July 2018

Evaluating Recharge Options for Phase Change Material Storage of a Personal Conditioning System

Rohit Dhumane

Center for Environmental Energy Engineering, University of Maryland, College Park, 4164 Glenn Martin Hall, MD 20742, USA, dhumane@umd.edu

Yiyuan Qiao

Center for Environmental Energy Engineering, University of Maryland, College Park, 4164 Glenn Martin Hall, MD 20742, USA, yyqiao@umd.edu

Jan Muehlbauer

Center for Environmental Energy Engineering, University of Maryland, College Park, 4164 Glenn Martin Hall, MD 20742, USA, muehlie@umd.edu

Jiazhen Ling

Center for Environmental Energy Engineering, University of Maryland, College Park, 4164 Glenn Martin Hall, MD 20742, USA, jjazhen@umd.edu

Vikrant Aute

Center for Environmental Energy Engineering, University of Maryland, College Park, 4164 Glenn Martin Hall, MD 20742, USA, vikrant@umd.edu

See next page for additional authors

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Dhumane, Rohit; Qiao, Yiyuan; Muehlbauer, Jan; Ling, Jiazhen; Aute, Vikrant; and Hwang, Yunho, "Evaluating Recharge Options for Phase Change Material Storage of a Personal Conditioning System" (2018). *International High Performance Buildings Conference*. Paper 302.

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Authors

Rohit Dhumane, Yiyuan Qiao, Jan Muehlbauer, Jiazhen Ling, Vikrant Aute, and Yunho Hwang

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Rohit DHUMANE¹, Yiyuan QIAO¹, Jan MUEHLBAUER¹, Jiazhen LING¹, Vikrant AUTE^{1*}, Yunho HWANG¹

¹Center for Environmental Energy Engineering, 4298 Campus Dr., 4164A Glenn Martin Hall, University of Maryland, College Park, MD, USA 20742

Phone: 301-405-7661, Email: {dhumane, yyqiao, muehlie, jiazhen, vikrant, yhhwang}@umd.edu

* Corresponding Author

ABSTRACT

Buildings consume about two-fifths of the primary energy in the United States with space heating and cooling taking up one-third of that fraction. Global warming and the associated climate changes have necessitated a rethinking of our energy consumption patterns. Roving Comforter (RoCo) is one such technology being developed with this objective. It enables buildings to save 10-30% of HVAC (Heating, Ventilating, and Air Conditioning) loads by enabling elevation of temperature set-points without compromising occupant comfort. The cooling operation involves an R134a vapor compression cycle operation, which stores the condenser heat into a phase change material (PCM) based thermal storage. Previous studies focused on enhancing the PCM to reduce the recharge time by thermosiphon operation from 8 hours for non-enhanced version by 40% without a significant increase in the weight of the thermal storage. This enhanced PCM storage needs to be recharged before the next cooling operation, which may be conducted in two different ways. A gravity-assisted two-phase thermosiphon operation consumes very less power takes a longer recharge time (6.5 hours) in comparison to a reverse heat pump operation which consumes more power in a much shorter operation time (2.5 hours). The current article uses a validated dynamic model of thermosiphon and heat pump in Modelica to evaluate the overall coefficient of performance for combined cooling and recharge operations. Reduced recharge time from the heat pump recharge leads to increased frequency of the cooling operation enabling savings on building HVAC energy. All these factors are considered in making design recommendations for future prototypes of RoCo, which will save additional energy and provide longer cooling operation to the current prototypes. The modeling framework discussed in the article being generic may be used by researchers investigating vapor compression cycle integration strategies with PCM thermal storages. The article provides interesting insights and quantification of benefits obtained from various strategies adopted for improvement over the first prototype.

Keywords: Dynamic Modeling, Modelica, Comfort, Thermosiphon, Heat Pump

1. INTRODUCTION

Personal conditioning systems (PCS) have potential to save building energy in the range of 10-30% by elevating building thermostat. Dhumane et al. (2017a) modeled four different PCS with potential to save building HVAC in the range of 10-30% by elevating building thermostat set-points by 4°F (2.2°C). Roving Comforter (RoCo) is one such PCS, which uses vapor compression cycle (VCC) to generate cooling. To avoid addition to the building loads from the heat released during the VCC, phase change material storage is used. Details of RoCo operation are discussed in Dhumane et al. (2017b) and Du et al. (2016). The PCM stores heat from the condenser as it melts and thus its latent heat of melting limits the cooling operation period for RoCo. A two-phase thermosiphon Dhumane et al. (2018) is used to re-solidify the PCM before its next operation.

The thermosiphon operation only requires the fan to operate and consumes about 7 W of power. Due to the low thermal conductivity of the PCM, the thermosiphon required about eight hours for completion in the first prototype made from a paraffin based PCM (Du et al., 2016). This limits RoCo to only one cooling operation per day. The thermosiphon operation period may be reduced significantly by enhancing the thermal conductivity of PCM using graphite (Mallow et al., 2016, 2018). To achieve more savings from the use of RoCo, the cooling operation period needs to be increased

and the recharge time decreased. With this objective, the PCM storage in newer versions of RoCo are developed using the graphite enhanced PCM. The amount of PCM is increased to support four hour cooling operation. Another option for solidification is to reverse the refrigerant flow in circuit and solidify the PCM by making the PCM-HX as an evaporator. This mode of operation will be referred to as heat-pump mode for recharge. The air-cooled heat exchanger now operates as a condenser and rejects heat to the ambient. Heat pump mode utilizes the compressor for its operation resulting in higher power consumption, but is expected to solidify the PCM much faster. The focus of the present study is to explore the advantages and disadvantages of thermosiphon and heat pump operation for solidification of PCM.

2. SYSTEM DESCRIPTION

The schematic of the two operations for recharge of RoCo is shown in Figure 1. RoCo uses vapor compression system to deliver cooling to its target occupant. The heat released during the vapor compression cycle is stored in the PCM thermal storage to avoid addition to building loads. The PCM melts as it stores this heat and needs to be recharged after its capacity is saturated. This recharge operation can be performed by thermosiphon operation or a heat pump operation.

The refrigerant circuit of RoCo using thermosiphon for recharge involves two ball valves, which by-pass the compressor and expansion device from the refrigerant circuit for cooling cycle. Details of this thermosiphon operation is available in Dhumane et al. (2018) and omitted here for brevity. As discussed in Dhumane et al. (2018) the recharge time of the thermosiphon can be significantly enhanced by addition of graphite to the phase change material (PCM). The systems discussed in the current research use these graphite enhanced PCM. For further value addition from RoCo, the cooling operation duration for the system is twice than those considered in the previous research (Dhumane et al., 2018; Du et al., 2016) and this is achieved by increasing the weight of the PCM used in the thermal storage. The higher weight of PCM leads to addition of recharge time.

Figure 1: Refrigerant circuits for the two different recharge options in RoCo

Consequently, a second mode of recharge called the heat pump mode is considered. A reversible four-way valve can be used to change the refrigerant flow direction from the compressor and recharge the PCM by operating the PCM storage as an evaporator. The compressor used in RoCo is of the variable-speed type and the recharge time may be controlled by adjusting its RPM. There are two TXV in the circuit for heat pump, each protected by a check valve. The bulbs of the TXV are connected to outlet of either the air to refrigerant heat exchanger or the PCM-HX. During the cooling operation, the outlet of air to refrigerant heat exchanger (marked as condenser in Figure 1) is connected to the bulb of TXV, while the outlet of PCM-HX is connected to the bulb of the second TXV. This second TXV is operational during the heat pump operation.

3. MODELING PCM STORAGE

The cross-section of the graphite enhanced PCM is shown on the left of Figure 2. It consists of a cylinder of diameter 0.254 m and height 0.305 m. Eight 0.0063 m copper tubes are inserted at a distance of 0.051 m from the outer diameter, with headers connecting at bottom and the top. The symmetric location of the refrigerant tubes is exploited to allow simplification in modeling. Only a single tube may be modeled and the behavior of the entire thermal storage may be captured by scaling the behavior of this single section by eight. The PCM surrounding each refrigerant tube is in the shape of a 1/8 pie. However, it is assumed to be a cylinder for modeling as shown on the right of Figure 2. Thermocouples located on the top and bottom surfaces of the cylindrical PCM negligible differences. Thus temperature variation along the length as well as the radius are ignored for the PCM control volume. This allows modeling of PCM as 1-D control volume. The radius of this imaginary cylinder (r_o) is taken such that the sum of the volumes of the eight cylinders equals the total volume of PCM in the experiment.

Figure 2: Cross section for modeling PCM

Finite volume method is used to model the cylindrical control volume of the PCM (Figure 2, right). A staggered grid is adopted with the mass and energy of the PCM assumed to be concentrated at the center (dotted lines in Figure 2, right), while the heat fluxes calculated the boundary (solid lines in Figure 2, right). The number of discretization for the control volume is 5. The phase change process is modeled using the temperature transforming model (Cao & Faghri, 1990, 1991). The graphite matrix present in the PCM prevents the circulation of liquid PCM by natural convection. As a result, the only mode of heat transfer for the phase change is conduction.

With the assumptions discussed above, the energy equation is written as shown in Equation (1). Temperature transforming model models specific enthalpy of PCM as a function of specific heat capacity (c) and source term (s) (see Equation (2)). The thermal conductivity of enhanced PCM is same for both solid and liquid phase, since it is primarily driven by the graphite (Dhumane et al., 2018). As a result, thermal conductivity (k) is treated as a constant and not a function of melt fraction. This assumption is not true in case of non-enhanced PCM and the variation of thermal conductivity may be captured by equations described in (Cao & Faghri, 1990, 1991). The PCM does not melt at a fixed temperature and has a temperature glide ($2\delta T$). The specific heat capacity and the source term are defined by Equations (3) and (4) during the phase change. The melt fraction of the PCM can now be evaluated from the specific enthalpy (Equation (5)).

$$\rho \frac{\partial h}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left(kr \frac{\partial T}{\partial r} \right) \quad (1)$$

$$h = c(T - T_m) + s \quad (2)$$

$$c = \begin{cases} c_s & T - T_m < -\delta T & \text{(solid)} \\ \frac{c_s + c_l}{2} + \frac{H}{2\delta T} & -\delta T \leq T - T_m \leq \delta T & \text{(two-phase)} \\ c_l & T - T_m > \delta T & \text{(liquid)} \end{cases} \quad (3)$$

$$s = \begin{cases} c_s \delta T & T - T_m < -\delta T & \text{(solid)} \\ \left(\frac{c_s + c_l}{2} \right) \delta T + \frac{H}{2} & -\delta T \leq T - T_m \leq \delta T & \text{(two-phase)} \\ c_s \delta T + H & T - T_m > \delta T & \text{(liquid)} \end{cases} \quad (4)$$

$$\lambda = \min \left(1, \max \left(0, \frac{h}{H} \right) \right) \quad (5)$$

As discussed earlier, the staggered grid approach employed in discretization involves having mass and energy evaluations at the center of volumes while the heat flux evaluations are at the border. The heat transfer between adjacent control volumes is evaluated using Equation (6). The index $i = 1$, for the refrigerant tube outer diameter and for discretization of 5, the outer diameter of PCM cylinder control volume gets $i = 6$. The solid lines at the border of control volume get integer indices 2,3,4 and 5, while the dotted lines at center have index of 1.5, 2.5 5.5. The thermal conductance between the refrigerant tube and first control volume is evaluated by Equation (7).

$$G_i = \frac{k(2\pi r_i L)}{\Delta r} \quad i = 2, 3, 4, 5 \quad (6)$$

$$G_1 = \frac{k \left(2\pi \left(\frac{r_1 + r_{1.5}}{2} \right) L \right)}{\Delta r / 2} \quad (7)$$

The discretized model of PCM is implemented in Modelica. The model diagram for it is shown in Figure 3. It consists of alternating grid of PCM capacitor and Thermal conductor (Modelica.Thermal.HeatTransfer.Components.ThermalConductor) components. Equations (2) to (5) are in the PCM capacitor while the thermal conductance evaluated from Equations (6) and (7) is given as a parameter input to the thermal conductor. The model has two HeatPorts at inner radius and outer radius of the cylinder, which enables heat transfer connections with refrigerant control volume and heat losses to ambient respectively. The combination of all these components is then wrapped up as a component with a model diagram for easy visualization in system simulation.

Figure 3: Model diagram for discretized PCM control volume

4. THERMOSIPHON RECHARGE

The PCM model developed in Section 3, is connected with components of a two-phase thermosiphon (riser, downcomer, air-cooled condenser and refrigerant tube) for simulation. Details of these component models and correlations used for heat transfers in them is presented in (Dhumane et al., 2018) and skipped here for brevity. The

Figure 5: Validation of thermosiphon model

The refrigerant pressure in the PCM-HX evaporator shows a good match with experiment data. The pressure gradient generated by gravity is used for circulation of refrigerant in the thermosiphon. These gradients are small and refrigerant pressures at all locations in the circuit have similar values. The thermocouples are located at inlet and outlet of PCM-HX evaporator and are compared with the temperature inside the refrigerant control volume (See Figure 4). The model is able to predict the temperatures fairly accurately.

Finally, thermocouples are located on the top and bottom faces of the PCM storage. Average value of all these readings is plotted versus the value of the PCM temperature obtained from the model. The model matches the trends qualitatively. The temperatures are slightly higher for the simulation as it represents averaged temperature over the entire length of the cylinder as compared to the top face where the thermocouples are located. Heat losses by convection and radiation from the surface may result in slightly lower measured temperature.

The total recharge time by thermosiphon operation for PCM starting at 50°C to the point of complete solidification is 390 minutes. Time taken by sensible portion of the recharge (liquid at 50°C to 40°C) is 28 minutes. Fan running at 7 W is used for the entire operation. The energy required for the recharge = 45 Wh.

5. HEAT PUMP RECHARGE

The total recharge time for thermosiphon is 6 hours even after enhancing the heat transfer of PCM with graphite for a 4-hour cooling operation. Ideally, RoCo should be able to operate the entire length of a typical office day of 8 hours. It may be desirable to operate a heat pump cycle and use the compressor for a faster recharge of PCM. The dynamic behavior of the heat pump operation is investigated by simulation. The compressor used is of variable speed type and a parametric study of its RPM is conducted to extent of trade-offs involved.

The model diagram for simulation of heat pump operation is shown in Figure 6. The PCM model discussed in Section 3 and validated with thermosiphon experiment data in Section 4, is used in the simulation. Other components like compressor, thermal expansion valve (TXV), refrigerant tube and condenser are discussed in Qiao et al. (2015). The pressure and temperature at the outlet of PCM-HX evaporator are given as input to the TXV model. The compressor efficiencies and valve parameters are available from model calibration with cooling operation experiment data Dhumane et al. (2017b). The compressor RPM is varied and simulations are conducted at RPM values of 2100, 2600, 3100 and 3600.

Figure 6: Model diagram for heat pump recharge

The melt fraction and compressor power consumption during the recharge operation at four different compressor RPM is shown in

Figure 7. It can be observed that higher RPM leads to faster recharge, but the power consumption by the compressor to deliver the high RPM is large. Since the time duration of recharge is different, it is necessary to compare integrated power consumption during the recharge time duration.

Figure 7: Comparison of melt fraction and compressor power consumption during heat pump recharge

A fan is also operational during the entire recharge cycle and so a 7-W power usage needs to be added to the power consumed by the compressor. These values are plotted in Figure 8. The increase in energy consumption for the entire recharge operation is increased by 3-4 times that of the thermosiphon.

Figure 8: Recharge performance by heat pump operation at different compressor RPM

From experiment data, the coefficient of performance (COP) for just the vapor compression cycle is obtained to be 4.25, with 754.0 Wh of cooling delivered at energy consumption of 177.4 Wh. For a complete cycle COP, the energy consumption of recharge cycle also needs to be added to the energy consumption during cooling operation. The energy additions for thermosiphon = 45 Wh, while for heat pump recharge is shown in Figure 9. We can observe 35-55% reduction in recharge time duration in comparison with the thermosiphon operation from the heat pump operation. The COP for complete cooling and recharge operation decreases by 30-35% on switching from thermosiphon to heat pump operation.

Figure 9: Comparison of various recharge options for RoCo

RoCo enables elevation of set-point temperatures of buildings without compromising occupant thermal comfort and can provide savings in the range of 10-30% on the building air conditioner (Dhumane et al., 2017a). The building air conditioners typically consume power in the range of kWh. Faster recharge from heat pump operation will provide longer cooling operation of RoCo and consequently higher energy saving on the building air conditioner.

6. CONCLUSIONS

RoCo enables elevation of set-point temperatures of buildings without compromising occupant thermal comfort and can provide savings in the range of 10-30% on the building HVAC. It stores the condenser heat during its vapor compression cycle based cooling operation into a PCM based thermal storage, which needs to be recharged before its next cooling operation. Thus, RoCo alternates between its cooling operation and recharge operation. The first prototype used pure paraffin based thermal storage, and required 8-hours for recharge after a 2-hour cooling operation. Newer prototype used graphite enhanced PCM, and delivers 4-hour cooling operation with 6.5-hour recharge operation. A physically based model of the graphite enhanced PCM is developed and validated with this experimental data. The model is then used in a system simulation for a reversible heat pump based recharge operation to investigate its potential benefits. The compressor RPM is varied to understand the heat pump operation at different operation conditions. The recharge period from heat pump operation is observed to be 35-55% lesser than that from thermosiphon. However, the heat pump recharge decreases the COP of RoCo by 30-35%. However, the power consumption is in the order of a few hundred Wh and is small compared to the savings from building HVAC, which typically consumes power in kWh. Thus, heat pump recharge shows promise in comparison to the thermosiphon and will be used in future prototypes.

NOMENCLATURE

c	Specific heat capacity	[J kg ⁻¹ K ⁻¹]
G	Thermal conductance	[W K ⁻¹]
h	Specific enthalpy	[J kg ⁻¹]
H	Latent heat	[J kg ⁻¹]
k	Thermal conductivity	[Wm ⁻¹ K ⁻¹]
r	Radius	[m]
s	Source term	[J kg ⁻¹]
t	Time	[s]
T	Temperature	[K]

Greek

δT	Half of temperature glide	[K]
λ	Melt fraction	[-]
ρ	Density	[kg m ⁻³]

Subscript

l	liquid
m	mid-point of temperature glide
s	solid

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ACKNOWLEDGEMENT

This research was supported by the Advanced Research Projects Agency - Energy (ARPA-E) under Award DE-AR0000530 and Center for Environmental Energy Engineering (CEEE). The authors acknowledge the support of the team members of the Roving Comforter Project.