PAPER • OPEN ACCESS

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To cite this article: S Sinha *et al* 2020 *IOP Conf. Ser.: Mater. Sci. Eng.* **778** 012103

View the article online for updates and enhancements.

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Abstract. This paper studies the technical feasibility of a novel solar roof tile (SRT) to be used for HVAC (heating, ventilation and air conditioning) in buildings. The SRT utilizes a phase change material (PCM) for thermal storage. The SRT works like a conventional thermal storage tank while featuring compact size and lower cost. Two heat exchangers are connected one each on the top and bottom of the PCM matrix. The top heat exchanger functions as an absorber of solar heat. Aqueous sodium hydroxide passes to the top heat exchanger from a storage sump. The bottom heat exchanger is the condenser and evaporator with water passing through the heat exchanger. The cooling system is integrated inside the customised roof tile interlocking system. This system uses roof tiles that have higher performance during extreme weather conditions and lower maintenance costs. In this paper, we have investigated the thermal performance and technical feasibility of the proposed solar SRT-HVAC system. This proposition eliminates conventional thermal storage tanks used in solar thermal collectors. The dynamic performance of the SRT under transient conditions is evaluated using the TRNSYS modelling package, which shows the influence of design variables like insulation points and charging/discharging durations on thermal storage performance.

Keywords: HVAC, phase change materials, solar energy, roof tile, latent heat battery

1. Introduction

Key statistical references from the International Energy Efficiency (IEA) and International Energy Outlook 2010 (IEO) show an increase in energy demand and energy consumption among the world nations [1]. Many research contributions addressing the issues related to global energy consumption and carbon dioxide (CO_2) emissions have been reported, including those looking to abate carbon dioxide emissions in buildings [2-6]. Buildings consume one quarter to one third of the overall energy generated. The energy saving potential of buildings reduces with time which also decreases their cost saving potential [7-11]. Studies [12-15] to improve energy savings of Heating, Ventilation, Air Conditioning and Refrigeration systems are ubiquitous. These studies focus on thermal comfort and improved indoor environmental conditions. Several authors [16-18] discussed advanced, intelligent & logic control mechanisms integrated in building management systems. Pérez-Lombard, L. et al. [19] discussed the scope and requirements of building energy regulations for Heating, Ventilation, Air Conditioning (HVAC) systems.

Different techniques may be used to improve energy efficiency of HVAC systems and thus reduce their environmental impact. Energy consumption rates of these systems have been improved by control and optimization strategies in recent years [20]. These approaches are very complicated and expensive and require constant monitoring [21]. Energy efficiency can be achieved by combining different components of HVAC systems, setting optimum process variables to achieve the desirable results for a comfortable indoor environment. For example, thermal storage systems (TSSs) may be used to mitigate changes in energy demand between on-peak & off-peak periods. The TSS offers various advantages compared to conventional HVAC system for heating and cooling such as performance improvements, increase in capacity, energy and capital cost savings, reduction of equipment size, and extensive range of applications.

The storage and release of heat energy based on fluctuating load demand at lower overall energy are described in [22-25]. The introduction of certain latent heat storage mediums into building materials has been used to minimize temperature fluctuations in indoor environments [26]. Use of latent heat energy storage (LHES) system to capture thermal energy from living space during day time and release this energy at night to the atmosphere has gained acceptance in recent years. Interesting research about pre-cooling or economizer ventilation techniques has been conducted to achieve enhanced performance in passive cooling and ventilation in buildings [27, 28].

In this context, developing agile solar roof tiles that can adapt simple HVAC principles alongside the typical functionalities of the normal roof tile seems to be an attractive research direction. Figure 1 illustrates the principles of an agile roof tile HVAC installed at the roof of a building. This agile roof tile may simultaneously fulfil heating and cooling requirements of a building. It can heat the room in winter and cool it in summer seasons. The hot water flowing to the desorber of this solar roof tile HVAC system can be used for domestic hot water supply and heating water in the pool.
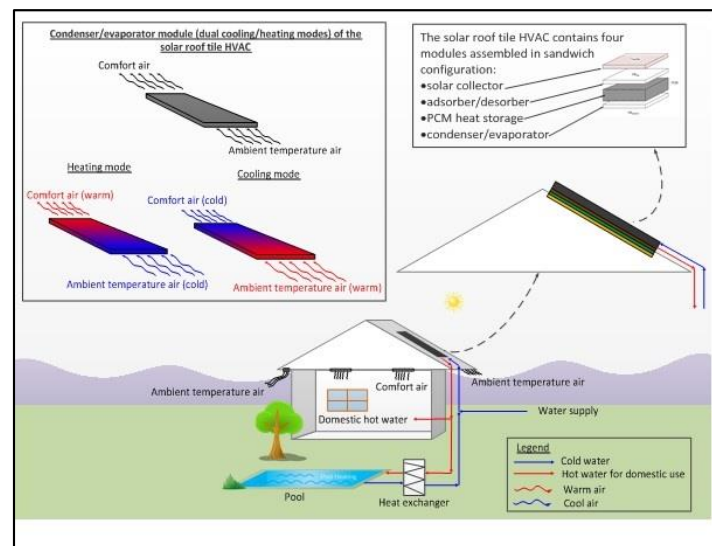

Figure 1. Solar roof tile HVAC.

This paper aims at the modelling of this novel solar roof tile for HVAC system with 100% renewable energy operation. The designed system is a compact unit which may be integrated in the roof or replace the typical roof tiles. The proposed roof tile can collect solar energy to produce hot water and flexibly

aid in the heating and cooling of a building. Additionally, this unique design using thermal storage where the phase change material (PCM) can bridge the gap between energy supply and demand. The model of this solar roof tile HVAC system is developed and simulated in TRNSYS simulation software using the weather profile of Sydney, Australia.

2. Technical aspects

2.1. System Operation
This TSS is a thermo-chemical heat pump operating under vacuum conditions. It allows evaporation of water at low temperature and transport of water vapour without a pump or fan. The principle is described below and shown in Figure .

1. Charging Phase (desorption/ drying of adsorbent)

Heat from the solar roof tile, a high temperature source is fed into the device. The adsorbent is heated and the water vapour released from the adsorbent. The desorbed water vapour is transported from the desorber Chamber into a separate reservoir where it condenses. The latent heat of condensation is rejected to the environment.

2. Storage Phase

Energy absorbed by the PCM during charging phase causes melting. During the discharging phase PCM solidifies and energy is released. Latent heat storage in PCM is attractive because of its high energy storage capacity in small volume and charging and discharging heat at constant temperature.

3. Discharging Phase (adsorption of water on adsorbent)

In the Evaporation Chamber water absorbs heat from the living space and evaporates. This water vapour is transported to the adsorption Chamber where it is adsorbed by the adsorbent. Heat of adsorption, at higher temperature, is released.
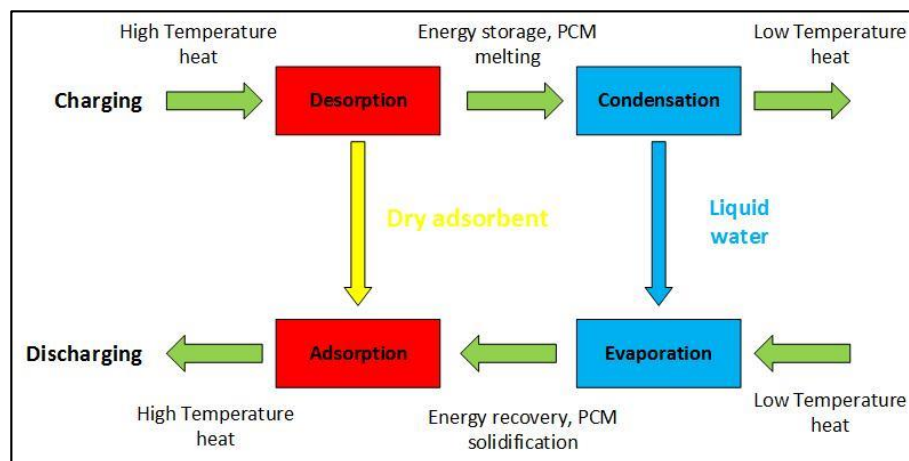

Figure 2. Working principle of a closed-cycle adsorption heat store.

2.2 Description of storage system

The process flow of the two Chambers – adsorption/desorption and condensation/evaporation – are – interconnected as shown in figure 3. Chamber 1 functions as desorber and adsorber. It contains the sorbent- aqueous sodium hydroxide. Chamber 2 is the condenser and evaporator containing, the sorbate, water. The reversible chemical reaction occurring in the sorbent is:

$$NaOH . (m + n)H_2O + Heat \leftrightarrow NaOH . mH_2O + nH_2O \tag{1}$$
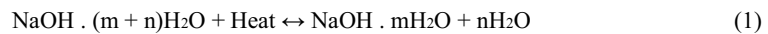
Charging or regeneration occurs by supplying heat from the roof tile to Chamber 1. Diluted sorbent is heated in Chamber 1 releasing water vapour. This water vapour is driven from the sorbent in Chamber 1 to Chamber 2, where it condenses. The latent heat of evaporation is released to the environment.

In the discharging phase Chamber 1 and 2 function as adsorber and evaporator. Water in Chamber 2 evaporates at low temperature by absorbing heat from the room/living space. Water vapour is driven from Chamber 2 to Chamber 1, where it is adsorbed by the sorbent, releasing latent heat of evaporation as well as heat of ratio at approximately 10 to 1. During discharging phase PCM solidifies and energy is released from the PCM. The output temperature depends on sodium hydroxide content in the sorbent in Chamber 1 as well as output temperature of water vapour in Chamber 2. The heating capacity depends on change in sorbent concentration between the charging and discharging phases.

2.3 Charging and discharging type

The charging and discharging process has the following assumptions:

- Constant power of 6 kW
- Overall charging efficiency 0.7
- Output concentration set to 0.5 kg sorbent to kg sorbate
- Constant mass balance
- Delta T of desorber and condenser set to 8K
- Temperature dependency of variables is neglected
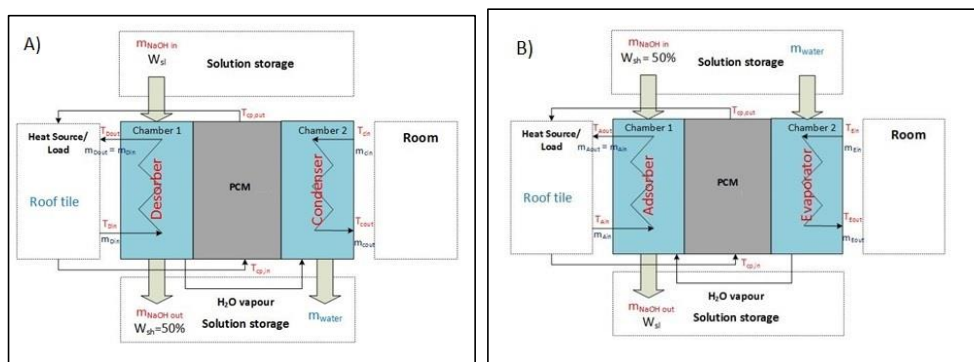

Figure 3. Schematics of heat and mass exchange: (A) charging and (B) discharging mode.

3. Mathematical Description

3.1. Energy Balance of PCM node

The energy balance of PCM node is described as in Figure 4-B:

$$\tag{2}$$

$$m_j \frac{dh_j}{dt} = P_{cond,j} + P_{loss,j} - P_{hx,j}$$

where, j is the PCM node; dh_j is the enthalpy difference of node j (kJ/kg); dt is the time difference (h); $P_{cond,j}$ is heat transfer due to conduction of node j (kJ/hr); $P_{loss,j}$ is the heat loss to ambient environment of node j (kJ/hr); $P_{hx,j}$ is the heat transfer via heat exchanger in node j(kJ/h). Convection between PCM nodes is neglected as there is no mass flow. Heat flow into storage Chamber and heat exchanger is assumed positive and outgoing heat flow is negative.
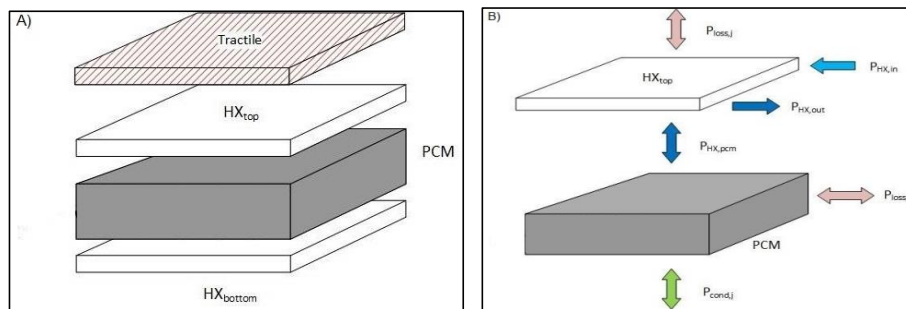

Figure 4. Solar driven roof tile cooler: (A) Zone model (B) Heat transfer model.

3.2. Melting of PCM case
The heat from the top heat exchanger is used to melt the PCM. Equation (3) is used to calculate final temperature of PCM by application of energy balance (temperature of PCM is above melting temperature).

$$\frac{dT_{PCM}}{dt} = \left(\frac{A}{mc}\right)_{PCM}\left\{S - U_T\left(T_p - T_a\right) - \left(\frac{k}{\Delta}\right)_{INS}\left(T_{PCM} - T_a\right)\right\} - \frac{Q_{conv}}{(mc)_{PCM}} \tag{3}$$

where, Q_{conv} is the heat transfer to the heat pipe (W) ; dT_{PCM} is the rate of change in the temperature of the PCM (K) ; A is the area of PCM (m2) ; m is the mass of PCM (kg) ; S is the absorbed solar radiation (W/m2) ; U_T is the top loss coefficient from heat exchanger to the ambient (W/m2.K); T_p is the temperature of the top heat exchanger (K); T_a is the ambient temperature (K) ; k is thermal conductivity (W/m.K) ; Δ is the change in insulation thickness (m) ; T_{PCM} is the temperature of PCM (K).

4. Analysis for dynamic characteristics of the system
This section discusses the dynamics of the latent heat battery HVAC system. The hot summer days of January were simulated based on the weather data for Sydney from TRNSYS. A period of three days was chosen to allow the system to establish a pattern of performance to show-case the summer season in Sydney. The switching time for: warming up, energy harvesting and energy releasing phases are 6am, 7am and 6pm respectively. The collective solar power profile was observed for these times of the day.

   Table 1 lists the design parameters of the system. The physical properties of the RT-50 PCM obtained from Rubitherm [29] is used in this work. The chosen PCM is a commercially available product with melting temperature of 47.5oC, suitable for cooling air in a room (Table 1). The mass of the PCM affects the amount of energy stored whereas the number of water tubes and tube length affects the quality of stored energy. Assuming that all energy supplied by the PCM is the latent heat energy, can subsume that total energy required in one day is the energy stored in the PCM (as latent heat energy). This is because solar radiation is unavailable before 9am and after 6pm.

Table 1. Design parameters for the proposed LHB - HVAC system.

Variable type/name	Proposed value	Unit
PCM used is RT-50 (commercially available from Rubitherm supplier [29])		
Phase change temperature	47.5	^{o}C
Thermal conductivity (both phases)	0.2	W/(m.K)
Heat capacity (both phases)	2	kJ/(kg.K)
Density (solid/liquid)	880/760	kg/m_3
Latent heat	168	kJ/kg
Volume expansion	12.5	%
Maximum operating temperature	70	^{o}C
Latent heat battery (LHB) unit		
Total PCM mass	53	kg
Hot water temperature constraints		
To load	45 - 46	^{o}C
During charge mode	$\geq$60	^{o}C
During discharge mode	45 - 46	^{o}C

Hot water at 45°C is required in discharge mode. At high collectible solar power (CSP), water at roof tile outlet may reach boiling point. The pump automatically switches off as the water temperature in the solar heating loop exceeds 95°C. The pump restarts only when CSP drops by 30% after shutting down. During shut down period, the latent heat battery will be utilized for supply hot water. Overheating protection method like using a drain-back system [30] can be used to the protect system when the temperature in the loop reaches near-evaporation boundary.

Another operational constrain of the RT-50 PCM is its maximum operating temperature of 70°C. The PCM can undergo degradation of thermo-physical properties. When the heat battery exposed to temperatures higher than 70°C. When the PCM reaches 70°C, the heat battery can be put in discharge mode to moderate the PCM temperature.

5. Operational mode
Figure 5 shows the temperature profile for the top heat exchanger (adsorber and desorber), PCM (RT-50) and the bottom heat exchanger (condenser and evaporator). When the top heat exchanger acts as a desorber, the lower heat exchanger acts as a condenser and the PCM melts while charging. However, when the top heat exchanger acts as an adsorber, the lower heat exchanger acts as an evaporator and the PCM solidifies while discharging. The desorber temperature ranges from 28.9^{o}C to 92.0^{o}C, PCM temperature increases from 47.5^{o}C to 55.2^{o}C while melting and condenser temperature increases from

2.3$\circ$C to 29.0$\circ$C. The system switches and the top heat exchanger acts as an adsorber and lower heat exchanger acts as an evaporator while PCM solidifies. The adsorber temperature ranges from 18.1$\circ$C to 75.0$\circ$C, PCM temperature decreases from 47.5$\circ$C to 39.8$\circ$C while it solidifies, and the evaporator temperature decreases from 20.3$\circ$C to 2.3$\circ$C.
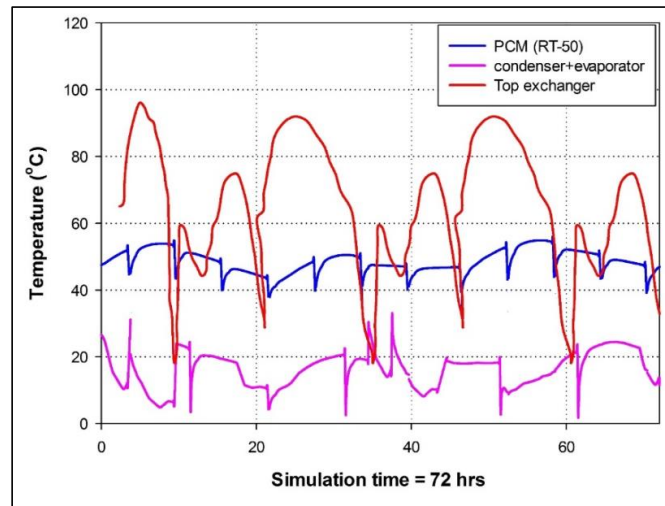

Figure 5. Temperature profile of top and bottom heat exchangers and PCM for 3 days.

6. Conclusion

In this work, we have integrated phase change materials (PCM) into solar-assisted HVAC system. This work describes the design of the latent heat battery with a top heat exchanger, PCM heat storage and bottom heat exchanger simulated in the charging, discharging and storage phases. The Rubitherm (RT-50) was used as the phase change medium. A dynamic model for the system was developed in TRNSYS. Weather data obtained from TRNSYS for Sydney, Australia was used for all simulations. The desorber temperature ranges from 28.9$\circ$C to 92.0$\circ$C, PCM temperature increases from 47.5$\circ$C to 55.2$\circ$C while melting of the PCM and condenser temperature increases from 2.3$\circ$C to 29.0$\circ$C. When the system switches over, the top heat exchanger acts as an adsorber and the lower heat exchanger acts as an evaporator while the PCM solidifies. In this case, the adsorber temperature ranges from 18.1$\circ$C to 75.0$\circ$C, the PCM temperature decreases from 47.5$\circ$C to 39.8$\circ$C while it solidifies and the evaporator temperature decreases from 20.3$\circ$C to 2.3$\circ$C.

Nomenclature

w_{sh}	charged sorbent concentration (kg/kg)
$w_{slstart}$	Initial sorbent concentration (kg/kg)
ΔT_C	condenser delta temperature (K)

$\varDelta T_{DC}$	desorber to condenser temperature (K)
$\varDelta T_E$	evaporator delta temperature (K)
$\varDelta T_{AE}$	adsorber to evaporator temperature difference (K)
P	Charging and discharging power (kW)
T_{stop}	charging stop temperature ($_o$C)
T_{charge}	charging start temperature ($_o$C)
T_{Din}	desorber input temperature ($_o$C)
T_{Cin}	condenser input temperature ($_o$C)
W_{sl}	discharged sorbent concentration (kg/kg)
$K_{disc,char}$	coefficient of efficiency
T_{Ain}	adsorber input temperature ($_o$C)
T_{Ein}	evaporator input temperature ($_o$C)
w_{sl}	discharged sorbent concentration (kg/kg)
m_{Ain}	adsorber input flow rate (kg/hr)
T_{Aout}	adsorber outlet temperature ($_o$C)
T_{Eout}	evaporator outlet temperature ($_o$C)
m_{Din}	desorber input flow rate (kg/hr)
T_{Dout}	desorber outlet temperature ($_o$C)
T_{Cout}	condenser outlet temperature ($_o$C)
m_{vap}	Desorber and evaporator water vapour (kg/hr)
m_{Cin}	condenser input flow rate (kg/hr)
m_{Ein}	evaporator input flow rate (kg/hr)
dh_j	enthalpy difference of node j (kJ/kg)
dt	time difference (hr)
$P_{cond,j}$	heat transfer due to conduction of node j (kJ/hr)
$P_{loss,j}$	loss to ambient environment of node j (kJ/hr)

$P_{hx,j}$	heat transfer via heat exchanger in node j (kJ/hr)
Q_{conv}	heat transfer to the heat pipe (W)
dT_{PCM}	rate of change of temperature of PCM (K)
A	area of PCM (m2)
m	mass of PCM (kg)
S	absorbed solar radiation (W/m2)
U_T	top loss coefficient from heat exchanger to ambient (W/m2.K)
T_P	temperature of top heat exchanger (K)
T_a	ambient temperature (K)
K	thermal conductivity (W/m.K)
Δ	change in insulation thickness (m)
T_{PCM}	temperature of PCM (K)

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