Model-based Co-Simulation of Heat Pump Water Heater with Embedded Phase Change Materials Thermal Energy Storage

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ABSTRCT

The study analysis the behavior of a new developed heat pump water heater technology which integrates a phase change materials storage with a standard heat pump water heater to maximize the performance parameters of the Unified Energy Factor (UEF) and First Hour Rating (FHR). A model-based control development co-simulation platform is developed to include equipment models, such as heat pump, standard water tank, phase change materials storage tank, and integrate them with control model to support controls design, analysis, verification, and validation. Simulation results are compared with lab test results to validate the accuracy of developed co-simulation platform.

1 INTRODUCTION

Although the water heaters market is dominated by conventional thermal energy storage tank-type electric or gas fired heaters, due to 2-3 times more energy-efficient compared to conventional water heating methods, and great potential on CO_2 emission reduction, heat pump water heaters (HPWH) is projected to reach $2.1 billion market size by 2026 at compound annual growth rate (CAGR) of 6.8%. The HPWH uses the vapor compressor cycle to pulls energy from the environment and transfer it to water in a storage tank through condenser coil which either wraps tightly around the tank or immerses inside of tank. Compared to conventional thermal energy storage tank-type electric or gas fired heater by industry standard performance test methods, the HPWH outperforms them significantly in the Unified Energy Factor (UEF), but far behind in FHR than a conventional gas storage tank. To improve the HPWH's FHR, a potential solution is to use a secondary tank with filling in phase change material (PCM) capsules since PCMs have high energy density and can deliver significant amount of heat when solidifying. In addition, HPWH with PCM thermal energy storage can also alleviate the issue of high peak electricity demand occurred in the morning and evening when using electric resistance water heater. This study is to develop model-based co-simulation platform for performance prediction to support integrated HPWH with PCM thermal energy storage system development.

2 SYSTEM DESCRIPTION

A commercially available HPWH was selected as the baseline system to be integrated with a small thermal energy storage tank which contains PCM spheres inside. This integrated HPWH with PCM thermal energy storage system with extensively instruments was shown in a detailed schematic (Figure 1). The refrigeration system side of HPWH was instrumented with pressure transducers, thermocouples and mass flow meter. Pressure transducers (Omega PX309) with a reading accuracy of 0.25% were installed in the refrigerant line at the compressor and electronic expansion valve inlet and outlet locations. Externally insulated T-type thermocouples were attached to the refrigerant lines in the same locations. A Micro Motion CMF025 Coriolis mass flow meter was installed between the condenser coil and the expansion valve. For the HPWH tank, six thermocouples were installed uniformly

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[1] This manuscript has been authored in part by UT-Battelle, LLC, under contract DE-AC05-00OR22725 with the US Department of Energy (DOE). The US government retains and the publisher, by accepting the article for publication, acknowledges that the US government retains a nonexclusive, paid-up, irrevocable, worldwide license to publish or reproduce the published form of this manuscript, or allow others to do so, for US government purposes. DOE will provide public access to these results of federally sponsored research in accordance with the DOE Public Access Plan (http://energy.gov/downloads/doe-public-access-plan)

throughout the height of the water tank. Thermocouples were also installed at the water heater inlet and outlet. The temperature measurement uncertainty is 0.5 °C, and for an energy balance of the water inflow and outflow, the combined uncertainties propagate to less than 3% relative error in terms of energy consumption. One thermocouple was set inside of small PCM tank to measure the PCM temperature change, and the PCM tank inlet and outlet temperatures was recorded as well.

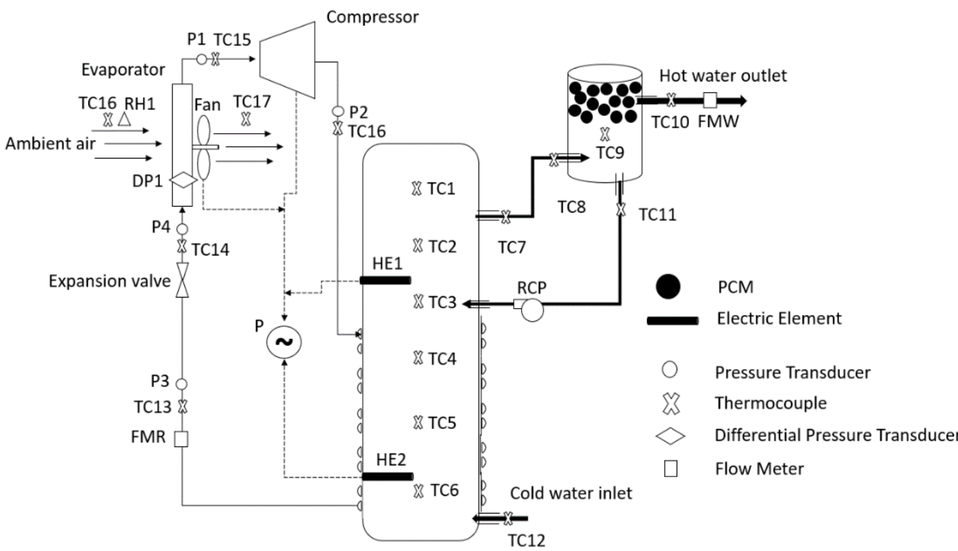

Figure 1: System diagram of HTHP with PCM thermal energy storage

3 MATHMATICAL MODEL

3.1 Water heater tank

Two types of heat pump water heater tanks are generally used: immersed coil based water tank and wrapped around coil based water tank.

The sensible water tank was simulated through one dimensional dynamic modeling based on conservation laws of mass, energy and momentum. the governing equations are referred to as Navier-Stokes equation,

$$\frac{\partial \rho}{\partial t} + div(\rho u) = 0 \tag{1}$$

$$\frac{\partial (\rho u)}{\partial t} + div(\rho u u) = -p\ div(u) + div(\mu\ grad\ T) + \emptyset\ + S_i \tag{2}$$

$$\frac{\partial (\rho i)}{\partial t} + div(\rho i u) = -\frac{\partial p}{\partial x} + div(k\ grad\ u) + S_i \tag{3}$$

Both mass and momentum balance are assumed to be steady state due to limited operating temperature range and negligible internal forces over the tank temperature. The governing equations in a control volume (Figure 2) can be rewritten as,

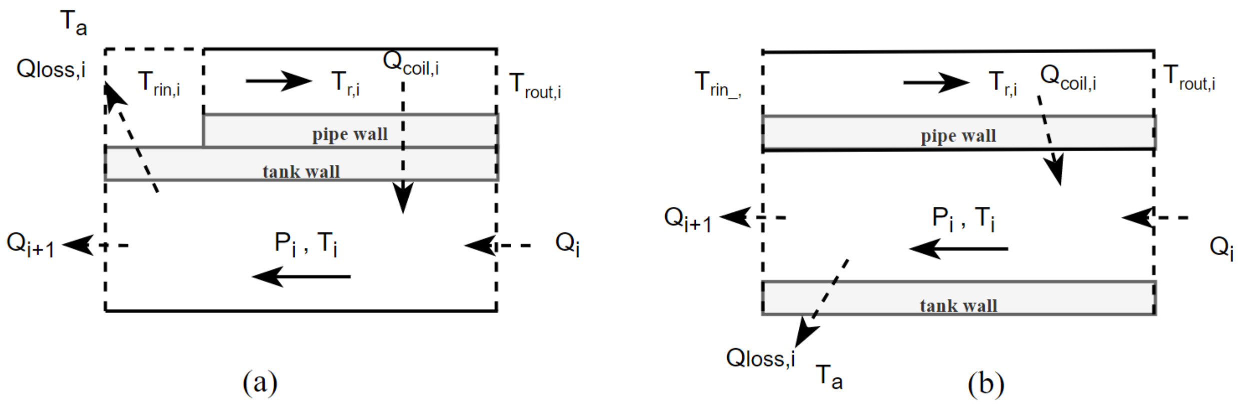

Figure 2: Control volume schematics of (a) wrapped around coil based water tank and (b) immersed coil based water tank.

$$0 = m_i - m_{i+1} \tag{4}$$

$$\frac{\partial(\rho u)}{\partial t} = (m_i c_p T_i - m_{i+1} c_p T_{i+1}) + Q_{coil,i} - Q_{loss,i} + (Q_i - Q_{i+1}) \tag{5}$$

$$0 = (p_i - p_{i+1}) + dp_{f,i} - \rho g z_i \tag{6}$$

$Q_{loss,i}$ is the heat transfer rate from the water heater tank to ambient through tank wall.

$$Q_{loss} = UA(T_i - T_a) \tag{7}$$

Where, U is the overall heat transfer coefficient, A is the overall heat transfer area. T_a is the ambient temperature

Q_i - Q_{i+1} is the net internal heat transfer rate of ith control volume, calculated by,

$$Q_i - Q_{i+1} = \frac{kA}{\Delta x}(T_{i+1} - 2T_i + T_{i-1}) \tag{8}$$

$Q_{coil,i}$ is the heat transfer rate from refrigerant to the water heater tank wall during charge period.

$$Q_{coil,i} = m_r C_{ri}(h_{rin,i} - h_{rout,i}) \tag{9}$$

The total heat transfer rate between refrigerant to the water heater tank wall is,

$$Q_{coil} = \sum_{i=1}^{n} Q_{coil,i} \tag{10}$$

As described in previous section, the dynamics of heat pump system is much faster than dynamics of water and PCM thermal energy storage tanks. The heat pump system is modeled as steady state operation. Nash et al (2017) noticed storage tank fluid experiences a quadratic increase in temperature during charge cycle, then assumed a quadratic temperature reduction profile for coil fluid. This quadratic temperature profile assumption is not applicable for refrigerant in HPWH coil if heat pump operates in subcritical region since refrigerant in most region of coil is in two-phase state and maintains at constant condensation temperature. for HPWH application, a quadratic enthalpy profile, instead of quadratic temperature profile, for coil inlet to outlet is used for refrigerant in coil.

3.2 PCM thermal energy storage tank

The PCM thermal energy storage tank is designed as a 6 to 10 gallons water tank with containing PCM capsules. For simplification, all these PCM capsules are considered to be equivalent to a PCM cylinder placed inside tank with water flowing over the surface, as shown in Figure 3.

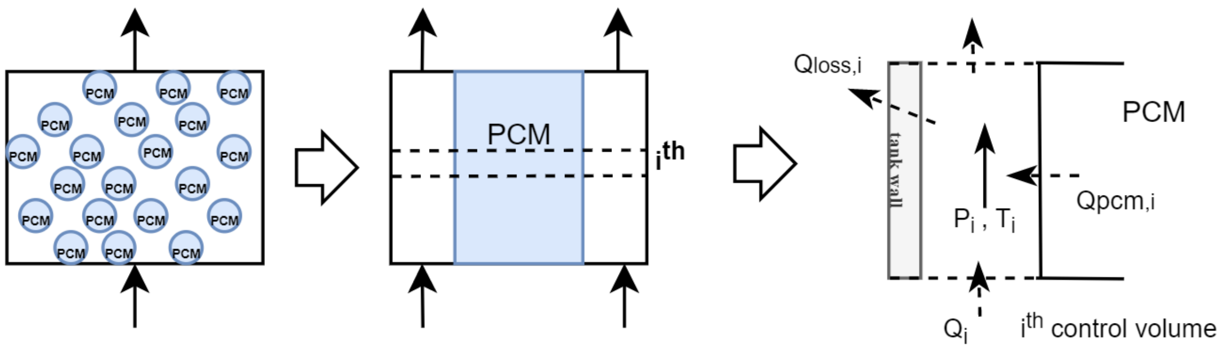

Figure 3: PCM tank modeling strategy

Similar as the water heater tank, the dynamics of PCM thermal energy storage tank are mainly captured through energy conservation. The mass and momentum conservation are assumed steady-state. The energy balance in the ith control volume is given by,

$$\frac{\partial(\rho u)}{\partial t} = (m_i c_p T_i - m_{i+1} c_p T_{i+1}) + Q_{pcm,i} - Q_{loss,i} + (Q_i - Q_{i+1}) \tag{10}$$

$$m_{pcm,i} C_{pcm,i} \frac{dT_{pcm,i}}{dt} = Q_{pcm,i} = UA_i (T_{pcm,i} - T_i) \tag{11}$$

Where, UA is overall heat transfer coefficient between PCM and tank water, which can be calculated through a function of fraction of PCM liquid and solid.

PCM can store large amount of latent heat in a relatively small temperature range, the enthalpy goes through a rapid increasing during the melting process. In general, three approaches are used to calculate the enthalpy of PCM (Buschle et al. 2006): linear interpolation method, Arc tangent method, and error function method. The arc tangent method is used in this study since it describes the enthalpy as a continuous and smooth function which is preferred to dynamic simulation. The arc tangent function is (Buschle et al. 2006, Leonhart & Muller, 2009),

$$h = h_{trans} \left\{ \frac{arctan\left[(T_{pcm} - T_{trans}) \times R_{trans} \right]}{\pi} + 0.5 \right\} + C_{p,s} \times (T_{pcm} - T_0) \tag{12}$$

Where, h_{trans} denotes specific enthalpy of transition, T_{trans} is temperature of transition, R_{trans} represents width of transition, C_p denotes specific heat capacity, T_0 is reference temperature.

The specific heat capacity of PCM can be calculated as (Leonhart & Muller, 2009),

$$C_p = \frac{h_{trans}}{\pi \left[\left((T_{pcm} - T_{trans}) \times r_{trans} \right)^2 + 1 \right]} \times r_{trans} + C_{p,s} \tag{13}$$

3.3 Heat pump system

Compared to water heater tank and PCM thermal energy storage tank, heat pump system experience much fast transit response to a change and can be represented with steady-state model in a HPWH-TES system. Except for the condenser, which is the water heater tank, the compressor, evaporator and expansion valve are simulated with below steady-state models.

Compressor

Semi-empirical modeling strategy is used for predicting the compressor performance. The volumetric efficiency, η_v, is calculated as a function of compressor pressure ratio by,

$$\eta_v = C_1 + C_2 \times (P_d / P_s)^{1/\gamma} \tag{14}$$

where P_s and P_d are compressor suction and discharge pressure, C_1, C_2 are constants, γ is isentropic exponent.

Accordingly, the mass flow at compressor suction port, m_r , is given by,

$$m_r = \eta_v \rho_s V_{\text{disp}} \tag{15}$$

Similarly, a semi- empirical 3-term model (Bourdhouxhe et al. 1999) is used to estimate the compressor input power consumption W,

$$W = \left[C_1 \left({}^{P_d}/_{P_s} \right)^{\gamma - 1/\gamma} + C_4 \right] \times P_s V_s + C_5 \tag{16}$$

Where C_3, C_4, C_5 are constants.

Finally, the energy balance of compressor is given by,

$$W = m_s (h_d - h_s) + Q_{loss} \tag{17}$$

Where h_s and h_d are compressor suction and discharge enthalpy, Q_{loss} is ambient heat loss, can be calculated through an empirical function of ambient temperature and compressor discharge temperature (Kim and Bullard 2002).

Evaporator

In this study, the evaporator simulation uses the zone modeling strategy which divides the evaporator into two zones: superheated zone and two-phase zone. In each zone, a lump model is used. the heat transfer between refrigerant and air is balanced with the capacities calculated based on refrigerant side and air side. For refrigeration side, the heat transfer rate of superheat zone Q_{ref_sh} and two phase zone Q_{ref_tp} are given by,

$$Q_{ref\_sh} = m_r \times (h_{ro} - h_{rs}) = U_{sh} f_{sh} A \Delta T_{m\_sh} \tag{18}$$

$$Q_{ref\_tp} = m_r \times (h_{rs} - h_{ri}) = U_{tp} f_{tp} A \Delta T_{m\_tp} \tag{20}$$

Where m_r is refrigerant mass flow rate, h_{ri} ,h_{ro} are refrigerant enthalpy at evaporator inlet and outlet, h_{rs} is the refrigerant enthalpy at saturated vapor refrigerant state (quality = 1), A is evaporator surface area, f_{sh} ,f_{tp} are ratio of superheat zone and two phase zone surface area to evaporator surface area, ΔT_{m_sh} , ΔT_{m_tp} is log-mean-temperature difference at superheat and two phase zone, U_{sh} , U_{tp} are overall heat transfer coefficients at superheat zone and two phase zone.

With considering the air side heat transfer, the energy balance becomes,

$$Q_{ref\_sh} + Q_{ref\_tp} = m_a \times (h_{ai} - h_{ao}) - m_w h_w \tag{19}$$

Where m_a is air mass flow rate, h_{ai} ,h_{ao} are evaporator entering and leaving air enthalpy, h_w is condensate enthalpy, m_w is condensate mass flow rate, can be given by,

$$m_w = m_a(W_{ai} - W_{ao}) \tag{20}$$

Where W_{ai},W_{ao} are evaporator entering and leaving air humidity ratio, which are calculated based on air pressure, air dry and wet bulb temperature. The enthalpy potential method with an apparatus dew-point temperature is used for the latent capacity calculation (Kuehn et al. 1998).

Expansion valve

The expansion valve performance is simulated based on a semi-thermodynamical model, which will balance the refrigerant mass flow rate prediction with compressor model to converge the system simulation. The expansion process is assumed to be isentropic process, the mass flow rate is function of pressure drop across the valve as,

$$m_v = C_v \sqrt{\rho_{in} \Delta P_v} \tag{21}$$

Where C_v is flow coefficient and can be either a constant or a function of opening position as below,

$$C_v = K_1 + K_2 \times Pos + K_3 \times Pos^2 \tag{22}$$

Where K_1, K_2, K_3 are constants, Pos represents the valve opening in percentage and is the input variable of the simulation, which can be obtained through controller output.

4 MODEL-BASED CO-SIMULATION PLAFORM

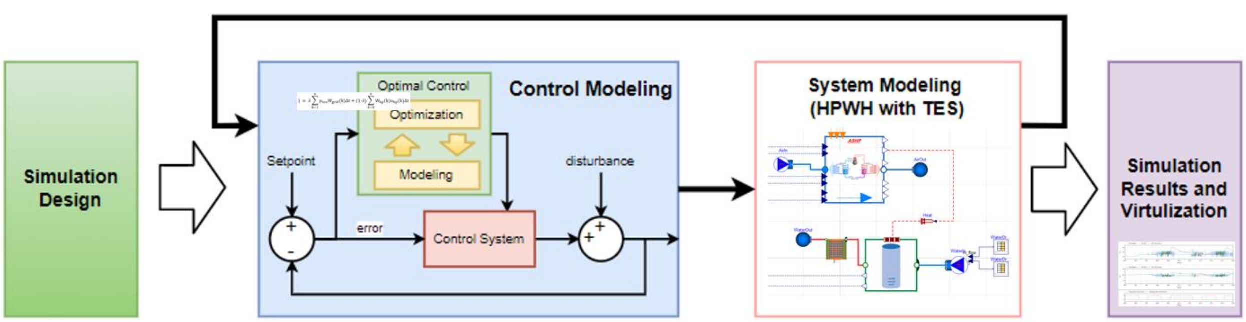

Figure 4: Co-simulation platform for HTHP-TES

The model-based co-simulation platform developed in this study is a closed loop modeling and simulation to integrate the heat pump water heater with thermal energy storage simulation with the control strategies. This platform can be used to design, develop, and evaluate the system design and control strategy before actual implementation and testing on the real system, which allows to quickly detect the controls error during evaluation and design stage. As shown in Figure 4, this platform is composed of four modules: simulation design, control strategy modeling, heat pump water heater with thermal energy storage simulation, and simulation results & virtualization. Simulation design is to define a series of simulation scenarios or user cases to automatically exam different control and design strategies according to predefined test requirements and targets. It will generate input signals to the control strategy modeling modular to allow closed loop simulation between control module and building module. A model representing the control logic will be developed based on control targets which define how the heat pump water heater system with PCM thermal energy storage tank should be operated. This control model can be simulated in closed loop with the models of HPWH-TES to enable rapidly and early identification of missing or inconsistent in control performance. As the core of the model-based co-simulation platform, steady state and /or dynamic modeling/simulation engine to represent the actual HPWH-TES system so that the design and control strategy can be tested and verified. The simulation results will be generated through closed loop simulation and be compared with predefine desirable control performance to determine if the system and control design achieve the targets

5 SIMULATION RESULTS

A 50-gallon HPWH was used in this case study to integrate with a 6-gallon PCM thermal energy storage tank. The PCM capsules are 2 cm in diameter with parameters listed in table 1.

Table 1: PCM parameters

Parameters	Units	Value
Latent heat of melting,	kJ/kg	201
Transition Temperature,	ºC	48.9
Solid specific heat capacity,	kJ/kg K	1.74
Liquid specific heat capacity,	kJ/kg K	2.03
Melting range coefficient, width of transition,	^{0}C	5.6

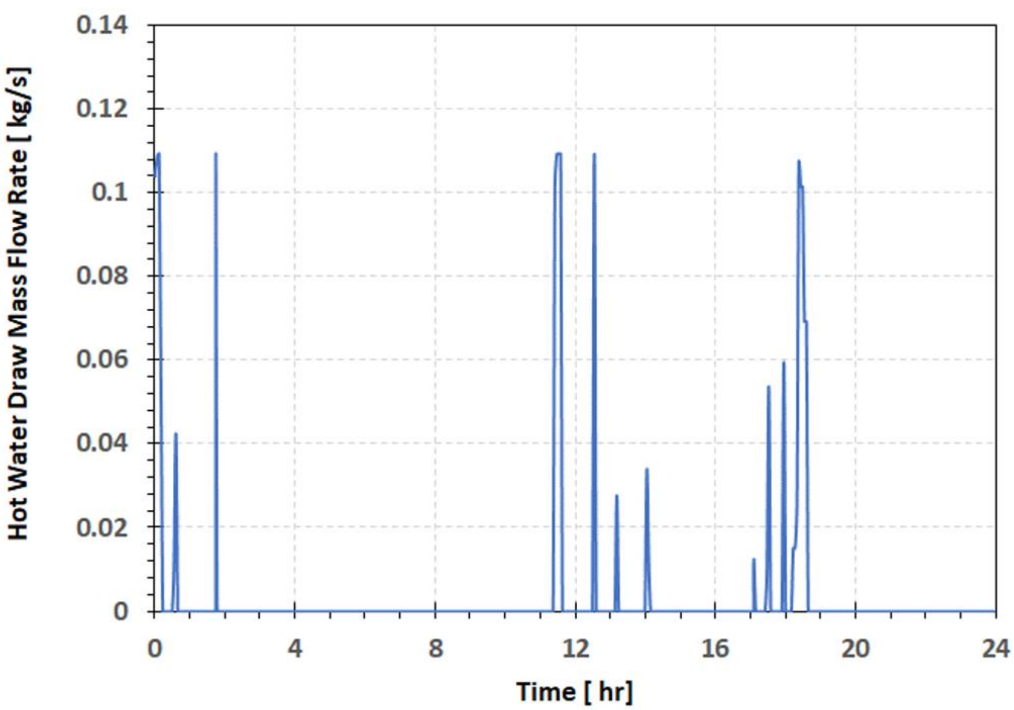

Figure 5. Heat pump water heat water draw schedule

The developed model-based co-simulation platform was used to predict a daily operational performance of the proposed HPWH with PCM thermal energy storage system under defined hot water draw schedule (Figure 5). The estimated tank outlet temperature and average temperature profiles are plotted in Figure 6. Compared with test data, the developed co-simulation platform maintains reasonable model accuracy and can be used for performance predict of the proposed HPWH with PCM thermal energy storage system under various operation condition.

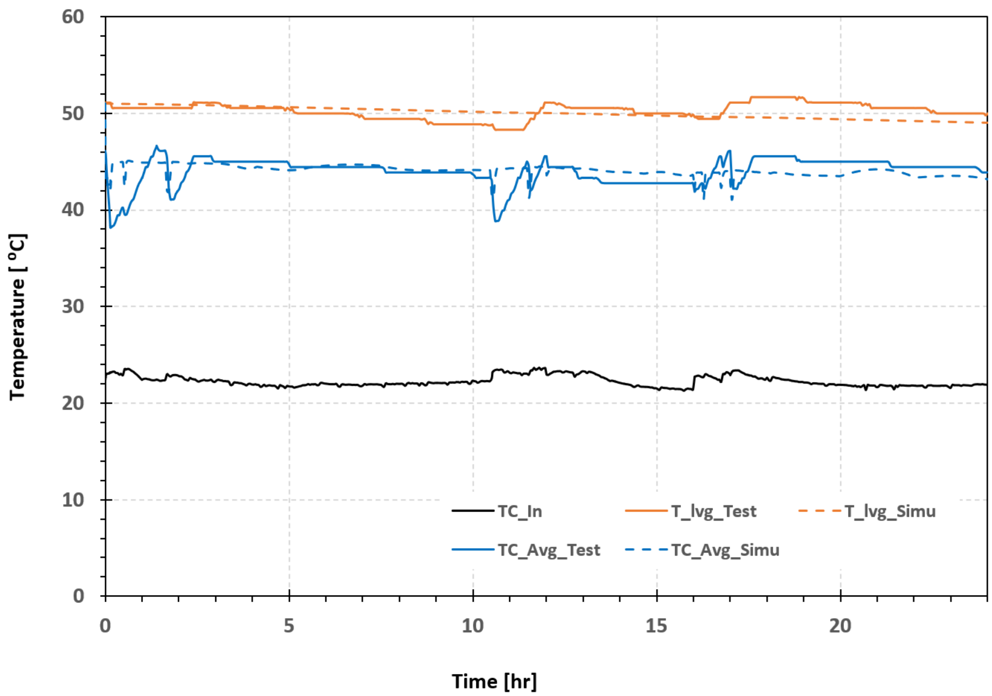

Figure 6. Leaving tank and average tank water temperature: test vs. simulation

A UEF simulation was also conducted using proposed co-simulation platform, and the simulation results are listed in table 2.

Table 2: UEF Simulation

Parameters	Test	Simulation	Error
UEF	3.61	3.93	8.9%
Standby heat loss coefficient of tank	1.89	2.11	11.6%
Daily water heating energy consumption, W	6076	5996	-1.3%

Compared to test results, the model prediction presents good match on daily water heat energy consumption (-1.3%), but relatively large error in UEF calculation (8.9%) and tank standby heat loss (11.6%), which indicates that further improvements are needed for the current model to better predict the temperature changes inside water tank. A few possible improvements are as below,

- Quadratic enthalpy assumption applied from HPWH coil in refrigeration side generates error and can be improved with more accurate profile based on lab test data.
- For heat pump refrigeration side, replacing steady state model with dynamic modeling can better capture the system transient performance to increase the accuracy of simulating heat transfer between vapor compression cycle and water tank.

- Introducing correction to the heat transfer between internal nodes inside of tank to improve the water temperature prediction inside tank.

6 CONCLUSTION

A model-based co-simulation platform was developed to predict the performance of a new developed heat pump water heater technology which integrates a phase change materials storage with a standard heat pump water heater. The co-simulation platform includes simulation design module, control strategy modeling module, HPWH with PCM thermal energy storage simulation module, and simulation results & virtualization module. In HPWH with PCM thermal energy storage simulation module, models are developed for heat pump, standard water tank, phase change materials storage tank. Performance predictions are compared with lab test results to validate the accuracy of developed co-simulation platform for supporting controls design, analysis, verification, and validation. Also, a few opportunities are identified to further improve the model accuracy for future work.

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ACKNOWLEDGE

Funding for this research was provided by the US Department of Energy (DOE), Office of Energy Efficiency and Renewable Energy (EERE). The authors would like to thank Mr. Antonio Bouza, Program Managers of Building Technology office (BTO), for his support of this work.