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SIMULATION OF THERMAL STRATIFICATION IN THE HEAT STORAGE FOR CHP PLANT

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Abstract. Thermal stratification phenomenon is successfully used to store and retrieve thermal energy from heat storages. The aim of this study is to compare two simulation models for describing thermal stratification in the heat storage tank used in the cogeneration (CHP) plant at any time. In this paper analytical and numerical analyses have been carried out to show temperature fields inside the heat storage. The first analytical model is done using an energy balance which is created and solved for each layer of the stratified heat storage. This model shows a simple one-dimensional approach to theoretical analysis of thermal stratification. The other numerical model is implemented using a multi-purpose computation fluid dynamics (CFD) software PHOENICS. Two-dimensional transient model was created and solved numerically. The validation of simulated results is made successfully by the comparison against the actual data gathered from Hvide Sande CHP plant in Denmark. A good agreement with actual data is observed. The average relative discrepancy varied from 0.22 % to 5.00 % in analytical modelling and from 0.21 % to 3.66 % in numerical simulation, when the duration of simulated processes lasts from 15 min to 6 hours.

Keywords: heat storage, thermal stratification, model, numerical simulation, analytical, CHP.

1. Introduction

One of the energy production technologies satisfying such requirements as high efficiency of energy conversion, minimal possible environmental impact and economic feasibility is the small-scale cogeneration based on gaseous fuels (Kostowski and Skorek 2005). The main feature of a cogeneration system is that it is meant to satisfy two separate needs or demands (heat and electricity) by a simultaneous production of the two forms of energy. Since the daily demand curves for electricity and heat are commonly distinct and seldom concomitant, a form of storage of one of the two products is necessary in order to succeed in providing both products when requested (Maifredi *et al.* 2000).

Introduction of the heat storage in the CHP plant may increase thermal efficiency and economy of the plant even more due to achieved compensation between the differences in the production and consumption of the heat (Bogdan and Kojar 2006). CHP plants with heat storages gain flexibility and may achieve improved economic results if managed properly. Moreover operators of CHP plants will gain an increase in security when planning their ahead schedules as fluctuation in the heat demand can be compensated with the storage (Streckienė *et al.* 2009).

It is well known and proven that efficiency of heat storage and all system can be improved if the water in the storage tank is stratified. Meanwhile, destratification always results in generation of entropy and loss of exergy (Rosen *et al.* 2004). There are four primary destratification factors, which contribute to the loss and degradation of the stored energy: heat losses to ambient, heat conduction from the hot layer to the cold layer, vertical conduction in the storage wall and mixing introduced during charge and discharge cycles (Kleinbach *et al.* 1993). By a comparison between fully stratified water storage and fully mixed storage employed in energy systems, it is found that the energy storage efficiency and the whole system may be increased up to 6% and 20%, respectively (Han *et al.* 2009).

Stratification of water is created by the difference in density between hot and cold water. Cold water remains at the bottom of the storage, while hot water flows to the top. The intermediate region is called the thermocline. The formation of the thermocline is determined by volume and geometry of the storage, the inlet, the hydrodynamics and thermal characteristics of the water flow in the storage. Water flows are a function of charging when hot water enters the storage and discharging when hot water is drawn out of the storage to service a load (Bouhdjar and Harhad 2002, Castell *et al.* 2010). Stratified heat storages have been studied for a long time. A

significant amount of experimental and theoretical works was done in this field from early 1970s (Gupta 1971, Castell *et al.* 2010).

In this paper, two simulations models for thermal stratification prediction are presented, compared and verified with actual operation results of the heat storage used in the CHP plant. These models can be used to determine temperature distribution in the heat storage installed in the CHP plant. Exact knowing of the temperature distribution can help to determine the fulfillment of the heat storage and plan the operation of the whole system.

The first analytical model is implemented using MS Excel and could be used for engineering purposes. The second numerical model is done with CFD software PHOENICS. Numerical simulation takes more time, however it can give more additional information, for example: velocity fields, temperature fields near inlet/outlet or in the whole domain and etc.

2. Analytical model

The heat storage considered in this analysis is a vertical cylinder of diameter D and height H. It is divided into N equal layers in the longitudinal direction with energy balances written for each layer. The analytical computation scheme is shown in Fig 1.

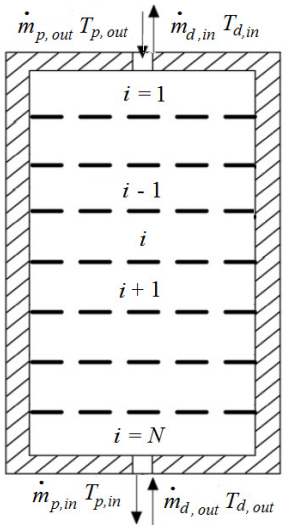

Fig 1. Analytical computation scheme of stratified heat storage

where: $\dot{m}$ – mass flow, T – temperature, subscripts: *p,out* – from production circuit, *d,in* – to demand circuit, *p,in* – to production circuit, *d,out* – from demand circuit.

The energy equation takes into account of the energy gain from the heat source, energy lost to surrounding, energy utilized by the load and convection flux with the thermal conduction. Exact mathematical modelling of the convection flux is complex. In the simplest approximation, thermal conduction and convection are summarized in an effective vertical heat conductivity λ_{eff}. The heat flow between layers i – 1 and i, or from i to i +1, is calculated using the Fourier equation. The net heat flow for layer i of height z cross section A_q results from the difference of the two heat flows (Eicker 2003):

$$Q_{free} = Q_{free,i-1\to i} - Q_{free,i\to i+1} = \\ = A_q \frac{\lambda_{eff}}{z_i}(T_{i+1} - 2T_i + T_{i-1}) \quad (1)$$

where: T – temperature, ºC.

The vertical temperature stratification in the storage is diminished by high effective heat conductivity. The effective heat conductivity with good storages without internal installations according to measurements taken by University of Stuttgart is in the area of the heat conductivity of water (λ = 0.644 W/m·K at 50 ºC) (Eicker 2003). Furthermore, amounts of heat are exchanged between the layers by forced convection, depending on the mass flow balance of the store. The heat flow by forced convection is calculated as:

$$Q_{for} = (\dot{m} c_p)_i (T_{i-1} - T_{i+1}) \quad (2)$$

where: c_p – specific heat capacity at constant pressure, J/kg·K. In this paper it is calculated as a function of temperature (Бажан *et al.* 1989):

$$c_p = 4{,}20511 - 0{,}136578 \cdot 10^{-2} \cdot T + \\ 0{,}152341 \cdot 10^{-4} \cdot T^2 \quad (3)$$

A positive effective mass flow $\dot{m}_i$ with energy entry from layer i – 1 is taken into account by parameter $\delta_i^+ = 1$ (otherwise $\delta_i^+ = 0$). A negative effective mass flow from layer i + 1, i.e. dominance of the load mass flow and thus cooling of layer i, is taken into account by parameter δ_i^-. Thus the general energy balance for temperature layer i reads (Dutré 1991, Eicker 2003):

$$(m_i c_p)\frac{dT}{d\tau} = \delta_i^p (\dot{m} c_p)_p (T_p - T_i) - \delta_i^d (\dot{m} c_p)_d (T_i - \\ T_{d,out}) - kA_{pav.i}(T_i - T_a) + \delta_i^+ \dot{m}_i c_p (T_{i-1} - T_i) + \\ \delta_i^- \dot{m}_{i+1} c_p (T_i - T_{i+1}) + A_i \frac{\lambda_{eff}}{z_i}(T_{i+1} - 2T_i + T_{i-1}) \quad (4)$$

where: k – overall heat transfer coefficient, W/(m^2·K); $A_{pav.i}$ – exterior heat transfer surface area of the respective layer, m^2; A_i – cross-section area of the respective layer, m^2; τ – time, s; T – temperature, K; d – demand circuit (discharging); p – production circuit (charging); *out* – outlet, a – ambient.

The mass of layer i is calculated as the product of density and volume. The density of water is calculated as a function of temperature (Maidment 1993):

$$\rho = 1000 \cdot \left(1 - \frac{(T + 288{,}9414)(T - 3{,}9863)^2}{(508929{,}2 \cdot (T + 68{,}12963))}\right) \quad (5)$$

Usually only charging process or discharging process occurs in the heat storage during operation of the CHP plant. Knowing the initial temperature distribution in the heat storage tank the (4) equation is applied to each layer and the analytical temperature distribution is obtained after respective time period. In this study presented analytical model allows to show temperature distribution evaluating the density and specific heat as the functions of temperature additionally.

3. Numerical model

Recently, numerical modelling has been applied in the heat and mass transfer processes more and more. Various CFD software packages can be used for this purpose: ANSYS, FLUENT, PHOENICS, TRNSYS etc.

In this study mathematical modelling of temperature distribution in the heat storage tank is performed with the PHOENICS 3.5.1 where the own code is created as Q1 file. The Q1 file is an American standard code for information interchange (ASCII) text file, which the PHOENICS SATELLITE module must read before it can do anything else. It tells the SATELLITE module whether to operate in the batch or interactive modes, and also how many runs to make. In general, PHOENICS performs three main functions (PHOENICS 3.5.1, 2002):

1) Problem definition (pre-processing), in which the user prescribes the situation to be simulated and the questions which are to be answered;
2) Simulation (data-processing), by means of computation, of what the laws of science imply in the prescribed circumstances;
3) Presentation (post-processing) of the results of the computation, by way of graphical displays, tables of numbers or other means.

In PHOENICS software, the Navier-Stokes equations for mass, momentum and energy equation derived for an elemental control volume are solved to obtain pressure, velocity and temperature using finite volume approach (Patankar 1980).

PHOENICS provides solution to the discretized version of sets of differential equations having the general form (PHOENICS 3.5.1, 2002):

$$\underbrace{\frac{\partial}{\partial \tau}\left(r_i \rho_i \Phi_i\right)}_{transient} + div\left(\underbrace{r_i \rho_i \vec{v}_i \Phi_i}_{convection} - \underbrace{r_i \Gamma_{\Phi i}\, grad \Phi_i}_{diffusion}\right) = \underbrace{r_i S_{\Phi i}}_{source} \quad (6)$$

where: τ – time, s; r_i – volume fraction of phase i; ρ_i – density of phase i; Φ_i – any conserved property of phase i, such as momentum per unit mass, turbulence energy, etc.; $\vec{v}_i$ – velocity vector of phase i; $\Gamma_{\Phi i}$ – the exchange coefficient of the entity Φ in phase i; $S_{\Phi i}$ – the source rate of Φ_i.

In this analysis Φ_i takes the following values, each of which gives rise to a particular conservation equation (Gajapathy *et al.* 2008):

1) $\Phi = 1$ gives the continuity equation;
2) $\Phi = v$ gives the radial direction momentum equation;
3) $\Phi = w$ gives the axial direction momentum equation and
4) $\Phi = c_p T$ gives the energy equation.

When a single-phase phenomenon is in question, the volume fraction r_i disappears from the equation, which thus becomes:

$$\frac{\partial}{\partial \tau}\left(\rho\Phi\right) + div\left(\rho\vec{v}\Phi - \Gamma_\Phi \mathrm{grad}\Phi\right) = S_\Phi \quad (7)$$

The differential equations presented above are the instantaneously – valid ones. PHOENICS solves these for laminar flows. For turbulent flows, this software can solve equations that are time averaged. Within simulation, PHOENICS solves sets of algebraic equations which represent the sequences of (PHOENICS 3.5.1, 2002):

- Integrating the differential equations over the finite volume of a computational cell and for transient problems over a finite time;
- Approximating the resulting volume, area and time averages by way of interpolation assumptions.

The governing equations of heat transfer and fluid flow considered in this work have been formulated using unsteady two-dimensional flow configuration and taking into account the following assumptions:

- Incompressible and laminar flow;
- Newtonian behaviour;
- Non-participant radiating medium.

The equations for temperature distribution modelling normally can be expressed in vector form as (Vaitiekūnas and Špakaukas 2003):

Continuity equation:

$$\frac{\partial \rho}{\partial \tau} + \nabla\left(\rho\vec{v}\right) = 0 \quad (8)$$

Momentum equation:

$$\rho\frac{\partial \vec{v}}{\partial \tau} + \left(\rho\vec{v}\cdot\nabla\right)\vec{v} = -\nabla p + \mu\nabla^2\vec{v} + \rho\vec{g} \quad (9)$$

Energy equation:

$$\frac{\partial T}{\partial \tau} + \vec{v}\cdot\nabla T = \frac{\lambda}{\rho c_p}\nabla^2 T \quad (10)$$

where: μ – dynamic viscosity, N·s/m^2; g – gravitational constant, m/s^2; p – pressure, Pa; λ – thermal conductivity, W/(m·K).

All equations in PHOENICS are solved iteratively. To evaluate the convective and diffusive fluxes at the interfaces, the HYBRID scheme is used.

Made model uses cylindrical coordinates. Because of the symmetrical location of inlet and outlet, the selected computational domain accounts for half of the storage. The schematic computational model is presented in Fig 2. It is assumed that storage walls are very thin and they are not modeled, only thermal insulation is taken into account. 40x58 grid resolution is used to calculate the velocity and temperature field. The mesh is finer near the inlet/outlet and walls region. In the transient calculations the velocity and the temperature of the inflow are specified from the first calculated time step.

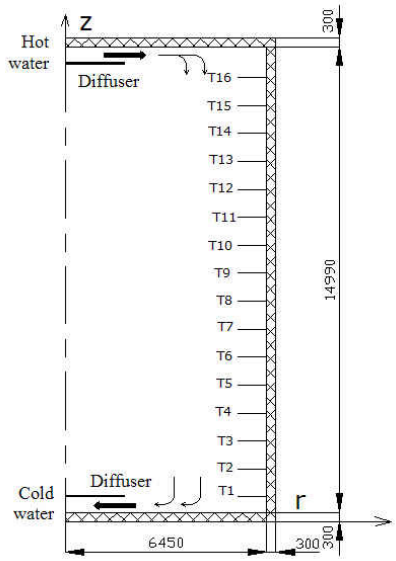

Fig 2. View of schematic computational model

Properties of used materials in numerical simulation are presented in Table 1.

Table 1. Properties of materials used in simulation

Property	Water (20 ºC)	Steel (27 ºC)	Fiberglass (27 ºC)
Density, kg/m^3	998.23	7800.0	48.0
Viscosity, m^2/s	$1.006\cdot10^{-6}$	–	–
Specific heat capacity, J/(kg·K)	4181.0	473.0	1100.0
Thermal conductivity, W/(m·K)	0.597	43.0	0.038
Thermal expansion coeff.; 1/K	$1.18\cdot10^{-4}$	$0.37\cdot10^{-5}$	$0.58\cdot10^{-5}$
Compressibility, m^2/N	–	$0.5\cdot10^{-11}$	$0.145\cdot10^{-10}$
Prandtl Number	7.033	–	–

The calculation domain was subdivided into zones corresponding to the existing different physical materials in the storage.

4. Description of heat storage installed in the CHP-plant

Simulation results are compared and validated using actual data gathered from Hvide Sande CHP plant in Denmark during the training in EMD International A/S company according Lifelong Learning Programme. This plant has two identical gas fired engines of 3.77 MW$_e$, two boilers: 10 MW and 4 MW and a heat storage. The analyzed heat storage is characterized by following features:

- construction: above ground level tank with thermal insulation (300 mm);
- storage medium: water;
- volume: gross – 1960 m^3, effective – 1750 m^3;
- charging/discharging: direct water exchange;
- maximum flow rate: 200 m^3/h.

During the charging, hot water of ~94–95 ºC temperature is supplied to the top of storage, while the same amount of colder water is taken off from the storage bottom. The charging of the storage starts when the heat production is higher than the heat consumption. During the discharge, an opposite process takes place.

The heat storage is usually filled with heat generated by gas engines in hours, where the spot price is high and it is rarely filled by a boiler. This happens, when some repairs are made for the gas engines.

Actually, 16 temperature sensors (PT100) are installed inside the heat storage, but there is possibility to get data only from 15 temperature sensors. The first meter is installed 0.5 m above the bottom and the last sensor is 0.5 m below the top diffuser. Used temperature sensors have tolerance class B. Error of the used temperature sensors ranges from ±0.50 ºC to ±0.78 ºC when the temperature of stored water is in range 40 ºC to 95 ºC.

Data are collected with SCADA system and registered every 15 min. The data were imported to Excel sheets.

5. Results

Analytical and numerical simulations are done using the main parameters of charging and discharging processes of Hvide Sande heat storage which are presented in Table 2, where the values of flow rate and ambient temperature (T_a) for 30 min – 6 h time period are calculated as average value. Both simulated processes last from 15 min to 6 hours. Numerical solutions are obtained for transient laminar flow.

A careful check for time-grid of the analytical and numerical solution has been made to ensure accuracy and validity of the models using different time steps. Analytical simulation used time steps of 45 s, 30 s, 20 s, 10 s, 5 s and 2 s.

Table 2. Specification of charging and discharging processes

Time	Charging		Discharging	
	T_a, °C	Q, m^3/h	T_a, °C	Q, m^3/h
15 min	6.7	40.2	5.8	93.2
30 min	6.8	38.9	5.9	93.4
1 hour	6.7	38.1	5.8	93.2
2 hours	6.8	43.0	5.7	93.9
4 hours	7.2	47.3	5.8	96.6
6 hours	7.6	51.5	5.9	95.0

In the numerical scheme the time of the respective process (charging or discharging) was divided into 5, 10, 20, 50, 100 and 200 uniform subdivisions.

5.1. Charging process

In the first the charging process of 15 min is simulated (Fig 3). Knowing the initial temperature distribution of Hvide Sande heat storage, the latter temperature values are calculated using different time steps.

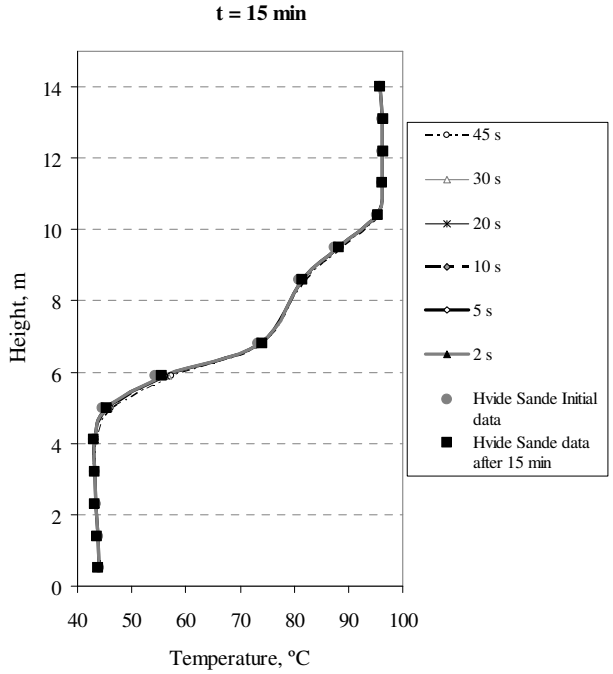

Fig 3. Analytically determined temperature distribution using various time steps, 15 min

From the figure, it is evident that all time steps of analytical solution show a good agreement with temperature data of Hvide Sande storage, when the simulated process time is short. All presented distributions are in the same position and agree with actual data. Fig 4 shows the analytical temperature distribution after 6 hours using different time steps.

Various time steps of analytical solution show the different temperature distribution in the heat storage, when the simulated charging process lasts long (Fig 4). Time step of 45 s shows the biggest temperature discrepancies in comparison with data of temperature sensors. Time steps of 2 s and 5 s are the closest to the actual data of Hvide Sande storage. For the comparison of accuracy using different time steps, the average relative discrepancy of temperature is calculated. The smallest relative discrepancy is desirable. The relative discrepancy of analytically simulated charging process is presented in Fig 5.

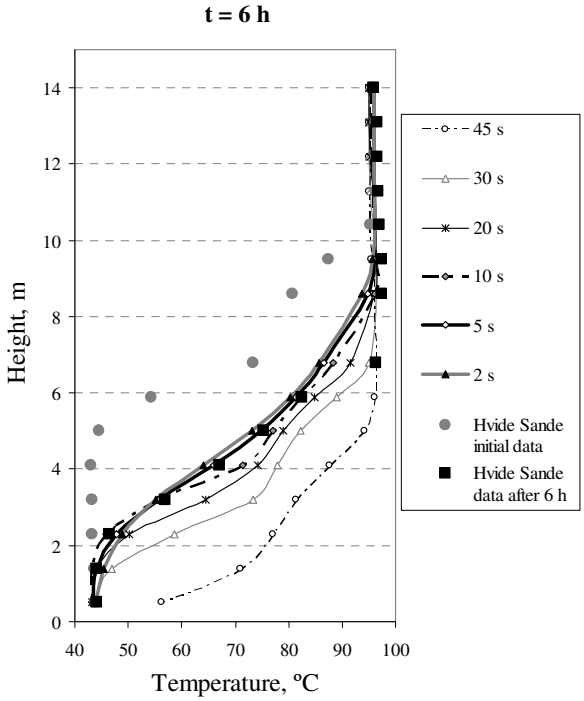

Fig 4. Analytically determined temperature distribution using various time steps, 6 hours of charging

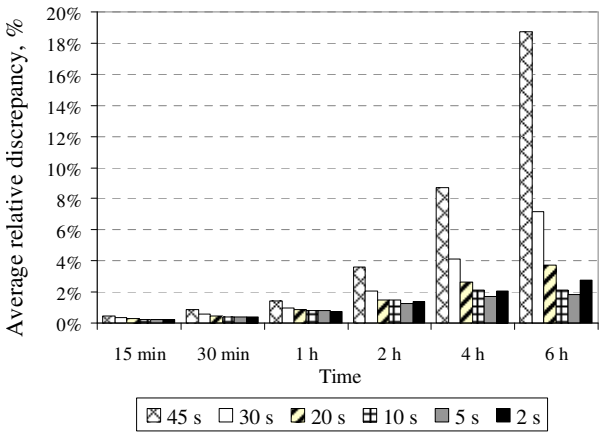

Fig 5. Average relative discrepancy of analytical charging simulation using different time steps

It is evident from Fig 5, that the time step of 5 s shows the best results of the analytical charging process, because the average relative discrepancy is the smallest in all cases simulated. Using this time step the average relative discrepancy varies from 0.22 % (15 min) to 1.82 % (6 hours). Time steps of 45 s and 30 s gives the biggest average relative discrepancies. Especially these time steps are not suitable for modelling of long duration charging

processes. Time steps of 45 s and 30 s make acceleration of the charging process and the temperature values are calculated much higher than they are in reality. Using the time step of 45 s gives the relative discrepancy from 0.48 % (15 min) to 18.74 % (6 hours).

The same charging processes of Hvide Sande heat storage were modeled numerically. Temperature distribution after 15 min and 6 hours simulated with PHOENICS are shown in Fig 6.

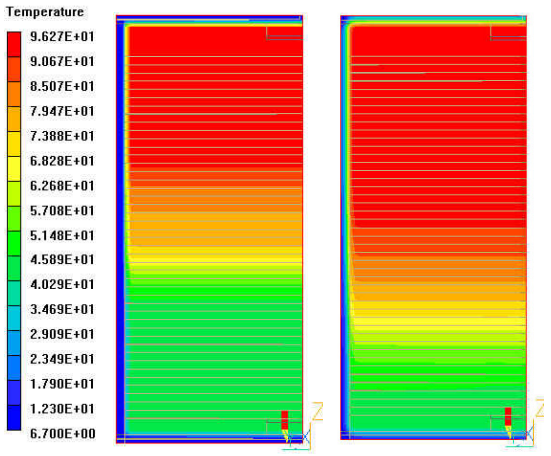

Fig 6. Temperature distribution after 15 min (left) and 6 h of charging (right)

The numerical simulations presented in Fig 6 were implemented using the 50 subdivisions of time period modeled. 10 iterative sweeps through the integration domain were performed for each time step. These options gave the even distribution of temperature in r (radius) direction. PHOENICS gives also a possibility to see velocity fields in the heat storage in VR Viewer and in RESULTS file. View of temperature distribution and velocity fields after 4 hours of charging is presented in Fig 7.

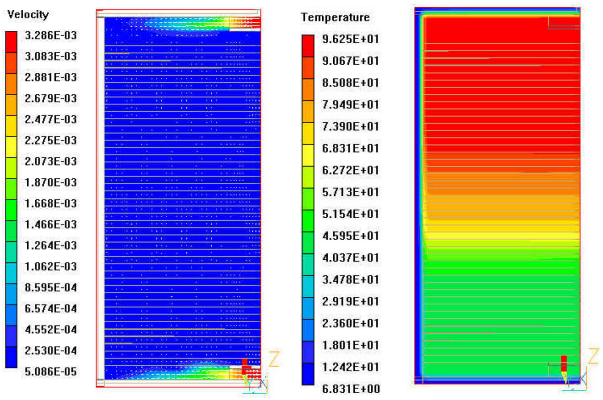

Fig 7. Field of velocity and temperature distribution after 4 h of charging

The maximal velocity is in inlet and outlet places. Velocity simulated in the major part of storage ranges from 10^{-4} m/s to 10^{-5} m/s. The time division of 50 equal parts in PHOENICS model showed a good agreement. The average discrepancy did not exceed 1.71 ºC, and the average relative discrepancy varied from 0.21 % (15 min) to 2.28 % (6 hours). Also, such time division was not time-intensive to perform the calculations.

5.2. Discharging process

At first the discharging process of Hvide Sande heat storage was modeled analytically. An analytical temperature distribution, using various time steps after 6 hours of discharging, is presented in Fig 8.

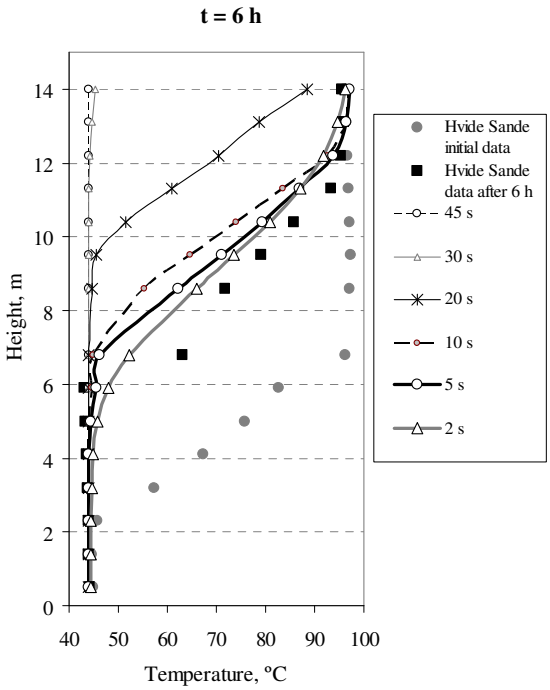

Fig 8. Analytically determined temperature distribution using various time steps, 6 hours of discharging

Fig 8 shows that the time step of 2 s has the closest temperature distribution in comparison to actual values of Hvide Sande storage, when discharge process of 6 hours is modeled. The worst agreement was detected using time steps of 45 s and 30 s. Using these time steps the discharging process progresses very fast, and all storage takes the temperature of cold water (~44ºC). The relative discrepancy of analytical discharging process using different time steps is presented in Fig 9. As it can be seen the time step of 2 s shows the best agreement of the analytical discharging process in all cases modeled. Using this time step the average relative discrepancy of temperature varies from 0.46 % (15 min) to 5.00 % (6 hours). Time steps of 45 s and 30 s give the biggest average relative discrepancies. The higher average relative discrepancy of analytical discharge processes in comparison to the analytical charging processes can be explained by the higher flow rate in discharging processes of Hvide Sande heat storage (Table 2).

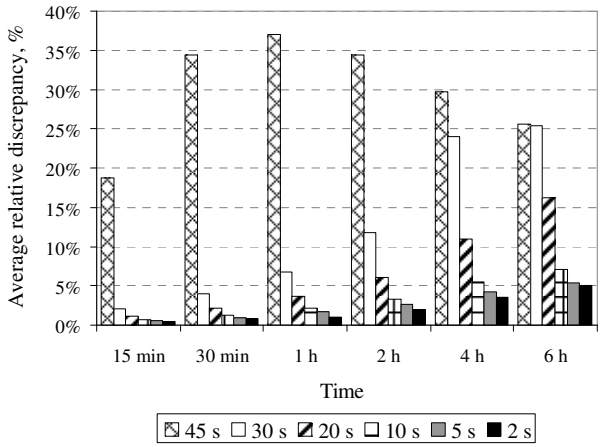

Fig 9. Average relative discrepancy of analytical discharging process simulation using different time steps

The same discharge processes of Hvide Sande heat storage were modeled numerically. Modeling of water density was made using Boussinesq approximation. It states that density differences are sufficiently small to be neglected, except where they appear in terms multiplied by *g*, the acceleration due to gravity. So the density was evaluated only in momentum equations in the simulation of discharge process. While the best modelling results of the charging process were achieved by assessment of density as function of temperature in all the heat and mass transfer equations. Numerically determined temperature distributions after 15 min and 6 hours of discharging are presented in Fig 10. Numerical simulation was implemented using the 50 subdivisions of time period modeled.

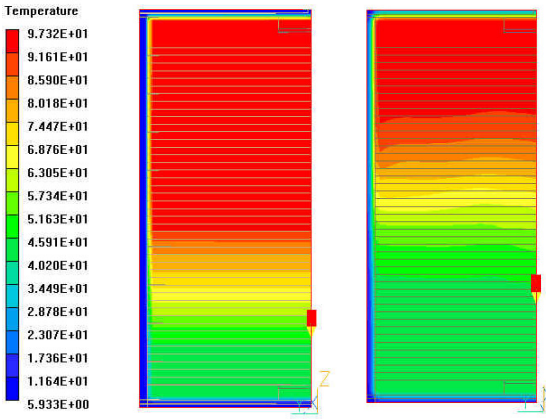

Fig 10. Temperature distribution after 15 min (left) and 6 h of discharging (right)

It was found that the number of time divisions had the direct influence on the computation time. This dependence is shown in Fig 11. From the figure, it is evident that computation time grows with increase of number of time divisions of the simulated process. In both cases (charging and discharging) linear dependency can be seen.

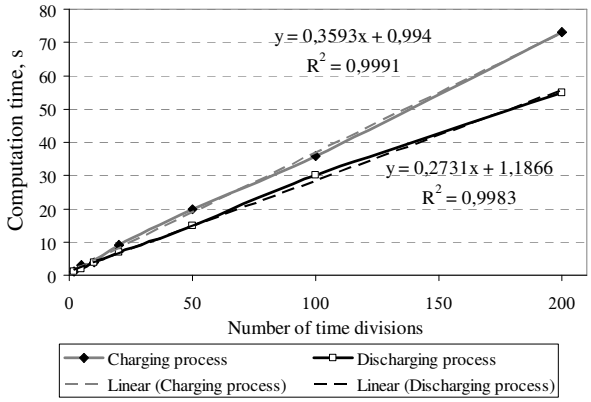

Fig 11. Dependence of computation time on number of time divisions

The simulation of charging process runs longer, because the water density is calculated in all differential equations solved by PHOENICS, while the modeling of discharge process uses the Boussinesq approximation.

In consideration of accuracy, reliability and computation time, the time division of 50 equal parts could be recommended for the simulation of discharging process. This number of time sub-divisions showed the average discrepancy of temperature from 0.26 ºC to 1.97ºC and the average relative discrepancy of temperature from 0.37 %to 3.66 %, when the discharge process lasts from 15 min to 6 hours.

After determination of time division parameters, the selected discharging process of 6 hours was simulated and compared using different models. The comparison is presented in Fig 12.

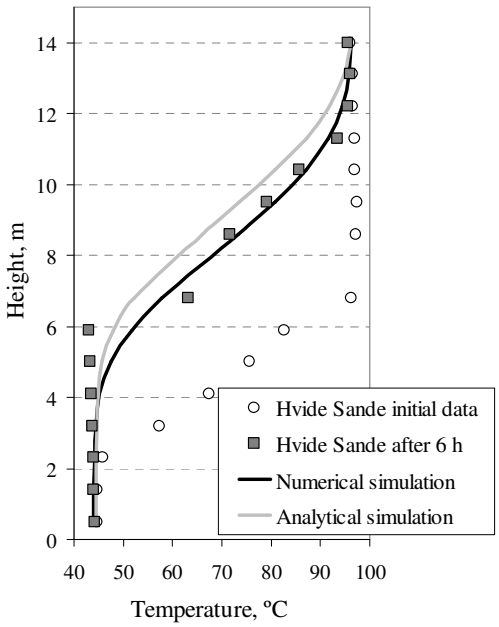

Fig 12. Comparison of modelling results of discharging process using the different models

As it can be seen the obtained temperature distributions using analytical and numerical models correspond well the actual temperature distribution in Hvide Sande storage tank. The greatest temperature discrepancy is obtained in height of 6.8 m at the 8th sensor, because no readings of the 9th sensor were registered. Thus the lacking temperature values in the relevant height in the models were calculated as average values. Besides, the 8th sensor falls into transitional temperature zone, where the greatest temperature gradient changes are.

6. Conclusions

In this work, the charging and discharging processes of a heat storage used in the CHP plant have been reproduced analytically and numerically. Both methodologies were presented. Prepared analytical model allows showing temperature distribution evaluating the density and specific heat as the functions of temperature. A particular emphasis is given to the validation and verification of the solutions. Results of the comparison of models lead to several conclusions:

- Time step of 5 s of analytical modeling was the most suitable for charging process. It showed the average relative discrepancy of temperature from 0.22 % (15 min) to 1.82 % (6 hours). For discharging process the time step of 2 s is the most suitable. In this case the average relative discrepancy varies from 0.46 % (15 min) to 5.00 % (6 hours).
- Application possibilities of the numerical model are more flexible than of the analytical model, however, this model is more time-intensive and demands greater analyzer's skills.
- The time division of 50 equal parts is recommended for numerical charging and discharging simulation in consideration of accuracy, reliability and computation time. Using this time division the average relative discrepancy was from 0.21 % to 2.28 % in simulation of charging process and from 0.37 % to 3.66 % in simulation of discharge process.

In common it was found that the simulated results using analytical and numerical models agreed well with actual data gathered from the heat storage, which is installed in the CHP plant. The created numerical model can be used for scientific purposes and investigation of specific conditions and parameters inside the heat storage. The used analytical model could be more suitable for designers, planners and engineers of energy systems who make decisions, because computations with CFD are costly and time-consuming. The exact knowing of the temperature distribution can help to determine the fulfillment of the heat storage and plan the operation of the whole system better. Also this model could be incorporated into software packages, which usually have models of fully mixed heat storages.

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