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Numerical analysis of Thermal Storage Systems for Residential and Small Commercial Retrofit Applications by using PCM

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Abstract

Environmental cooling loads comprise 11% of overall electric usage in the United States, and during peak cooling months in hot/humid climates these cooling loads can comprise more than 50% of peak electric energy loads. Housing and small commercial buildings generate much of the cooling responsible for peak electric load, but these infrastructures may not have the space or facilities to support conventional sensible energy thermal storage systems, such a chilled water thermal storage tanks. For these applications a compact, high-density, retrofit-table the change material) based latent thermal storage system for these applications. The outcome show that a simple tube-encapsulated, tetradecane PCM-based thermal storage system can reduce the size of a thermal store by at a factor of between 2 and 4 while providing acceptable energy recovery rates.

Keywords: Thermal storage; PCM (Phase Change Material); retrofit; tetradecane

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1. Introduction

Electric energy, without any utility-scale or distributed energy storage system, must be generated exactly as it is needed. To meet this requirement, electric utilities dispatch generation capacity in stages. These stages consist of baseload, intermediate, and peak generation power plants [1]. Baseload power plants are generally inexpensive to operate, but have long ramp-up periods and cannot respond quickly to changes in demand; these are typically run continually at their rated capacity. Intermediate power plants have moderate start-up periods, but their dispatch can be predicted and they can be brought on-line over the course of a day in anticipation of demand. Peak power plants are fast-response systems capable of mediating the uncertainty between generation and demand; these are generally more costly to operate, less efficient, and higher polluting that baseload or intermediate power plants.

Intermittent renewable energy resources, such as wind and solar, add uncertainty to generation capacity. Solar and wind power generation is expected to provide 20% of all U.S. electrical power by 2040 [2]. As the use of these intermittent resources increases, the added uncertainty in generation requires an increased use of less-efficient and costly fast-response peak power plants. Shifting peak electric energy loads to non-peak hours or periods when intermittent renewable resources are available can make better use of intermittent renewable resources, as well as baseline generation capacity, while reducing the need for inefficient, costly, and polluting peak power plants [3].

Environmental cooling loads comprise 11% of overall electric usage in the United States, and during peak cooling months in hot/humid climates these cooling loads can comprise more than 50% of peak electric energy loads. Data from the Electric Reliability Council of Texas [4] confirms this claim, as shown in Figure 1. This figure also shows that the largest contributors to weather sensitive load are residential and small commercial users.

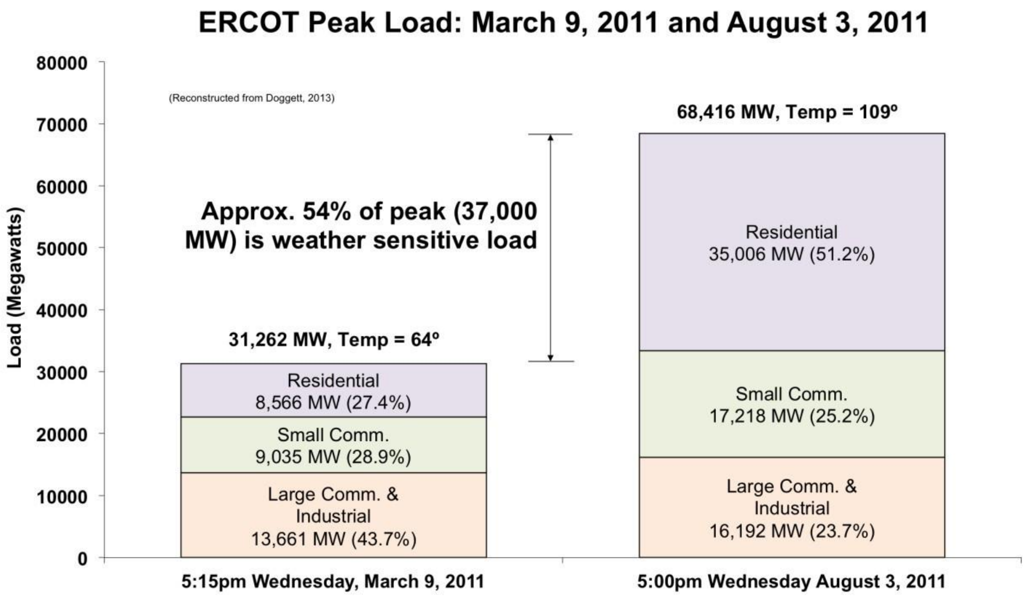

Fig. 1. ERCOT Peak loads for March 3 and August 3 of 2011. This figure demonstrates that the weather-sensitive portion of peak summer electric load is primarily due to residential and small commercial users, ostensibly as HVAC demand. Recreated from [4].

Thermal storage systems can shift these environmental cooling loads to non-peak hours. While residential and small commercial buildings generate much of the cooling responsible for peak electric load, these buildings may not have the space or facilities to support conventional sensible energy thermal storage systems, such as chilled water tanks. In addition, residential and small commercial structures have long service lifespans that can average 37 and 28 years respectively [5,6]. For these applications a compact, high-density, retrofittable thermal storage system is needed.

Thermal energy can be stored sensibly, through a change in temperature, or it can be stored latently, as a change in phase of a material – for instance, from a solid phase to a liquid phase. Phase change processes are characterized by large changes in enthalpy at constant or near-constant temperatures. Phase change latent energy storage systems store energy at greater density and over a smaller temperature range than sensible energy storage systems, such as conventional chilled water tanks [7]. For example, the energy required to melt ice at 0 C is roughly equal to the energy required to heat the same amount of water to almost 80 C (334kJ/kg at a nearly constant 0 C, vs. 4.2kJ(/kg K) [8]).

The freezing point of water is lower than the cooling capabilities of a typical HVAC chiller system; using water as a PCM therefor requires separate systems for the HVAC and thermal energy storage functions, making such systems cost-prohibitive. In addition, water expands when frozen, making containment difficult. However, paraffins of the chemical form CH_3-(CH_2)n-CH_3 have been well-investigated as a possible phase change material for HVAC applications [9,10]. While paraffins generally suffer from poor conductivity, some paraffins exhibit the necessary characteristics of high heat of fusion, acceptable thermal conductivity, compatibility with encapsulation materials, and chemical stability [11,12], as well as freeze-melt temperatures appropriate for use in HVAC systems (5 - 12 C) [13]. One such candidate PCM is Tetradecane, $C_{14}H_{30}$, which has a freezing point of 5.6 C and a heat of fusion of 226 kJ/kg.

PCM may require encapsulation when used in a thermal storage system. Spherical encapsulation provides a greater surface area to volume ratio, potentially improving external heat transfer rates, but for retrofittable thermal storage systems energy density is of greater importance. Cylindrical encapsulation offers a higher theoretical maximum PCM fraction of 90.7% for hexagonal packed cylinders versus 74.0% for hexagonal close-packed spheres. In addition, cylindrical packing results in a unidirectional heat transfer fluid flow that simplifies numeric modeling.

This paper investigates the development, testing, and modeling of a compact, scalable, paraffin PCM (phase change material) based latent thermal storage system for retrofitting into existing residential and small commercial building applications. This system uses tube encapsulated Tetradecane as a PCM. A test thermal store is constructed and modeled to determine the initial feasibility of the design.

2. Methods

An experimental thermal store design is constructed utilizing PCM encapsulated in CPVC tubes arranged in a pseudo annular-ring configuration inside of an insulated PCV tank. Cylindrical encapsulation is used due to its higher potential density of encapsulated volume over spherical encapsulation, and also to minimize manufacturing costs. The thermal performance of this design can be manipulated by altering the diameter, length, spacing, and number of the encapsulation tubes.

A finite-volume numeric model, suitable for use in thermal storage applications, is also developed to demonstrate that simplified models for design and real-time operation/control are feasible for this thermal store design.

2.1. Experimental Apparatus

Tetradecane is used for the phase change material. A lab grade product of approximately 99% purity is selected. Water bath tests indicate that the tetradecane melts over a range of between 4.5 and 6.5 C., which is a suitable range for HVAC applications.

The tetradecane PCM is encapsulated in CPVC tubes, which are then packed into a larger containment tank. Standard 1/2 inch (12.7mm) CPVC tubing is used for PCM encapsulation due to cost and availability. Tubes are cut to a finished length of 1.14m, and are filled with 0.116ml of PCM before being fitted with end caps. The completed tubes contain approximately 0.167m of clear space above the liquid PCM to allow for expansion/contraction of the PCM during the freeze/melt cycle. Two PCM/tank configurations are investigated: a high density configuration utilizing 31 tightly-packed encapsulated PCM tubes, and a moderate-density configuration utilizing 19 evenly-spaced encapsulated PCM tubes.

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The tank is constructed of standard 4 inch (0.102m) schedule 40 PVC pipe utilizing standard PVC fittings to allow for water flow and instrumentation. A PVC pipe flange is installed at a level coincident with the top of the PCM encapsulation tubes to facilitate installation and re-placement of the PCM tubes and instrumentation. The tank assembly is placed within a 12 inch (0.305m) cardboard tube, with the 4 inch (0.102m) gap between the tank and tube filled with polyurethane spray foam. The upper flange and top portion of the tank is manually insulated with polyurethane foam batting and covered by a 12 inch (0.305m) diameter cardboard cylinder.

A Polyscience 5706T portable chiller unit provides chilled HTF (heat transfer fluid, 35% propylene glycol 65% water). The tank is supplied from a high-volume-flow loop that delivers the HTF at constant temperature to points very near the tank inputs. Small diameter polyethylene tubing connects the supply loop with the in-put/output terminals of the tank, minimizing residence times in the supply tubes. All tubes are insulated using foam rubber pipe insulation having a minimum R-value of 0.5 (SI). A schematic of the tank and its connections is shown in Figure 2.

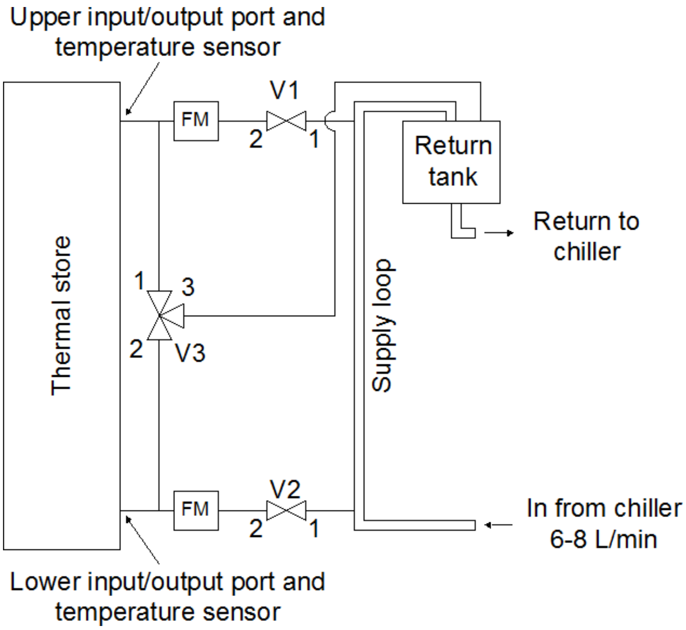

Fig. 2. Schematic of the experimental apparatus. The supply loop has a high flow rate to minimize temperature losses. The flow from the thermal store back to the chiller passes through the return tank to allow for flow rate confirmation.

The temperature measurement is accomplished using Omega 44033 epoxy-encapsulated thermistors. The thermistors are connected to a GW Instruments iNet-100 A/D data acquisition system, which provides excitation current for each thermistor through a precision 4.7k ohm resistor (+/- 0.025% 20ppm/C). This configuration has a temperature measurement accuracy of +/- 0.1 C. Temperature data is recorded at 10 second intervals during each test run.

HTF flow rates are measured using Omega FLR1009 Pelton-type turbine wheel flow meters with a range of 50 to 500 ml/min. These devices have a repeatability of 0.2% of full scale. One flow meter is used per tank port (at the top and bottom of the PCM tank), although only one is used for each test run. These devices are affected by fluid viscosity, and so require calibration to the fluid in use. The flow meter in use for each run is manually calibrated by intercepting the flow at the return tank and measuring the actual flow rate (see Figure 2).

During the charge cycle, chilled water at approximately 2 C is pumped into the bottom of the tank to freeze the PCM. During the discharge cycle, chilled water at approximately 11 C is pumped into the top of the tank to melt the PCM. The 2 C charge temperature is selected as low enough to fully freeze the PCM while also being within the capability of typical chilled water HVAC systems. The 11 C discharge temperature is selected as representative of a typical cooling coil chilled water return temperature. HTF flow rates for the charge cycle are fixed at 300ml/min. Flow rates for discharge are constant during each run, but vary between 100ml/min to 200ml/min. These values were chosen as appropriate for the scale of the experiment such that the discharge period of the thermal store would be on the order of 2 hours.

2.2. Numeric Model

A finite-volume numeric model is developed. The numeric model is based on an annular ring configuration of cylinder (tube)-encapsulated PCM surrounded by HTF, as shown in Figure 3. The annular ring configuration is the result of assuming evenly-spaced tubes placed within a larger thermal storage tank. The annular ring is actually a hexagonal cylinder; for modeling purposes it is replaced by an equivalent circular cylinder of equal x-sectional area. The annular ring analogy is used even when the spacing between the tubes is reduced to zero; in this case an equivalent annular area replaces the free area formed by the confluence of the tubes. A 60 degree segment of a single PCM encapsulation tube is used as the repeating element to accommodate the zero-spacing scenario.

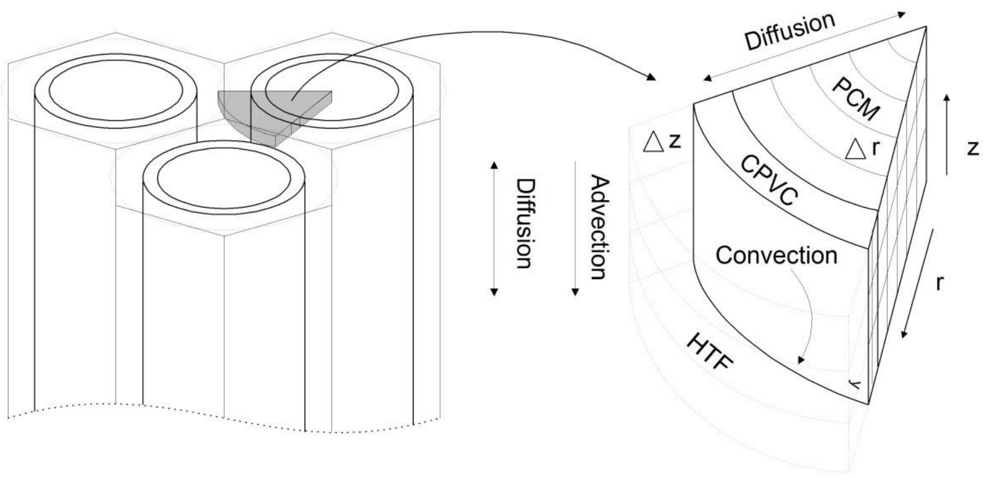

Fig. 3. Basis of the finite volume numeric model. Advection, convection, and vertical diffusion are considered for the HTF, convection between the HTF and the outside of the CPVC, and vertical as well as radial diffusion internal to the CPVC and PCM.

The model accounts for advection, convection, and vertical diffusion in the HTF; convection between the HTF and the CPVC encapsulation; and radial as well as vertical diffusion in the CPVC and PCM. Angular diffusion in the CPVC and PCM are not considered due to the assumption that the thermal energy is distributed evenly around the encapsulation tube by the HTF. Potential free convection in the HTF and PCM is ignored in favor of pure forced convection (HTF to CPVC) and conduction (CPVC and PCM). The HTF flow rates and velocities are relatively slow, resulting in Reynolds numbers on the order of one. All HTF flow is considered laminar due to the low Reynolds numbers of the HTF flow.

3. Results

Figure 4 shows the results of the experiment and the numeric model when using the 19 tube, moderate-density tank configuration and a median HTF flow rate of 150ml/min. These results show that 80% of the recoverable

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sensible and latent energy contained in the thermal store can be recovered at an output temperature of less than 6.5 C over an approximately 2-hour time interval. The 6.5 C temperature limit is required so that the output can be utilized by typical chilled water cooling coil systems. The two hour recovery time frame is suitable for peak load reduction, as peak periods typically last between 2 and 4 hours. Test runs at 100ml/min gave recovery rates of ~88% over ~3.5 hours, while the 200ml/min HTF flow rate gave recovery rates of ~70% at just under 1.5hours.

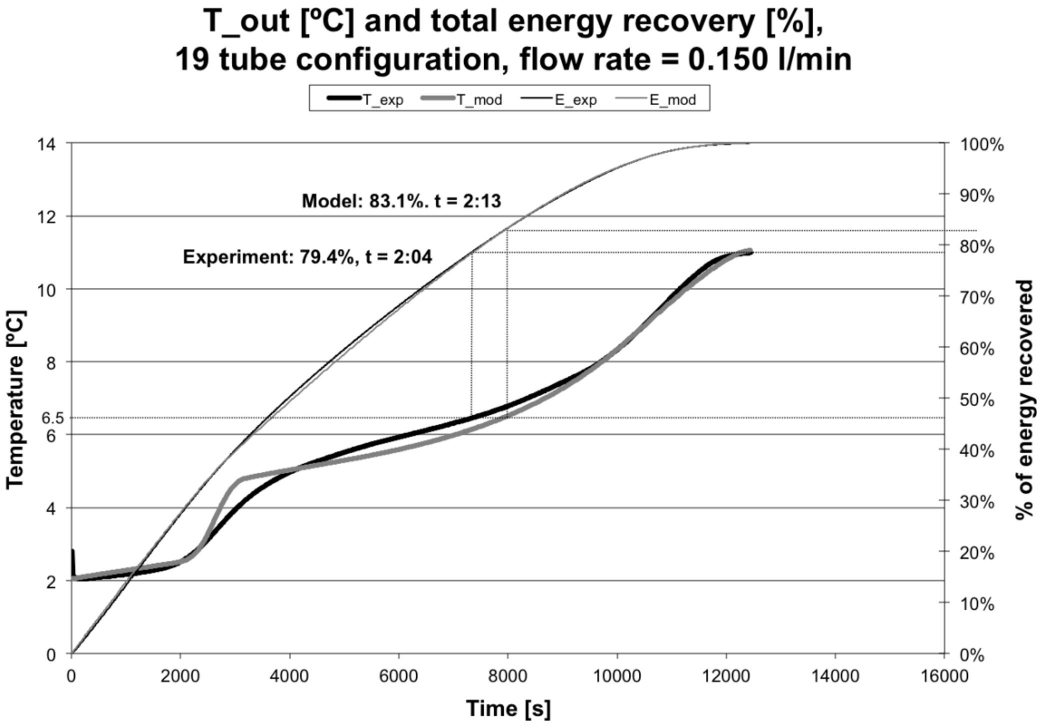

Fig. 4. Experiment and numeric model results for the moderate-density 19-tube configuration at moderate flow rate (150ml/min). The numeric model tracked the experimental output to within 5% for the energy recovery rates at Tout < 6ºC.

Figure 5 shows the results of the high-density configuration using 31 PCM-filled tubes in a hexagonal packed arrangement in the same thermal storage tank. In this experiment the data did not track the model predictions. It is theorized that the poor performance of the experiment compared to the model prediction is due to the model assumption that the thermal energy is evenly distributed around the circumference of the PCM encapsulation tube. In reality, the narrow, triangular-shaped flow paths created by the confluence of the encapsulation tubes cause the thermal energy to be concentrated near the center of these flow paths. This requires the thermal energy to be redistributed by either the CPVC encapsulation material or by the PCM itself. Unfortunately, these materials both exhibit poor thermal conductivities of ~ 0.2 and 0.15 W/(m K) respectively. This redistribution, which is not anticipated by the model, would explain the poor performance of the experiment when compared to the numeric model.

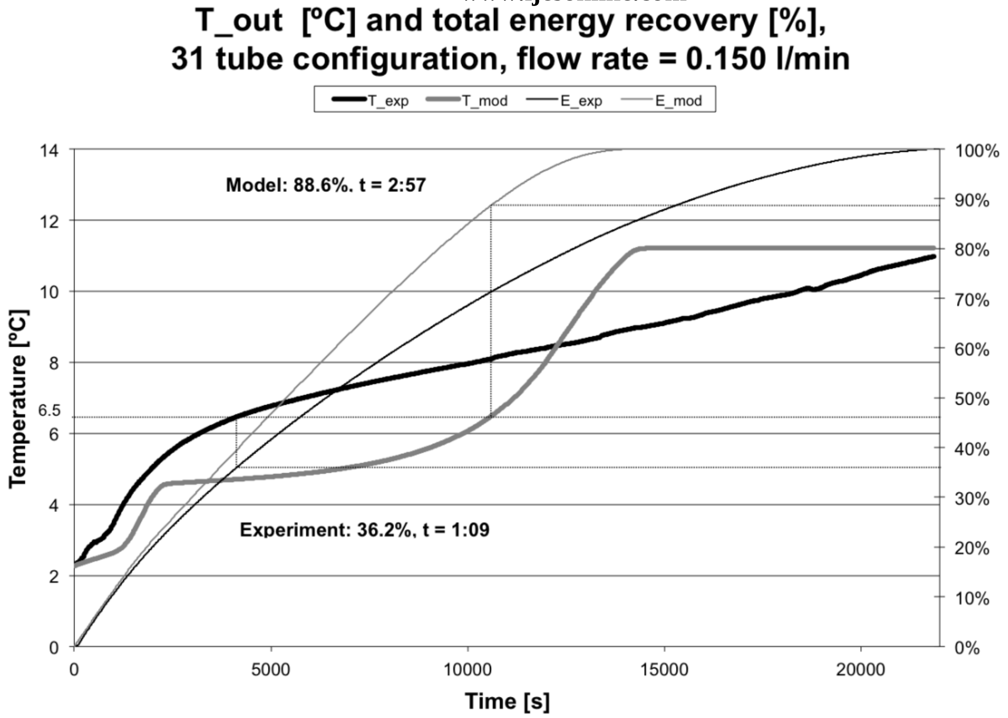

Fig. 5. Experiment and numeric model results for the high-density 31-tube configuration at moderate flow rate (150ml/min). Note that the model predicts good performance for this configuration, but the experimental results disagree. This is thought to be due to the uneven distribution of thermal energy entering the encapsulation tube.

4. Discussion

The 19-tube configuration of the thermal store performs well, and compared well with the results predicted by the numeric model. In this configuration, the thermal store has a capacity of 2.1 times that of a comparable chilled water tank operating over the same temperature range, and approximately 80% of the recoverable energy is recovered at a temperature at or below 6.5 C, making it usable by chilled-water HVAC systems.

Unfortunately, the high-density hexagonal packed 31-tube configuration did not perform as expected. The output temperature became higher than 6.5 C at only 36% energy recovery. The poor performance is believed to be the result of poor distribution of the thermal energy around the outside of the encapsulation tubes in the hexagonal-packed configuration, forcing the thermal energy to be redistributed by the CPVC encapsulation tube and the PCM, both which have poor thermal conductivity. This can be resolved by utilizing either a highly conductive encapsulation material or by adding thermal enhancements to the PCM.

Future experiments will investigate the use of thin-wall metallic encapsulation materials to improve the performance of the high-density hexagonal-packed configuration. The use of a modified Biot number, which takes into account the convection surface area as well as the x-sectional area of conduction along the encapsulation tube, indicates that typical thin-wall copper or aluminum tubing will redistribute the thermal energy evenly around the encapsulation tube. Thin-wall metallic encapsulation may enable the higher capacity hexagonal-packed configuration, leading to a four-fold increase in energy density over that of a chilled water tank.

5. Conclusions

Tube-encapsulated Tetradecane-based PCM thermal stores can reduce the size of a thermal storage unit, when compared to a similar-performing chilled water tank, by at least a factor of 2. However, the potential for a greater than 4 fold improvement over chilled water tanks is possible if issues with the high-density hexagonal-packed configuration of this thermal store design can be resolved. Further research is required to confirm that thin-wall metallic encapsulation tubes can resolve issues with the high-density hexagonal packed configuration.

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