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DEVELOPMENT OF A STRATIFIED TANK MODEL WITH IMMERSED HEAT EXCHANGERS IN ESP-R

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ABSTRACT

A stratified tank model with an immersed helical heat exchanger has been implemented in the plant domain of ESP-r. The model was developed in two phases: first a simple stratified tank model was developed; then a model for immersed heat exchanger was added to the simple model.

The stratified model divides the tank into up to 100 segments, or nodes; heat and mass transfer equations between these nodes are solved. In addition, a mixing algorithm simulates the effects of natural convection in situations where warmer segments are located below colder ones; this algorithm groups nodes in fully-mixed sections; heat and mass transfer equations are then solved for sections rather than nodes.

The immersed heat exchanger (HX) model is implemented as a subroutine within the stratified tank model. Heat transfer is governed by forced convection inside the coil, conduction through the coil wall, and a combination of free and forced convection outside the coil. The correlations used were chosen based on an extensive literature search.

The paper summarizes model development and provides comparisons of model performance against TRNSYS and against experimental data.

INTRODUCTION

The Sustainable Built Environment Group of Natural Resources Canada is currently working on the development of the next generation building energy simulation software HOT3000, which is based on the ESP-r simulation engine. Part of this development is the implementation of the capability to model four commonly available solar DHW (SDHW) systems, which will allow users of the HOT3000 software to assess and compare the performance of these different systems (Haddad *et al.*, 2007). Until now, the ESP-r plant modeling environment did not have a reliable model for the simulation of a stratified storage tank, which is needed for the proper modeling of solar DHW systems that employ a side-arm natural convection heat exchanger (that is, a heat exchanger mounted outside the tank, and working by natural convection). Another common type of solar DHW system features a stratified tank with an immersed heat exchanger (HX). Again, no such model existed in ESP-r. The work presented in this paper addresses these two modeling needs. First, a stratified storage tank model was added to ESP-r; it is described in the first part of this paper. Then, an immersed HX model was added; it is described in the second part of the paper. The last part of the paper presents a comparisons of the model predictions with TRNSYS model and with experimental data for a solar DHW system.

STRATIFIED TANK MODEL

Formulation

After a review of existing models from the literature, it was decided to base the stratified tank model on the TRNSYS Type 60 model (TRNSYS, 2006), which features up to 100 fully-mixed equal volume segments. To create an equivalent model in ESP-r, it was immediately apparent that the segments could not be modeled as individual 'nodes' in the ESP-r plant model. This would indeed significantly increase the size of the plant matrix, and hence the resolution time. It may also require to solve the whole plant model with very small time steps. Instead, it was decided to develop the model independently of the ESP-r formalism, then to 'wrap' an ESP-r component around it. The stratified tank model considered here features two inlet and two outlet ports. It was therefore natural to use a two-node ESP-r component, the two nodes representing the outlet temperatures of the tank. The model nodal representation is shown in *Figure 1*. It computes outlet temperatures θ_1 and θ_2 on sides 1 and 2, given the inlet temperatures, the flow rates, and the temperature of the environment:

$$\theta_1 = f_1(\dot{m}_i, \dot{m}_2, \theta_j, \theta_{j+1}, \theta_e, \Theta) \quad (1)$$

$$\theta_2 = f_2(\dot{m}_i, \dot{m}_2, \theta_j, \theta_{j+1}, \theta_e, \Theta) \quad (2)$$

where $\dot{m}_1$ and $\dot{m}_2$ are the incoming flow rates on sides 1 and 2, θ_j and θ_{j+1} represent the temperatures of the components feeding into the tank, θ_e is the temperature of the environment and Θ is an internal variable that describes the internal state (temperature profile) of the tank.

The corresponding matrix formulation:

$$\begin{pmatrix} 1 & 0 \\ 0 & 1 \end{pmatrix} \begin{pmatrix} \theta_1 \\ \theta_2 \end{pmatrix} = \begin{pmatrix} f_1 \\ f_2 \end{pmatrix} \quad (3)$$

can be plugged into the ESP-r solver, while marking θ_1 and θ_2 for iteration. This formulation leads directly to the calculation of θ_1 and θ_2 by the ESP-r solver.

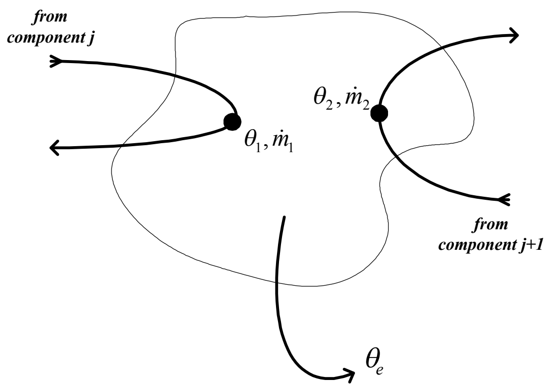

Figure 1 – ESP-r tank model.

Nodes and sections

The basic formulation consists in dividing the tank in a number of equal elementary segments, or *nodes*[1]. Each node is assumed to be homogeneous and can be represented by a single temperature. The state of the tank is obtained by solving the equations representing heat and mass transfer between nodes and their neighbours and between nodes and the environment. To obtain better accuracy, and also simply because the distance traveled by the forced motion of the fluid within a single simulation time step may exceed the distance between two nodes, the equations are solved using an *internal time step* δt smaller than the ESP-r plant solver *simulation time step* Δt used for the of the rest of the system.

Once the equations are solved, the temperature profile in the tank is obtained. In some cases, this profile may indicate temperatures rising steadily from the bottom to the top of the tank: such a configuration corresponds to a fully stratified tank, and is stable. However in other cases, such as when cold water is added to the top of a warm tank, the calculation may indicate that some nodes have a temperature lower than a node below them. Such temperature inversions are unstable and will lead to mixing through natural convection. The model handles mixing by lumping all nodes that are part of the inversion into one single, homogeneous *section*. The tank is therefore represented as a succession of homogeneous sections, each comprising several adjacent nodes; each section is fully mixed and can be represented by a single temperature. The sections are such that section temperatures are monotonically increasing from the bottom to the top of the tank.

The distinction between nodes and sections is fundamental to the understanding of this model. Nodes are defined once and for all and are the atomic building block of the model. The grouping of nodes into sections, however, is recalculated at every time step, and even several times during a single time step: when temperature inversions are discovered, the corresponding sections are lumped together and the system of equations is solved again. Note that the grouping of nodes in sections is something new compared to Type 60 in TRNSYS. The need for sections became apparent during the testing of the model, as without them the energy balance of the system was not satisfied. But a similar idea can be found in the work by B.J. Newton (1995), whose M.S. Thesis served as the basis for the development of Type 60:

> *Ideally, all nodes that get mixed should be combined into one large node, thereby allowing the energy balance equation (...) to account for the energy input and mixing phenomena simultaneously (...) It is recommended that the idea of combining nodes together be explored further because the mixing algorithm used by the model is the primary cause of errors when tank energy balances are solved.*

Energy balance at the node level

The tank consists of N identical nodes. An energy balance for node i is shown in *Figure 2*. The node is subject to:

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[1] In the context of the stratified tank model description, a *node* is an elementary volume of the tank. It should not be confused with the *nodes* used to describe ESP-r plant components, as used in the previous section.

- heat losses to the environment, $UA_S(T_e - T_i)$, where A_S is the surface area of the node in contact with the wall tank, and U the overall heal loss coefficient of the tank;

- conductive heat transfer between the node and the two nodes immediately above and below it, $\frac{(k_w + k_t)}{\Delta z} A_c (T_{i-1} - T_i)$ and $\frac{(k_w + k_t)}{\Delta z} A_c (T_{i+1} - T_i)$, where A_c is the cross-sectional area of the tank and k_w is the thermal conductivity of water. The term k_t represents an additional thermal conductivity equivalent to heat conduction along the wall;

- forced convective heat transfer from or to the segments above and below, that is, net fluid motion due to circulation of water brought by the two inlets and drawn through the two outlets. This is represented by the $\dot{m}_{up} C_p T_i \text{ or } \dot{m}_{down} C_p T_{i+1}$ and $\dot{m}_{up} C_p T_{i-1} \text{ or } \dot{m}_{down} C_p T_i$ terms. $\dot{m}_{up}$ should be understood as the resulting flow rate if flowing upwards, and zero otherwise; and $\dot{m}_{down}$ as the resulting flowrate if flowing downwards, and zero otherwise. The resulting flowrate is calculated algebraically, i.e. a flow upwards is positive, and downwards negative, from the flow rates imposed through the tank's inlets and outlets;

- water flow from the inlet or to the outlets themselves, represented by the $\dot{m}_{in} C_p T_{in}$ and $\dot{m}_{out} C_p T_i$ for those nodes where the inlets and outlets are located.

The energy balance of the node then becomes:

$$\begin{aligned}(M_i C_p)\frac{dT_i}{dt} &= UA_S(T_e - T_i) \\ &+ \frac{(k_w + k_t)}{\Delta z} A_c (T_{i-1} - T_i) \\ &+ \frac{(k_w + k_t)}{\Delta z} A_c (T_{i+1} - T_i) \\ &- \dot{m}_{up,i\to i+1} C_p T_i \text{ or } - \dot{m}_{down,i\to i+1} C_p T_{i+1} \\ &+ \dot{m}_{up,i-1\to i} C_p T_{i-1} \text{ or } \dot{m}_{down,i-1\to i} C_p T_i \\ &+ \dot{m}_{in} C_p T_{in} - \dot{m}_{out} C_p T_i\end{aligned} \tag{4}$$

where M_i is the mass of water in node i (the $M_i C_p$ term could be replaced by $\overline{M_i}\overline{C_p}$, where the bar denotes averaging of fluid and tank properties). Equation (4) is valid for all nodes except the bottom and top nodes. For these, the conduction term to the upper or lower nodes, respectively, disappear, and A_S is increased by the area of the bottom and top of the tank.

Equation (4) is discretized using the Crank-Nicholson scheme. The use of this method gives the possibility to switch between the explicit forward Euler scheme (first order, conditionally stable), implicit backward Euler scheme (first order, unconditionally stable) and the full Crank-Nicholson scheme (second order, unconditionally stable). The full Crank-Nicholson implementation is normally the one used, except when it is necessary to revert to another scheme such as forward Euler, for example because of the presence of internal heat exchangers (see later). The internal time step of the model, δt, in most cases will be smaller than the time step used to solve the plant equation, as this latter may be too large and include the motion of water through several nodes; typically, the time step used to solve the tank model is chosen such that vertical motions of liquid do not exceed a fraction of one node.

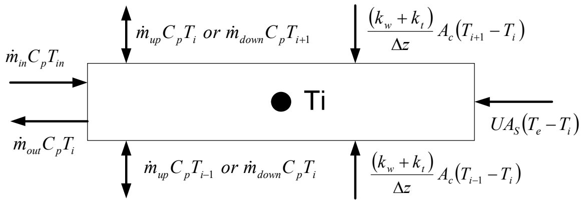

Figure 2 – Energy balance of tank node.

Mixing and energy balance at the section level

Once the various node temperatures are computed, the algorithm needs to mix nodes that show inversions. That is, nodes with temperatures out of ascending order (from bottom to top) are mixed together in a section and a new average temperature for the section as a whole is calculated. The mixing algorithm is an approximate and efficient way to simulate natural convection, the effect of which would be to homogenize parts of the tank that show an unstable temperature profile.

The simplest way to calculate the mixed section temperature is to equate it to the average temperature of the nodes it contains. However this raises a major difficulty: the node temperatures thus re-calculated do not satisfy the original equations (4) from which they were derived. The program handles this by re-solving the system of equations (4), written for sections instead of nodes. The system of equations (4) is therefore replaced by a similar system written for sections instead of nodes (note that some sections may have only one node), the various parameters taking on a new meaning:

i becomes the number of the section, M_i becomes the mass of the fluid in the whole section, A_S the surface area of the section, $\dot{m}_{in}$ and $\dot{m}_{out}$ are the sum of inlet and outlet flows within the section, etc.

At the beginning of an elementary time step δt, there are as many sections as there are nodes (the tank is assumed fully stratified). The equations are solved, inversions lead to nodes being grouped into sections; the equations are solved again, inversions lead to sections being grouped into larger sections, etc.

IMMERSED HEAT EXCHANGER MODEL

There are several possible immersed heat exchanger (HX) configurations for use in solar water heating systems. They are influenced by the intended use, the necessity or not to use the heat exchanger in a retrofit situation, and cost considerations. They include several shapes of the HX, several orientations or positions, and the possible presence of a baffle or outside fins. The work presented in this paper concentrates on the simple coil geometry summarized in *Figure 3*, and will be mostly applicable to a collector-side or load-side, vertical heat exchanger, although they may be applicable to other configurations as well.

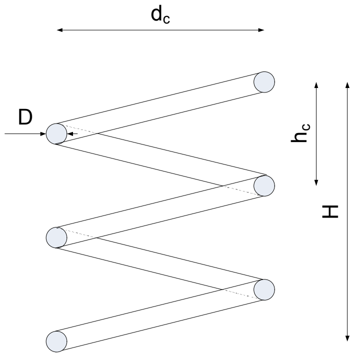

Figure 3 – Coil dimensions.

General form of heat transfer equation

The heat transfer coefficient between the HX fluid and the tank fluid is expressed (per unit length of coil) as:

$$\frac{1}{(U_c P)_i} = R_i + R_w + R_o \tag{5}$$

where P is the coil tube perimeter [m], U_c is the overall heat transfer coefficient of the coil [W/m^2/K], and R_i, R_w, and R_o are the thermal resistances due to forced convection inside the coil, thermal diffusion through the wall of the coil, and free or forced convection outside the coil. These resistances are placed in series, hence the form of equation (5).

A literature review was conducted to find the best estimates of R_i, R_w, and R_o. It can be found in Thevenard (2009) and its findings are summarized below. As is customary in fluid dynamics problems, correlations are best explained as relationships between non-dimensional numbers; in particular, heat transfer is expressed through the Nusselt number (Nu) rather than through U_c. A summary of non-dimensional numbers used in this paper can be found in the Appendix section.

Correlations for internal convection

Fluid flow inside the HX is classified as forced convection since the flow is imposed by external means – either the collector pump or the load draw. The heat transfer coefficient inside the pipe is a function of the velocity of the fluid, the cross-section of the pipe (circular, elliptic, inside finned), and the relative values of the coil and pipe diameters.

In the laminar regime, correlations developed for straight pipes are inappropriate; centrifugal forces create a secondary flow pattern consisting of two vortices perpendicular to the axial flow direction, which enhances significantly the rate of heat transfer (Kreith and Bohn, 2001). The Hemisphere Handbook of Heat Exchanger Design (1990) suggests the following equation due to Schmidt for laminar flow:

$$\mathrm{Nu} = 3.65 + 0.08 \cdot \left[1 + 0.8 \cdot \left(\frac{D}{d_c}\right)^{0.9}\right] \cdot \mathrm{Re}^m \cdot \mathrm{Pr}^{1/3} \cdot \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_w}\right)^{0.14} \tag{6}$$

$$m = 0.5 + 0.2903 \cdot \left(\frac{D}{d_c}\right)^{0.194} \tag{7}$$

where Pr is evaluated at the mean fluid temperature and $\mathrm{Pr_w}$ is evaluated at the wall temperature. Transition to turbulent flow occurs at:

$$\mathrm{Re_D} = 2{,}300 \cdot \left[1 + 8.6 \cdot \left(\frac{D}{d_c}\right)^{0.45}\right] \tag{8}$$

For fully turbulent flows, the Gnielinski correlation is recommended (Incropera and DeWitt, 1990):

$$\mathrm{Nu_D} = \frac{(f/8) \cdot (\mathrm{Re}_D - 1000) \cdot \mathrm{Pr}}{1 + 12.7\sqrt{f/8}\left(\mathrm{Pr}^{2/3} - 1\right)} \tag{9}$$

$$f = \frac{1}{\left(0.79 \cdot \ln(\mathrm{Re}_D) - 1.64\right)^2} \qquad (10)$$

Flow is fully turbulent for Re > $2.2 \cdot 10^4$. Between the onset to turbulence given by (8) and the fully turbulent flow, a combination of eq. (6) and (9) is used.

Conduction

Resistance to heat flow by conduction per unit length of pipe is expressed as (Kreith and Bohn, 2001; Haddad et al., 2007):

$$R_w = \frac{1}{h} = \frac{\ln(r_o / r_i)}{2 \cdot \pi \cdot k_w} \qquad (11)$$

where r_o is the outer radius of the coil, r_i is its inner radius, and k_w is the thermal conductivity of the coil.

Correlations for external convection

Fluid flow outside the HX is mostly driven by natural convection. When water is drawn from the tank, forced convection also exists as the fluid inside the tank is subject to an externally imposed motion. Natural convection fluid flows inside the tank are extremely complex and have been the subject of numerous studies, both experimental and numerical. Correlations try to capture general trends about heat transfer but do not claim to capture the complexity of the flows themselves.

General Nusselt number correlations for natural convection correlations take the form:

$$\mathrm{Nu} = C \cdot \mathrm{Ra}^n \qquad (12)$$

with typically C ~ 0.52 and n ~ 0.25 (Arora *et al.*, 2001; Liu *et al.*, 2003; Incropera and DeWitt, 1990). The length scale to use for Ra is the distance that a fluid particle travels in the boundary layer on the body (for a horizontal cylinder: $\pi \cdot D / 2$ - See Liu *et al.*, 2003).

For forced convection, Incropera and DeWitt (1990) suggest the Zhukauskas correlation:

$$\mathrm{Nu} = C \cdot \mathrm{Re}_D^m \cdot \mathrm{Pr}^n \cdot \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_c}\right)^{1/4} \qquad (13)$$

where Pr_c is evaluated at the coil wall temperature. Coefficients *C* and *m* are summarized in *Table 1*. *n* is 0.37 if Pr < 10 and 0.36 if Pr > 10.

In mixed mode, when both natural and forced convections are present, the overall Nusselt number can be estimated as (Incropera and DeWitt, 1990):

Table 1 – Coefficients for Zhukauskas correlation.

Re	C	*m*
1-40	0.75	0.4
40-1000	0.51	0.5
1000-20000	0.26	0.6

$$\mathrm{Nu}^3 = \mathrm{Nu}_F^3 + \mathrm{Nu}_N^3 \qquad (14)$$

where Nu_F and Nu_N represent the Nusselt numbers for forced and natural convection.

Implementation in ESP-r

The stratified tank with immersed HX is simply a variation of the regular stratified tank model described earlier; in particular, the formulation of eq. (1)-(3) still holds. The first node is still represents the tank, however the second node is replaced by the immersed HX; θ_2 now represents the outlet temperature of the HX, and θ_{j+1} represents the temperature of the fluid feeding into the immersed HX. A 3-node model can also be used to represent tanks with two immersed HXs (Thevenard, 2009).

The correlations outlined above result in the simple addition of a heat exchange term Q_{HX} to the right-hand side of the stratified tank model equations (eq. (4)). This term is calculated as:

$$Q_{HX} = \dot{m}_{HX} \cdot C_{p,HX} \left(T_{in,HX} - T_{out,HX}\right) \qquad (15)$$

where $\dot{m}_{HX}$ is the flow rate through the heat exchanger, $C_{p,HX}$ is the specific heat of the fluid through the heat exchanger, and $T_{in,HX}$ and $T_{out,HX}$ are the temperatures of the heat exchanger fluid for the inlet and outlet of the segment of heat exchanger comprised within the node.

The heat exchanger is divided into segments, each segment corresponding to a node of the tank. The calculation of $T_{in,HX}$ and $T_{out,HX}$ proceeds by exploring each segment of the heat exchanger, from inlet to outlet. $T_{in,HX}$ for the first segment is the inlet temperature of the heat exchanger, and for subsequent segments is the outlet temperature of the previous segment. The number of transfer units (NTU) is then defined as:

$$NTU = \frac{U_c \cdot \pi \cdot D \cdot L}{\dot{m}_{HX} \cdot C_{p.HX}} \qquad (16)$$

where U_c is the overall heat transfer coefficient (eq. (5)), *D* is the inside diameter of the coil and *L* is the length of the heat exchanger segment. The effectiveness is calculated as:

$$\varepsilon = 1 - e^{-NTU} \tag{17}$$

and the temperature of the fluid at the outlet of the segment is simply:

$$T_{out,HX} = T_{in,HX} - \varepsilon \cdot \left(T_{in,HX} - T_s\right) \tag{18}$$

where T_s is the temperature of the storage for the node in question.

Ideally the estimation of U has to be done not at $T_{in,HX}$ but at a heat exchanger fluid temperature representative of the whole segment. This representative temperature can be taken as

$$T_{HX} = T_s + LMTD \tag{19}$$

where the log mean temperature difference LMTD is defined as:

$$LMTD = \frac{\Delta T_{in,HX} - \Delta T_{out,HX}}{\ln\left(\Delta T_{in,HX} / \Delta T_{out,HX}\right)} \tag{20}$$

with $\Delta T_{in,HX} = \left(T_{in,HX} - T_s\right)$ and $\Delta T_{out,HX} = \left(T_{out,HX} - T_s\right)$. T_{HX} depends on the heat transfer between the HX segment and the tank node, which in turn depends on T_{HX}, so an iterative solution is required.

The computation of Q_{HX} for all the nodes is done by a procedure called from within the stratified tank calculation. It should be noted that the evaluation of Q_{HX} requires the knowledge of the tank temperature. This precludes the use of implicit (backward Euler) or semi-implicit (Crank-Nicholson) methods to solve for tank temperatures: the explicit method (forward Euler) has to be used. Since Q_{HX} is calculated using the temperatures from the previous time step, it can be calculated only once at the beginning of the step, before the mixing algorithm groups nodes into sections. This speeds up the calculation considerably.

TEST OF THE MODEL

The tank model, both with and without immersed heat exchanger, was tested extensively against theoretical calculations, the TRNSYS program, and experimental data. Comparisons to TRNSYS show that the heat transfer predicted by this model is slightly higher than what is estimated by TRNSYS, due to the use of different correlations for heat transfer between the HX and the tank (these correlations are thought to be more up-to-date and accurate than those used in TRNSYS; see Thevenard, 2009). Details about these tests can be found in Thevenard (2007, 2009). In this paper only the test against experimental data is shown, as well as a study showing the need for stratification for model accuracy.

Table 2 – Simulation parameters for the test against experimental data.

Tank	
Capacity	227 L
Inside height	1.403 m
Height of flow inlet	0 m
Height of flow outlet	1.403 m
Heat loss coefficient	0.695 W/m2/K
Coil	
Volume	2.7 USG
Heat transfer area	14.2 ft^2
Inlet	0.521 m
Outlet (a)	0.101 m
Outside coil diameter	0.0254 m
Inside coil diameter (c)	0.0274 m
Diameter of coil (d)	0.4 m
Pitch	0.0570
Conductivity of coil mat.	300 W/m/K
Mains water	
Temperature	16.3 °C
Ambient temperature	
Collector	Measured
Tank	18.0 °C

Test against experimental data

Experimental data came from test performed at the National Solar Test Facility NSTF for a complete solar DHW system, for a 'test day' and with a load of 300 L/day. The system includes two collector panels, a storage tank with an immersed heat exchanger, and a circulation pump between the collectors and immersed HX. Additional parameters representative of the system are summarized in Table 2. *Figure 4* compares simulation results to measured data (both were obtained by repeating the same days several times; only the last day is shown). It is apparent that the predicted temperature profile is closer to the observed temperature profile during the charge period and shortly after. The agreement is less satisfactory at night, where the tank seems to experience a faster temperature loss than is predicted; this may be due to an improper modeling of tank losses (one can only hypothesis the cause, but possible culprits could be an inaccurate ambient temperature, or inaccurate heat loss coefficient, or losses not taken into account by the model, such as losses through manifolds, or radiative losses exceeded what is modelled). Over the whole day, measured gross energy supplied by the system is 27.8 MJ; the simulated equivalent is 28.1 MJ. This illustrates that despite the complexity of the phenomena at play, the model captures reasonably well the stratification in the tank and the heat transfer between the various layers and the heat exchanger.

Effect of stratification on immersed HX

Finally, several runs were devoted to examining if, in the case of an immersed heat exchanger, the mixing effect induced by the presence of the heat exchanger was such that the tank could be considered as fully mixed. Year-long simulations were run, with the immersed heat exchanger connected either to the collector-side or to the load side of the system, and with either one node (fully mixed tank) or ten nodes (fully stratified tank). Details can be found in Thevenard (2009). The comparative runs show that the stratified tank model predicts values of heat transfer that are higher by 4 to 8% on an annual basis. This indicates that in most cases one cannot make the economy of running a fully stratified model when considering an immersed HX.

CONCLUSIONS

A stratified tank model was developed for ESP-r. The model is based loosely on TRNSYS type 60. It includes a TRNSYS-like function, a wrapper that serves as interface between that function and ESP-r, and a dummy two-node ESP-r plant component. The two nodes represent the outlet temperatures of the stratified tank. These temperatures are calculated entirely outside of the ESP-r plant solver and are therefore marked for iteration inside ESP-r.

The calculation method relies on the division of the tank into up to 100 segments, or *nodes*; heat and mass transfer equations between these nodes are solved. In addition, a mixing algorithm simulates the effects of natural convection in situations where warmer segments are located below colder ones; that algorithm groups nodes in fully-mixed sections; heat and mass transfer equations are then solved for sections rather than nodes.

This paper also summarized the development of an immersed heat exchanger model within the stratified tank model of ESP-r. The model is implemented as a subroutine, called by the stratified tank model, which calculates the energy transferred from the HX to the tank. Heat transfer is governed by forced convection inside the coil, conduction through the coil wall, and a combination of free and forced convection outside the coil. The correlations used were chosen based on an extensive literature search.

The model has been tested against TRNSYS and against one-day rating experimental data for a commercially available solar DHW system. The heat transfer predicted is slightly higher than what is estimated by TRNSYS, but seems to match reasonably well experimental data.

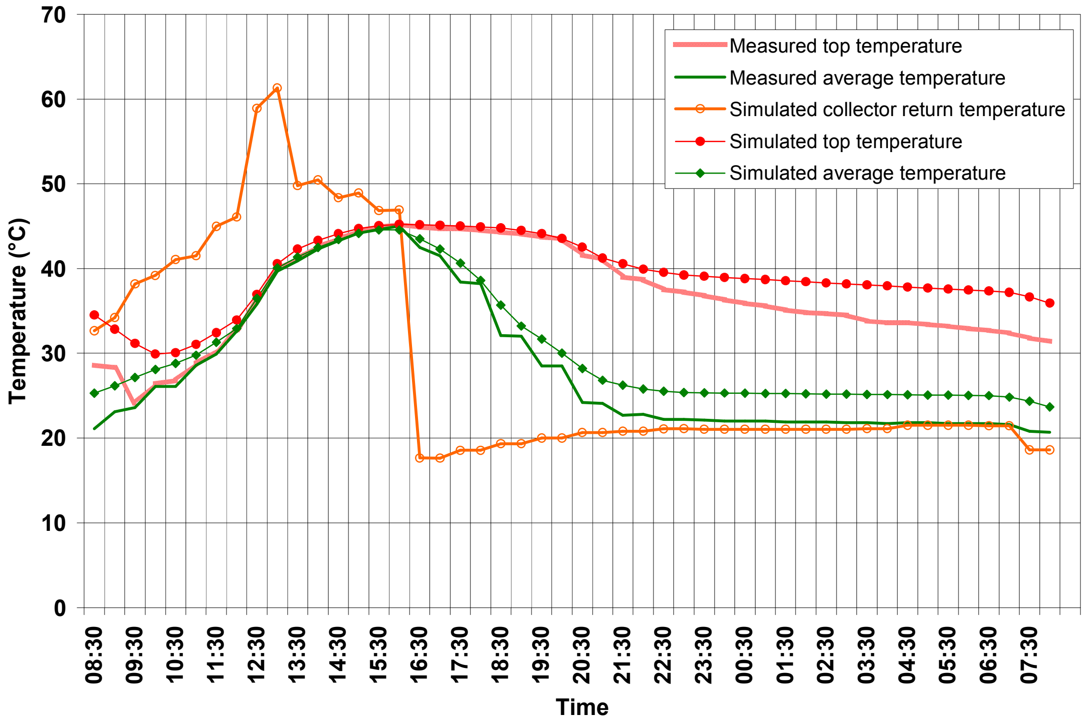

Figure 4 - Comparison of the model against experimental data.

ACKNOWLEGEMENTS

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APPENDIX: NON-DIMENSIONAL NUMBERS FOR HEAT TRANSFER CALCULATIONS

Heat and mass transfer in fluids is best expressed by relationships between non-dimensional numbers. The definition of some of these numbers that appear in the body of the article is summarized here (see Incropera and DeWitt, 1990; Kreith and Bohn, 2001). In those definitions, the following variables being defined as:

D	characteristic dimension of the object under consideration [m]
g	acceleration of gravity [m/s^2]
k	thermal conductivity of fluid [W/m/K]
U	heat transfer coefficient [W/m^2/K]
V	velocity of the fluid [m/s]
α	thermal diffusivity of fluid [m^2/s]
β	coefficient of volumetric thermal expansion [1/K]
ΔT	temperature differential [K]
μ	dynamic viscosity of fluid [N·s/m^2 or Pa·s or kg/m/s]
ν	kinematic viscosity [m^2/s]
ρ	density [m/s^2]

Nusselt number Nu

$$\mathrm{Nu} = \frac{U \cdot D}{k_f} \tag{21}$$

The Nusselt number is a dimensionless temperature gradient at the surface.

Prandtl number Pr

$$\mathrm{Pr} = \frac{\nu}{\alpha} \tag{22}$$

The Prandtl number is the ratio of the momentum and thermal diffusivities.

Reynolds Number Re

$$\mathrm{Re} = \frac{V \cdot D}{\nu} \tag{23}$$

or

$$\mathrm{Re} = \frac{V \cdot \rho \cdot D}{\mu} \tag{24}$$

The Reynolds number represents the ratio of the inertia and viscous forces.

Grashof number Gr

$$\mathrm{Gr} = \frac{g \cdot \beta \cdot D^3 \cdot \Delta T}{\nu^2} \tag{25}$$

The Grashof number is the ratio of buoyancy to viscous forces.

Rayleigh number Ra

$$\mathrm{Ra} = \frac{g \cdot \beta \cdot D^3 \cdot \Delta T}{\nu \cdot \alpha} = \mathrm{Gr} \cdot \mathrm{Pr} \tag{26}$$

The Rayleigh number is the product of the Grashof and Prandtl numbers.