EXPERIMENTAL INVESTIGATION ON PACKED BED COOL STORAGE SYSTEM FOR SUPPLY-DEMAND MANAGEMENT IN BUILDING AIR-CONDITIONING SYSTEM SUITABLE FOR MICRO THERMAL GRID

by

Rajamani NARAYANASAMY[a], Pandiyarajan VELLAICHAMY[b], Mangat Ram SHARMA[c], and Velraj RAMALINGAM[a*]

[a] Institute for Energy Studies, Anna University, Chennai, India
[b]Department of Chemical Engineering, Anna University, Chennai, India
[c] Higher Education Department, Secretariat, Chennai, India

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In most of the developed nation, the increase in percentage share of renewable power in the total power generation causes major concerns over the integration of these renewable power with the grid resulting grid instability. Energy storage is a new frontier technology which is considered as the ultimate solution in developing micro-smart grid with distributed renewable power generation. Most of the hot countries like India spend nearly 24% of the electricity generated on air-conditioning and food preservation. Under such scenario, among the various types of storage systems, the cool thermal storage plays a vital role to promote renewable power in an economical way. Considering the importance in the present renewable energy scenario, in the present work, an experimental investigation was performed on a packed bed cool storage system integrated with a chiller system which has major advantages in central air-conditioning system for demand management strategies. The storage system consists of encapsulated spherical balls filled with a mixture of distilled water and pseudomonas (nucleating agent) as phase change material and a mixture of distilled water and mono-ethylene glycol as heat transfer fluid. The essential parameters such as reduction in subcooling, instantaneous, and cumulative heat transfer during the charging process are presented for the efficient operation.

Key words: *cool thermal energy storage, demand side management, building cooling, phase change material, pseudomonas, nucleating agent, thermal micro-grid*

Introduction

The development and economy of any nation depend on the quantum of power generation, effective utilization of energy and environmental protection. The building sector consumes about 40% of the energy generated in most of the developed and developing countries. In hot countries, major energy consumption in buildings is due to large capacity air-conditioning systems that consumes nearly 60% of the energy spent on building. In most of the buildings, the air-conditioning system is usually designed to meet the peak loads, which are required only for a short duration immediately after the noon. Hence most of the duration in a day the chillers are either in idle condition or in part loads operation. Cool thermal energy storage (CTES) is

* Corresponding author, e-mail: velrajr@gmail.com

one of the key technologies that have prospects of playing an important role through demand side management, particularly in countries like India where 24% of the total electrical energy generated is being used for air-conditioning in the building sector. Further, the CTES system helps efficient utilization of the renewable energy for building applications, with an appreciable reduction in the operating cost of the air-conditioning system. Among the various types of cool thermal storage systems, the packed bed storage system, in which the phase change material (PCM) encapsulated in spherical capsules, is best suited for large building air-conditioning applications. Velraj *et al.* [1] studied in detail, the CTES system coupled with a large building air-conditioning system. The authors have listed the advantages of the system and also reported that the integration of CTES system resulted in a saving of INR* 2.26 million per annum for the building considered. A summary of literature review pertaining to the various aspects of cool storage systems is presented in this section.

Li *et al.* [2] reviewed the use of various materials used for cool storage in air-conditioning application. The authors have summarized the thermal and physicochemical properties of different PCM like latent heat, thermal conductivity, supercooling, phase separation, and corrosion including the phase change slurries characteristics. Osterman *et al.* [3] reviewed various PCM based cooling technologies that find application in buildings. The variations in indoor air temperature in buildings arising as a result of the addition of PCM and the issues in solidification of PCM due to problems encountered in heat transfer have been discussed.

Gunther *et al.* [4] conducted an experimental study of the subcooling in hexadecane emulsions as PCM for different droplet sizes. They reported that there is an increase in the subcooling for smaller droplets and suggested reduction in the subcooling in emulsions by addition of nucleating agents. Gunther *et al.* [5] explained the experimental data on the basis of the theory of subcooling and nucleation in emulsions. They indicated the possibility of domination over nucleation rate in small droplets by homogenous nucleation. Kumaresan *et al.* [6] conducted experimental study of the charging behavior of water based nanofluid phase change material (NFPCM) encapsulated in a spherical container. They predicted an energy saving potential of 6-9% in the CTES using the NFPCM. Chandrasekaran *et al.* [7, 8] investigated the thermal performance of two different water based NFPCM using multiwalled carbon nanotubes and CuO as nanoparticles encapsulated in a spherical capsule. They reported that the supercooling is the major hurdle in using water as PCM and it was eliminated with the use of a pseudomonas as the nucleating agent. The duration of solidification was considerably reduced by the addition of nanoparticles in the PCM due to its enhanced thermal transport properties.

Chandrasekaran *et al.* [9] performed a solidification characteristics study to analyze the influence of spherical capsules of three different sizes with water as PCM filled with 90% of its fill volume, maintained at various bath temperatures. They concluded that the effect of subcooling decreases following an increase in size of the PCM capsules and the subcooling was avoided with increased temperature difference between the heat transfer fluid (HTF) and the PCM. Chandrasekaran *et al.* [10] investigated the effect of fill volume to study the solidification characteristics of spherical capsule filled with water as PCM. They conducted experiments with PCM encapsulated in different fill volumes and reported that the increase in fill volume had a significant influence in reducing the degree of subcooling and the subcooling was eliminated with 95% fill volume. Further they reported that the heat flux was increased several fold particularly at 95% fill volume making it highly suitable for applications that demand large cooling load in a short duration.

* INR = 0.013 $

Chen and Yue [11] have studied the thermal performance of a packed bed storage system with spherical capsule filled with water as PCM for air-conditioning applications and pointed out that more than 75% of the cool energy was stored as latent heat and the remaining was stored as sensible heat. They concluded that the percentage of sensible heat stored should be reduced for efficient utilization of CTES system. Regin *et al.* [12] developed a numerical model for a packed bed latent heat thermal energy storage (TES) using paraffin wax as PCM encapsulated in a spherical capsule and studied the thermal behavior for charging/discharging modes. They reported that a longer solidification time than melting time which was the result of a very low thermal conductivity of the solidified layer. They have also reported higher charging and discharging rates for the capsule with a smaller radius than for a larger size. Bellan *et al.* [13] developed a numerical model for a latent TES system with sodium nitrate as PCM encapsulated in spherical capsules for focus on solar power plant applications. The influence of the capsule size and the flow rate of HTF on temperature distribution, fluid-flow, melting and solidification characteristics have been analyzed.

Chen *et al.* [14] investigated the thermal performance of water and nucleating agents as PCM encapsulated in a cylindrical capsule of cool thermal storage tank during the charging process by varying the inlet coolant temperature and flow rate. They reported that a higher efficiency is achieved at reduced inlet coolant temperature and at increased flow rate. Investigation done by Cheralathan *et al.* [15] on the performance of an industrial refrigeration system together with an encapsulated PCM based CTES system used distilled water with heterogeneous nucleating agents as PCM. They reported that decrease in evaporator temperature by 1 °C results in about 3-4% increase in SEC and 1 °C decrease in condensing temperature leads to 2.25-3.25% decrease in specific energy consumption. Wang *et al.* [16] developed a prototype for a refrigeration system through direct integration of PCM heat exchanger as the pre-condenser, to save energy and found increase in COP arising from the use of PCM technology. Bedecarrats *et al.* [17, 18] made an experimental study on the performance of encapsulated spherical capsules containing water with a nucleation agent as PCM and a mixture of mono--ethylene and water as HTF, during the charging/discharging processes. They have reported the benefit of incomplete discharge mode followed by a charge mode that helps easy crystallization. They have also developed a numerical model for cold storage to analyze the delay of the crystallization of the PCM.

Oro *et al.* [19] attempted the application of PCM in low temperature storage like frozen food transportation, without refrigeration system and this technology is gaining importance in the recent years due of low cost energy production possible with low energy. Comodi *et al.* [20] showed that it is possible to enhance the efficiency of the building air-conditioning system through the integration of cool storage system and thereby achieving both the economy and energy savings. Al-Aifan *et al.* [21] performed a combined variable refrigeration volume and CTES air-conditioning system was experimentally investigated for summer and winter design conditions.

The concept of smart thermal grids can be regarded as being parallel to smart electricity grids. Rivarolo *et al.* [22] investigated the impact of uncontrollable generation of power from solar and wind integrated with 100 kWe micro gas turbine, 20 kWe internal combustion engine to produce both electrical and thermal energy and 100 kWth adsorption chiller to produce cool energy. The results are presented to show the interaction among the fossil and renewable resources and the importance of integrating suitable storage system in the grid. Lund *et al.* [23] defined the concept of smart energy and smart thermal grids. Their major objective was to identify the challenges associated with the future renewable based heat supply in the overall sustainable energy systems. Marrasso *et al.* [24] studied a system consisting of a ground-source heat pump, a low temperature thermal network and a series of electric heat pumps. The thermal grid considered was the heat source/sink for the water to water electric heat pumps installed in a six floors building located in Naples (Southern part of Italy). Analysis was performed using TRANSYS 17 to compare the performance of this system with a conventional one installed in each dwelling consisting of an air to water air-conditioner for cooling operation and a boiler for heating. They reported interesting global and local environmental advantages.

Literature shows the extensive use of water as PCM in the CTES system due to its good thermal properties, good stability in the repeated cycling, affordable cost, eco-friendly and compatibility with the refrigeration equipment compared to any other PCM. Charging the system at a faster rate will be helpful by two ways:

– Reducing the charging time will reduce the operational duration of the chiller and also the pumping power.
– In the emerging renewable energy scenario, the tariff for the electric power will be dynamic in a day if the charging is done during the low tariff period which may prevail for a shorter duration faster charging is required.

Higher mass-flow rate of the HTF and higher temperature difference between the HTF and the phase transition temperature of the PCM seems to be better solution for the aforementioned. However, it is to be noted that for every 1 °C reduction in the evaporator temperature, the COP of the chiller system decreases by 2.5%. Hence the chiller set point temperature should be as high as possible to achieve better efficiency. Further if the PCM experiences subcooling, the HTF temperature also has to be reduced. This again decreases the COP of the chiller. Further the higher mass-flow rate may immediately reduce the temperature difference between the HTF and the PCM. However very low temperature difference may affect the freezing characteristics due to subcooling and other internal conductive resistance within the PCM during solidification. Hence optimum inlet HTF temperature and flow rate is essential and freezing should occur without subcooling.

There is only a rare or limited presence of the literature on the selection of the volume of the nucleating agent to avoid the subcooling and the suitability of packed bed cool thermal storage system for industrial application. Considering the aforementioned, the present research is focused on reduction of subcooling by mixing a suitable fraction of pseudomonas with water and to develop a pilot scale PCM based packed bed cool storage system. Further investigation is performed to arrive suitable inlet HTF temperature and the mass-flow rate of the HTF for the selected packed bed PCM based storage system. Extensive promotion of cool storage system could play a predominant role in micro thermal grid to achieve grid stability in the near future while promoting renewable power generation on a large scale.

Experimental investigation

A major part of the present research deals with development of an experimental set up consisting of a packed bed storage tank with a capacity of 200 L filled with PCM encapsulated spherical capsules to study the charging behavior under different operating conditions. Initially experiments were conducted to study the effect of the addition of a small concentration of pseudomonas with PCM on eliminating the subcooling characteristics of the PCM in order to find the optimum volume of pseudomonas to eliminate the subcooling. The selected concentration of pseudomonas was used for preparing the PCM filled in the spherical capsule. The construction of the experimental set-up and the experimental procedure adopted are also explained in this section.

Investigation on subcooling

Figure 1 is the schematic diagram of a test set-up to study the effect of addition of pseudomonas on subcooling. A vapor compression refrigeration (VCR) system of capacity 1 ton of refrigeration (TR) was used for setting the required bath temperature of HTF in the evaporator tank. The tank had five spherical balls (test modules) filled with a mixture of distilled water and pseudomonas as PCM and these balls were kept in a liquid bath at a constant temperature. The first PCM container was filled with 150 mL of distilled water but without pseudomonas. The remaining four balls contained a mixture of 150 mL of distilled water and different volumes of pseudomonas ranging from 0.1 mL to 0.4 mL. A stirrer is kept inside the bath to maintain the surrounding bath temperature at a near uniform level during the entire experimentation period. It also helps to achieve a convective heat transfer coefficient on the surface of the PCM balls approximately equivalent to a real time packed bed storage tank where the HTF flows across the balls with certain velocity. Further the absence of stirrer may lead to stratification inside the bath and the low surface convective heat transfer coefficient by the stationary fluid may give resistance more than the conductive resistance offered during initial stages of solidification inside the ball which may not prevail in the real case. Hence the stirrer is introduced and it is in operation during the entire duration of the experiment . An resistance temperature detector (RTD) (PT 100 – Class A) with an accuracy of ±0.1 °C was placed at the center of each spherical balls to monitor the temperature variations in the PCM, with the help of data logger during the testing of these five spherical balls.

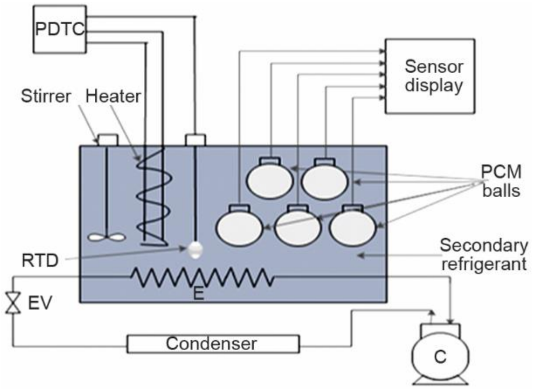

Figure 1. Schematic diagram of the test set-up: E – evaporator, EV – expansion valve, C – compressor

The test set-up was initially at a room temperature of 32 ±0.2 °C and the surrounding bath temperature was kept constant, always lower than the freezing temperature of the PCM through the proportionate differential temperature controller (PDTC). The solidification experiment was conducted by keeping the bath temperature at –10 °C by means of VCR. The temperature of the PCM was continuously observed for every 20 seconds till the complete PCM was solidified. The effect of pseudomonas on the subcooling was investigated and the results are discussed in chapter *Results and discussion*.

Experimental set-up for charging analysis

Figures 2 and 3 show the photographic view and schematic diagram of the experimental set-up, respectively. The set-up consisted of two parts, namely the CTES tank and the VCR system.

The CTES tank of capacity 212 L was made of stainless steel material and integrated with the evaporator of VCR system used in the earlier experiment by using distribution pipes made of chlorinated polyvinyl chloride material. The CTES tank was packed with spherical balls made of high density polyethylene with an outer diameter of 75 mm, filled with a mixture of distilled water and pseudomonas as PCM. Each PCM ball contained 150.2 mL PCM (distilled water with 0.13% pseudomonas) and this mixture was occupying 95% of the volume of spherical balls. Chandrasekaran *et al.* [10] have reported that 95% fill volume has the maximum heat transfer effect. The total mass of PCM kept in all the 600 PCM balls was 90.12 kg. The tank was filled with a mixture of distilled water (60%) and mono-ethylene glycol (40%) normally used as the secondary refrigerant in CTES system for large scale air-conditioning applications which surrounded the PCM capsules and acting as the HTF. The CTES tank was kept in a vertical position as recommended by Bedecarrats *et al.* [17].

Figure 2. Photographic view of experimental set-up

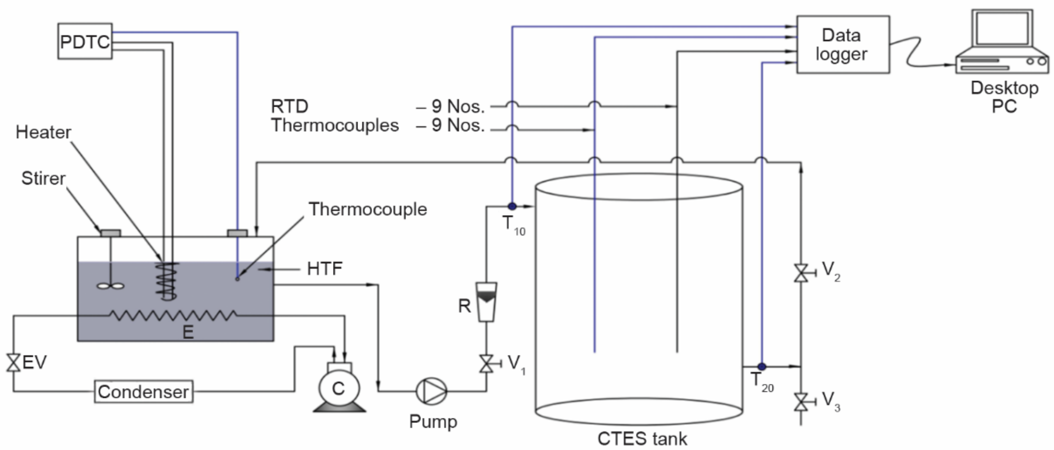

Figure 3. Schematic diagram of experimental set-up; E – evaporator, EV – expansion valve, C – compressor, R – rotameter, T_{10}-T_{20} – thermocouples, V_1, V_2, V_3 – valves

The VCR set-up consisted of a stainless steel tank insulated with polyurethane material of capacity 11 L, a heating coil of capacity 2000 W and a PDTC used for variations in the heating oil output on the basis of the temperature of the bath measured by RTD sensor. A mechanical stirrer operated by a 9 W electrical motor (speed 1280 rpm) was kept at the top to achieve a near uniform temperature of the HTF in the bath. The evaporator coil in the bath was completely immersed in the HTF.

The specifications of major components used in the experimentation are shown in tab. 1. The maximum quantity of the HTF filled in the CTES tank was 82.4 L with an estimated porosity value of 33.3% based on the balls packed in the storage tank. The HTF was filled with the help of a measuring jar. The quantities of HTF in the evaporator tank and the CTES tank were 10.9 kg and 82.4 kg, respectively, and an additional 2 kg of HTF was filled in the pipeline.

Table 1. Technical specifications of major components with present system

Utilities/Devices	Rated Capacity/Range
VCR system	1 TR
HTF circulation pump	0.25 horse power
CTES tank	212 L
Flow meter	0-1000 L per hour
Evaporator tank	11 L

All the RTD were connected to a data acquisition system of Agilent make 34972A with an accuracy of 0.004% for storing the data generated during the experiment on a continuous basis. The HTF was allowed to circulate through the CTES tank using a pump and the HTF from the storage tank was allowed to enter the evaporator tank. The HTF flow rate was controlled by the valves and a rotameter was used for measuring the flow rate. A temperature controller attached to the evaporator was used for maintaining the evaporator tank bath temperature at any desired constant temperature between 0 and –20 °C during the charging process. The RTD (PT 100 – Class A) with an accuracy of ±0.1 °C was located at the center of the selected PCM balls and connected to a data logger through which the temperature of the PCM inside the spherical balls was monitored. The temperature of the HTF inside the CTES tank was monitored through data logger using thermocouples of *T*-type which were kept at various locations in the storage tank. In addition, two thermocouples were fixed at the inlet and the outlet of the storage tank.

Experiments

During the experiments initially, the HTF mass-flow rate circulated in the system was adjusted to 360 Lph. The HTF bath temperature was maintained at –6 °C using PDTC controller. The VCR system was operated continuously and, after a certain duration, the entire PCM in the balls attained crystallization. Experiments were continued until the PCM reached frozen ice temperature of –2±0.5 °C. The temperature readings of both HTF and PCM were recorded for every 10 seconds using a data logger throughout the experiment. The experiments were repeated for HTF bath temperatures of –9 °C and –12 °C with the mass-flow rate of 360 Lph.

Table 2. Results of uncertainty analysis

Measured quantities	
Diameter of spherical capsule	±0.02 mm
Temperature	±0.15 °C
Volume (100 mL)	±0.015 mL
Mass-flow rate	±2.7%
Derived quantities:	
Instantaneous heat transfer	±1.85%

Experiments were repeated by varying the HTF flow rate as 400 Lph and 440 Lph by adjusting the valves and with bath temperatures of –6, –9, and –12 °C (for both the cases). The uncertainties of the measured and derived data were calculated and the values are presented in tab. 2.

Data analysis

Equations used to determine the performance parameters such as instantaneous heart transfer and cumulative heat transfer using the measured temperature values are presented in this section.

The instantaneous heat transfer (the rate at which the thermal energy is stored in the storage tank) during the charging process is evaluated using:

$$Q_{\text{ins}} = \dot{m}C_p\,(T_{\text{out}} - T_{\text{in}}) - Q_{\text{loss}} \tag{1}$$

where $\dot{m}$ is the mass-flow rate of the HTF through the storage tank, C_p – the specific heat capacity of HTF, T_{in} – the inlet temperature of the HTF in the storage tank at any instant, T_{out} – the outlet temperature of the HTF in the storage tank at any instant, and Q_{loss} – the teat loss from the storage tank to the ambient at any instant during the charging process evaluated using:

$$Q_{\text{loss}} = UA_{\text{s}}\Delta T \tag{2}$$

where ΔT is the temperature difference between the HTF in the storage tank and the ambient air at any instant, A_{s} – the outer surface area of the storage tank, and U – the overall heat transfer coefficient evaluated using a separate experiment.

The cumulative heat transfer, Q_{cum}, is evaluated by integrating the instantaneous heat transfer, Q_{ins} using:

$$Q_{\text{cum}} = \sum_{k=1}^{n} Q_{\text{ins}}^{k}\Delta T \tag{3}$$

where n is the number of time steps and ΔT is the size of each time step (10 seconds considered in the present evaluation).

Results and discussion

The results of the experiments conducted with different mass-flow rates and at different HTF inlet temperature are presented and discussed in this chapter.

Effect of pseudomonas concentration on subcooling characteristics of PCM

Figure 4 shows the temperature variation of the PCM in four different balls with various volumes of pseudomonas (0, 0.1, 0.2, and 0.3 mL). It is seen from the figure that the PCM kept in the spherical capsule without pseudomonas is subjected to large subcooling by 5 °C before the freezing is initiated. When the pseudomonas is added with the PCM, there is a reduction in subcooling. The subcooling is reduced to 3 °C when 0.1 mL of pseudomonas was added in the PCM and when the volume concentration of pseudomonas is increased to 0.2 and 0.3 mL, the subcooling is totally eliminated.

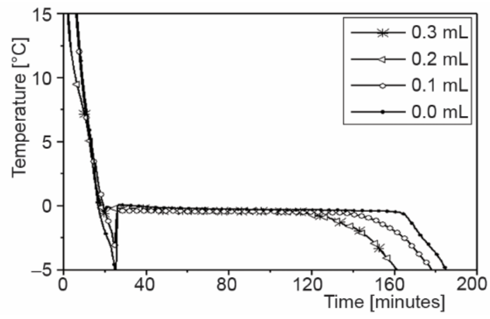

Figure 4. Effect of pseudomonas concentration on temperature variation during freezing of PCM

The reduction in subcooling also helps in reducing the freezing duration. It is observed that the freezing was completed within a duration of 120 minutes in the case with 0.2 mL and 0.3 mL of pseudomonas, whereas the freezing was completed after a time interval of 170 minutes in the case without pseudomonas. The presence of pseudomonas in the PCM helps to start the nucleation process that reduces the freezing duration appreciably. Hence in the present experimental investigation to study the charging behavior of the packed bed cool storage system, the spherical capsules placed inside the storage tank are filled with distilled water as PCM along with 0.13% (0.2/150) of pseudomonas to study the charging characteristics of the storage system.

Temperature-variation of the PCM in the storage tank

Figure 5 shows the temperature-time history of the PCM at center of the storage tank when the inlet HTF from the constant bath temperature set point is kept at –6, –9, and –12 °C when the mass-flow rate is kept at 440 Lph. The figure shows that the variations in the temperature from the initial condition of approximately 32 °C to the start of the phase change temperature as linear and then a near constant temperature was obtained during the phase change process. The figure shows the time taken to remove the sensible heat is approximately 500, 450, and 440 minutes for the HTF set point temperature of –6, –9, and –12 °C, respectively. When the chiller is set to on position, the high thermal mass in the storage system allows the HTF to reduce its temperature slowly irrespective of set point temperature as the capacity of the chiller is same, which shows that there is not much effect due to inlet HTF set point temperature. However, there was an appreciable variation in the time taken during the phase change process. The duration of the phase change process was 250 minutes for the case with –6 °C and nearly150 minutes for the cases with –9 °C and –12 °C.

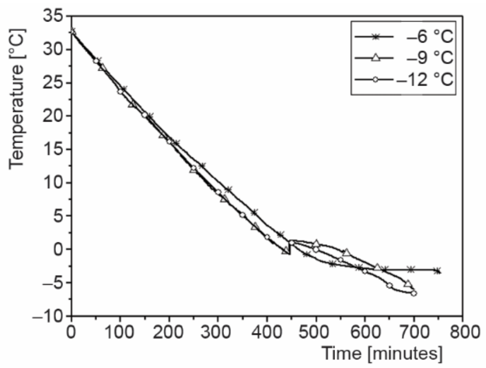

Figure 5. Temperature time history of PCM at three different inlet HTF temperatures (mass-flow rate = 440 Lph)

Further the figure shows an increase in temperature of PCM during the start of solidification process at –9 °C and –12 °C set point temperatures which showed the start of crystallization of the PCM after a marginal subcooling. There would not to be any resistance normally for the PCM to change its phase following the start of crystallization. However, the phase change process depended on the rate at which the heat was removed from the surface to the surrounding HTF. The time was larger for the case with –6 °C, compared to –12 °C, since the temperature difference available between the HTF and PCM is lower. Hence it is understood that the temperature difference played a major role during the solidification process.

Effect of inlet HTF temperature on instantaneous and cumulative heat transfer

Figure 6 shows the instantaneous heat transfer of approximately 1.6 kW when the inlet set point temperature is –6 °C and nearly 2 kW when the inlet set point temperature is –9 °C and –12 °C. The instantaneous heat transfer at all set point temperature, decreases with respect to time as the HTF in the CTES tank temperature increases. When the inlet set point temperature is kept at –9 °C and –12 °C, the instantaneous heat transfer is marginally higher throughout the charging process compared to the HTF inlet set point temperature of –6 °C. Hence the total time required for completely charging was around 750 minutes at –6 °C and it is nearly

600 minutes when the set point temperature is –9 °C and –12 °C. It is inferred from the previous results that when the set point temperature is increased from –6 °C to –9 °C there is an appreciable increase in instantaneous heat transfer and also there is a reduction in time for complete solidification. However, this enhancement in heat transfer was not observed when the inlet HTF set point temperature is increased from –9 °C to –12 °C.

Figure 7 shows the cumulative energy stored in the storage tank with respect to time when the mass-flow rate was kept at 440 Lph. The figure shows an observable reduction in the duration of complete charging when the bath temperature set point was kept at –9 °C and –12 °C compared to –6 °C. However, there was not much difference in the time taken for complete charging between the cases of –9 °C and –12 °C. This shows a minimum temperature driving potential required for efficient transfer of the heat, for a given mass-flow rate beyond which the temperature difference may not have significant effect. Considering the reduction in COP of 2.5% for every 1 °C reduction in evaporator temperature, setting at an optimal temperature difference between the HTF and the PCM solidification temperature is essential at a given mass-flow rate of the HTF to achieve a higher operational efficiency. It is construed that for the present configuration, the HTF inlet set point temperature need not be decreased beyond –9 °C since the chiller operation is expensive if there is a decrease in set point temperature.

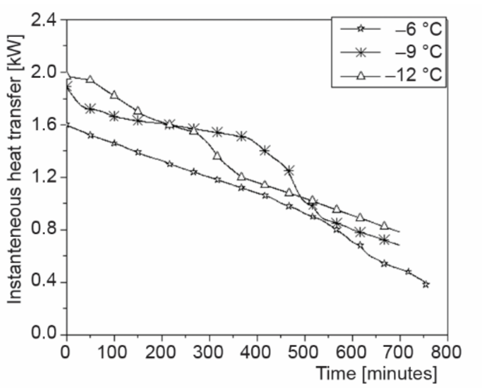

Figure 6. Instantaneous heat transfer in the storage tank for various bath temperatures (mass-flow rates of $\dot{m}$ = 440 Lph)

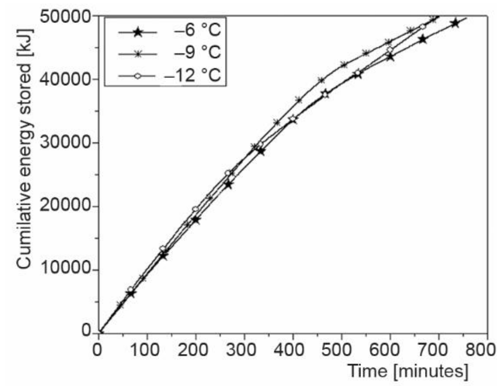

Figure 7. Cumulative energy stored in the storage tank for various bath temperatures (mass-flow rate $\dot{m}$ = 440 Lph)

Effect of mass-flow rate on instantaneous and cumulative heat transfer

Figure 8 shows that the effect of mass-flow rate on instantaneous and cumulative heat transfer when the HTF temperature was set at –9 °C. It is seen from the figure that the instantaneous heat transfer starts decreasing from 1.8 kW and an average instantaneous heat transfer of 1.6 ± 0.2 kW is maintained for a longer duration at all mass-flow rates. After 400 minutes the heat transfer is around 1.3 kW at all flow rates and then decreases appreciably with respect time. At a higher mass-flow rate of 440 Lph only a small decrease in instantaneous heat transfer is observed till 500 minutes and then the instantaneous heat transfer decreases appreciably with respect time compared to the mass-flow rates of 360 Lph and 400 Lph. It is also observed from the figure that there is a linear increase in cumulative heat transfer until 400 minutes and then the increase in cumulative heat transfer decreases with respect to time. Further it is observed that both the instantaneous and cumulative heat transfer do not vary much with the variation in the mass-flow rate of the HTF. It is construed from the above results that nearly 80% of the heat available in the storage tank could be charged nearly at a uniform charging rate.

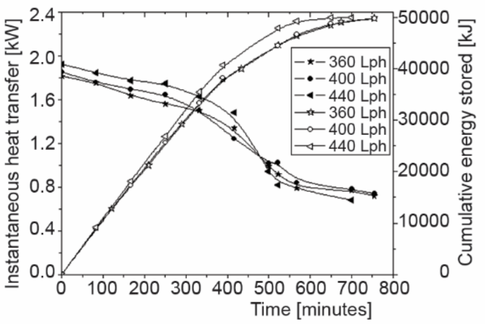

Figure 8. Effect of mass-flow rate on instantaneous and cumulative heat transfer (HTF set point temperature –9 °C)

Conclusions

In the present research, initially an experiment was performed for evaluating the subcooling behavior of the PCM with various pseudomonas concentration. Further experiments were performed to determine the heat transfer behavior during the charging process in a cool thermal storage tank of capacity 50000 kJ. The results of the experiments conducted at different HTF inlet temperatures and with various mass-flow rates are presented. The following conclusions are arrived at from the investigation:

- The presence of pseudomonas in the PCM helps to start the nucleation process at its phase change temperature that reduces the freezing duration appreciably and it was concluded that 0.13% of pseudomonas in the distilled water eliminates the complete subcooling process when the surrounding temperature is maintained at –10 °C.
- It was observed from the temperature variation of PCM that the time for complete solidification is nearly same when the inlet HTF set point temperature was –9 °C and –12 °C. However, at –6 °C HTF set point temperature, the time for complete solidification is relatively higher and instantaneous heat transfer is also lower compared to the set point temperature of –9 °C and –12 °C. Hence considering the operational expense at lower temperature, it is recommended to circulate the HTF at the set point temperature of –9 °C to achieve higher energy efficiency.
- The mass-flow rate of HTF has only a marginal effect on the instantaneous and cumulative heat transfer.

The operational characteristics of the cool thermal storage are very essential to reduce the operational cost in the central air-conditioning system in large capacity buildings. In the recent years, in many countries, the electricity tariff is dynamic with respect to time in a day. Particularly in the countries where renewable energy is promoted in a large scale, the electricity tariff is very low during the time of large availability of renewable power and the demand is very low. Hence in the present increasing renewable energy scenario, this kind of storage system has large financial benefits. The results given in the present work are very useful to eliminate the subcooling of PCM and to design the energy efficient operational condition of the cool thermal storage system.

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References

[1] Velraj, R., *et al.*, S., Energy Management through Encapsulated PCM Based Storage System for Large Building Air Conditioning Application, *Int. Energy Journal*, *7* (2006), 4, pp. 253-59

[2] Li, G., *et al.*, Radermacher, R., Review of Cold Storage Materials for Air Conditioning Application, *Int. J. Refrig., 35* (2012), 8, pp. 2053-2077
[3] Osterman, E., *et al.*, Review of PCM Based Cooling Technologies for Buildings, *Energy and Buildings, 49* (2012), June, pp. 37-49
[4] Gunther, E., *et al.*, Subcooling in Hexadecane Emulsions, *Int. J. Refrig., 33* (2010), 8, pp. 1605-1611
[5] Gunther, E., *et al.*, Subcooling in PCM emulsions – Part 2: Interpretation in Terms of Nucleation Theory, *Thermochimica Acta, 522* (2011), 1-2, pp. 199-204
[6] Kumaresan, V., *et al.*, Role of PCM Based Nanofluids for Energy Efficient Cool Thermal Storage System, *Int. J. Refrig., 36* (2013), 6, pp. 1641-1647
[7] Chandrasekaran, P., *et al.*, Solidification Behavior of Water Based Nanofluid Phase Change Material with a Nucleating Agent for Cool Thermal Storage System, *Int. J. Refrig., 41* (2014), May, pp. 157-163
[8] Chandrasekaran, P., *et al.*, Enhanced Heat Transfer Characteristics of Water Based Copper Oxide Nanofluid PCM (Phase Change Material) in a Spherical Capsule during Solidification for Energy Efficient Cool Thermal Storage System, *Energy, 72* (2014), Aug., pp. 636-642
[9] Chandrasekaran, P., *et al.*, Influence of the Size of Spherical Capsule on Solidification Characteristics of DI (Deionized Water) Water for a Cool Thermal Energy Storage System – An experimental Study, *Energy, 90, Part 1* (2015), Oct., pp. 807-813
[10] Chandrasekaran, P., *et al.*, Effect of Fill Volume on Solidification Characteristics of DI (Deionized) Water in a Spherical Capsule – An Experimental Study, *Energy, 90, Part 1* (2015), Oct., pp. 508-515
[11] Chen, S. L., Yue, J. S., Thermal Performance of Cool Storage in Packed Capsules for Air Conditioning, *Heat Recovery Syst CHP, 11* (1991), 6, pp. 551-561
[12] Regin, A. F., *et al.*, An Analysis of a Packed Bed Latent Heat Thermal Energy Storage System Using PCM Capsules: Numerical Investigation, *Renewable Energy, 34* (2009), 7, pp. 1765-1773
[13] Bellan, S., *et al.*, Numerical Investigation of PCM-Based Thermal Energy Storage System, *Energy Procedia, 69* (2015), May, pp. 758-768
[14] Chen, S. L., *et al.*, An Experimental Investigation of Cold Storage in an Encapsulated Thermal Storage Tank, *Exp. Therm. and Fluid Sci., 23* (2000), 3-4, pp. 133-144
[15] Cheralathan, M., *et al.*, Performance Analysis on Industrial Refrigeration System Integrated with Encapsulated PCM-Based Cool Thermal Energy Storage System, *Int. J. Energy Res., 31* (2007), 14, pp. 1398-1413
[16] Wang, F., *et al.*, The Novel Use of Phase Change Materials in Refrigeration Plant. Part 1: Experimental Investigation, *Appl. Therm. Eng., 27* (2007), 17-18, pp. 2893-28901
[17] Bedecarrats, J. P., *et al.*, Study of a Phase Change Energy Storage Using Spherical Capsules. Part I: Experimental Results, *Energy Convers. Manage., 50* (2009), 10, pp. 2527-2536
[18] Bedecarrats, J. P., *et al.*, Study of a Phase Change Energy Storage Using Spherical Capsules. Part II: Numerical Modeling, *Energy Convers. Manage., 50* (2009), 10, pp. 2537-2546
[19] Oro, E., *et al.*, Thermal Analysis of a Low Temperature Storage Unit Using Phase Change Materials without Refrigeration System, *Int. J. Refrig., 35* (2012), 6, pp. 1709-1714
[20] Comodi, G., *et al.*, Application of Cold Thermal Energy Storage (CTES) for Building Demand Management in Hot Climates, *Appl. Therm. Eng., 103* (2016), June, pp. 1186-1195
[21] Al-Aifan, B., *et al.*, Performance Evaluation of a Combined Variable Refrigerant Volume and Cool Thermal Energy Storage System for Air Conditioning Applications, *Int. J. Refrig., 76* (2017), Apr., pp. 271-295
[22] Rivarolo, M., *et al.*, Thermo-Economic Optimization of the Impact of Renewable Generators on Poly-Generation Smart-Grids Including Hot Thermal Storage, *Energy Convers. Manage., 65* (2013), Jan., pp. 75-83
[23] Lund, H., *et al.*, 4th Generation District Heating (4GDH) Integrating Smart Thermal Grids into Future Sustainable Energy Systems, *Energy, 68* (2014), Apr., pp. 1-11
[24] Marrasso, E., *et al.*, Global and Local Environmental and Energy Advantages of a Geothermal Heat Pump Interacting with a Low Temperature Thermal Micro Grid, *Energy Convers. Manage., 172* (2018), Sept., pp. 540-553

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