  - liquid phase $\rho_L = 750$ kg/m^3.

Sixteen K-type thermocouples are installed inside the paraffin at various locations as shown in Figure 3. Two additional thermocouples are place at the inlet and outlet of the inside tube. All thermocouples are connected to a data acquisition system. To minimize the effects of natural convection on melting, i.e. to maintain axisymmetric melting around the inside tube, the test unit is oriented in the vertical direction.

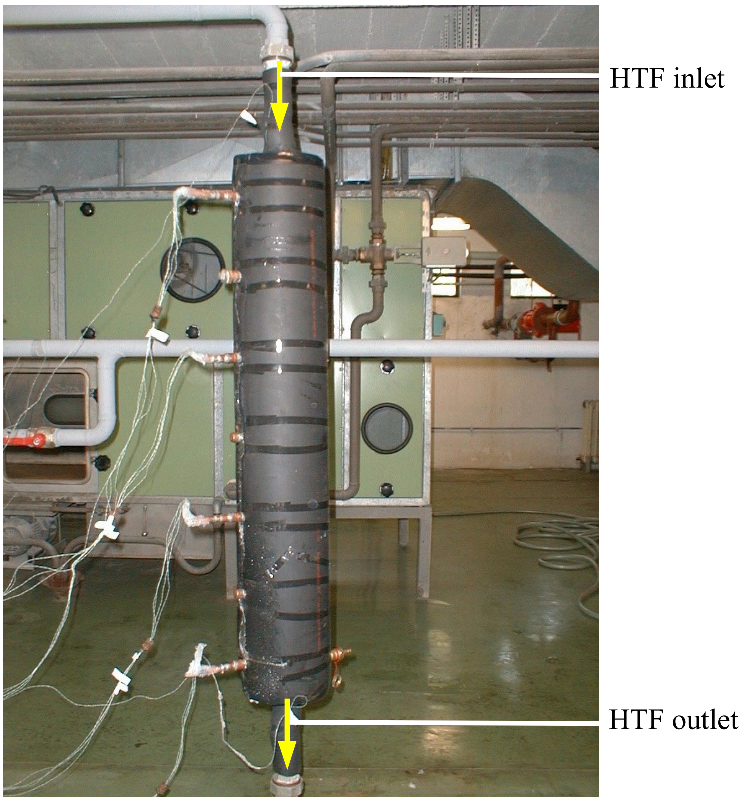

Figure 2. Experimental test unit

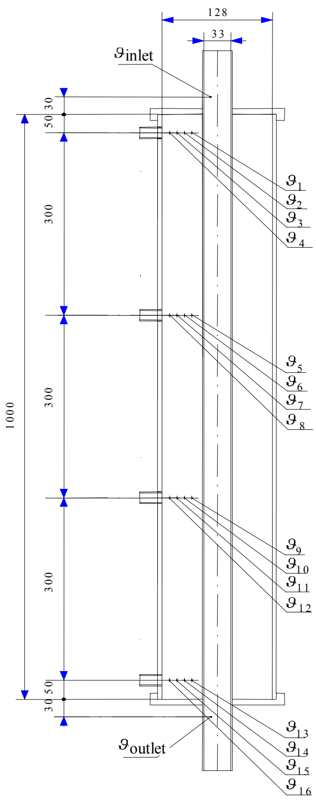

Figure 3. Thermocouples position in experimental test unit

Several melting and solidification experiments were performed for different mass flow rates and inlet temperatures of the heat transfer fluid. The computational model was set up to reproduce these experimental conditions. Numerical simulations were carried out for a grid size of 250 nodes in the axial direction and 73 nodes in a radial direction that includes HTF, wall and PCM, and a fixed time step of 0.1 s. Figures 4 and 5 show the timewise variations of the measured and calculated values of the PCM temperature at location 9 (x = 0.65 m and r = 0.0265 m) during melting of the PCM and at location 15 (x = 0.95 m and r = 0.0445 m) during solidification of the PCM. Mass flow rates and inlet temperatures of the HTF as well as inlet temperatures of the PCM are indicated on diagrams.

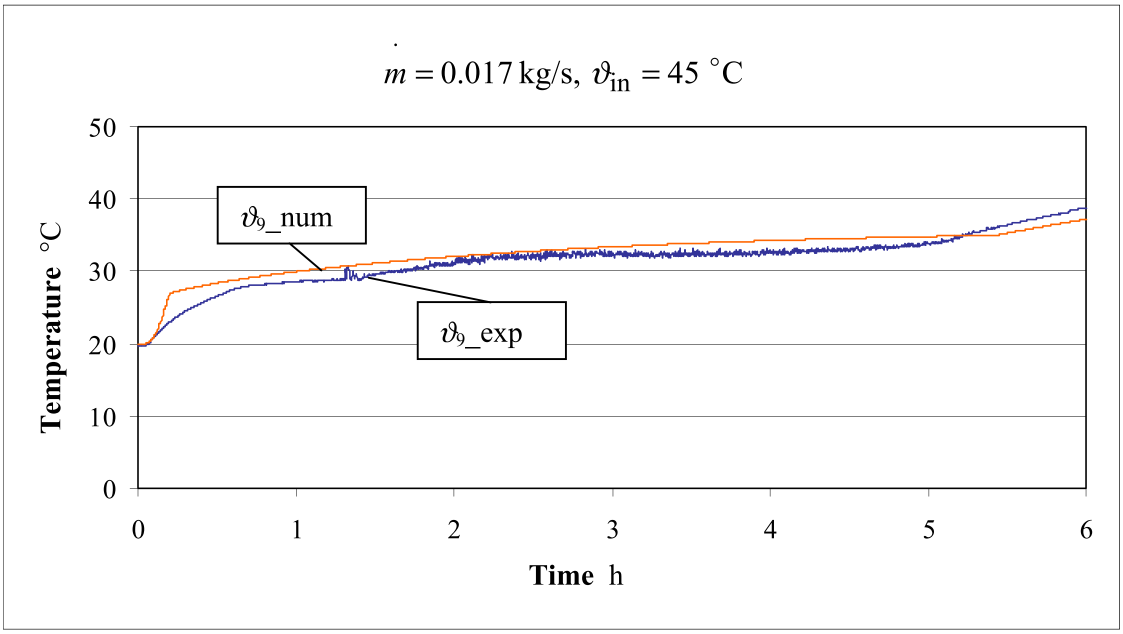

Figure 4. Timewise variation of calculated and measured values of the PCM temperature at location 9 during melting

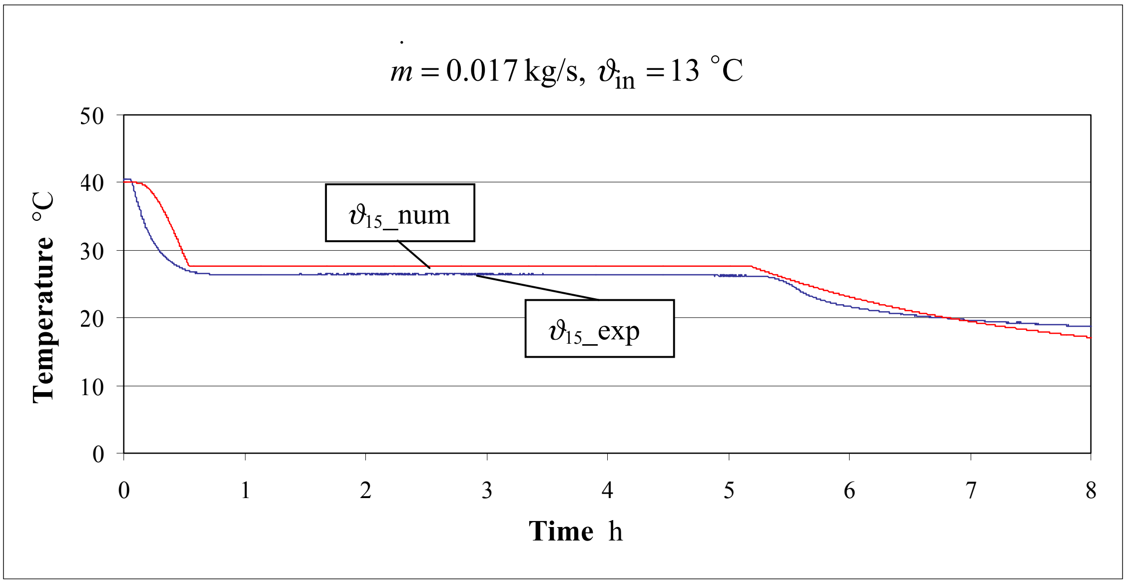

Figure 5. Timewise variation of calculated and measured values of the PCM temperature at location 15 during solidification

From the figures, it can be seen that agreement between calculated and measured values is well within certain experimental uncertainty. It was noticed that melting of used paraffin was performed within short temperature interval (not at a constant temperature), so the computational model has been modified. It can be concluded from the results obtained that a demonstrated numerical procedure could be efficiently used for simulation of transient heat transfer in a shell-and tube latent heat storage unit.

5. NUMERICAL RESULTS

A series of numerical calculations were performed in order to analyse transient heat transfer during melting and solidification of commercial paraffin used in experimental investigations. Some of the results obtained for selected case are illustrated below.

Melting of the PCM, i.e. storing of energy, has been observed. The mass flow rate of the HTF is 0.017 kg/s and inlet temperature of the HTF is 318 K. PCM is initially at the solid phase with temperature of 293 K. Corresponding Reynolds and Stefan numbers are Re = 657 and St = 0.2. The numerical results obtained show that a steady state of the HTF velocity inside the tube was reached quickly, while the temperature field did not reached a fully developed state. The temperature field changes as the melting interface progresses. Unsteady temperature distributions obtained by numerical calculation are shown in the following figures.

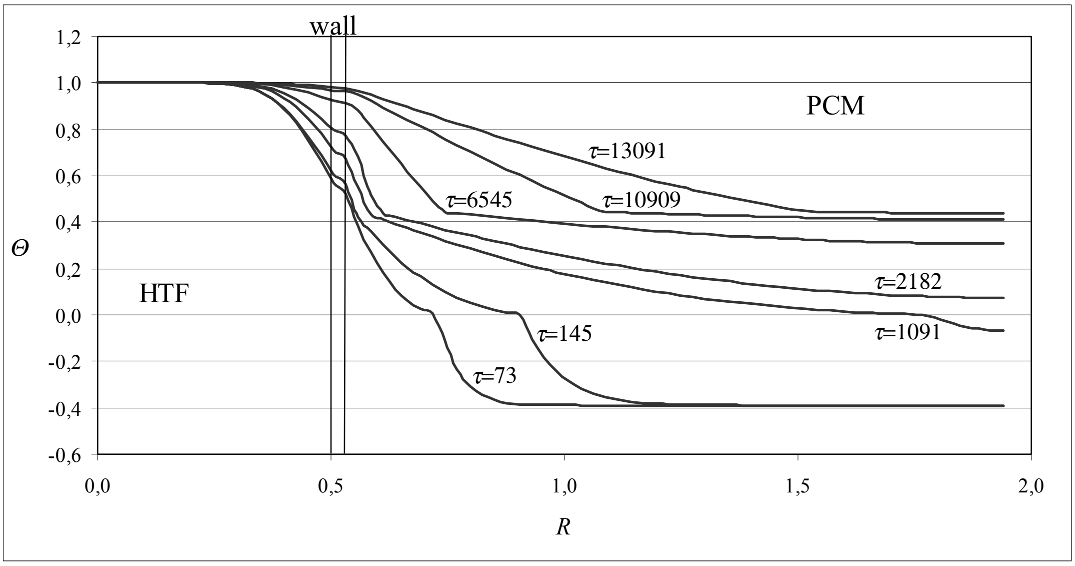

Figure 6. Radial dimensionless temperature distribution at $X = 12.73$ for different time periods

Figure 6 shows the radial temperature distribution at $X = 12.73$ for different time periods. The three regions in the radial direction (HTF, wall and the PCM) are indicated in the figure. Melting of the PCM was performed within a dimensionless temperature interval 0 to 0.42. Melting starts around the wall surface of the inner tube and spreads inside the outer tube. It can be seen that as melting interface progresses, the temperature curve moves upward accordingly.

Isotherms of dimensionless temperatures for different time periods are shown in Figure 7. Filled areas on diagrams are melting areas. As the melting interface progresses, the isotherms move upward. It can be seen that the melting interface moves from inlet to outlet of the storage unit faster than to the inner diameter of the outer tube. The reason is that since the Prandtl number of the HTF is relatively large (the thermal conductivity is small), a large amount of heat is carried downstream, while a relatively small amount of heat is transferred directly to the PCM upstream. At dimensionless time $\tau = 10909$, only a small amount of the PCM is in the melting process, while at dimensionless time $\tau = 13091$, heating of the liquid phase of the whole amount of the PCM is performed.

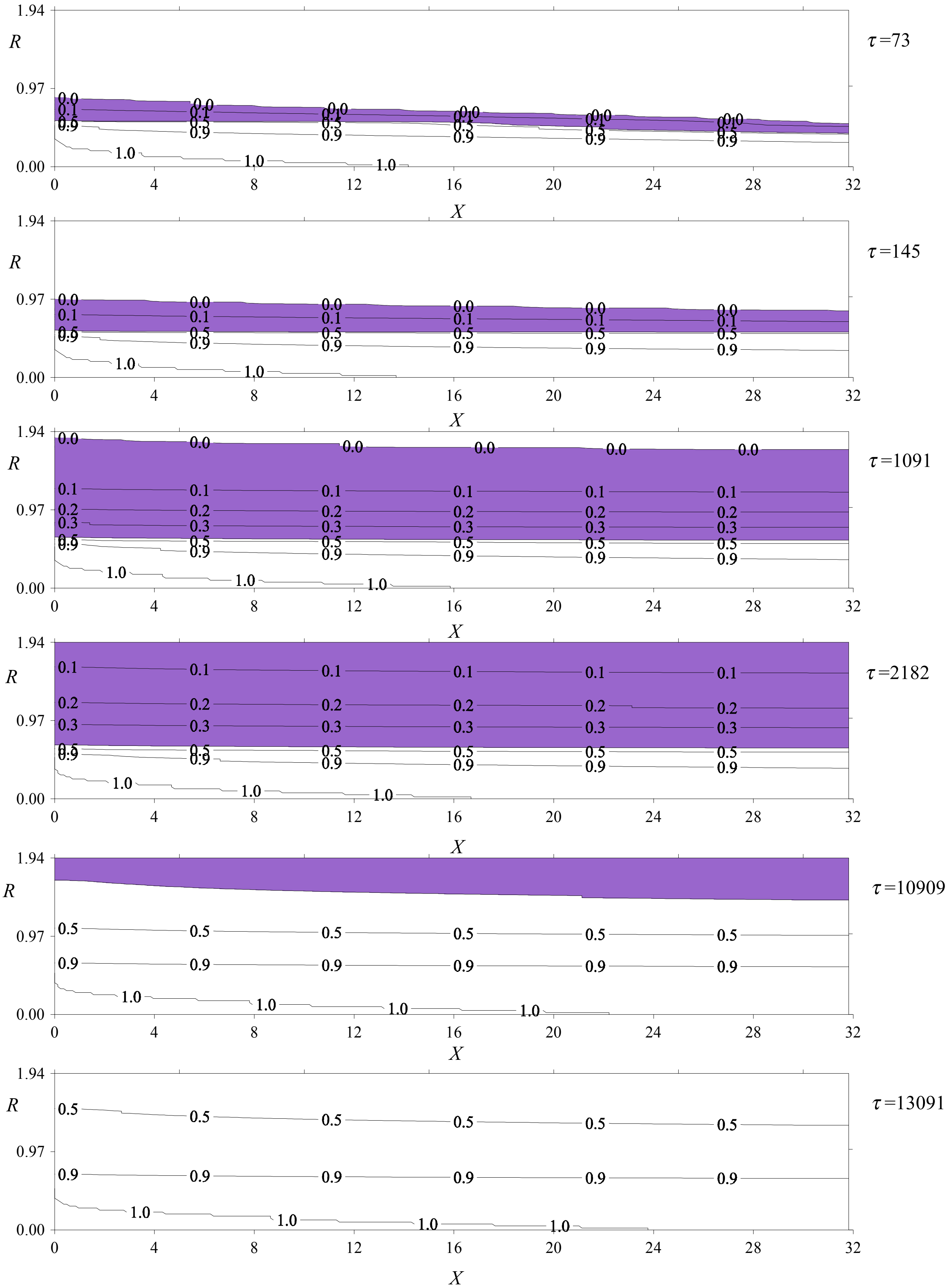

Figure 7. Isotherms of dimensionless temperatures for different time periods

6. CONCLUSION

A numerical model was developed to predict the transient phase change heat transfer behaviour of a shell-and-tube solar thermal heat storage unit with the PCM filling the shell side and HTF circulated inside the inner tube. An experimental test unit was constructed and the numerical predictions were verified with experimental data. The results of numerical analysis show that the fluid velocity field reaches a steady state quickly, while the temperature field changes with progression of the melting interface. Because of that, it is very important to treat the phase change and fluid flow as a conjugate problem and solve them simultaneously. Unsteady temperature distributions of heat transfer fluid, wall and phase change material have been obtained by numerical calculations and thermal behaviour of latent heat storage unit during charging and discharging has been simulated. For observed case of water as HTF and paraffin as PCM, it can be concluded that heat transfer from the HTF to the PCM is slow due to the relatively large Prandtl numbers of the HTF. Because of that, a large amount of heat is carried downstream with the HTF, while a relatively small amount of heat is transferred directly to the PCM upstream. The numerical results obtained provide guidelines for design optimisation of the latent heat storage unit.

7. LIST OF SYMBOLS

a	thermal diffusivity [m^2/s]
c	specific heat [J/(kgK)]
D	diameter of the tube [m]
H	volume enthalpy [J/m^3]
L	length of the tube [m]
$\dot{m}$	mass flow rate [kg/s]
Nu	Nusselt number
P	dimensionless pressure
p	pressure [Pa]
Pr	Prandtl number
q	latent heat of melting [J/kg]
R	dimensionless coordinate along the radial direction
r	coordinate along the radial direction [m]
Re	Reynolds number
St	Stefan number
T	thermodynamic temperature [K]
t	time [s]
W	dimensionless velocity
w	velocity [m/s]
X	dimensionless coordinate along the axial direction
x	coordinate along the axial direction [m]
α	convective heat transfer coefficient [W/(m^{2}K)]
χ	dimensionless enthalpy
ϑ	temperature [°C]
λ	thermal conductivity [W/(mK)]
Θ	dimensionless temperature
ρ	density [kg/m^3]
τ	dimensionless time

υ kinematic viscosity [m^2/s]

Subscripts:

f HTF
i inside radius of the tube
in inlet
init initial
L liquid phase of the PCM
m melting
o outer surface of the latent storage unit
out outlet
p PCM
R dimensionless coordinate along the radial direction
r coordinate along the radial direction
s solid phase of the PCM
w wall
X dimensionless coordinate along the axial direction
x coordinate along the axial direction

REFERENCES

[1] Voller, V., Cross, M.: *Accurate solutions of moving boundary problems using the enthalpy method*, International Journal of Heat and Mass Transfer, 24 (1981) 3, 545-556.
[2] Voller, V. R., Cross, M., Markatos, N. C.: *An Enthalpy Method for Convection/Diffusion Phase Change,* International Journal for Numerical Methods in Engineering, 24 (1987), 271-284.
[3] Voller, V. R.: *Fast Implicit Finite-Difference Method for the Analysis of Phase Change Problems,* Numerical Heat Transfer, 17 (1990), 155-169.
[4] Hunter, L. W., Kuttler, J. R.: *The Enthalpy Method for Heat Conduction Problems With Moving Boundaries,* Journal of Heat Transfer, 111 (1989), 239-242.
[5] Cao, Y., Faghri, A., Chang, W. S.: *A numerical analysis of Stefan problems for generalized multi-dimensional phase-change structures using the enthalpy transforming model*, International Journal of Heat and Mass Transfer, 32 (1989) 7, 1289-1298.
[6] Lee, S. L., Tzong, R. Y.: *An Enthalpy Formulation for Phase Change Problems with a Large Thermal Diffusivity Jump Across the Interface,* International Journal of Heat and Mass Transfer, 34 (1991) 6, 1491-1502.
[7] Bellecci, C., Conti, M.: *Phase change thermal storage: transient behaviour analysis of a solar receiver / storage module using the enthalpy method*, International Journal of Heat and Mass Transfer, 36 (1993) 8, 2157-2163.
[8] Bellecci, C., Conti, M.: *Latent heat thermal storage for solar dynamic power generation*, Solar Energy, 51 (1993) 3, 169-173.
[9] Bellecci, C., Conti, M.: *Transient behaviour analysis of a latent heat thermal storage module*, International Journal of Heat and Mass Transfer, 36 (1993) 15, 3851-3857.
[10] Lacroix, M.: *Numerical Simulation of a Shell-and-tube Latent Heat Thermal Energy Storage Unit,* Solar Energy, 50 (1993) 4, 357-367.
[11] Lacroix, M.: *Study of the heat transfer behaviour of a latent heat thermal energy storage unit with a finned tube*, International Journal of Heat and Mass Transfer, 36 (1993) 8, 2083-2092.

[12] Laouadi, A., Lacroix, M.: *Thermal performance of a latent heat energy storage ventilated panel for electric load management*, International Journal of Heat and Mass Transfer, 42 (1999) 2, 275-286.
[13] Cao, Y., Faghri, A.: *Performance characteristics of a thermal energy storage module: a transient PCM / forced convection conjugate analysis*, International Journal of Heat and Mass Transfer, 34 (1991) 1, 93-101.
[14] Cao, Y., Faghri, A.: *A Study of Thermal Energy Storage Systems With Conjugate Turbulent Forced Convection*, Journal of Heat Transfer, 114 (1992), 1019-1027.
[15] Zhang, Y., Faghri, A.: *Heat transfer enhancement in latent heat thermal energy storage system by using the internally radial finned tube*, International Journal of Heat and Mass Transfer, 39 (1996) 15, 3165-3173.
[16] Zhang, Y., Faghri, A.: *Semi-analytical solution of thermal energy storage system with conjugate laminar forced convection*, International Journal of Heat and Mass Transfer, 39 (1996) 4, 717-724.
[17] Trp, A., Franković, B., Lenić, K.: *An Analysis of Phase Change Heat Transfer in a Solar Thermal Energy Store,* Proceedings of the ISES 1999 Solar World Congress, Vol. III, Jerusalem, Israel, 1999, 484-489.
[18] Patankar, S. V.: *Numerical Heat Transfer and Fluid Flow,* Hemisphere Publishing Corporation, Taylor & Francis Group, New York, 1980.
[19] Versteeg, H. K., Malalasekera, W.: *An Introduction to Computational Fluid Dynamics: The Finite Volume Method,* Longman Scientific & Technical, Essex, 1995.
[20] *RUBITHERM GmbH,* (http://www.rubitherm.com), Hamburg, 2001.

NUMERIČKA I EKSPERIMENTALNA ANALIZA NESTACIONARNE IZMJENE TOPLINE UNUTAR CILINDRIČNOG LATENTNOG SPREMNIKA S PARAFINOM KAO AKUMULATOROM TOPLINE

Sažetak: *U ovom je radu numeričkim i eksperimentalnim putem analiziran fizikalni proces nestacionarne izmjene topline unutar cilidričnog latentnog spremnika. Izmjena topline razmatrana u radu složeni je fizikalni proces nestacionarne međuovisne izmjene topline putem prisilne konvekcije između fluida i stijenke cijevi, provođenja topline kroz stijenku cijevi te izmjene topline pri promjeni agregatnog stanja akumulatora topline. Bezdimenzijske diferencijalne jednadžbe izmjene topline za fluid, stijenku i akumulator topline, s definiranim početnim i rubnim uvjetima, diskretizirane su primjenom metode kontrolnih volumena te riješene iteracijski kompjuterskim programom napisanim u programskom jeziku Fortran-u. Numerički je model provjeren usporedbom s eksperimentalnim rezultatima. U eksperimentalnim je istraživanjima kao akumulator topline korišten trgovački parafin RUBITHERM RT 30. Usporedbom vremenskih promjena temperatura dobivenih numeričkim proračunom i eksperimentalnim putem utvrđena je zadovoljavajuća podudarnost rezultata. Numeričkim su proračunom zatim dobivena nestacionarna temperaturna polja fluida, stijenke i akumulatora topline čime je simulirano toplinsko ponašanje spremnika za vrijeme njegova punjenja i pražnjenja tj. za vrijeme spremanja i korištenja topline.*

Ključne riječi: nestacionarna izmjena topline, latentni spremnik topline, numerička analiza, eksperiment