258: 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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Outline

- Introduction
  - » Motivation and Objectives
  - » Storage heat pump concept
- System model
  - » Modeling approach for heat pump water heater; Component models
- Experimental facility
- Model validation using experimental results
- Comparing storage heat pump system performance with conventional system
  - » Conventioanl system using back-up electric coil
  - » Conventioanl system in heat pump only mode with variable speed compressor

Project motivation

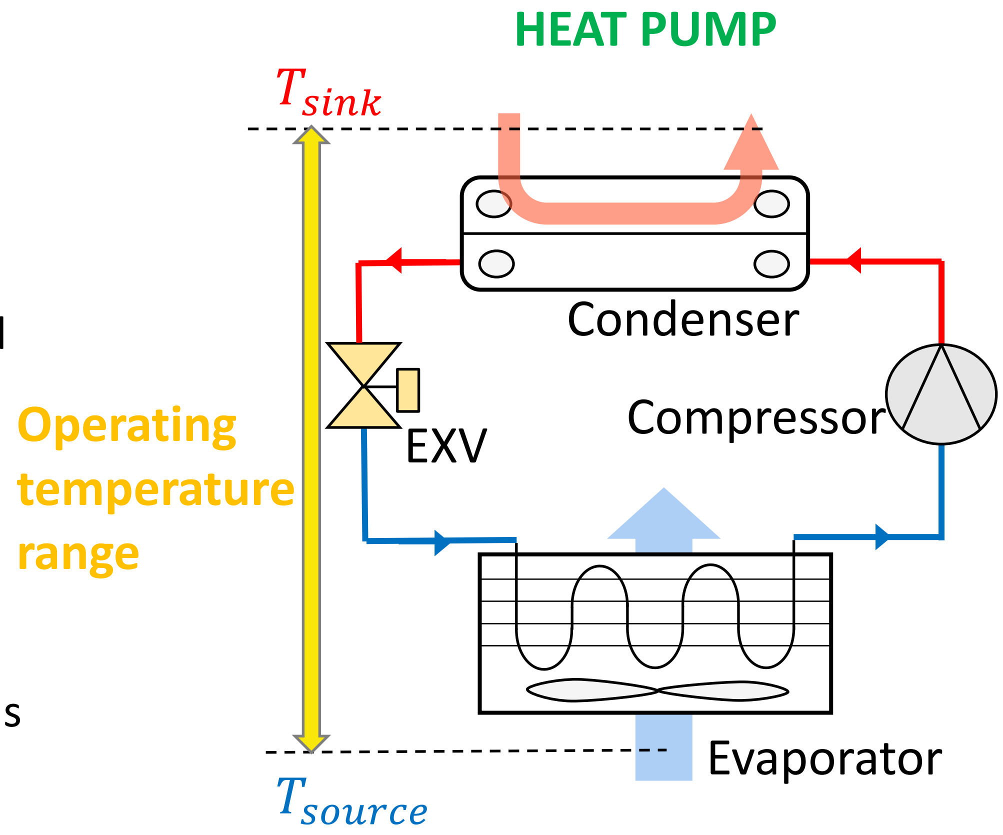
» Heat pump systems support decarbonization goals

» Achievable temperature is often limited by compressor operating envelope

- Example: Cold climate heat pumps face issue of high pressure ratio and discharge temperature

» Mitigation currently achieved through back-heat source (electric or gas)

- Other technologies: Vapor injection scroll compressor, Cascade systems

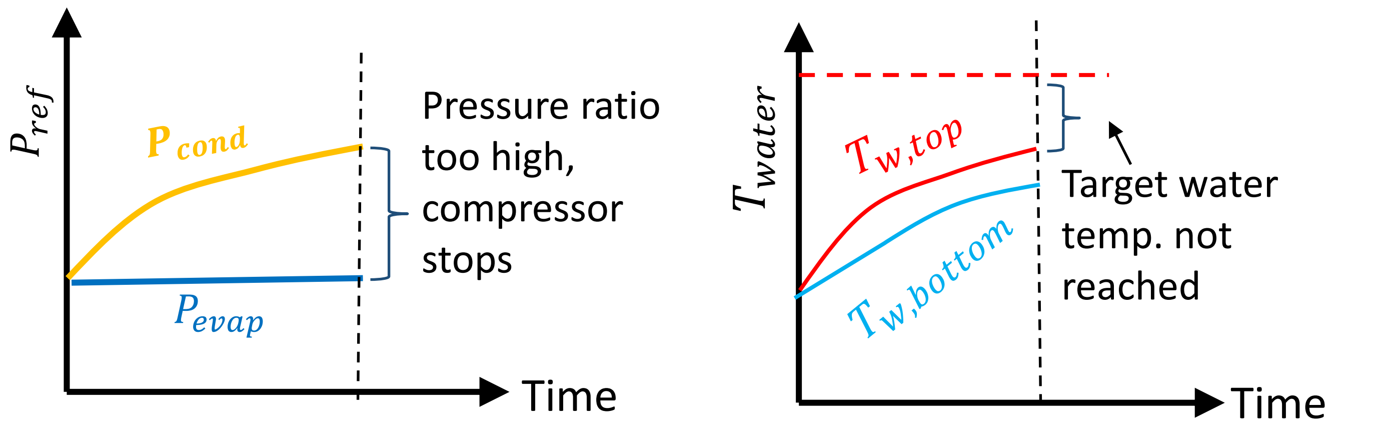


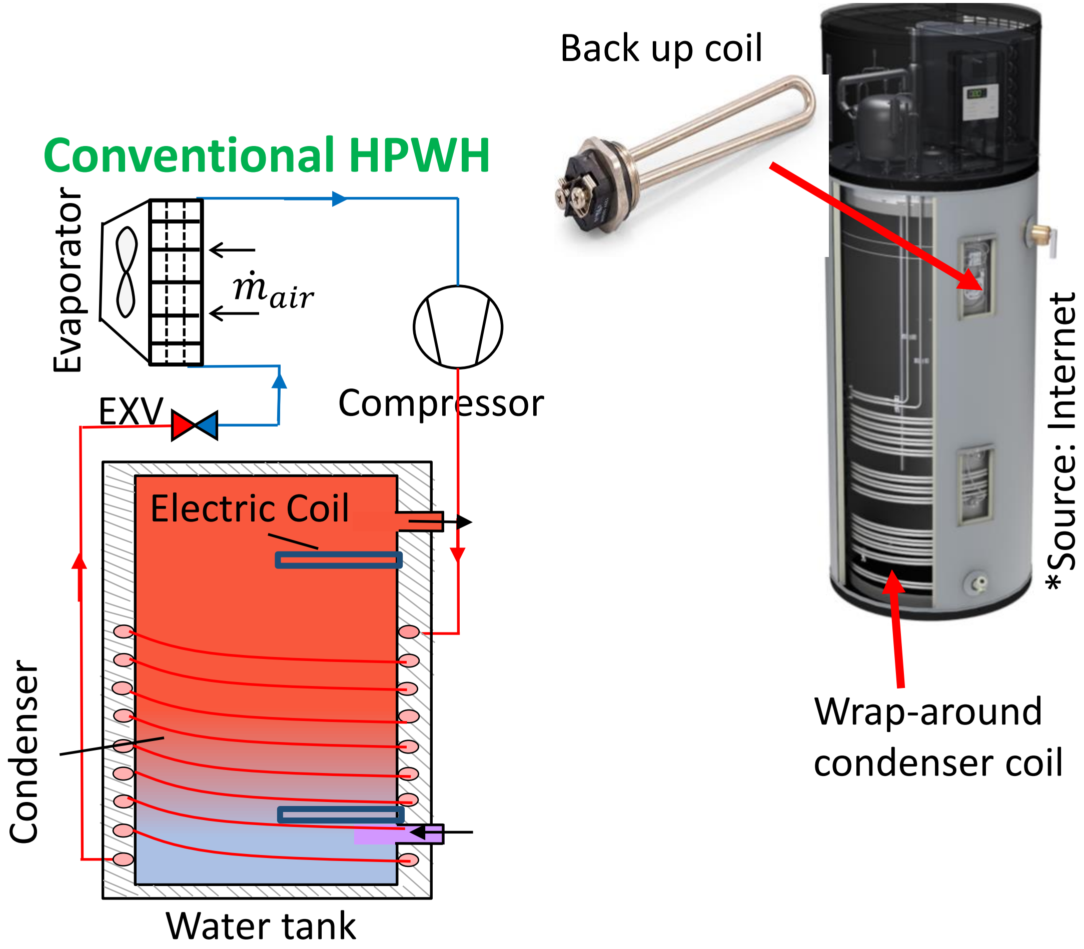
Project objectives

» A new **storage heat pump concept** to increase output temperature of various heat pump systems

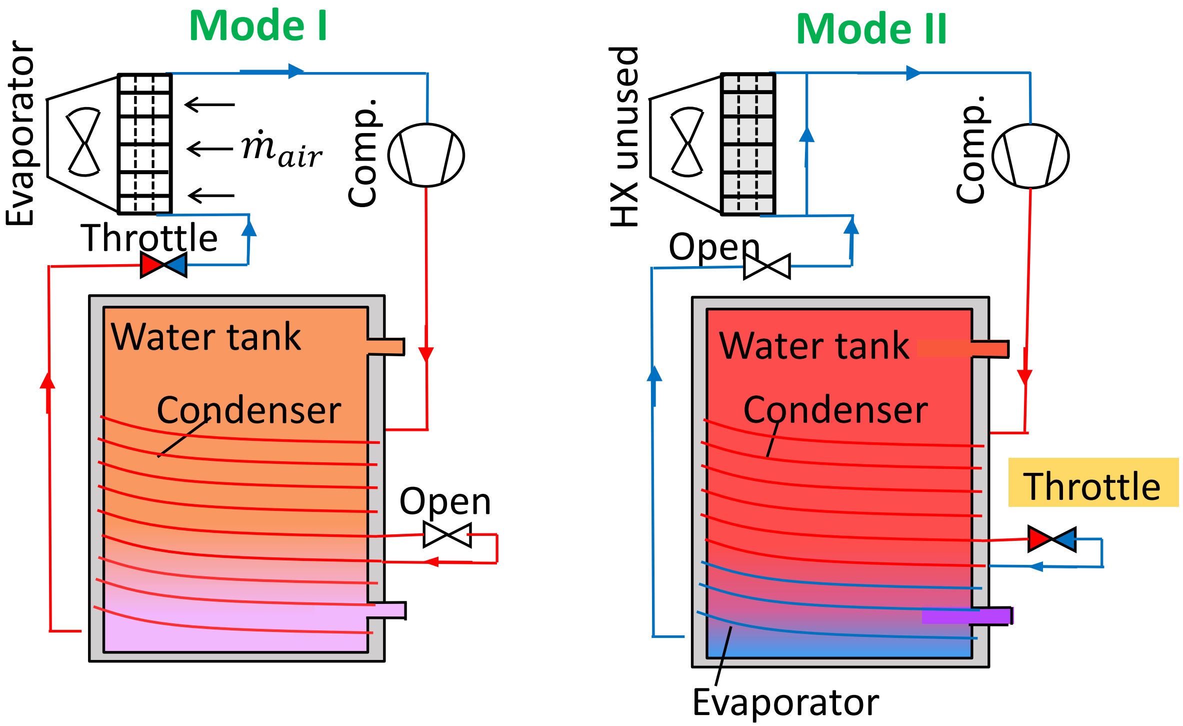
» Implementation of new concept for a Heat Pump Water Heater (HPWH) system

Refrigerant pressure, water temperature profiles



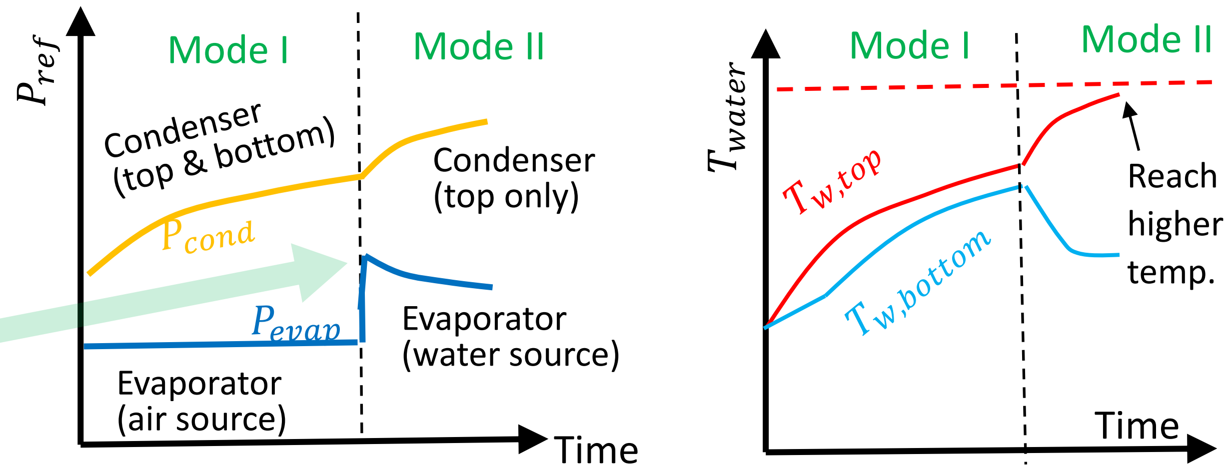


Storage heat pump system



» System operation in two modes to get the high temperature lift
» Mode I: Same as conventional system operation
» Mode II: Throttle valve splits wrap-around coil into condenser and evaporator

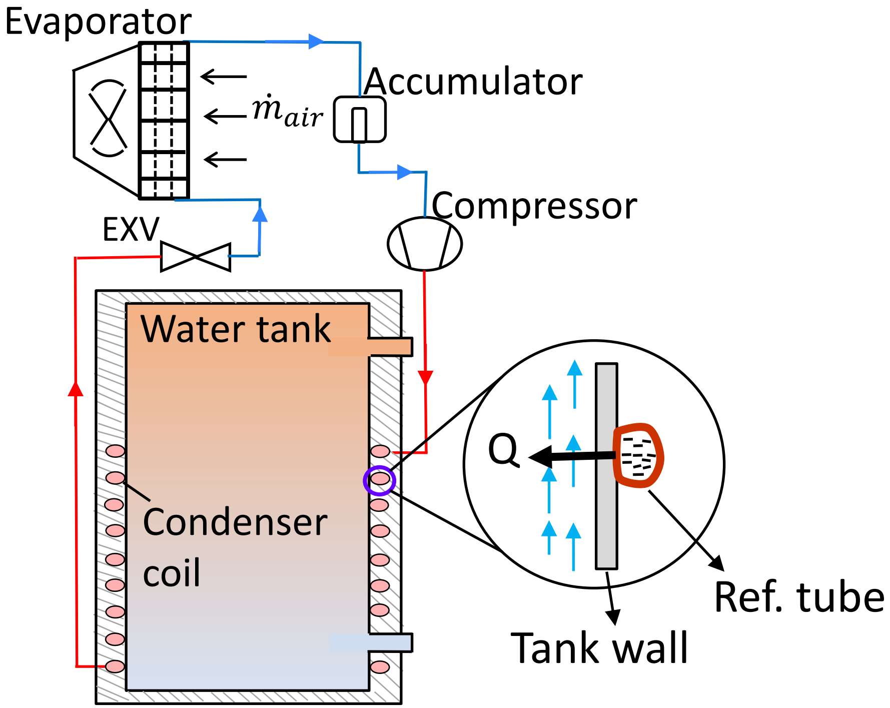
Refrigerant pressure, water temperature profiles



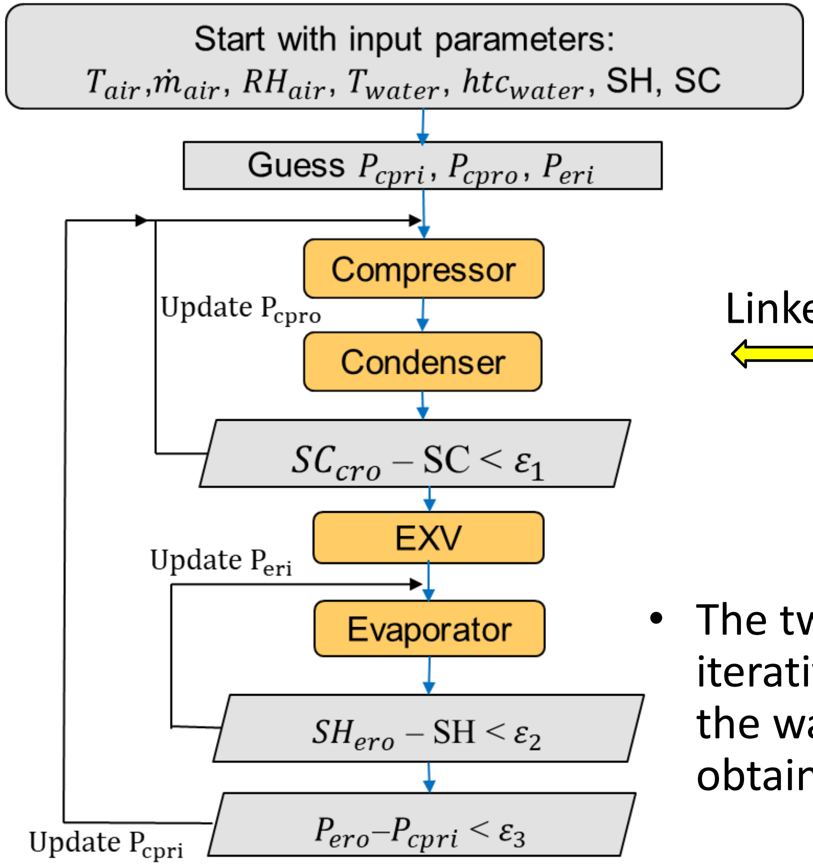
» Switching to Mode II lifts up evaporating pressure (reduces pressure ratio)

Modeling approach for HPWH system

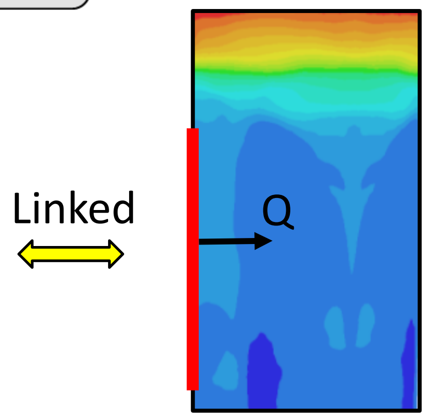
- Water temperature and velocity affect performance of vapor compression cycle.
- Linked modeling approach from Li and Hrnjak (2018)



Quasi-steady cycle model



Transient tank CFD model

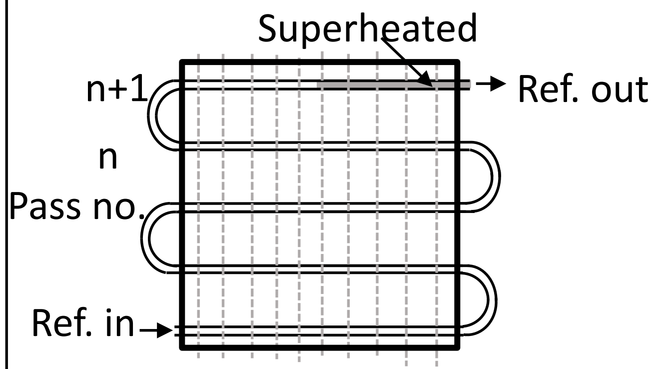


- The two models are solved iteratively until convergence in the wall heat flux profile is obtained

Component models

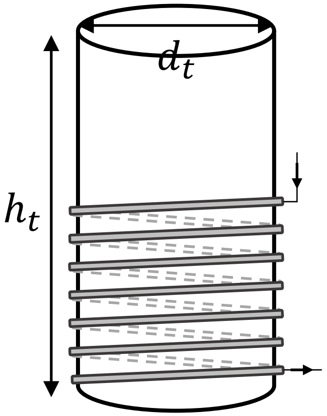
Heat exchanger models
- Finite element discretization
- Heat transfer and pressure drop equations solved for each element sequentially

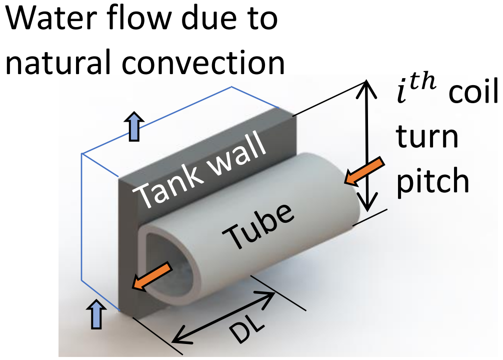
Fin tube HX: Evaporator in Mode I



- 2 rows, 13 passes in each row, air in crossflow

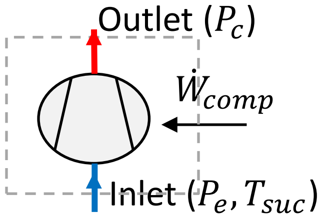
Wrap around coil HX: Split type condenser





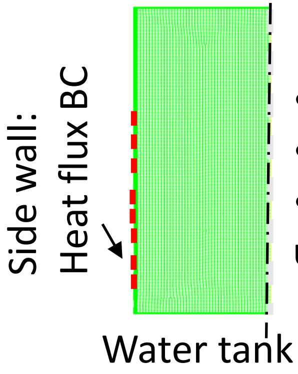
$$\frac{1}{UA} = \frac{1}{(\eta_f hA)_w} + \frac{\delta_{wall}}{(kA)_{wall}} + \frac{1}{(\eta_f hA)_{ref}}$$

Compressor model
- Variable speed, recip. type
- Input: P_e, P_c, SH , Output: h_{dis}, $\dot{W}$, $\dot{m}$



$$\eta_{vol} = f_1(p_c/p_e)$$
$$\eta_{isen} = f_2(p_c/p_e)$$
$$\eta_{com} = f_3(p_c/p_e)$$

Water tank CFD model



- Transient
- 2D-axis symmetric
- Heat flux BC using user defined function

Modeling results: Determining wrap-around coil split ratio (Ne/Nc) and compressor speed in Mode II

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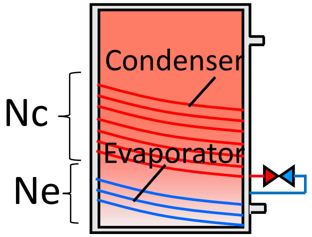
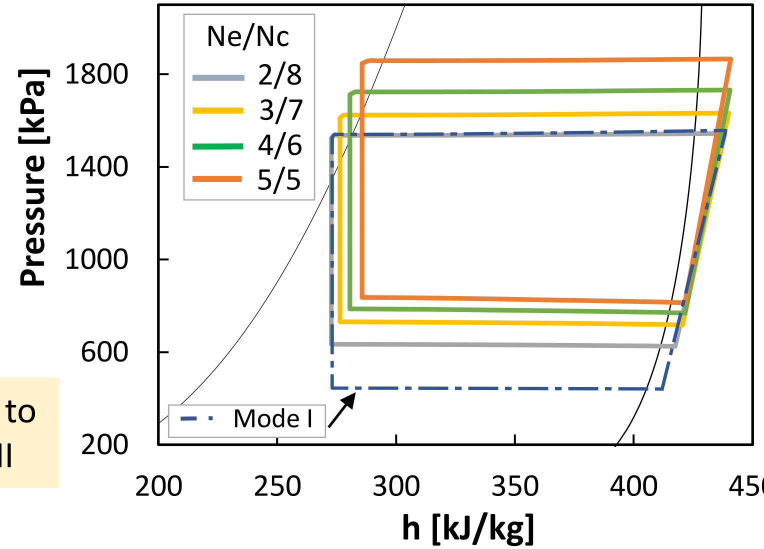
Operating conditions

- $T_{water,initial} = 25°C, T_{ambient} = 27°C$
- Refrigerant: R134a, SH = 10 °C, SC = 10 °C

Mode II

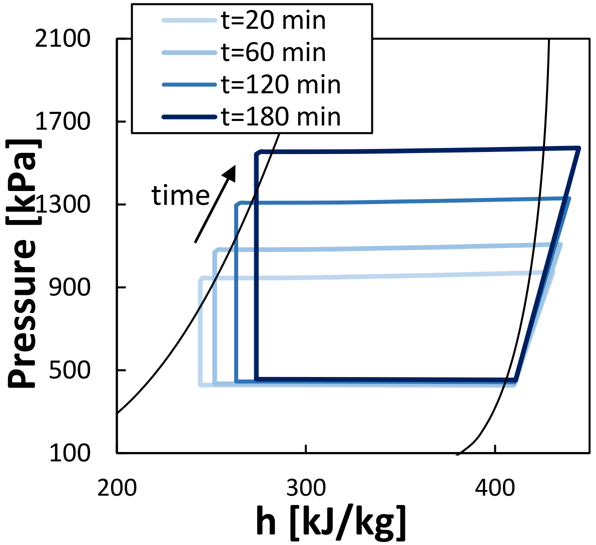


P-h plot upon switching to Mode II

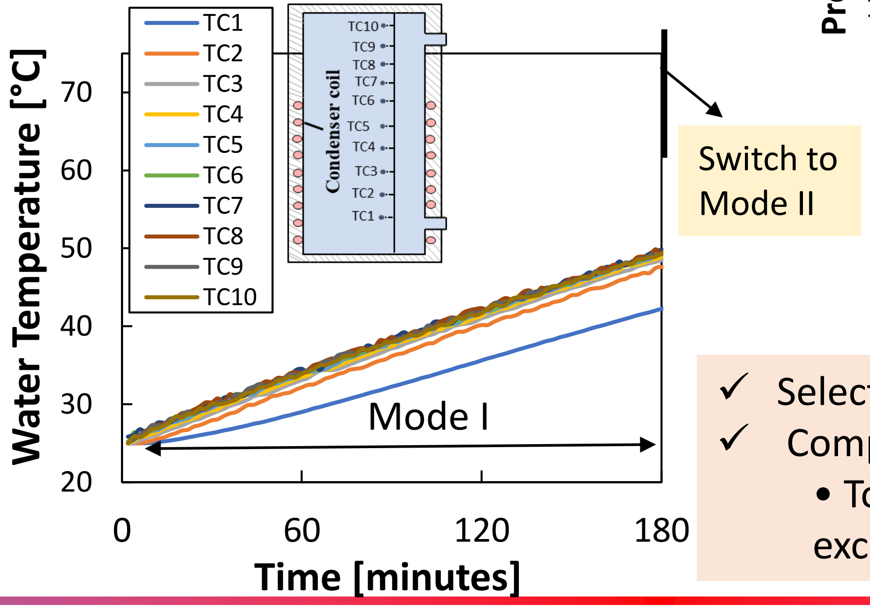


- Higher Ne/Nc
  - -> Greater evap. size
  - -> Smaller cond. Size
  - -> Leads to higher heating capacity
  - -> But a higher P_{cond} and $T_{discharge}$

P-h plot for conventional system operation (Mode I)

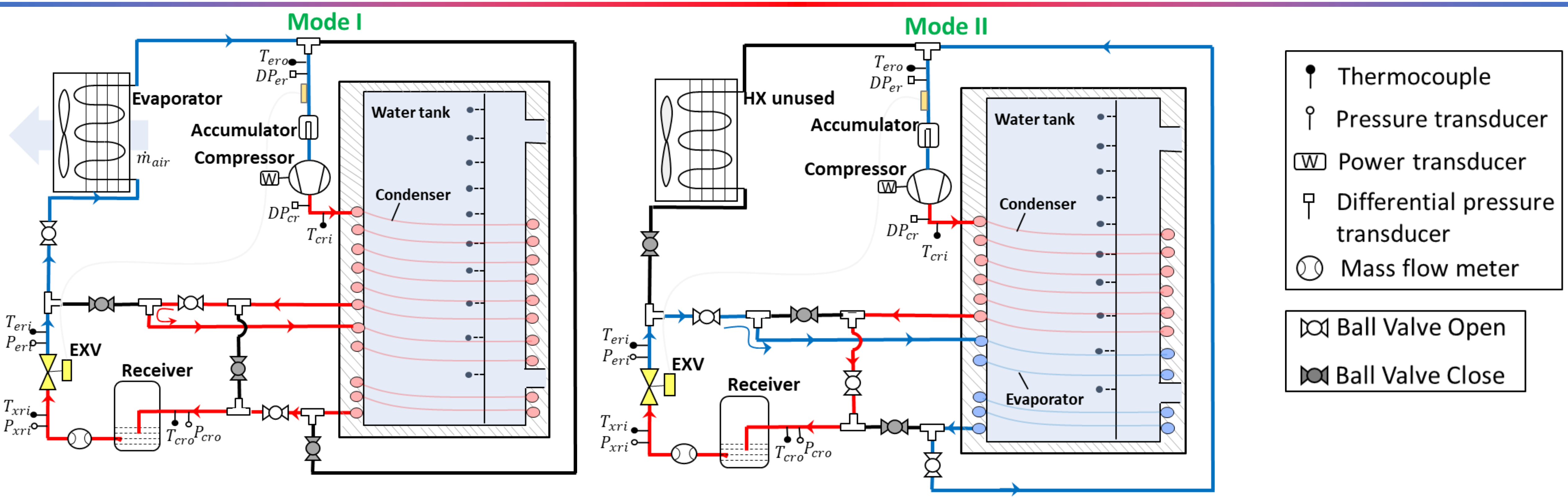


Water temperature vs time



- ✓ Select intermediate split ratio Ne/Nc = 3/7 or 4/6
- ✓ Compressor speed in Mode II lower by 50% compared to Mode I
  - To match compressor capacity with smaller size of heat exchangers in Mode II

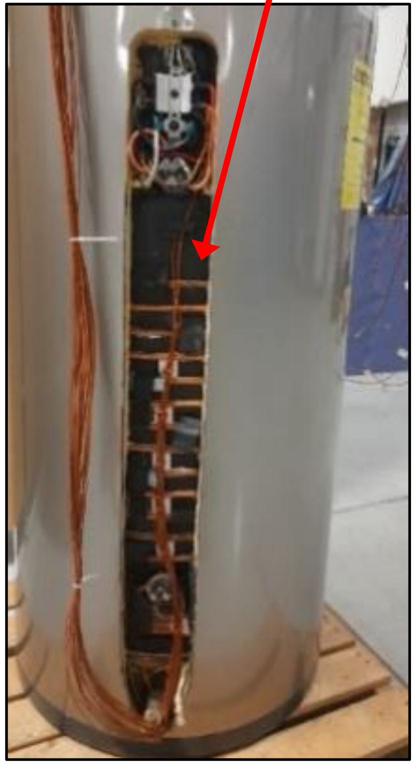
Experimental facility



- » Ball valves used to change refrigerant flow configuration and switch between the modes
- » Receiver installed in liquid line helps to manage the charge variation between Mode I and II

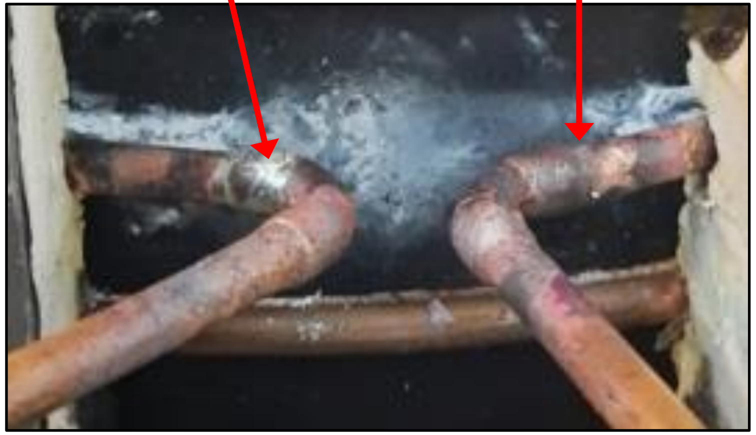
Splitting of wrap-around coil

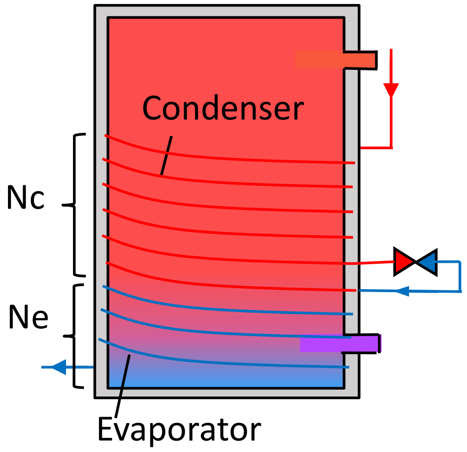
Tank with a section of insulation cut open



Cutting of coil to separate the top and bottom portion

Brazing pipes at cutting point to make connection with system



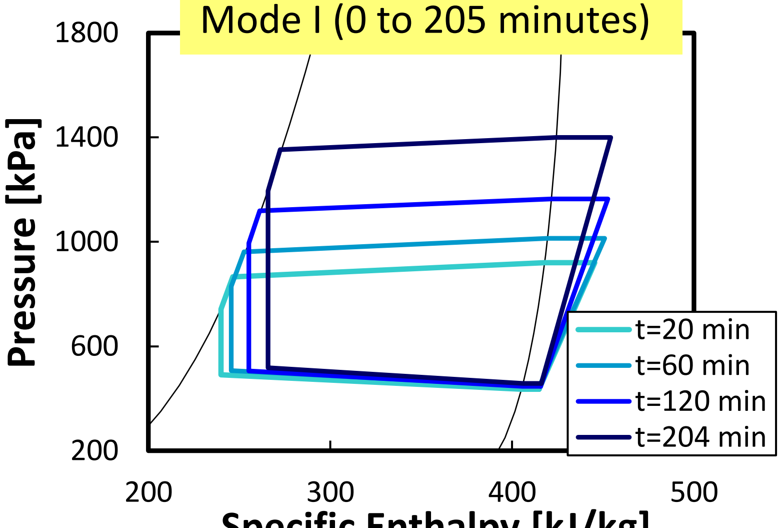


- Wrap-around coil is split at a certain split ratio (Ne/Nc) in Mode II
- Here, $\mathrm{Ne}/\mathrm{Nc} \cong 4/6$ (determined from modeling study)
  - Compressor speed in Mode II is lowered by 50% compared to Mode I

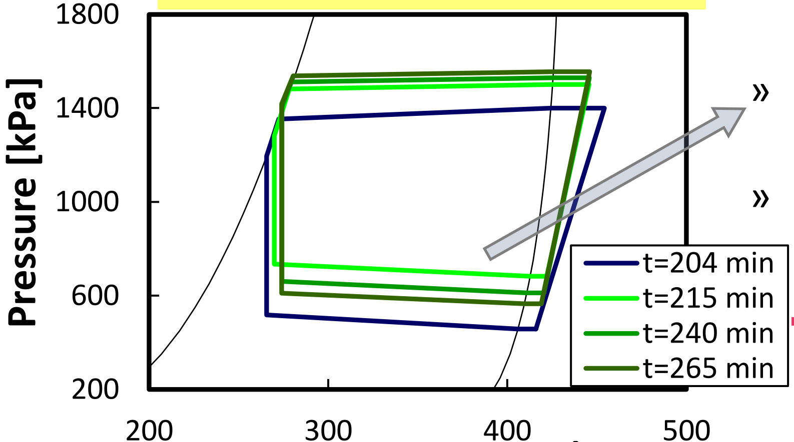
Experimental results

Operating Conditions	
Warm-up test (no water draw)	
$T_{ambient}$	27℃
$T_{water,initial}$	26℃
Split ratio	Ne/Nc=4/6
Compressor speed	Mode I: 4500min^{-1} Mode II: 2250min^{-1}
Time (minutes)	**Mode**
t=0 to 205 (until T_{water} = 46℃)	I
t=205 to 285	II
t=285 to 320	I
t=320 to 380	II
t=380 to 405	I
t=405 to 465	II

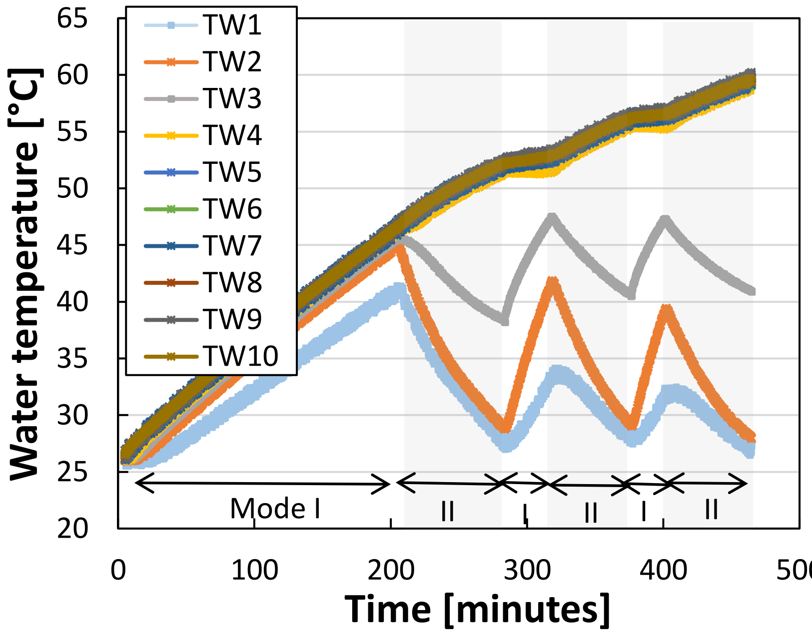
Mode I (0 to 205 minutes)



Mode II (205 to 285 minutes)



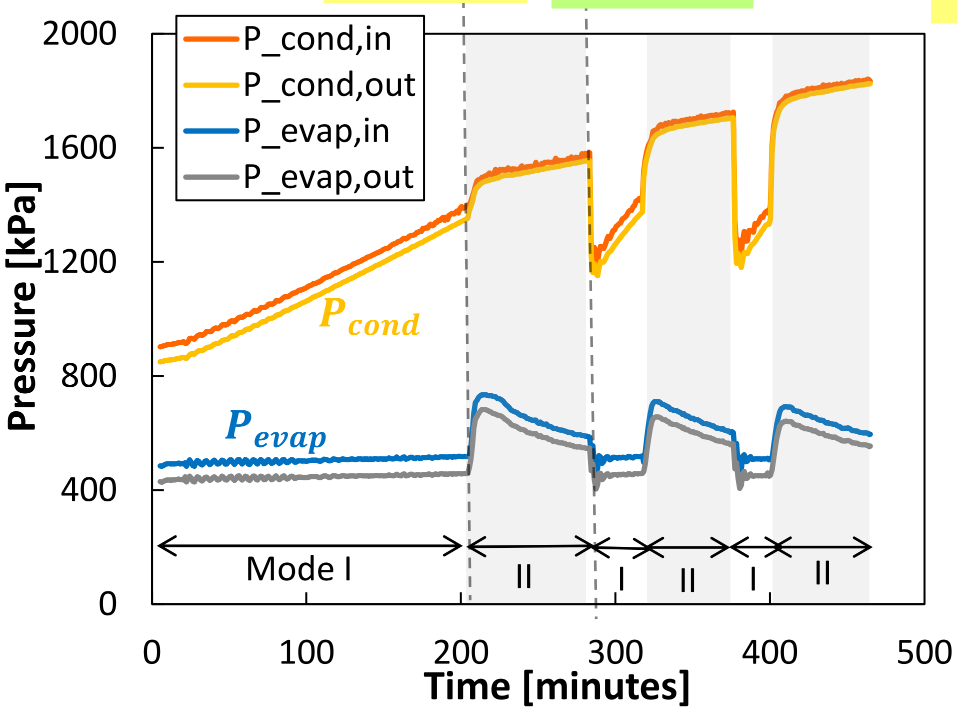
Water temperature vs time



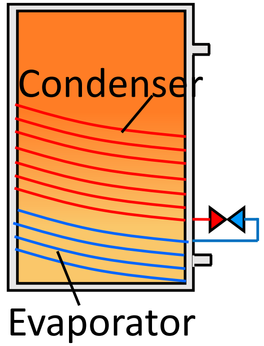
» P_{evap} lifts up when switching to Mode II
- Gives reduction in pressure ratio

» P_{evap} reduces with time in Mode II
- As water in lower tank region gets cooled

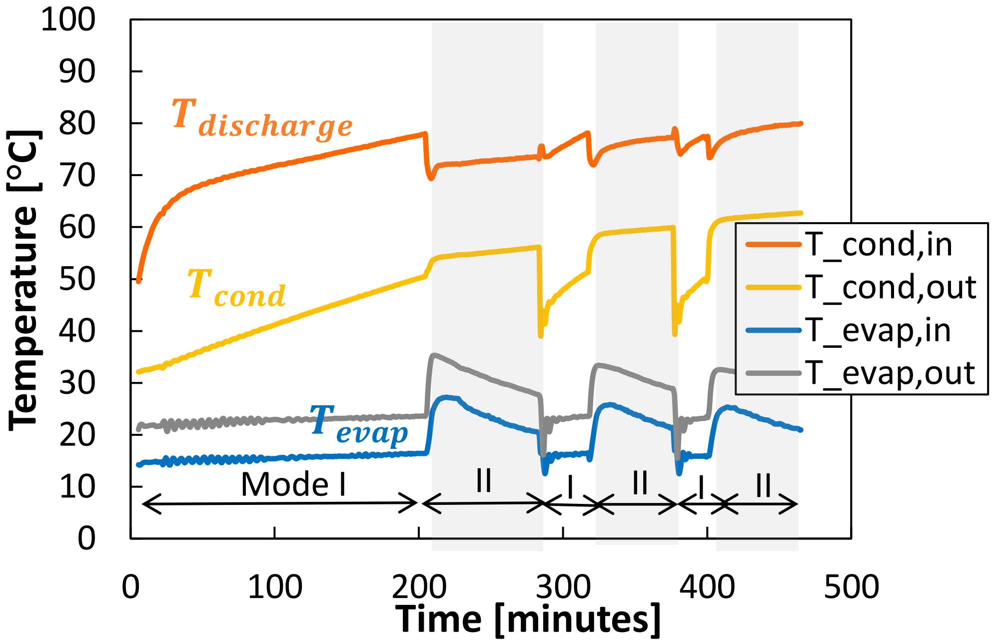
Experimental results: Refrigerant side pressure and temperature



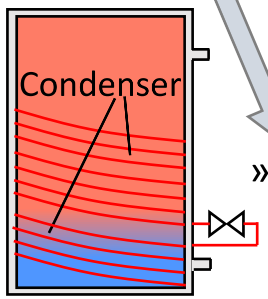
At t=205 minutes (switching to Mode II)



» Water in lower tank region becomes heat source for Evap.
- This lift up P_{evap} , T_{evap}



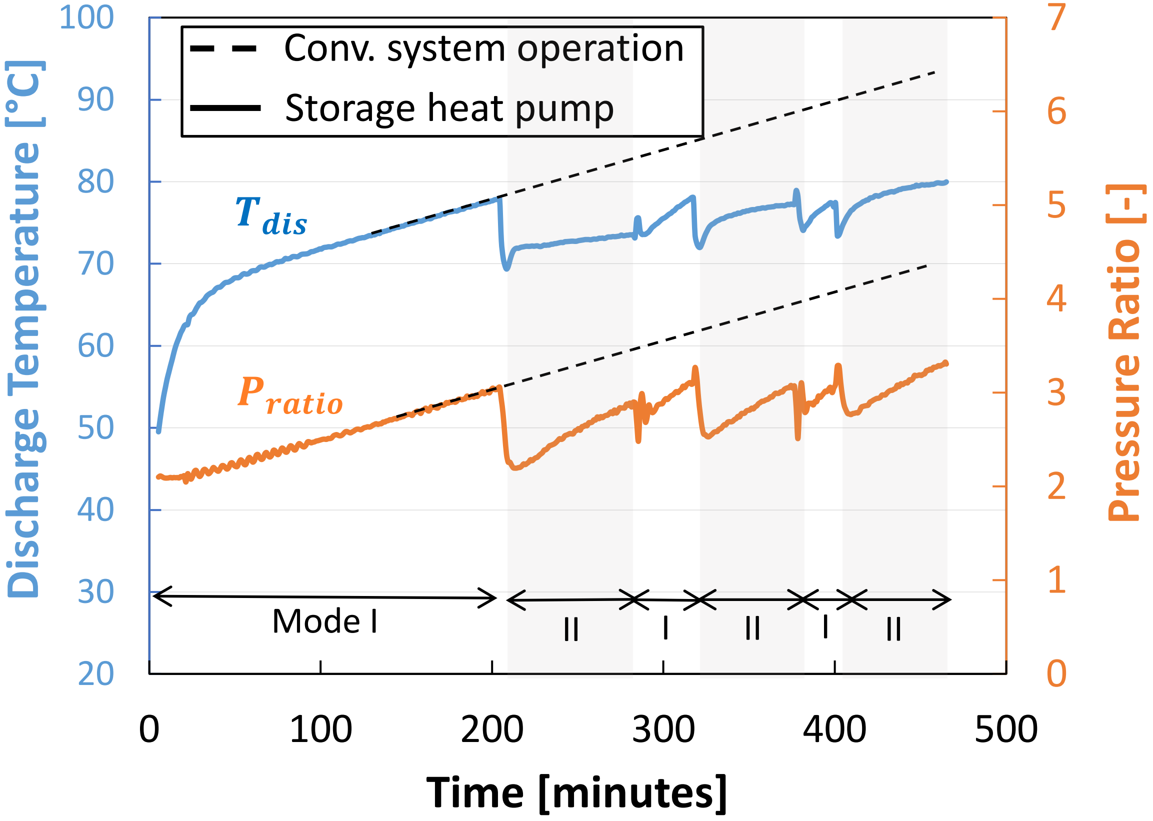
At t=285 minutes (switching back to Mode I)

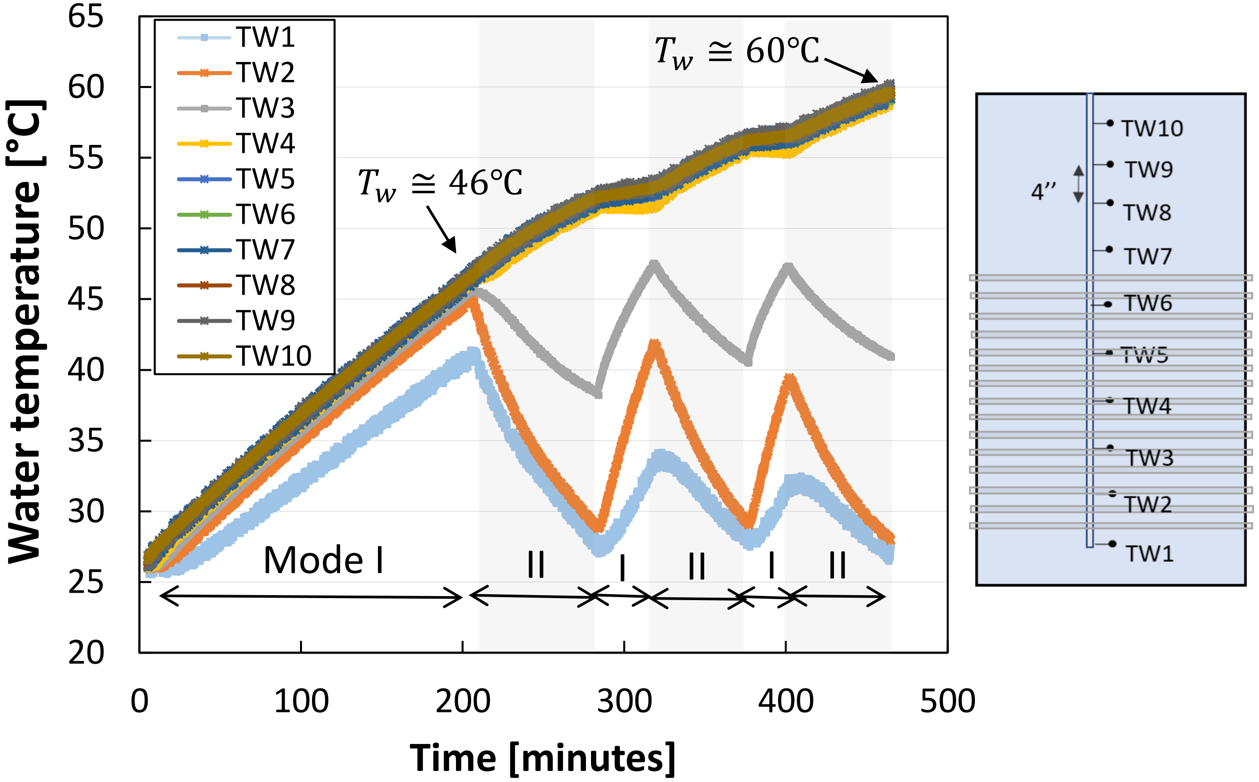


» Water in lower tank region is colder
- This gives a drop in P_{cond}, T_{cond}

*Note: Local fluctuations in the pressure and temperature observed during switching are due to sudden change in refrigerant flow configuration

Experimental results: Discharge temperature, pressure ratio, water temperature

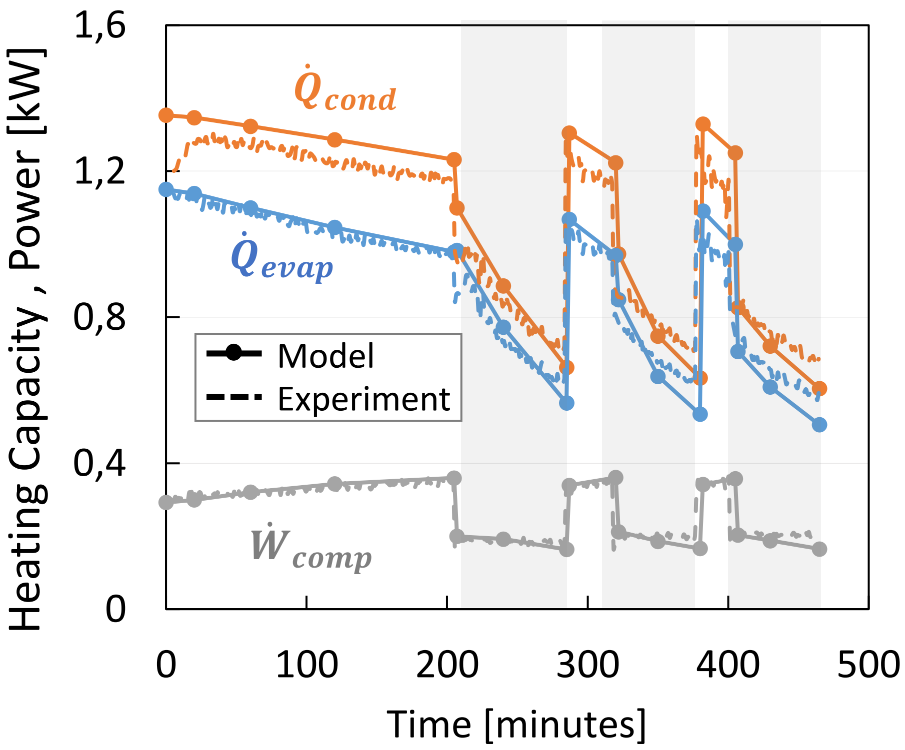


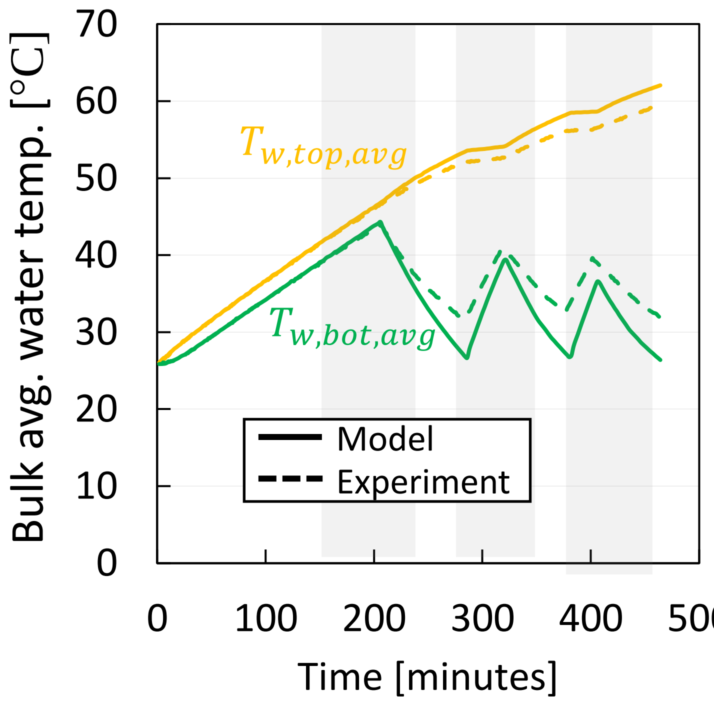


- Water temperature in upper region keeps increasing
- Water temperature in lower tank region (TW1 to TW3) reduces in Mode II
- Conduction within water layers in vertical direction is negligible

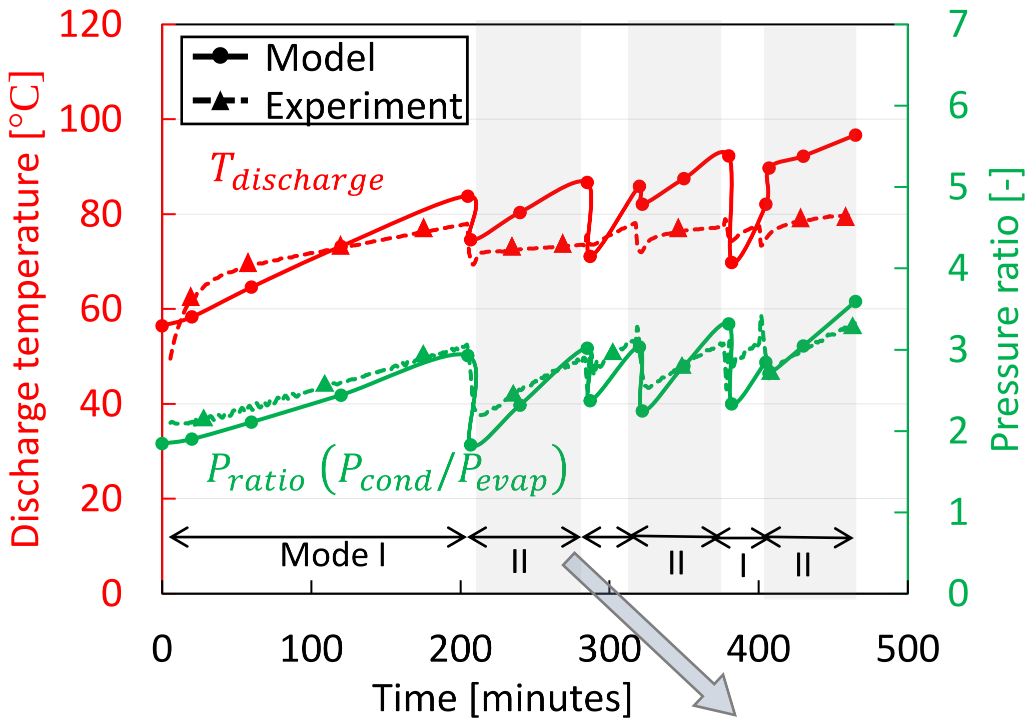
Model vs Experiment: Heating capacity, water temperature

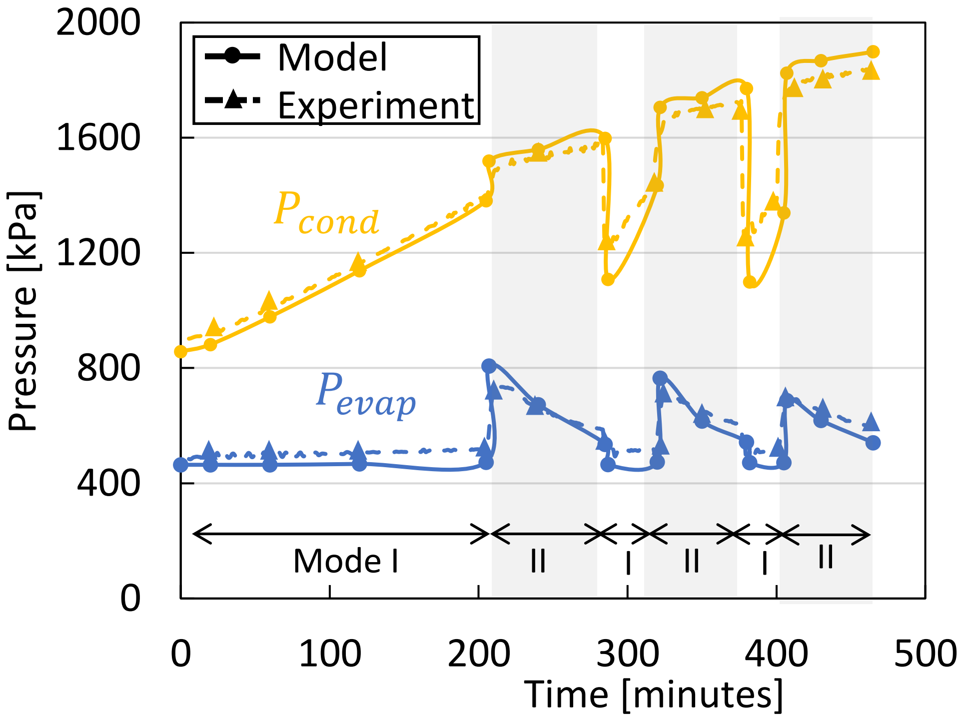
Operating Conditions	
Warm-up test (no water draw)	
$T_{ambient}$	27°C
$T_{water,initial}$	26°C
Split ratio in Mode II	Ne/Nc=4/6
Compressor speed	Mode I: 4500min^{-1} Mode II: 2250min^{-1}
Time (minutes)	**Mode**
t=0 to 205	I
t=205 to 285	II
t=285 to 320	I
t=320 to 380	II
t=380 to 405	I
t=405 to 465	II





Model vs Experiment: Refrigerant side pressure, discharge temperature and pressure ratio

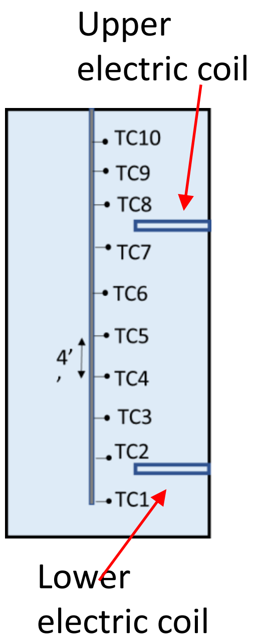


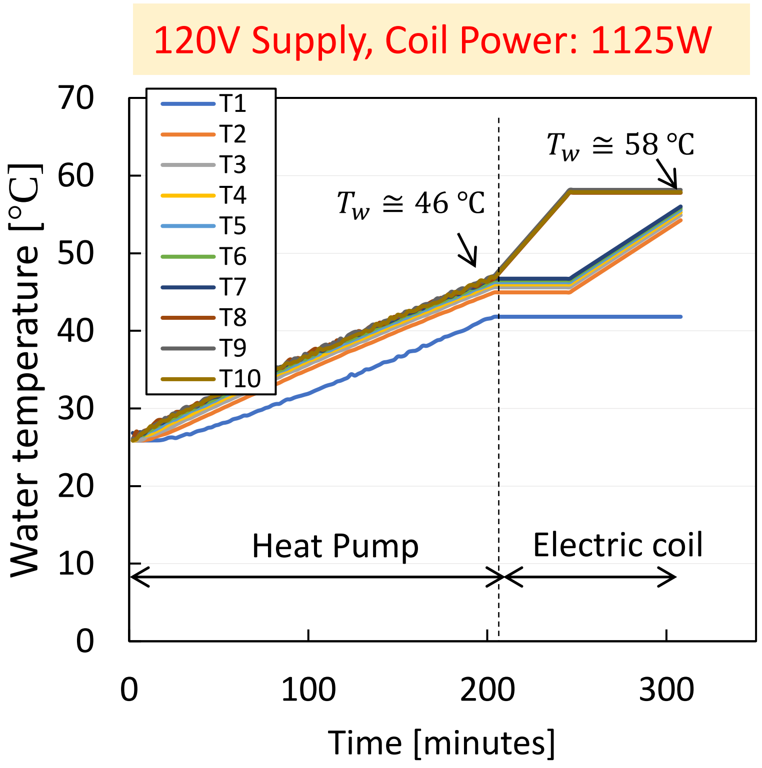


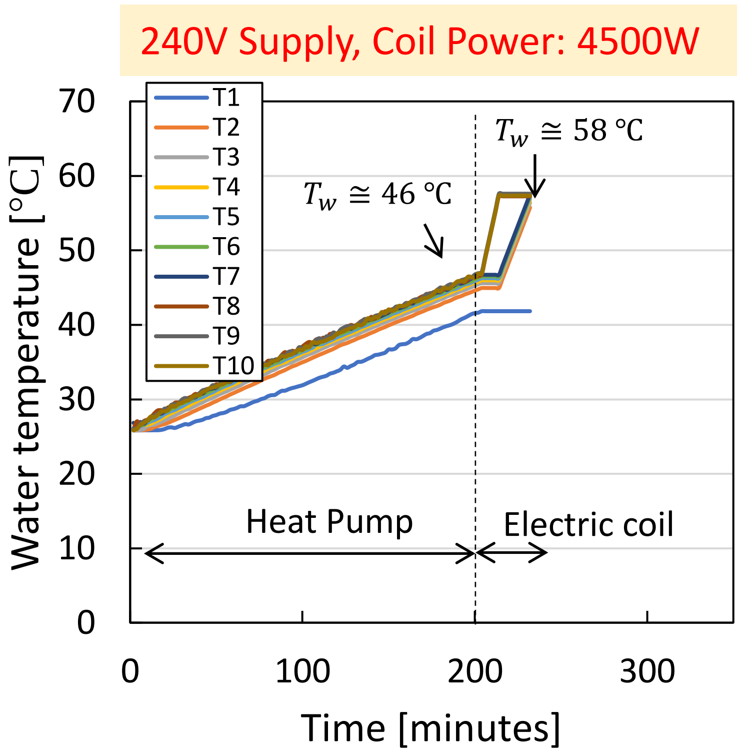
- In Mode II, model predicted a lower compressor isentropic efficiency and consequently a higher discharge temp. than experimental results
- Observed experimentally:
  - The compressor isentropic efficiency increased significantly upon reduction of the compressor speed (this can be compressor specific)

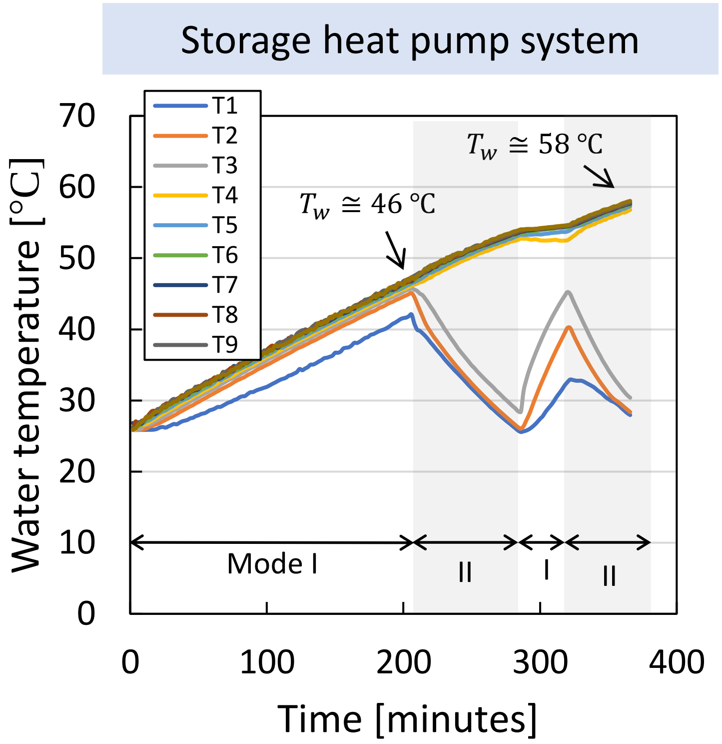
Modeling results: Comparison with conventional system that uses back-up electric heating elements

Conventional system using electric coil to reach the higher water temperature









Modeling results: Comparison with conventional system that uses back-up electric heating elements (cont.)

		Conventional system		Storage HP
		120V Electrical Supply : 1125W electric coil	240V Electrical Supply: 4500W electric coil	-
Energy consumed by heat pump compressor or electric coil	T_{water} **from 26°C to 46°C (t = 0 to 205 minutes)**	1.04 kWh (using heat pump)	1.04 kWh (using heat pump)	1.04 kWh (Mode I)
	T_{water} **from 46°C to 58°C (205 minutes onwards)**	2.25 kWh (using electric coils)	2.25 kWh (using electric coils)	0.59 kWh (Mode II & I)
Time taken for the additional increase in water temperature (T_{water} from 46°C to 58°C)		120 minutes	30 minutes	160 minutes
Hot water available at end of *warm-up* (Total tank capacity = 50 gallon)		50 gallons	50 gallons	40 gallons
Hot water available per unit energy consumed		15.1 gallon/kwh	15.1 gallon/kwh	24.5 gallon/kwh

Modeling results: Comparison with conventional heat pump system using variable speed compressor

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Storage heat pump system

Operating Conditions		
Warm-up test (no water draw)		
Switch between modes using the split type condenser		
Time (minutes)	**Mode**	**Compressor speed**
t=0 to 205 (until T_{water} = 46°C)	I	$4500 min^{-1}$
t=205 to 285	II	$2250 min^{-1}$
t=285 to 320	I	$4500 min^{-1}$
t=320 to 380	II	$2250 min^{-1}$

Conventional system (only heat pump mode) With compressor speed reduction

Operating Conditions		
Warm-up test (no water draw)		
System continuously operates in Mode I		
Time (minutes)	**Mode**	**Compressor speed**
t=0 to 205	I	$4500 min^{-1}$
t=205 to 380	I	$2250 min^{-1}$

» Compressor speed is lowered in Mode II for Storage HP
- To match compressor capacity with smaller size of condenser and evaporator in Mode II compared to Mode I

Modeling results: Comparison with conventional heat pump system at medium ambient temperature ($T_{ambient} = 27°C$)

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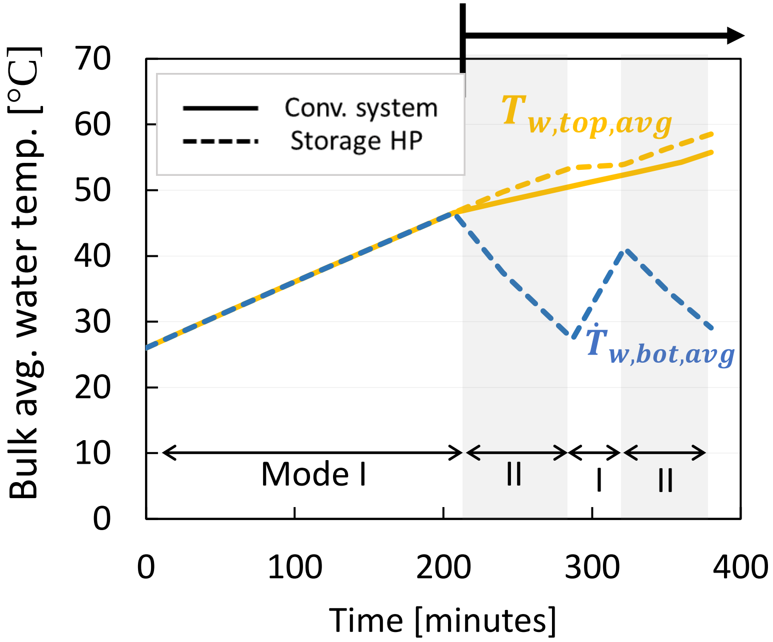
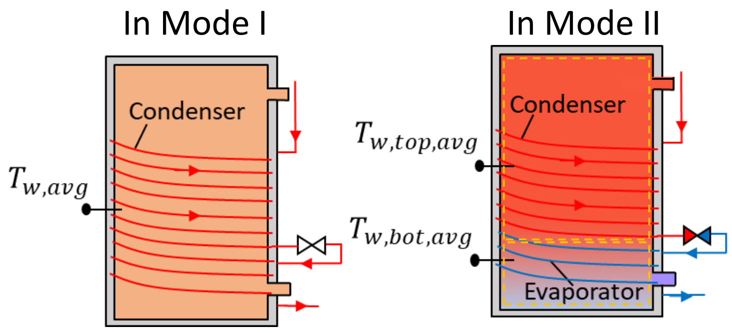
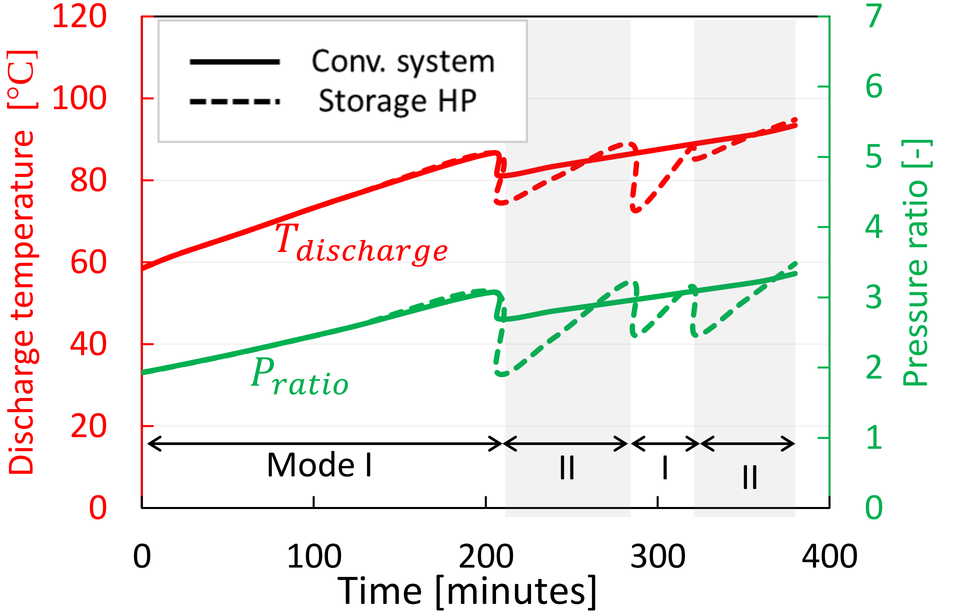
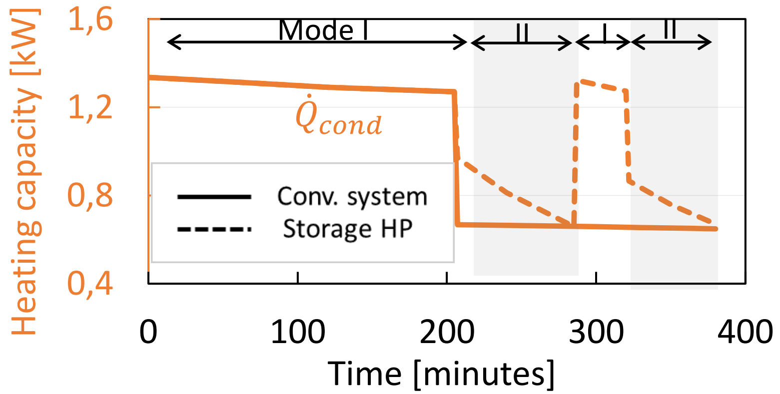


- Storage heat pump system switches to Mode II
- Conv. system continues in Mode I, but compressor speed is reduced

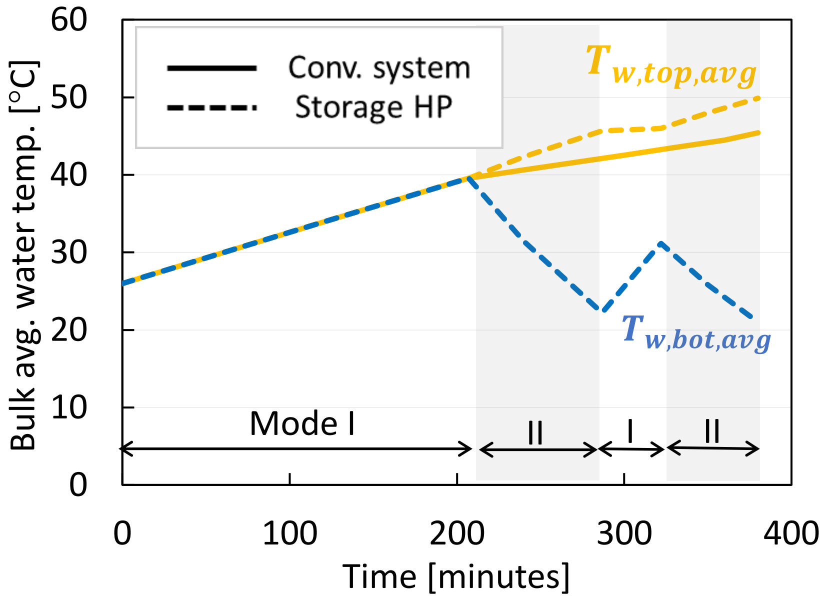


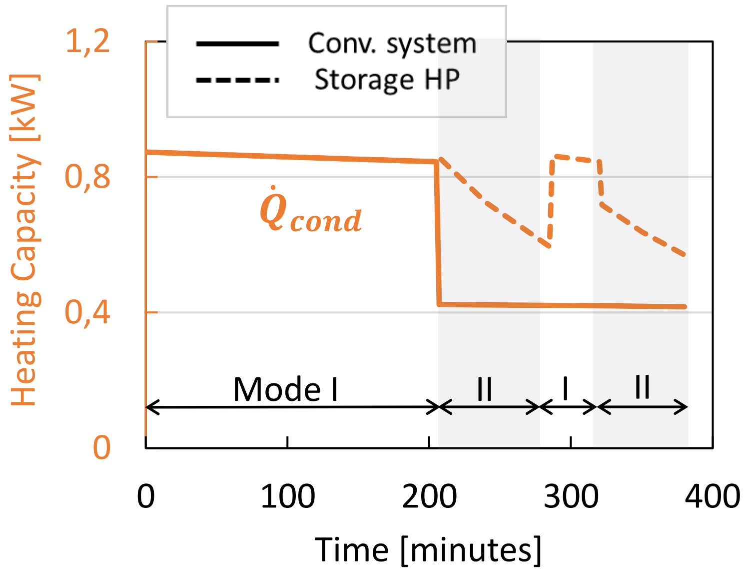


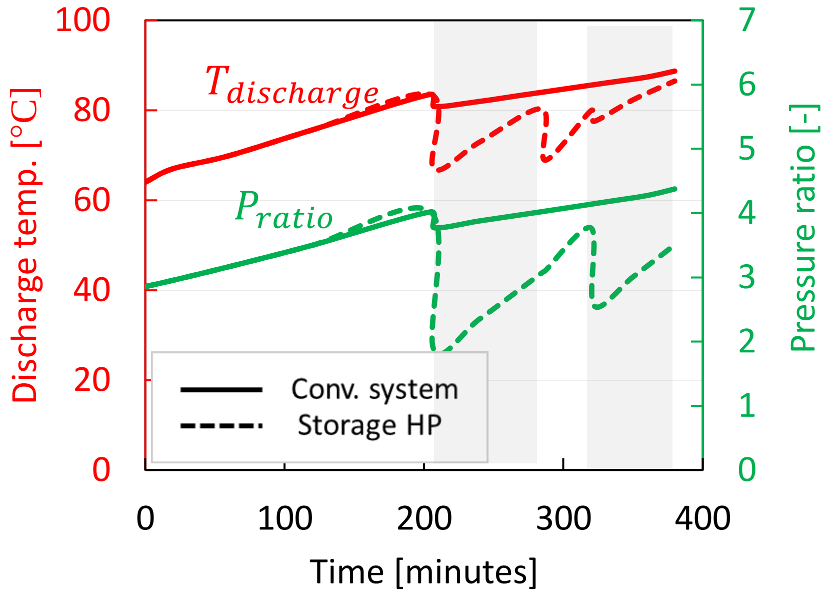


Observations	
$T_{discharge}$ and P_{ratio}	Storage HP gives slightly greater reduction in $T_{discharge}$ and P_{ratio} than conv. system
$T_{w,avg}$ and hot water quantity at end of warm-up	Storage HP : 58.5°C, 40 gallons Conv. system: 55.5°C, 50 gallons
Average $\dot{Q}_{cond}$ after switching to Mode II	Higher by 14% for Storage HP than conv. system

Modeling results: Comparison with conventional heat pump system at lower ambient temperature ($T_{ambient} = 10$℃)







When comparing system performance at lower ambient temperature	
$T_{discharge}$ and P_{ratio}	Storage HP gives much greater reduction in T_{dis} and P_{ratio} than conv. system
$T_{w,avg}$ and hot water quantity at end of warmup	Storage HP : 50°C, 40 gallons , Conv. HP: 44°C, 50 gallons
Average $\dot{Q}_{cond}$ after switching to Mode II	Higher by 75% for Storage HP than conv. system

Conclusions

» Storage heat pump concept is introduced

- **Mode I:** Same as conventional system operation, **Mode II:** Wrap-around coil is split into condenser and evaporator
- Water contained in lower tank region acts as energy storage element

» System model is developed: Wrap-around coil split ratio (Ne/Nc=4/6) and compressor speed in Mode II are obtained

» Experimental facility is built by modifying a conventional HPWH system

- Experimental results are used to validate the model

» Comparison with conventional system that uses electric heating elements

- Storage HP takes longer time to get the additional rise in water temp. (5 times more compared to 240V conv. system (4500W coil) and 35% more compared to 120V conv. system (1125W coil)).
- For the additional rise in water temperature, the storage HP has lower power consumption (70% less than conv. system).

» Comparison with conventional system (heat pump only) that uses variable speed compressor like the storage HP

- $T_{ambient} = 27°\text{C}$: Conventional system can give the required reduction in $T_{discharge}$ and P_{ratio}, however storage HP shows slightly higher heating capacity (14% average) when switching to Mode II.
- $T_{ambient} = 10°\text{C}$: Storage heat pump system gives much greater reduction in $T_{discharge}$ and P_{ratio} and a higher (75% average) heating capacity when switching to Mode II.

Thank you for your attention and questions

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