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# Performance Analysis of Heat Pump Water Heater System Operating on a New Storage Heat Pump Cycle to Achieve Higher Operating Range

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**Abstract**

Heat pump systems are often restricted to operate within a certain temperature range due to the limits on pressure ratio and discharge temperature. A new storage heat pump concept aimed at improving the operating temperature range of a heat pump system is proposed. Performance of an R134a heat pump water heater (HPWH) system with wrap-around condenser coil operating based on the new concept is studied by means of a system model. The storage heat pump cycle involves system operation in two modes to achieve the high temperature lift. In Mode I, ambient air acts as heat source for the evaporator and full length of wrap-around coil is used as condenser, whereas in Mode II, a throttling valve splits the wrap-around coil into a condenser and evaporator. Here, the intermediate temperature water in the lower tank region acts as heat source for the evaporator. This lifts up the evaporating pressure which enables the heat pump to operate at a lower pressure ratio and reach a higher water end temperature. Modeling results obtained for the storage heat pump system are also compared with a conventional HPWH system that uses back-up electric elements to reach the required water temperature when the heat pump is unable to operate.



*Keywords: heat pump water heater; operating range; energy storage; modeling*

## 1. Introduction

Heat pump systems operate on the principle of vapor compression cycle and thus offer a much higher energy utilization efficiency as compared to conventional methods like electric resistance heating. The maximum possible temperature difference between the heat source and heat sink for the system is referred as the operating temperature range of the system and it is limited by the compressor operating envelope in conjunction with the refrigerant used. Exceeding the operating temperature range can cause issues such as degradation of lubricant as well as lower heating capacity due to the high pressure ratio and discharge temperature. The heat pump water heater (HPWH) system faces this situation when it is required to provide hot water in cold ambient conditions. A new storage heat pump concept aimed at improving the operating temperature range of a heat pump system is introduced in this paper. The system under consideration is an R134a HPWH with wrap-around condenser coil.

Implementation of the new concept requires a split-condenser design, an energy storage element and the system operation in two different modes. Mode I is same as conventional system operation whereas as the wrap-around coil is split into condenser and evaporator in Mode II. The water contained in the lower portion of tank itself acts as energy storage element for the system, wherein energy stored by the system in Mode I is utilized in Mode II to enable the system to operate at a lower pressure ratio and discharge temperature compared to a conventional heat pump system. Objective of this study is to predict the performance of a storage heat pump system by the means of a system model. Model is used to obtain the system performance under different scenarios and determine the suitable wrap-around coil split ratio along with compressor speed and

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time duration for system operation in Mode II. System model consists of a quasi-steady vapor compression cycle model linked with a transient water tank CFD model. Performance of the storage heat pump system is also compared with a conventional HPWH that uses back-up electric heating elements to reach the target water temperature when heat pump is unable to operate.

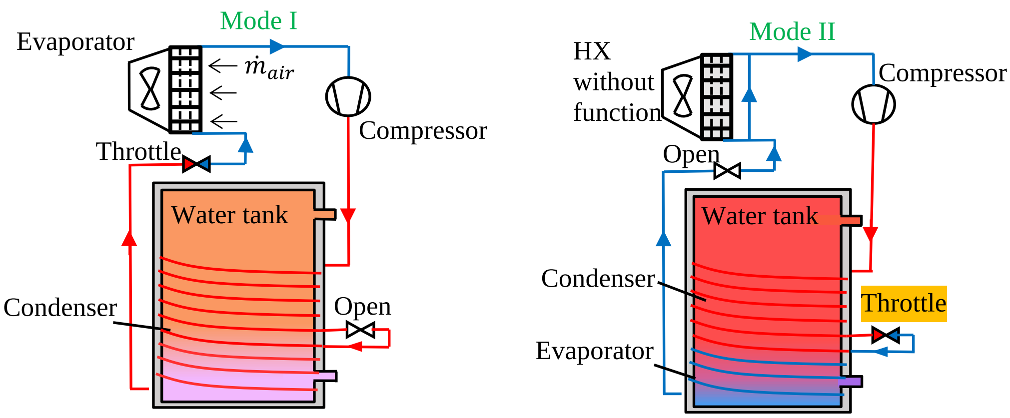
Several authors have studied methods such as refrigerant injection in HPWH system (Liu *et al.*, 2008) and usage of cascade heat pump systems (Wu *et al.*, 2012) with an aim to overcome issues of high discharge temperature and low heating capacity when operating the heat pump in cold climate. Refrigerant injection can cause issues such as compressor slugging while the cascade system can lead to increased complexity and cost for the heat pump. The storage heat pump concept introduced in this paper is based on energy storage at an intermediate temperature and can be implemented through a single stage heat pump design.

### *1.1. New Storage Heat Pump Concept*

A schematic of the storage heat pump system operating in two modes is shown in Figure 1. The system operation in Mode I shown in Figure 1a is same as the conventional system operation. For a conventional heat pump system, the ambient air is used as heat source for the fin-tube evaporator and water contained in tank is heated by heat rejection from the condenser. The schematic also shows the water tank temperature stratification which occurs due to the effect of natural convection. As the water contained in the tank heats up, the temperature difference between the ambient (heat source) and water (heat sink) increases. This results in an increase in the pressure ratio and discharge temperature for vapor compression cycle. If the pressure ratio becomes too high, then the compressor has to be cycled off and the system is unable to reach the target water temperature while using the heat pump. Conventional HPWH system often uses back-up electric heating elements to reach the higher water temperature when the heat pump is unable to operate.

The storage heat pump system involves switching between the two modes to achieve the high temperature lift. In Mode I, the water contained in the tank is heated from a low to intermediate temperature while using the full length of wrap-around coil as condenser. Then the system operation switches to Mode II, in which the wrap-around coil is split by means of a throttle valve to use top portion of coil as condenser and the bottom portion as evaporator. The fan attached to the fin-tube heat exchanger is kept off so that the heat exchanger is unused in this mode. In Mode II, the intermediate temperature water present in lower region of tank would act as heat source for the evaporator. This increases the evaporating pressure and reduces the pressure ratio which allows the system to continue to operate and reach a higher water temperature in the upper region of the tank.

Thus, instead of using the back-up electric elements to reach a higher water temperature the storage heat pump system in Mode II would use the heat pump compressor to pump the energy contained in the water in lower tank region to that in the upper region. While evaporator received energy from the ambient in Mode I, the energy received by evaporator in Mode II comes from within the system and the temperature of water in lower tank region reduces. It should be noted that the tank will not heat up faster than in the conventional HPWH because the heating capacity of vapor compression cycle is lower than the heating capacity of back up electric elements used in conventional HPWH system. Instead, the storage heat pump concept allows the system to reach a higher water end temperature that could otherwise not be reached with a conventional system while only using the heat pump.




(a)

(b)

Figure 1. Heat pump water heat system operating in the two modes: a) Mode I and b) Mode II

## 2. Model Description

HPWH systems are commonly modeled by combining a quasi-steady vapor compression cycle model with a transient water tank model. Ibrahim *et al.* (2014) developed a dynamic simulation model for an air source HPWH with immersed coil condenser to obtain system performance under different climatic conditions. Vapor compression system was modeled under quasi-steady assumption and the water tank was modeled through a lumped parameter approach. Shen *et al.* (2018) developed a quasi-steady HPWH model with wrap-around condenser coil. Water tank was modeled by dividing the tank into several nodes in 1D to model the temperature stratification in the tank. Deutz *et al.* (2018) used the zonal tank model approach in their HPWH model. The zonal water tank model takes into account the thermal and inertial boundary layer forming along the tank wall when heated by the condenser. Li and Hrnjak (2018) presented an experimentally validated HWPH model in which the vapor compression system model was linked with a transient water tank CFD model. The CFD model of the water tank could give accurate estimation of the water side temperature and flow field inside the water tank.

The linked modeling approach from Li and Hrnjak is used to model the storage heat pump system in this work. System model consists of a quasi-steady vapor compression system model linked with a transient water tank CFD model. The linked modeling algorithm is shown in Figure 2b. System is simulated with water in *warm-up* condition, where the water in the tank is heated from an initial to set point temperature without any water draw from the system. The vapor compression system model consists of a compressor model, condenser and evaporator models and an expansion valve modeled by isenthalpic process. This model is solved by using an iteration algorithm (shown in Figure 2a) based on the Newton Raphson method adopted from Inampudi *et al.* (2021). Pressure and specific enthalpy are taken as connecting variables between the various components and the system model is solved iteratively until the refrigerant states at connecting points do not need further modification.

### *2.1. Compressor Model*

For the given refrigerant inlet pressure, inlet specific enthalpy and outlet pressure, the compressor model uses three variables: volumetric efficiency, compression efficiency and isentropic efficiency to compute the refrigerant mass flow rate, discharge specific enthalpy and the electric power consumed by the compressor.

$$\eta_{vol} = \frac{\dot{m}}{\rho_{suc} * V_{disp} * f} \tag{1}$$

$$\eta_{comp} = \frac{(h_{cpro,isen} - h_{cpri})}{(h_{cpro} - h_{cpri})} \tag{2}$$

$$\eta_{isen} = \frac{\dot{m} * (h_{cpro,isen} - h_{cpri})}{\dot{W}_{electric}} \tag{3}$$

Curve fitting equations that relate efficiencies to pressure ratio were obtained from experimental data. A linear curve fitting between efficiencies and pressure ratio was found to give a reasonable result. Mass flow rate, discharge specific enthalpy and electric power consumption is obtained by solving equations (1), (2) and (3) respectively.

### *2.2. Heat Exchanger Models*

Heat exchangers in the system are modeled by using the finite volume approach. The heat exchanger is divided into discrete elements along the refrigerant flow circuit and the heat transfer and pressure drop in each element is obtained by applying equations for conservation of energy and momentum. Refrigerant inlet pressure and specific enthalpy, mass flow rate and external fluid (air for fin-tube evaporator and water for wrap-around coil heat exchanger) conditions are taken as inputs for the heat exchanger model. Discrete elements are solved sequentially to obtain the refrigerant specific enthalpy and pressure at exit of each element. The model finally gives the refrigerant specific enthalpy and pressure at the outlet of the heat exchanger. The $\varepsilon - NTU$ method is used for calculating the heat transfer in each element considering a cross flow configuration between external fluid and refrigerant. Table 1 shows the correlations used in calculating pressure drop and heat transfer coefficient for the evaporator and condenser. Water side heat transfer coefficient for the wrap-around coil heat exchanger is obtained from the water tank CFD model.

Table 1. Correlations used to obtain heat transfer coefficient and pressure drop in each element for heat exchanger models

| | **Condenser** | | **Evaporator** | |
|---|---|---|---|---|
| | HTC | $\Delta P$ | HTC | $\Delta P$ |
| Single phase refrigerant | Gnielinski (1976) | Churchill (1977) | Gnielinski (1976) | Churchill (1977) |
| Two-phase refrigerant | Dobson and Chato (1998) | Souza *et al.* (1992) | Wattelet *et al.* (1994) | Souza *et al.* (1992) |
| Air | - | - | Wang *et al.* (2009) | |

## *2.3. Water Tank CFD Model*

A transient water tank CFD model is developed in ANSYS Fluent. The three-dimensional cylindrical water tank is simplified to a two-dimensional axis-symmetric model by assuming the heat flux from each coil turn to be uniform in the circumferential direction. Each turn of the wrap-around coil is represented by a line segment (of the length equal to the pitch of coil turn) at the tank wall. The heat flux from each coil turn is implemented as a time varying boundary condition by using User Defined Functions in Fluent. The natural convection of water is modeled by computing the density based on Boussinesq approximation. Laminar flow model is selected for the simulation and the SIMPLE scheme is selected for the pressure velocity coupling.

## *2.4. HPWH System Model Solving Procedure*

The system model consists of a quasi-steady vapor compression cycle model linked with a transient water tank CFD model using the linked modeling approach of Li and Hrnjak (2018). Starting from a guess value of heat flux profile q''(n,t) (here n is the coil turn number and t is time) at the tank wall, the CFD model is solved for the full water heating period. The resulting water side temperature and heat transfer coefficient are used as inputs for the vapor compression system model which is then run at various time instances to obtain the new temporal heat flux profile. The water tank model is run again with the new heat flux profile as input and the iterations continue until convergence in heat flux profile through tank wall is obtained. The vapor compression system model takes the air side mass flow rate and temperature (for the fin-tube evaporator), water side temperature and heat transfer coefficient (for the wrap-around coil heat exchanger), compressor speed, superheat (set by EXV) and subcooling (set by system charge) as inputs.

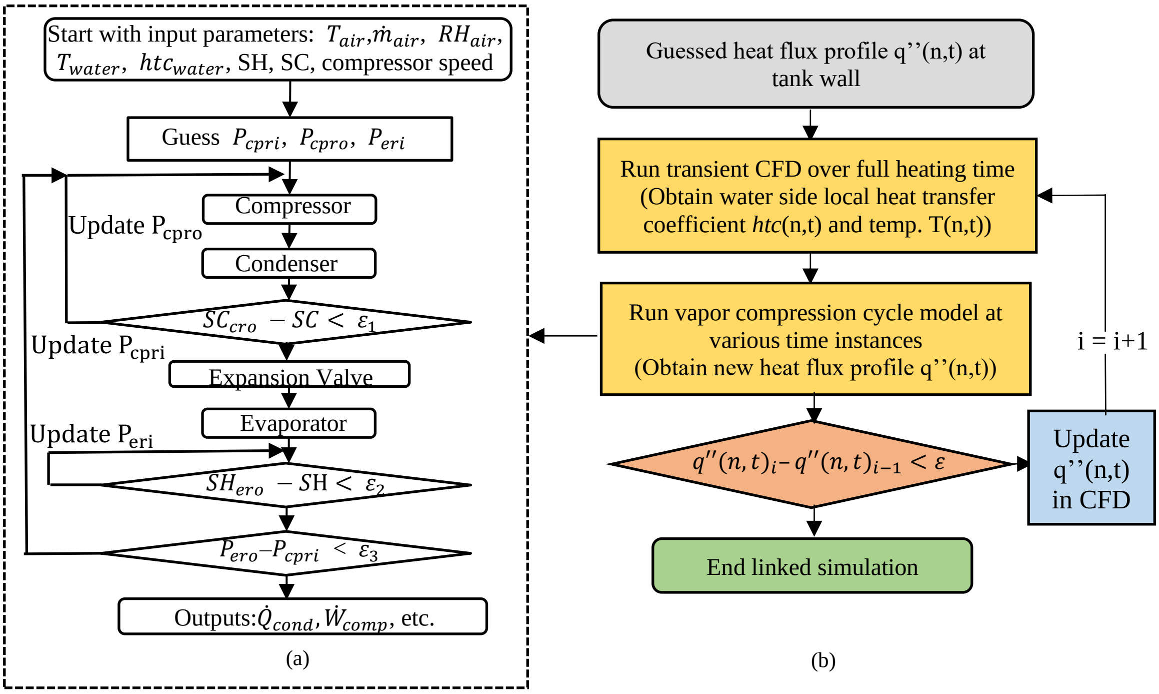



Figure 2. a) Vapor compression cycle model solving algorithm, b) Linked modeling algorithm

## 3. Modeling Results

### *3.1. Storage Heat Pump System*

A previous experimental study (Patel and Elbel, 2022) carried out on a conventional HPWH showed that for an initial water temperature of 25℃ the system could heat water only to a bulk average temperature of 48℃ by using the heat pump. The heat pump compressor is cut-off when the water temperature reaches 48℃ and the conventional HPWH would use a back-up electric coil to reach a higher water temperature. The time taken for the system to heat the water from 25℃ to 48℃ in warm-up condition (no water draw from tank) was about 180 minutes.

The storage heat pump cycle involves system operation in two modes to reach a higher water temperature than a conventional HPWH could reach while using the heat pump. Performance of the storage heat pump system is obtained for the operating conditions shown in Table 2. System operates in Mode I until the water contained in the tank is heated to bulk average temperature of 48℃. Then the system operation switches to Mode II in order to get the reduction in pressure ratio and discharge temperature.

Table 2. Operating conditions for the storage heat pump system model

| | |
|---|---|
| $T_{ambient}$=25℃, $T_{water,initial}$=25℃, Subcooling = 5℃, Superheat = 5℃ | |
| Compressor Speed in Mode I = 3600$min^{-1}$, Mode II = 1800$min^{-1}$ | |
| Split ratio for wrap-around coil in Mode II (no. of coil turns used as evaporator/ no. of coil turns used as condenser = Ne/Nc) = 3/7 | |
| Time | Mode |
| t=0 to 180 minutes (until $T_{water}$ = 48℃) | I |
| t=180 to 240 minutes | II |
| t=240 to 265 minutes | I |
| t=265 to 300 minutes | II |

As show in Table 2, the compressor speed in Mode I is 3600$min^{-1}$, while in Mode II it is taken as 1800$min^{-1}$. The compressor speed for storage heat pump system operating in Mode II is lowered by 50% compared to speed in Mode I. The compressor speed is reduced in order to match the compressor capacity with the smaller size of heat exchangers in Mode II. The system uses fin-tube evaporator and full length of wrap-around coil as condenser in Mode I, while splitting the wrap-around coil into condenser and evaporator in Mode II. Due to this, there is a change in the size of the heat exchangers in the system upon switching to the Mode II. The $\Delta T$ $\left(\Delta T_{hx} = T_{ri,sat} - T_{ext,fluid}\right)$ for a heat exchanger is defined as the difference between the refrigerant saturation temperature and the external fluid temperature. The $\Delta T$ values of condenser and evaporator for the system operating in Mode I are listed in Table 3. The $\Delta T$ values predicted for the system upon switching to Mode II and operating with two different compressor speeds are also shown in the Table 3.

It can be observed that if the compressor speed in Mode II is kept the same as the speed in Mode I then the $\Delta T$ values for the heat exchangers in Mode II are much higher than the corresponding values in Mode I. This indicates that the compressor is oversized for the size of heat exchanger in Mode II and the high $\Delta T$ values will be detrimental to reducing the pressure ratio in Mode II. Thus, to keep the $\Delta T$ for heat exchanger in Mode II closer to the value in Mode I, the compressor speed is reduced by 50% in Mode II. The reduction in compressor speed will result in a lower heating capacity for the vapor compression cycle in Mode II compared to Mode I.

Table 3. The $\Delta T$ values for condenser and evaporator in Mode I and Mode II

| | **Mode I (at t=180min)** | **Mode II (at t=182min)** | |
|---|---|---|---|
| **Compressor speed** | 3600$min^{-1}$ | 3600$min^{-1}$ | 1800$min^{-1}$ |
| $\boldsymbol{\Delta T_{cond}}$ [℃] | 8.4 | 15.7 | 10.5 |
| $\boldsymbol{\Delta T_{evap}}$ [℃] | 11.9 | 20.5 | 14.6 |

The P-h plot obtained for the storage heat pump system for various time instances from t=0 to 240 minutes is shown in Figure 3. The system achieves a lift in evaporating pressure upon switching to Mode II at t=180 minutes. This is because the evaporator in Mode II will use the water contained in the bottom region of the tank as the heat source. The water contained in the tank (heat source in Mode II) would be at a higher temperature than the ambient air (heat source in Mode I) as the water temperature was elevated during system operation in Mode I. The condensing pressure for the system also increases upon switching to Mode II. This is due to a slightly higher $\Delta T_{cond}$ (resulting from a smaller condenser size in Mode II compared to Mode I) for system in Mode II. However, the lift in evaporating pressure achieved upon switching to Mode II enables the system to maintain a lower pressure ratio and discharge temperature while the water in the upper tank region continues to heat up.

Figure 4a shows the variation in water temperature with time (at 10 different vertical locations spaced uniformly) as system switches operation between the two modes. The water tank CFD model showed that the water temperature in the tank remains almost uniform in the radial direction other than within a small boundary layer region near the tank wall. The water tank temperature contour at t=240 minutes is shown in Figure 4b. It can be observed from Figure 4a that the water temperature in the upper tank region (4/5$^{th}$ of total tank height for wrap-around coil split-ratio Ne/Nc = 3/7) continues to increase while the water temperature in the lower region increases in Mode I and reduces when operating in Mode II. Water contained in the lower tank region is used an energy storage element, wherein the energy stored in Mode I is utilized by the system in Mode II.

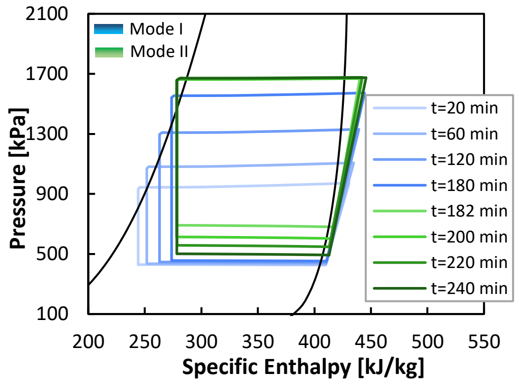



Figure 3. Pressure vs Specific enthalpy plot for system at different time instances

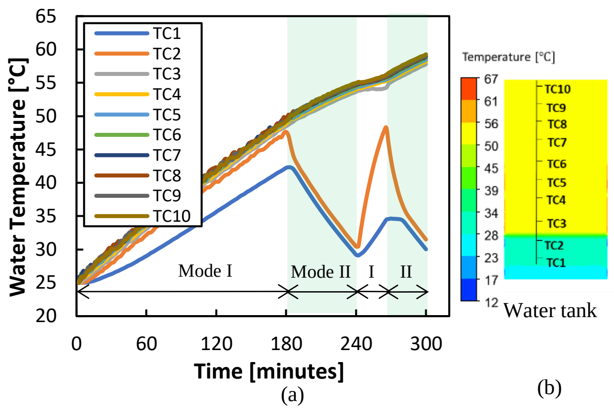



Figure 4. a) Water temperature measured at different vertical locations in the tank b) Water temperature contour at t=240 minutes

While operating in Mode II (from t=180 to 240 minutes), the water temperature in the bottom tank region reduces due to the heat extraction by the evaporator. The lift in evaporating pressure achieved upon switching to Mode II also diminishes with time. This limits the time duration for which it is advantageous (in terms of reduced pressure ratio and discharge temperature) for the system to operate in Mode II. However, the system can again switch back to Mode I (at t=240 minutes) as the intermediate operation in Mode II would have resulted in lowering the temperature of water in the bottom tank region. This enables the system to maintain a lower pressure ratio and discharge temperature compared to a system with continuous operation in Mode I. Now when the system operates in Mode I (from t=240 to 265 minutes), most of the heat transfer from the condenser coil would happen from the coil turns located around bottom region of tank. The temperature of water in bottom tank region increases at a much faster rate and once the water in bottom tank region is heated up to an intermediate temperature level, the system can again switch back to Mode II (at t=265 minutes) to use this water as heat source for the evaporator.

The variation in discharge temperature and pressure ratio with time for the storage heat pump system are shown in Figure 5. When operating in Mode I (between t=0 to 180 minutes) the pressure ratio increases as the water heats up and results in an increase in condensing pressure while the evaporating pressure remains almost same due to the ambient temperature being constant. The pressure ratio reduces as the system switches to Mode II (at t=180 minutes). This is due to the lift in evaporating pressure achieved upon switching to Mode II. The reduction in pressure ratio would contribute to lowering the discharge temperature for the system while the higher condensing pressure in Mode II will contribute to increasing the discharge temperature. The combined effect causes the discharge temperature to reduce slightly upon switching to Mode II

When operating in Mode II (between t=180 and 240 minutes) the pressure ratio increases as the water in upper tank region (heat sink) heats up while the water in the lower tank region (heat source) cools down. The discharge pressure also increases when operating in Mode II. As the system again switches to Mode I (at t=240 minutes) and uses full length of wrap-around coil as condenser, the discharge temperature and pressure ratio drop due to the lower condensing pressure for the system caused by the low temperature of water in the bottom tank region. When operating in Mode I (between t=240 and 265 minutes) the pressure ratio and discharge temperature increase as water in the tank is heated up. Now, when the system switches again to mode II (at t=265 minutes) the pressure ratio drops while the combined effect of lower pressure ratio and higher condensing pressure causes the discharge temperature to increase slightly. Compared to a conventional heat pump system (which is equivalent to a continuous operation in Mode I) the storage heat pump system can maintain the lower discharge temperature and pressure ratio. The heating capacity and compressor power consumption are shown in Figure 6. The heating capacity is lower in Mode II due to the reduction in size of heat exchangers and compressor speed in Mode II as compared to Mode I.

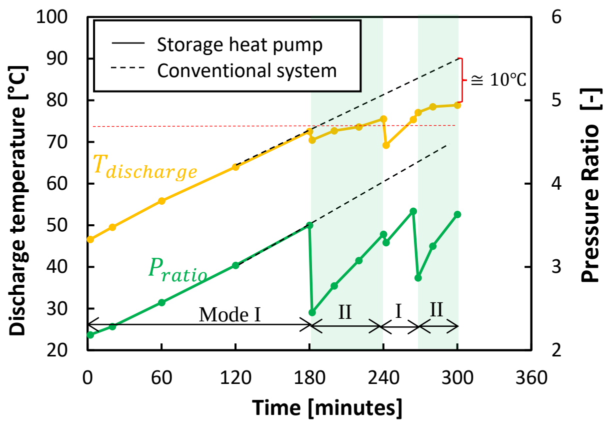



Figure 5. Discharge temperature and pressure ratio vs time for the storage heat pump system

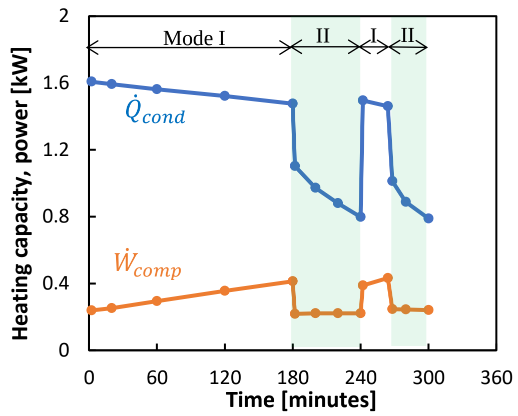



Figure 6. Heating capacity and compressor power consumption vs time for the storage heat pump system

### *3.2. Effect of Wrap-Around Coil Split Ratio on System Performance*

The storage heat pump system uses full length of wrap-around coil as condenser in Mode I while splitting the wrap-around coil to use top portion of coil as condenser and bottom portion as evaporator in Mode II. The split ratio for wrap-around coil in Mode II is defined as (Ne/Nc) = number of coil turns used as evaporator/ number of coil turns used as condenser in Mode II. The split ratio determines the relative sizes of condenser and evaporator in Mode II and thus impacts the system performance. The model is used to study the system performance in Mode II for four different split ratios. Other than varying the split ratio all the other conditions for the four cases were kept same as listed in Table 4.

Table 4. Operating conditions for the system model to study effect of wrap-around coil split ratio on system performance in Mode II

| |
|---|
| $T_{ambient}$=25°C, $T_{water,initial}$=25°C, Subcooling = 5°C, Superheat = 5°C |
| System operates in Mode I from t=0 to 180 minutes and in Mode II from t=180 to 240 minutes |
| Compressor Speed in Mode I = 3600$min^{-1}$, Mode II = 1800$min^{-1}$ |
| Four different cases considered are: Split ratio for wrap-around coil in Mode II (Ne/Nc) = 2/8, 3/7, 4/6 and 5/5 |

Figure 7 shows the comparison between P-h plot of the different cases upon switching Mode II. It can be observed that a system with a higher split ratio achieves a higher lift in the evaporating pressure upon switching to Mode II. This is because the system with a higher split ratio will have a larger size of evaporator compared to the lower split ratio case and this results in a lower $\Delta T_{evap}$ $(T_{w,lower} - T_{ref,evap})$ for the system with higher split ratio. The system with a higher split ratio also shows a higher condensing pressure in Mode II because of the relatively smaller condenser size that results in a higher $\Delta T_{cond}$ $(T_{ref,cond} - T_{w,upper})$ for the system. The discharge temperature and pressure ratio for the different cases is shown in Figure 8. The pressure ratio is almost the same for all the cases while the system with higher split ratio shows a higher discharge temperature due to the higher condensing pressure in Mode II.

The average heating capacity for the system in Mode II (average $\dot{Q}_{cond}$ from t=180 to 240 minutes) for the different cases is listed in Table 5. The compressor speed in Mode II is kept same for all the cases due to which there is not much difference between the heating capacity among the different cases. With the pressure ratio being almost same for all the cases, the system with a higher split ratio has a higher evaporating pressure which results in higher refrigerant mass flow rate. This leads to a higher heating capacity for system with higher split ratio in Mode II. The average value of the ratio of heating capacity and compressor power consumption in Mode II is also listed in the Table 5. This system with intermediate split ratio such as

Ne/Nc=3/7 and Ne/Nc=4/6 have a higher heating capacity per unit power consumption as compared to other cases which is due to better matching between the compressor capacity and size of condenser and evaporator in these systems. It should be noted that the heating capacity of system is the rate of energy transfer to the water in upper region of the tank while water in lower tank region acts as heat source for the evaporator. Thus, energy received by the evaporator comes from within the system and temperature of water in lower tank region (within bottom of tank to the height at which wrap-around coil is split) reduces when operating in Mode II.

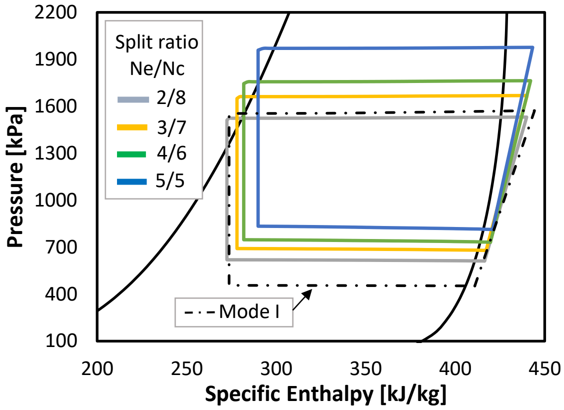



Figure 7. P-h plot upon switching to Mode II (at t=182 minutes) for different split ratio cases

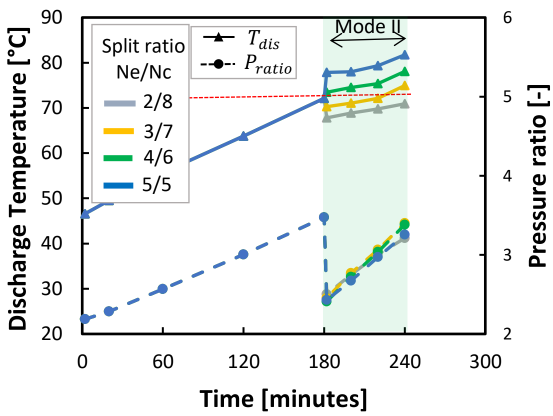



Figure 8. Discharge temperature and pressure ratio vs time for different split ratio cases

Table 5. Average heating capacity and heating capacity per unit compressor power consumption in Mode II (from t=180 to 240 minutes) for the different cases

| Split ratio Ne/Nc | Average $\dot{Q}_{cond}$ [kW] in Mode II | Average $\dot{Q}_{cond}/\dot{W}_{comp}$[-] in Mode II |
|---|---|---|
| **2/8** | 0.93 | 4.41 |
| **3/7** | 0.98 | 4.49 |
| **4/6** | 1.04 | 4.50 |
| **5/5** | 1.13 | 4.40 |

While the higher split ratio of Ne/Nc=5/5 gives a higher heating capacity, the absolute value of discharge temperature for this case becomes even higher than its value before in Mode I (at t=180 minutes). The lower split ratio case of Ne/Nc=2/8 will have a lower value of condensing pressure and discharge temperature but the heating capacity for this case is lowest and it would take a longer time to reach the required hot water temperature. Thus, an intermediate split-ratio such as Ne/Nc=3/7 can be selected for the wrap-around coil in the storage heat pump system. This split ratio will result in the system having a medium heating capacity (when compared with other split-ratio cases) while also maintaining the value of discharge temperature lower than the value in Mode I.

### *3.3. Effect of Compressor Speed in Mode II on System Performance*

The storage heat pump system operates with a reduced compressor speed in Mode II in order to match the compressor capacity with the smaller size of heat exchangers in Mode I. The compressor speed in Mode I is 3600$min^{-1}$(corresponding to full compressor capacity) while it is reduced in Mode II. For a certain split ratio, the upper limit on the compressor speed in Mode II is based on the consideration that the compressor should not be oversized compared to condenser and evaporator in the system.

In the previous section, the system performance was obtained for system with four different values of split ratio and with a compressor speed of 1800$min^{-1}$ in Mode II. The system with higher wrap-around coil split ratio (Ne/Nc=5/5) in Mode II showed a higher heating capacity due to its higher evaporating pressure compared to system with lower split ratio. However, the higher split ratio could not be selected for the system because the condenser is undersized compared to the compressor capacity in Mode II which leads to system having a much higher condensing pressure and discharge temperature. In order to bring down the condensing pressure and discharge temperature for the high split ratio case the compressor speed would have to be reduced.

Table 6 shows the discharge temperature (at t= 182 minutes) for the system with split ratio Ne/Nc=5/5 and two different compressor speeds in Mode II. By lowering the compressor speed from 1800 $min^{-1}$ to 1200$min^{-1}$ in Mode II, the reduction in condensing pressure and discharge temperature is obtained but the heating capacity of the system also reduces. The system with split ratio Ne/Nc=5/5 and compressor 1200$min^{-1}$ perhaps shows a lower heating capacity than system with split ratio Ne/Nc=3/7 and with compressor speed 1800$min^{-1}$ in Mode II. This shows that the split ratio affects size of heat exchangers and the absolute value of evaporating and condensing pressure for the system in Mode II, but the heating capacity of the system in Mode II is much more sensitive to the compressor speed than the split ratio. Thus, it is more advantageous in terms of heating capacity to keep a higher compressor speed in Mode II with the upper limit on speed being that the reduction in discharge temperature and pressure ratio is achieved upon switching from Mode I to II.

Table 6. Discharge temperature and heating capacity for the system upon switching to Mode II

| | **Storage heat pump system in Mode II (at t=182 minutes)** | | |
|---|---|---|---|
| **Wrap-around coil split ratio** | Ne/Nc=5/5 | | Ne/Nc=3/7 |
| **Compressor speed** | 1800$min^{-1}$ | 1200$min^{-1}$ | 1800$min^{-1}$ |
| $T_{discharge}$ ($T_{discharge}$ = 72.1℃ in Mode I at t=180minutes) | 77.8℃ | 71.7℃ | 70.3℃ |
| Heating Capacity ($\dot{Q}_{cond}$ =1.48 kW in Mode I at t=180minutes) | 1.24 kW | 0.93 kW | 1.10 kW |

### *3.4. Comparing System Performance with Conventional System*

Conventional heat pump water heater (HPWH) systems often use back-up electric elements to heat the water to the target temperature when the heat pump is unable to operate. Performance of the storage heat pump system obtained for the operating conditions shown in Table 2 is compared to a conventional HPWH that uses heat pump to heat water from 25℃ to 48℃ (t=0 to 180 minutes) and then uses back-up electric elements to achieve the water heating from 48℃ to 58℃ (t>180 minutes). The water contained in the tank is heated without any water draw from the system.

The conventional HPWH system usually contains one upper and one lower heating element mounted on the tank wall. Both the elements have the same power rating and only one of the elements is operated at a time to avoid exceeding the limits on the allowable current draw. When using the electric element, the heating capacity for the system will be equal to the power of the electric element. Most conventional HPWH operate on 240V with current draw limited to about 20A and having 4500W heating element. Assuming that the system is powered by a limited electric supply of 120V the power of the electric element reduces to 1125W. Systems having 1125W electric element are not common and the example is used only for comparing storage heat pump system performance with two conventional systems having different power rating of electric elements. The Figure 9 shows the water temperature vs time predicted for conventional HPWH operating on two different electric supplies - a) operating on 120V with limited current draw (1125W electric elements) and b) operating on 240V (4500W electric elements).

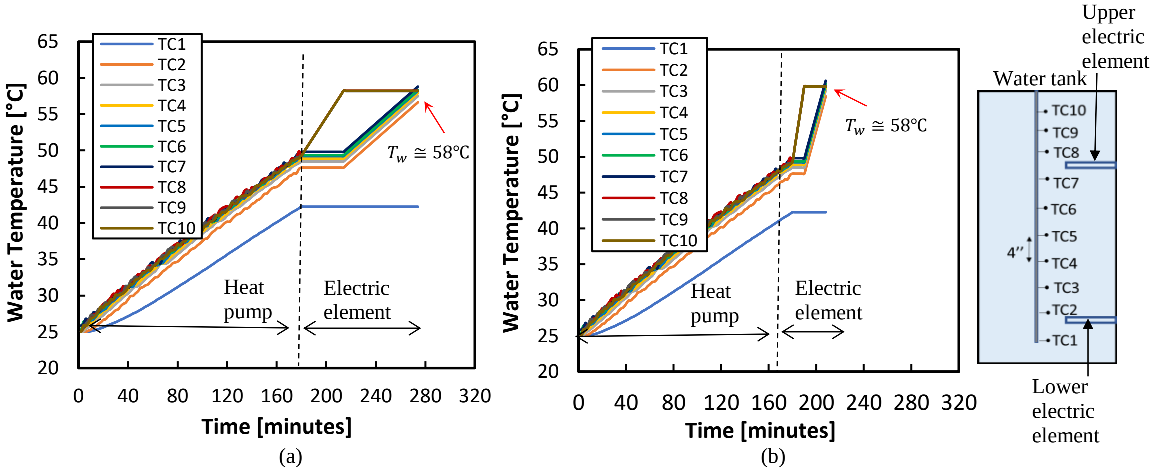



Figure 9. Water temperature vs time for conventional HPWH: a) 1125W electric elements b) 4500 W electric elements

The conventional heat pump water heater system uses the heat pump to heat water until 48℃ and then uses the upper and lower electric heating elements until the water temperature reaches 58℃. The system uses the upper electric element first and then uses the lower electric element. When the upper electric element operates, the water contained in the region between upper element and top wall of tank will heat up due the effect of natural convention. Thus, locations T8, T9 and T10 in Figure 9 show an increase in temperature. The upper electric element is turned off once the water in this region reaches the target temperature. The rate of heat conduction between the water layers in vertical direction is negligible and hence the water contained in the tank below the height of upper element stays at almost the same temperature. Then the lower electric element is turned on to finish heating up the remaining water. When the lower electric element operates, water contained in the tank above the height of lower electric element shows an increase in temperature. Thus, locations T2 to T7 in Figure 9 show an increase in temperature. The natural convection in the tank causes the hot water to remain above the relatively cold water and thus a small volume of water located between lower element and bottom of tank does not show a rise in temperature when the lower element operates. The water temperature vs time for the storage heat pump system was shown in Figure 4. Table 7 shows the comparison between storage heat pump system and conventional heat pump water heater system.

Table 7. Comparison between storage heat pump system and conventional heat pump water heater system

| | | **Conventional HPWH** | | **Storage HP** |
|---|---|---|---|---|
| | | 1125W electric elements | 4500W electric elements | - |
| **Overall energy consumed** | $T_{water}$ from 25℃ to 48℃ (t = 0 to 180 minutes) | 0.9 kWh (Using heat pump) | 0.9 kWh (Using heat pump) | 0.9 kWh (Mode I) |
| | $T_{water}$ from 48℃ to 58℃ (t > 180 minutes) | 1.8 kWh (Using electric coil) | 1.8 kWh (Using electric coil) | 0.53 kWh (Mode II, I) |
| **Time taken for $T_{water}$ to reach from 48℃ to 58℃** | | 96 minutes | 24 minutes | 120 minutes |
| **Hot water available at end of warm-up (Total tank capacity = 50 gallons)** | | 44.6 gallons | 44.6 gallons | 40 gallons |

The performance of storage heat pump system is same as conventional system for water heating from 25℃ to 48℃ (i.e. from t=0 to 180 minutes). The storage heat pump system operates in Mode I until water is heated to 48℃ and then switches to Mode II for the first time to get reduction in pressure ratio and discharge temperature. The conventional system uses heat pump to heat water from 25℃ to 48℃ and then uses the back-up electric elements to heat the water from 48℃ to 58℃.

The storage heat pump system takes additional time of 120 minutes to heat the water from 48℃ to 58℃. The time taken is 25% more compared to conventional system operating on 120V (1125W electric elements) and 5 times more compared to conventional system operating on 240V (4500W electric elements). This is because of the lower heating capacity of vapor compression cycle in the storage heat pump system compared to the heating capacity (power) of the electric elements used by the conventional system. The overall energy consumed by the storage heat pump system is about 70% less than that consumed by the conventional system. For a total tank capacity of 50 gallons, the total hot water available at end of the warm-up is lower than the total tank capacity for all the systems. For the conventional system, the water contained in tank located below the height of lower heating element does not get heated. For the storage heat pump system, the water contained in the tank at a height below which the wrap-around is split into evaporator and condenser is not heated up. The storage heat pump system uses this water as the heat source for evaporator in Mode II.

## 4. Conclusions

System model for the heat pump water heater system operating based on a new storage heat pump cycle is developed. Storage heat pump cycle involves system operation in two modes to reach the required water temperature while using the heat pump compressor instead of the back-up electric elements. Water contained in lower region of tank is used as energy storage element by implementing a split-condenser design. Mode I is same as conventional system operation, while the wrap-around coil is split to use the top portion as condenser and the bottom portion as evaporator in Mode II. The intermediate temperature water in the lower tank region acts as heat source for the evaporator which lifts up the evaporating pressure and reduces pressure ratio upon switching to Mode II.

Model is used to study the effect of wrap-around coil split ratio and compressor speed on system performance in Mode II. The wrap-around coil split ratio determines the size of condenser and evaporator in Mode II which affects the absolute values of condensing and evaporating pressure for the system in Mode II. System performance for four different split ratios is obtained and it is shown that an intermediate split ratio such as Ne/Nc=3/7 which gives medium heating capacity while giving a reduction in the discharge temperature and pressure ratio can be selected for the split condenser design. The split ratio also limits the compressor speed in Mode II as the speed has to be reduced to match compressor capacity with the smaller size of condenser and evaporator in Mode II compared to Mode I.

Modeling results for storage heat pump system with wrap-around coil split ratio Ne/Nc=3/7 and compressor reduced by 50% in Mode II show that the system can reach a higher water end temperature while maintaining relatively lower pressure ratio and discharge temperature than a conventional system. System performance is also compared with conventional heat pump water heater system that uses back-up electric elements to reach the higher water temperature when the heat pump is unable to operate. The additional time taken by the storage heat pump system to heat the water to a higher temperature is 25% more compared to conventional system operating on 120V (1125W electric elements) and 5 times more compared to conventional system operating on 240V (4500W electric elements). However, the energy consumed by the storage heat pump system is about 70% lower than the energy consumed by the electric elements in the conventional system. The power consumption for storage heat pump system is lower as it uses the vapor compression cycle to heat water to higher temperature instead of using electric elements. The storage heat pump concept is to be evaluated experimentally by modifying a conventional heat pump water heater system to implement the system operation in two modes.

## Nomenclature

***Symbols***

| | |
|---|---|
| f | Operating frequency of compressor [revolutions per second] |
| h | specific enthalpy [kJ/kg] |
| htc | heat transfer coefficient [W/m$^2$K] |
| $\dot{m}$ | mass flow rate [kg/s] |
| Nc | number of wrap-around coil turns used as condenser in Mode II |
| Ne | number of wrap-around coil turns used as evaporator in Mode II |
| P | pressure [kPa] |
| $\dot{Q}$ | heat transfer rate/heat exchanger capacity [kW] |
| $q''$ | heat flux [W/m$^2$] |
| SC | subcooling [°C] |
| SH | superheat [°C] |
| T | temperature [°C] |
| $T_{w,lower}$ | bulk average water temperature in lower tank region [°C] (the region between bottom tank wall to height at which coil is split in Mode II) |
| $T_{w,upper}$ | bulk average water temperature in upper tank region [°C] (the region between top tank wall to height at which coil is split in Mode II) |
| $\Delta T_{hx}$ | difference between ref. saturation and external fluid temperature [°C] |
| $\dot{W}$ | power consumption [kW] |

***Subscripts***

| | |
|---|---|
| comp | compressor |
| cond | condenser |
| cpri | comp. inlet |
| cpro | comp. outlet |
| cro | cond. outlet |
| disp | displacement |
| dis | discharge |
| eri | evap. inlet |
| ero | evap. outlet |
| evap | evaporator |
| hx | heat exchanger |
| isen | isentropic |
| ref | refrigerant |
| ri | ref. inlet |
| sat | saturation |
| suc | suction |
| vol | volumetric |
| w | water |

***Greek***

| | |
|---|---|
| $\varepsilon$ | epsilon [-] |
| $\eta$ | efficiency [-] |
| $\rho$ | density [kg/m$^3$] |
| $\Delta$ | difference [-] |

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