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Experimental validation of a tool for the numerical simulation of a commercial hot water storage tank

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Abstract

This work focuses on the experimental validation of a numerical tool realized to simulate a commercial hot water storage tank. The tool implements unsteady 1D models to simulate the temporal evolution of the temperature field inside the hot water storage tank, and the one relative to the heat transfer fluid flowing through the immersed coil heat exchanger. It has been implemented by means of the Simulink tool of Matlab.

The first part of the paper is dedicated to the description of the indoor experimental facility used to realize the experimental test. Successively, the analytical models, and the numerical schemes and algorithms used to perform the numerical simulations are described. Finally, the results of the experimental validation of the tool, accomplished by comparing the experimental temperature profiles inside the tank, and the measured temperatures at the coil heat exchanger exit section over the entire experimental test duration, with the numerical results obtained from simulations performed using different correlations for the evaluation of the heat transfer rate between the tank water and the heat transfer fluid through the coil, are reported and discussed.

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Keywords:

1. Introduction

Hot water storage systems are widely employed for the sensible thermal energy storage, as they can allow, for example, to maximize the profit of cogeneration systems, which, by integrating a hot water storage system, can produce electricity when it is more economically convenient without or with a limited waste of heat, and to enhance the exploitation of the thermal energy produced by solar thermal systems. To date, due to the high heat capacity and practically null cost of water, hot water storage systems are likely the most suitable ones for relatively low temperature applications (60°C – 100°C) [1].

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The above statements explain why hot water thermal energy storage systems have been massively studied in the last decades, with many works focused on the analytical modelling of such systems [2-5], which plays a key role in the their design optimization.

The present paper deals with the experimental validation of a numerical tool realized to simulate a commercial hot water storage tank. The tool implements unsteady 1D models to simulate the temporal evolution of the temperature field inside the hot water storage tank, and the one relative to the heat transfer fluid (HTF) flowing through the immersed coil heat exchanger. It has been implemented by means of the Simulink tool of Matlab. The above tool has been derived from a home-made numerical code, written in Matlab, developed for the numerical simulation of solar thermal systems [6,7].

The first part of the paper is dedicated to the description of the indoor experimental facility used for the experimental test. Successively, the tool characteristics such as the analytical models, and the numerical schemes and algorithms used to perform the numerical simulations are described. Finally, the results of the experimental validation of the tool, performed by comparing the experimental temperature profiles inside the tank, and the measured temperatures at the coil heat exchanger exit section over the entire experimental test duration, with the numerical results obtained from simulations performed using different correlations for the evaluation of the heat transfer rate between the tank water and the heat transfer fluid through the coil, are reported and discussed.

2. Experimental facility

Figures 1 and 2 show a picture and a schematic layout of the indoor facility used to perform the experimental test, respectively.

Fig. 1. Experimental facility

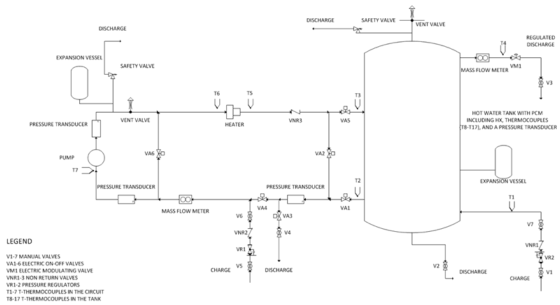

Fig. 2. Layout of the experimental facility

The commercial insulated hot water storage tank has a vertical cylindrical shape, a height of 1.27 m, an internal diameter of 0.65 m, a 0.05 m thick polyurethane insulation, and a total capacity of about 420 liters. It is equipped with a 1” coiled-tube heat exchanger having a total heat exchange area of 1.9 m^2, and with 10 type-T thermocouples with the hot junctions uniformly distributed along the tank axis. Moreover, the hot water tank is connected to an expansion vessel in order to perform tests at different pressure values. The thermal charging of the thermal energy storage tank is realized by means of a fluidic circuit connected to the tank heat exchanger, while the thermal discharging is done by means of an electric modulating valve that allows to regulate the set-point of the water mass flow rate. The heat exchanger inflow and outflow sections are located at about 0.73 m and 0.28 m from the tank bottom, respectively.

Relatively to the charging circuit, the main components are represented by the pump, modulated by an inverter in order to control the mass flow rate through the circuit, and the electric heater providing heating up to 24 kW. A thermoregulator controls the heater in order to regulate the temperature at the tank heat exchanger inflow section. The data acquisition and control are made by means of NI modules mounted on

NI cRIO-9066 controller. The control panel has been realized by means of the NI software Labview, and all the controlled parameters are regulated using PID controllers.

3. Models of the hot water storage tank components

One-dimensional analytical models have been developed for the hot water storage tank components, to evaluate the temperature field of the heat transfer fluid (HTF) flowing through the coil heat exchanger (HEX) immersed in the tank, as well as the water temperature along the tank vertical axis. Fig. 3 shows a sketch of the system analyzed. Each component has been discretized and solved using the finite volume method.

3.1. Immersed coil heat exchanger model

The main data of the coil heat exchanger are reported in Table 1. As can be seen in Fig.3, the heat exchanger can be considered formed by two connected parts, namely the vertical oriented upper part, and the lower one, which is inclined of 25° from the vertical in order to enhance the heat transfer efficiency. The discretization of the coil volume is dependent on the discretization of the tank one. Indeed, for a given tank volume discretization, the coil is discretized so that each coil section, corresponding to a coil node, intersects only one tank layer corresponding to a tank node.

The energy balance equation in transient regime for each node is formulated as:

$$\rho_{HTF} c_{HTF} V_{HTF} \frac{dT_{HTF}}{dt} = \frac{T_{w,tank} - T_{HTF}}{R_{hex}} + \rho_{HTF} c_{HTF} \dot{V}_{HTF} \left(T_{HTF}^{in} - T_{HTF}^{out}\right) \qquad (1)$$

where ρ_{HTF} and c_{HTF} are the density and the specific heat of the HTF, respectively, V_{HTF} is the volume, T_{HTF} is the HTF temperature, $T_{w,tank}$ is water temperature of the corresponding node in the tank, $\dot{V}_{HTF}$ is the volumetric flow rate, T_{HTF}^{in} and T_{HTF}^{out} are the inlet and outlet temperatures of the HTF at the boundaries of the volume corresponding to the node, respectively, and R_{HEX} is the total thermal resistance between the HTF in the heat exchanger and the corresponding water node in the tank.

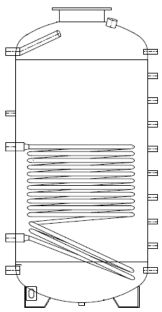

Fig. 3. Sketch of the hot water tank without insulation

Table 1. Main data of the coil heat exchanger

Thermal conductivity (W/m/K)	30.0
External diameter (m)	0.0334
Internal diameter (m)	0.0301
Helix length (m)	18.10
Helix height (m)	0.455
Helix diameter (m)	0.480

This latter is equal to the series of the internal convective thermal resistance, the conductive thermal resistance of the heat exchanger material, and the external convective thermal resistance:

$$R_{HEX} = R_{HEX,conv}^{i} + R_{HEX,conv}^{e} + R_{HEX,cond} \qquad (2)$$

The internal convective thermal resistance $R_{HEX,conv}^{i}$ is evaluated as:

$$R^{i}_{HEX,conv} = 1/\left(A^{i}_{HEX}\bar{h}^{i}_{HEX}\right) \tag{3}$$

with

$$\bar{h}^{i}_{HEX} = \left(\overline{Nu}^{i}_{HEX} k_{HTF}\right)/D^{i}_{HEX} \tag{4}$$

where A^{i}_{HEX} is the area of internal surface, and $\bar{h}^{i}_{HEX}$ is the mean internal convective heat transfer coefficient based on the thermal conductivity of the HTF k_{HTF}, the internal diameter D^{i}_{HEX}, and the mean Nusselt number $\overline{Nu}^{i}_{HEX}$, evaluated through the Gnielinsky's correlation for coiled tube heat exchangers [8].

The external convective thermal resistance $R^{e}_{HEX,conv}$ is evaluated as:

$$R^{e}_{HEX,conv} = 1/\left(A^{e}_{HEX}\bar{h}^{e}_{HEX}\right) \tag{5}$$

with

$$\bar{h}^{e}_{HEX} = \left(\overline{Nu}^{e}_{HEX} k_{w,tank}\right)/L_{HEX} \tag{6}$$

where A^{e}_{HEX} is the area of external surface, and $\bar{h}^{e}_{HEX}$ is the mean external convective heat transfer coefficient based on the thermal conductivity of the corresponding node in the tank, $k_{w,tank}$, the coil length L_{HEX}, and the mean Nusselt number $\overline{Nu}^{e}_{HEX}$, evaluated through different correlations, namely the Morgan's correlation [9] and the one of Churchill and Chu [10] for natural convection in horizontal tubes, the Ali's correlation for helical coiled tubes [2,11]

$$\overline{Nu}^{e}_{HEX} = 0.106\ \overline{Ra}_{L}^{0.335} \tag{7}$$

where the Rayleigh number is calculated using the tube length, and the one of Prabhanjan et al. [12]

$$\overline{Nu}^{e}_{HEX} = 0.009759\ \overline{Ra}_{L}^{0.3972}. \tag{8}$$

In the following section, the numerical results relative to each of the above correlations are presented and compared with experimental results.

For each time-step and for each node, Eq. (1) is solved by using the implicit Euler method. The input to the model are the initial HTF temperature field, the HTF temperature and mass flow rate at the inlet section at each time-step, which are provided by experimental data, and the temperature distribution of water in the tank at each time-step. The output is the HTF outlet temperature relative to each node and at each time-step.

3.2. Thermal storage tank model

The water in the tank is considered subdivided into isothermal layers, characterized by the same water volume. The energy balance equation in transient regime for each node is formulated as:

$$\rho_w c_w V_w \frac{dT_w}{dt} = \frac{\overline{T}_{HTF} - T_w}{R^*_{HEX}} + \dot{Q}_{cond} - \frac{T_w - T_{amb}}{R_{tank}} \quad (9)$$

where ρ_w and c_w are the density and the specific heat, respectively, V_w is the volume, T_w is the water temperature, $\dot{Q}_{cond}$ is the conductive heat rate with the adjacent nodes, T_{amb} is the ambient temperature, and R_{tank} is the total thermal resistance related to the tank wall. As to the parameters $\overline{T}_{HTF}$ and R^*_{HEX} , for each tank node, $\overline{T}_{HTF}$ is evaluated as the weighted average of the nodes temperature relative to the coil sections intersected by the tank layer relative to the tank node, using as weights the corresponding R_{HEX} values evaluated by means of equation (2), while R^*_{HEX} is calculated as the sum of the R_{HEX} values relative to the coil sections intersected by the tank layer relative to the tank node. This is because at each tank node relative to a tank layer intersecting the diagonal part of the coil does not correspond only one coil node, as it is can be argued from Fig. 3. Indeed, in the diagonal part of the coil, the tank layers can intersect up to four coil sections.

For each time-step and for each node, Eq. (9) is solved by using the implicit Euler method. The empirical reversion-elimination algorithm [13,14] is implemented to take into account the effects of natural convection between the water layers at different heights in the thermal stratification inside the tank

3.3. Components coupling

The models formulated above are simulated by means of the Simulink tool of Matlab. The system components are coupled by using an iterative approach. At each time-step, the immersed coil heat exchanger is simulated by using the measured HTF temperature at the inlet section and mass flow rate, whereas the temperature distribution of water in the tank is that evaluated at the previous time-step. Then, the water in the tank is simulated by using the updated HTF temperature distribution in the coil heat exchanger. The process restarts from the simulation of the coil heat exchanger by using the updated water temperature distribution and goes on till convergence is achieved.

4. Results

The experimental results used to validate the developed simulation tool have been obtained by a thermal energy charging test of the tank water of 5244 s, with the experimental measures recorded with a sample time of 1 s. The charging test has been performed using water as HTF, with the water mass flow rate through the coil maintained at 0.37 kg/s, and with the temporal evolution of the measured water temperature at the coil inflow section shown in Fig. 4. The indoor ambient temperature has been regulated to 23 °C for the entire experiment duration. The above mass flow rate and temperatures have been imposed as boundary conditions in the unsteady numerical simulations.

The numerical results presented hereafter refer to four simulation runs, realized using 100 nodes for the tank water discretization, and a time-step of 1 s. Grid independence of results has been assured in all the reported cases. The four simulation runs differ from one another by the correlation used for the evaluation of the thermal resistance $R^e_{HEX,conv}$, relative to the convective heat transfer by natural convection at the coil external surface, introduced in section 3.1. In the following, the Ali's correlation is referred to as correlation 1, the one of Prabhanjan et al. as correlation 2, the Morgan's one as correlation 3, and the one of Churchill and Chu as correlation 4.

Figure 5 shows the experimental and simulated temperature profiles at different time instants. It can be noticed that, in all cases, the numerical results are very different from the experimental one in correspondence of the diagonal section of the coil. This can be explained considering that the experimental temperatures are measured along the tank axis, i.e. at the centre of the tank horizontal sections passing through the thermocouples hot junctions, and that in the low part of the tank, where the diagonal section of the coil is located, water is far from being well mixed. Since the measured water temperatures in this part of the tank are supposed to be lower than the average ones relative to the corresponding horizontal tank water sections, due to the fact that the highest temperatures are located close to the coil external surface where the convective plume develop, the numerical temperatures, which represent the average temperatures in the corresponding layers volume, result to be higher than the corresponding experimental ones. It can be seen that, in the upper part of the tank, where the water can be considered fully mixed, the results obtained with correlation 2 are in very good agreement with the experimental ones, while with the other ones the experimental temperatures are overestimated, especially with correlation 4, as confirmed by the root mean square error values relative to the tank water mean temperature, evaluated using all samples of the experimental temperature over the entire test duration, reported in Table 2.

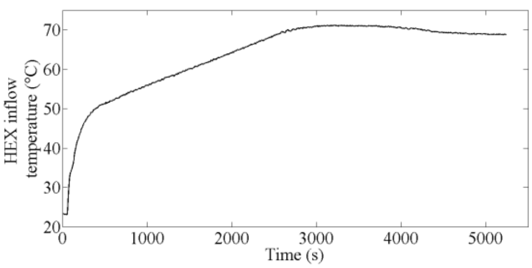

Fig. 4. Temporal evolution of water temperature at the HEX inflow

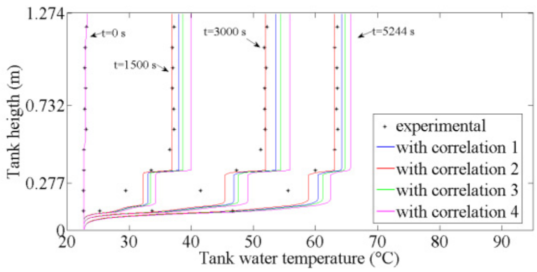

Fig. 5. Water temperature profiles in the tank

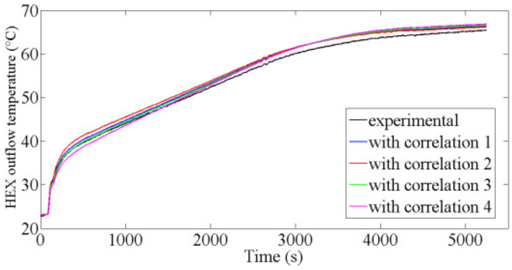

Fig. 6. Temporal evolution of water temperature at the coil outflow

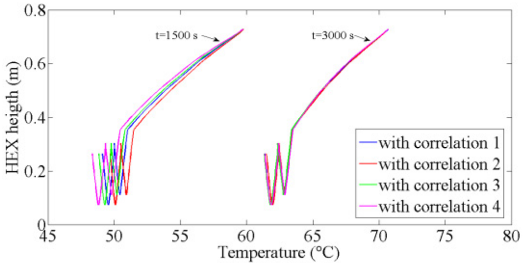

Fig. 7. Water temperature profiles in the coil

Table 2. Root mean square error relative to tank water mean temperature

Correlation	RMSE (°C)
1	0.84
2	0.66
3	1.36
4	2.45

Table 3. Root mean square error relative to the coil water outflow temperature

Correlation	RMSE (°C)
1	1.09
2	1.32
3	1.05
4	1.19

Figures 6 and 7 show the temporal evolution of the experimental and simulated water temperature at the coil outflow section, and the simulated water temperature profiles in the coil, respectively. It can be noticed that the quality of simulation of the water temperature at the coil outflow section is about the same for all the implemented correlations, and this is confirmed by the root mean square error relative to the coil water outflow temperature reported in table 3. Finally, owing to the above results, it is clear that correlation 2 represents the better choice for the present numerical tool.

5. Conclusion

In the present paper, the experimental validation of a numerical tool realized to simulate a commercial hot water storage tank has been presented and discussed. The details of the 1D models adopted for the numerical simulation of the temporal evolution of the tank water temperature, and of the HTF flowing through the immersed coil heat exchanger, as well as the main characteristics of the experimental facility, have been reported. The numerical results, obtained by using different correlations for the evaluation of the heat transfer rate between the tank water and the heat transfer fluid through the coil, have been compared with the experimental ones. The correlation performing the best fitting of the experimental data, and that has been finally selected for the present tool, has been indicated.

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Biography

Luigi Mongibello, PhD. He graduated in aerospace engineering at the University of Naples "Federico II" (Italy) in 2002, where he completed, in 2005, a doctorate in aerospace engineering with a thesis on actuators for flow control. His research activities are focused on the experimentation, simulation, and optimization of components and systems for solar collectors, CSP and DES plants, and on the thermal design of PV, BIPV and CPV systems.