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ABSTRACT

A theoretical analysis has been introduced on vapor compression systems integrated with thermal energy storage (TES) using phase change materials (PCM) as the storage media. The analysis is based on the first principle and the coefficient of performance definition for heat pumping systems. A general equation has been derived to describe the overall system COP for systems integrated with TES, which equation is linked to the COPs when charging the TES and discharging the TES. The model of a 10.5kW (3T) R410A residential system from production has been used as an example to derive the overall system COP when integrated with TES. Data generated from the modeling calculations have been presented to help answer whether TES integrated with vapor compression systems offers energy savings and under what conditions.

1. INTRODUCTION

Thermal energy storage (TES) using phase change material (PCM) as storage media has been widely studied in recent years. Even though water/ice thermal storage has been used in large commercial scales for decades, question regarding whether TES is capable of offering energy savings is still lingering around. MacCracken has attempted to clarify the many "myths" associated with TES for large scale commercial applications. In the discussions, however, many complex issues associated with the application of thermal storage have been lumped together obscuring the essential question on whether TES offers any energy savings.

The report by Kung et al. has discussed an evaluation framework for TES integrated with packaged air conditioning (AC) systems. In their report, they have concluded that TES integrated AC systems can increase or decrease energy consumption depending on site-specific conditions. Their report has also indicated that for a comprehensive assessment of TES impacts, site metrics must then be converted to source metrics with discussions on identified methods and examples.

Recently, Ma et al. have disclosed simple configurations for the integration of TES/PCM with vapor compression systems in a patent application publication, in which TES systems are designed as add-on component to be integrated with AC systems during installation. The add-on concept from this publication can provide a general framework for the evaluation of overall energy efficiencies for AC systems integrated with TES.

In the current paper, a simple approach based on energy efficiency analysis has been taken to attempt to provide a theoretical base for site metrics development independent of the source metrics. The conceptual framework published by Ma et al. is used to evaluate system efficiencies of vapor compression systems integrated with TES in terms of an overall system COP. Modeling study of a 3T heat pump system integrated with an add-on design of TES/PCM is presented at the end of the discussions as an illustration.

2. THEORY

A vapor compression system integrated with TES is assumed to operate in two steps, charging the TES either when the cost of energy is lower or the system operation is more efficient, and discharging the TES when the energy cost is high or when the system efficiency is significantly low. In the charging step, a vapor compression system would transfer the heat from a TES unit to the environment. In the discharging step, the vapor compression system would transfer the heat from the cooling space to the TES, which could serve as a condenser (see Ma et al.). With this system configuration, a theoretical analysis to evaluate overall system energy efficiency can be developed.

In the following analysis, COP_{ch} stands for the coefficient of performance (COP) when TES is charged, COP_{ds} stands for the COP when TES is discharge, COP_{di} stands for the COP without *TES*, COP_{sys} stands for the overall system COP for a system integrated with a *TES* component, Q_{TES_C} is the capacity of the TES unit when it is charged, Q_{TES_d} is the capacity of the TES when it is discharged.

The COP, when the TES having capacity of Q_{TES} is charged, can be expressed as in equation (1).

$$COP_{ch} = \frac{Q_{TES\_c}}{Q_{dc} - Q_{TES\_c}} \tag{1}$$

Similarly, the COP during discharging can then be expressed as in equation (2), recognizing that the TES/PCM phase change temperature has to be between the evaporation and condensing temperatures.

$$COP_{ds} = \frac{Q_c}{Q_{TES\_d} - Q_c} \tag{2}$$

The overall COP for systems integrated with TES having a capacity of Q_{TES} can be written as equation (3), total cooling capacity is divided by the work done during charge and discharge cycles.

$$COP_{sys} = \frac{Q_c}{(Q_{dc} - Q_{TES\_c}) + (Q_{TES\_d} - Q_c)} \tag{3}$$

Rearrange all equations with proper equation substitution, the following can be derived,

$$COP_{sys} = \frac{COP_{ch} COP_{ds}}{COP_{ch} + COP_{ds}(Q_{TES\_c}/Q_{TES\_d})} \tag{4}$$

If the capacities of the TES during charging and discharging were identical, equation (4) would become simplified as in equation (5).

$$COP_{sys} = \frac{COP_{ch} COP_{ds}}{COP_{ch} + COP_{ds}} \tag{5}$$

However, this is not what happens in real world systems. The charging and discharging capacities are not identical because the evaporation temperatures are different. The overall system efficiency should be calculated in terms of total energies, i.e. the system COP equals to the ratio of total cooling energy to the total energy spent in charging and discharging steps. With this definition, the overall system efficiency can be linked to the efficiencies during charging and discharging cycles as in the following equation,

$$COP_{sys\_energy} = \frac{COP_{ch} COP_{ds}}{1 + COP_{ch} + COP_{ds}} \tag{6}$$

Note that in deriving equation (6), the only assumption is zero energy storage loss, which is reasonable in practice, because TES units would be more or less thermally insulated.

Equation (6) can further be developed with more assumptions to see how a realistic system might behave. Consider cases where Carnot cycles at charging and discharging steps are assumed, the charging and discharging efficiencies can be described by equations (7) and (8), respectively, where T_c is the cold temperature, $T_h$is the hot temperature, and T_{PCM} is the phase change temperature, which is the cold temperature during charging and the hot temperature during discharging.

$$COP_{ch} = \frac{T_{PCM}}{T_h - T_{PCM}} \tag{7}$$

$$COP_{ds} = \frac{T_c}{T_{PCM}-T_c} \tag{8}$$

The overall system COP becomes,

$$COP_{sys\_energy} = \frac{T_c T_{PCM}}{(T_h-T_{PCM})(T_{PCM}-T_c)+(T_{PCM}-T_c)T_{PCM}+(T_h-T_{PCM})T_c} \tag{9}$$

For a system without TES, the efficiency from a Carnot cycle is,

$$COP_{di} = \frac{T_c}{T_h-T_c} \tag{10}$$

Note that (9) reduces to (10) when T_{PCM}=T_c or T_{PCM}=T_h.

Equation (9) can be further developed to include the effect of heat transfer between the PCM and heat transfer fluid (HTF or refrigerant). During the freezing of PCM (charging TES), the HTF temperature has to be below the freezing point of the PCM. During melting of PCM (discharging TES), the HTF temperature has to be higher than the melting point of the PCM. If the melting temperature of a PCM is T_{PCM}, the HTF temperature during charging process has to be $T_{PCM} - \Delta T$, and the HTF temperature during discharging process has to be $T_{PCM} + \Delta T$ to account for the temperature difference required by heat transfer. Equation (9) becomes equation (11) as follows.

$$COP_{sys\_energy} = \frac{(T_{PCM}-\Delta T)T_c}{(T_h-T_{PCM}+\Delta T)(T_{PCM}+\Delta T-T_c)+(T_{PCM}+\Delta T-T_c)(T_{PCM}-\Delta T)+(T_h-T_{PCM}+\Delta T)T_c} \tag{11}$$

Note that equation (9) should be a special case of equation (11), when ΔT is zero.

3. RESULTS AND DISCUSSIONS

3.1 Analysis of Carnot Cycle

Figure 1 shows the comparison of system level efficiencies when temperature difference for heat transfer is considered

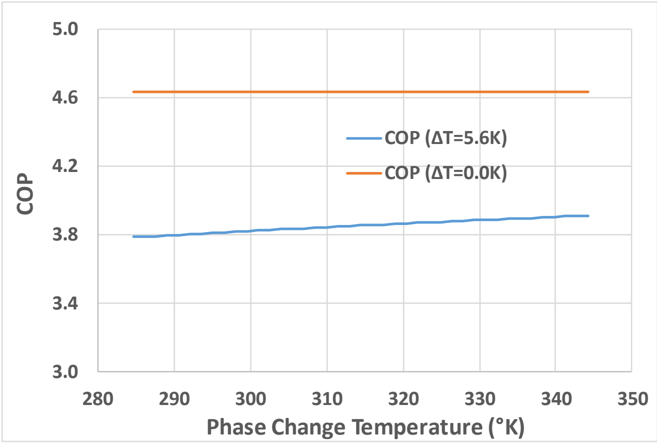

Figure 1. Overall system COP versus phase change temperature for Carnot cycles. T_h=344K and T_c=283K are used in the calculation.

with that when the temperature difference is not considered. In case of latter, the overall efficiency is not affected by TES phase change temperature at all.

Note that the COP monotonically increases when the phase change temperature increases from low to high. In reality, ΔT value is always none zero due to necessary heat transfer between the heat transfer fluid (refrigerant) and PCM, therefore, the COP of real world systems integrated with TES/PCM is less than that without TES.

Some might have thought that the advantage of using PCMs is its property of being at a constant temperature during phase change which could be beneficial to heat transfer. However, a careful examination of the heat transfer processes in typical TES/PCM designs reveals that to gain the benefit requires insightful design of the heat exchanging mechanism between refrigerant and the PCM, which determines the final cost of a TES/PCM system. For example, the thermal conductivities of the PCMs studied in publications (solid/liquid phase change) are significantly lower than metals. Organic based PCMs have a range of 0.15 W/m-K to 0.23 W/m-°K. Inorganic based PCMs have thermal conductivity similar to that of water, 0.53 W/m-K. These values are orders of magnitude less than, say, 401 W/m-K for copper, 205 W/m-K for aluminum. These low thermal conductivity values could limit heat exchanging system designs having temperature difference such that potential benefit of TES/PCM is diminished in order to reach meaningful heat transfer rate in practical systems.

3.2 Baseline System Modeling

In order to understand how a practical system integrated with TES might behave, a 3T residential system is used as an example. The system configuration proposed in the patent publication by Ma et al. has been used. And a simulation tool developed in-house was used to generate the system performances. The model setup is from a product performance model with the addition of an ideal TES/PCM model as the evaporator during charging step and as the condenser during discharging step. The heat transfer rate between refrigerant and PCM in the TES/PCM unit is assumed to be similar to Round Tube Plate Fin (RTPF) evaporator coils in production systems.

Figure 2 shows the diagram of the R410A 3T system conditions with input/output values on refrigerant side between each component and the air side conditions for outdoor and indoor airs. The model results in a COP of 3.51 and a capacity of 10.6kW (36390 btu/hr) at the conditions of 308.2K (35°C) outdoor air and 287.5K (14.4°C) indoor air delivery temperatures. The evaporation temperature is 284.7K (11.5°C). Typical compressor power is 2326W, 438W for the indoor fan, and 272W for the outdoor fan. This system is used as a reference baseline system for subsequent discussions.

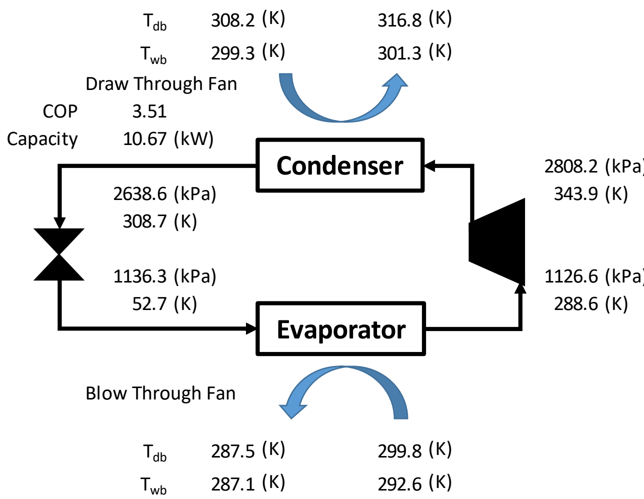

Figure 2. Typical conditions for a R410A residential system as baseline

3.3 Charge Cycle Modeling

Figure 3 shows the diagram of the baseline system with the refrigerant/air evaporator model replaced by an ideal evaporator model of a refrigerant/PCM heat exchanger, ideal model being a fixed temperature difference between refrigerant evaporation temperature and the PCM phase change temperature assumed without detailed description on

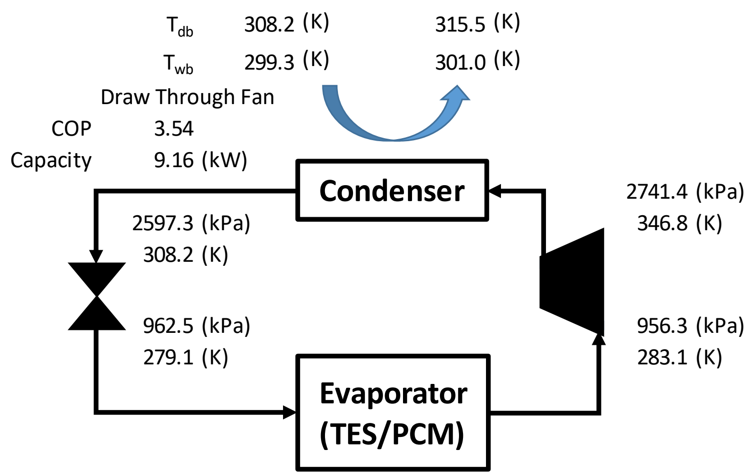

Figure 3. Conditions when TES/PCM charged at 308.2K outdoor temperature

heat exchange process. The phase change temperature is assumed as the evaporation temperature in the baseline system. Therefore, the evaporation temperature is set to 5.5K (5.55°C) lower than the phase change temperature to match the heat transfer rate for the baseline system. The model has resulted in outputs of 3.54 for COP and 9.12kW (31270 btu/hr) for the cooling capacity. The compressor power is now 2319W with same outdoor fan power and zero indoor fan power.

Note that, during charging, the refrigerant flow is directed to the TES/PCM storage unit in which solid/liquid PCM is cooled by directly expanding the refrigerant through metal tubes with fins, which is in contact with the PCM material. The electrical power consumption is decreased and zero heat is generated from the indoor fan motor slightly increasing the cooling capacity, resulting in an increased COP. However, the evaporation temperature is set at lower value than that in the baseline system to match the heat transfer rate for an add-on design, which is why the COP increases only slightly.

3.4 Discharge Cycle Modeling

Following similar logic, the model for discharging is setup from the baseline system model with the refrigerant/air

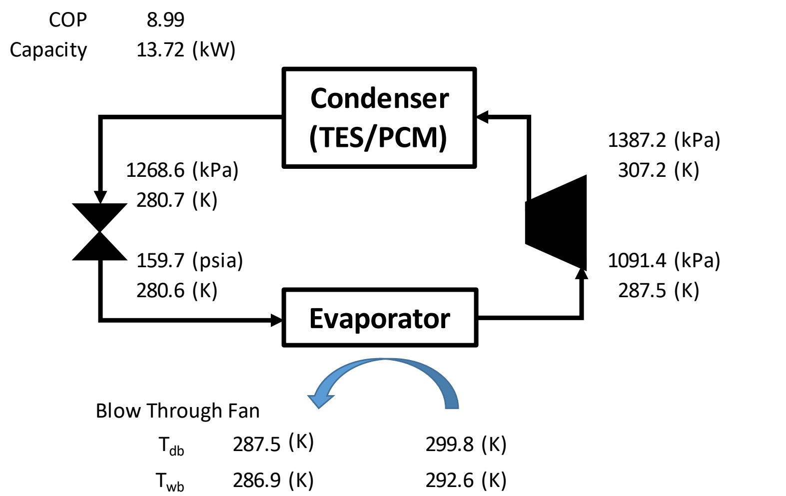

Figure 4. Conditions when system with TES discharged

condenser replaced by an ideal condenser model of refrigerant/PCM heat exchanger in the TES/PCM unit (see Figure 4). A condensing temperature with 5.5K (5.5°C) higher than the phase change temperature in Figure 3 to provide the driving force needed for heat transfer from the refrigerant to the PCM. During discharging, the indoor conditions are similar to that in Figure 2 and the condensing temperature is set as 290.2K (17.06°C). The model has an output of 8.99 for COP and a cooling capacity of 14.18kW (48600 btu/hr). The compressor power is 1089W, outdoor fan power is zero and indoor fan is the same as in the baseline system. With the baseline system performance data (Figure 2), the charging cycle performance data from the system integrated with TES/PCM (Figure 3), and the discharging cycle performance data from the same system integrated with TES/PCM (Figure 4), the overall COP defined by equation (6) can be compared with that from the baseline system to explore at what conditions TES could offer energy savings.

From the data shown in Figure 3 and 4, the charging COP is 3.54 and the discharging COP is 8.99 for a system integrated with TES. Equation (6) results in an overall COP of 2.35. This indicates that the system integrated with TES/PCM is less efficient than the baseline system. This is because the outdoor conditions are the same at the baseline system when the charging cycle performances are calculated. One would argue that charging of the TES/PCM should be done at night when the outdoor temperature is low and the vapor compression system runs more efficiently.

3.5 Analysis of Charge Conditions to Achieve Energy Savings

Suppose charging TES happens during the early morning hours of a summer day with conditions, say, 298.2K (25°C) outdoor air with a diurnal temperature swing around 10K (10°C), the charging COP from the model output is 4.79 and the overall COP is calculated with equation (6) as 2.91. This system level COP is still significantly less than 3.51 from the baseline system. In order to figure out the outdoor temperature at which the TES is charged such that the system COP is the same as the baseline system, equation (6) is used to set the system level at 3.51 and calculate the COP during charging step using the discharging COP of 8.99. A COP of 6.40 during the charging step is required for breaking an even COP compared with the baseline system. The charge model is subsequently used to find out the outdoor temperature at which the charging COP equals to 6.40, which results in an outdoor air temperature of 287.9K (14.71°C) (Figure 5) and compressor power of 1485W. Therefore, in order to have energy savings, the TES has to be charged at outdoor temperatures less than the temperature of 287.9K (14.71°C).

The exercise has demonstrated that a diurnal temperature swing from the day time 308.2K (35.0°C) to a night time 287.9K (14.71°C) has to happen with certain frequency during cooling seasons in order to achieve meaningful energy savings with vapor compression systems integrated with TES/PCM as add-on component.

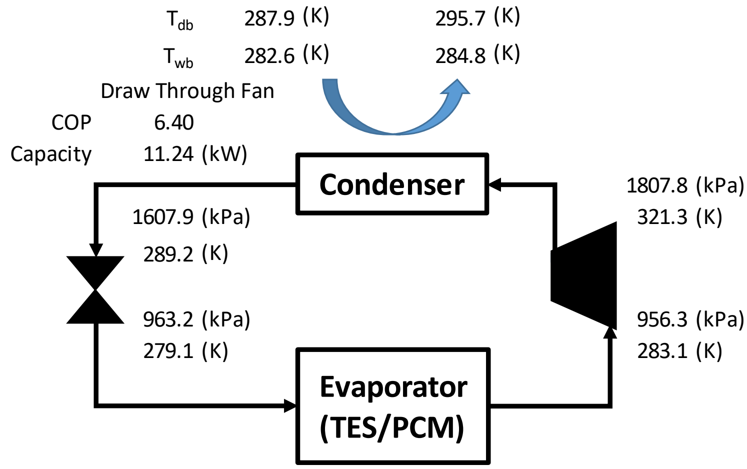

Figure 5. Conditions when TES/PCM charged at 287.9K outdoor temperature

4. CONCLUSIONS

The analysis of the TES system presented in this manuscript has demonstrated that TES integrated with vapor compression systems has potential to offer energy savings at the right conditions. However, the right conditions have to involve diurnal temperature swing as large as more than 20 Celsius. Such a large diurnal temperature swing are rare in geographic locations that cooling is necessary most of the hours in a summer day. The analysis has also shown why energy savings is hard to achieve with TES integrated with small systems due to the heat transfer resistance to and from the thermal storage media (e.g. PCMs).

NOMENCLATURE

COP	coefficient of performance	(-)
Q	cooling capacity	(W)
T	temperature	(K)

Subscript

ch	charge cycle
ds	discharge cycle
di	cycle without TES
sys	system
c	cooling, cold
h	heating, hot
C	cooling, cold
H	heating, hot
PCM	phase change material
TES_d	TES discharge cycle
TES_c	TES charge cycle

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