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International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2014

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Experimental Study of a CO_2 Thermal Battery for Simultaneous Cooling and Heating Applications

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ABSTRACT

This paper presents experimental investigations of the dynamics of a transcritical CO_2 heat pump system with two thermal storages for simultaneous cooling and heating application. The preliminary results of the thermal battery are provided using a small-scale test bed that shows the accelerated penetration of renewable energy sources for building heating and cooling applications. The experimental system consists of a CO_2 heat pump system with a compressor of 3 kW (1.02x104 BTU/hr) cooling capacity and two water tanks. During operation, the compressor and expansion valve are considered quasi-static. Thermal sensors are located in each of the two tanks to monitor the temperature gradient of water along the vertical orientation of the tank which impacts the overall system performance. Experiments are carried out under different water circulation flow rates for both the gas cooler and the evaporator in the heat pump, as well as under various discharge pressure conditions controlled by different charging rates and expansion valve openings. The impacts of water circulation flow rate and valve opening are reported in an effort to find the optimum coefficient-of-performance (COP). The results show that increasing the water inlet temperature in the gas cooler raises the discharge pressure significantly and drops the COP, whereas increasing the water temperature of the evaporator raises the discharge pressure relatively moderately. Although a larger water flow rate enhances the heat exchanger capacity and system COP, a smaller water flow rate seems to be preferable to maintain the thermal profile of the water tanks and to provide a more stable COP. At higher gas cooler water inlet temperature, the COP tends to increase with closing expansion valve. In this particular setup, the best COP is found to be approximately 7.0 at a specific expansion valve opening and at a discharge pressure between 75 and 83 bars (1088 to 1204 psia). The heating COP negatively corresponds to the water temperature at the gas cooler inlet. Experiments suggest the need of a proper control strategy and a matched tank capacity design. Based on these results, a 20% power enhancement may be possible by controlling the hot and cold water flow rates.

1. INTRODUCTION

The refrigeration, air conditioning and heat pumping systems are changing rapidly because of the phase-out of hydro chlorofluorocarbon (HCFC) refrigerants and the anticipated phase-out of hydro fluorocarbon (HFC) refrigerants. Utilization of HCFC refrigerants is prohibited by the Montreal Protocol (1987) and London and Copenhagen amendments (1990, 1992). Legislation has recently been introduced in the European Parliament to start the phase-out of HFC refrigerants. As a consequence, researchers have been focusing on the development of new refrigerants to replace HCFCs and HFCs, as well as utilizing natural refrigerants, such as ammonia and carbon dioxide (CO_2), as refrigerants. CO_2 offers an advantage as a refrigerant due to its high heat transfer capacity, and shows good performance in heat pump applications, such as air source heat pumps (Rieberer et al., 1998) and water source heat pumps (Saikawa et al., 2001). However, Sarkar et al. (2010) shows that the system performance is significantly influenced by ambient temperature. In addition, Byrne et al. (2009) show that high ambient temperatures during the air conditioning season limits the applications of CO_2 in residential systems for year round usage.

Blarke et al. (2012) proposed to use the water source CO_2 heat pump system as a ‘Thermal Battery’ with smart grid option, which has the capability of simultaneous cooling and heating, along with minimizing operational cost and CO_2 emissions. This kind of thermal batteries can be applied to buildings, such as hospitals, hotels, and data centers, which require both heating (for hot water supply or space heating) and cooling (for electrical equipment or space cooling) purposes.

Fornasieri et al. (2008) has proved the necessity of optimized design and control strategy for CO_2 heat pump systems. In addition, Sarkar et al. (2009) presented results showing the influence of the water mass flow rate and water temperature at the inlet of the gas cooler. However, the influence of the finite volume storage and build-in analysis of dynamic coefficient of performance (COP) for water source heat pumps is still unknown.

In this study, the dynamic COP of a thermal battery system was investigated as a function of different gas cooler pressures, water flow rates and expansion valve (ExV) openings.

2. EXPERIMENTAL SETUP

A bread board test setup was built to simulate simultaneous heating and cooling use. Figure 1 and Figure 2 shows a schematic and an actual picture of the experimental setup, respectively. The system consists of a CO_2 heat pump system and two tanks (D61cm*H120cm) filled with water. The heat pump system comprises of the following components: a CO_2 compressor with a cooling capacity of 3 kW and a maximum discharge pressure of 120 bars (1740 psia); plate heat exchangers that are used as gas cooler and evaporator; water pumps that are installed adjacent to the tanks for circulation of water between heat exchangers and tank which can operate at maximum flow rate of 3.78 L/min (1 gpm) with 12 VDC power input; and a manually controlled expansion valve (ExV) that is used for controlling the gas cooler pressure. In addition, pressure transducers (±1%) and K-Type thermocouples (±0.5 ºC) are installed at points 1, 2, 3 and 4 as indicated in Figure 1. Water flow meters (±0.1 L/min) are added after each water pump to provide the transient water flow reading.

In addition to the heat pump system, the setup consists of two 265 Liters thermal storage tanks. Water volumes in the thermal storage tanks are different from each other to maintain appropriate energy balance. The cold tank has larger volume due to smaller target temperature difference. During operation, the system is considered in the quasi steady state. In the hot tank, water is pumped out from the bottom into the gas cooler and flows back into the tank at the top. In the cold tank, water is pumped out from the top of the tank and flows back to the bottom of the tank. This inlet/outlet arrangement helps to maintain the temperature gradients inside the tanks and avoids unwanted vertical mixtures in the tanks. The temperature gradient along the vertical orientation is monitored using five thermocouples (±0.5 ºC) in both the thermal storage tanks. The thermocouples are placed at a distance of 10 cm from each other. A conventional data acquisition system connected to a personal computer is used for collecting data.

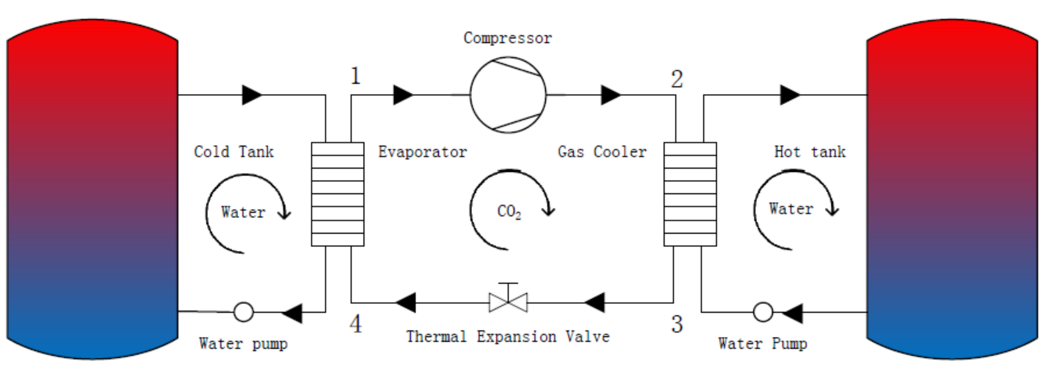


Figure 1: Schematic of Experimental Setup

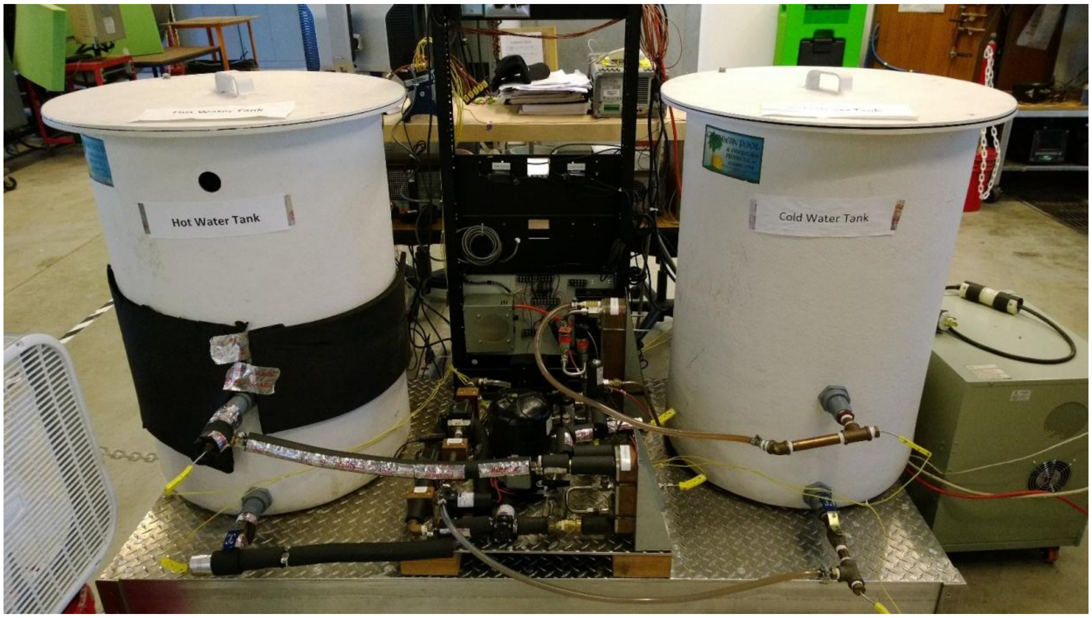


Figure 2: Photo of Experimental System

3. CONTROL PARAMETERS

3.1 Water flow rate

The water flow rates of the hot side and the cold side influence the heat transfer rates in the gas cooler and the evaporator. The water pumps were operated under three flow rates: 1 L/min (0.26 gpm), 2 L/min (0.53 gpm), and 3 L/min (0.79 gpm). It was observed that increasing the pump flow rate requires more pumping power at the same temperature of 24ºC (75.2ºF). During operation, the water temperature was maintained at 24ºC (75.2ºF) up until the system reaches a quasi steady-state. The system took approximately 3 minutes to reach the quasi steady-state.

3.2 Refrigerant charge

In order to find the optimum refrigerant charge, the system was first over-charged. Then, CO_2 was continuously released from the system until the optimal high-side pressure was found. The pressure was initially set to 90 bars (899 psia) with the ExV opening set at 2 ½ turn. The water temperature was set at 24ºC (75.2ºF) in both hot and cold tanks. The water flow rate was chosen to be the minimum flow rate of 1 L/min (0.26 gpm) for both heat exchangers to minimize the distribution of water gradient in the tank. While the compressor was running, CO_2 was released from the suction tube. During the release, the discharge pressure reduced at a rate of 15 bars (217.6 psia) per hour and simultaneously data were collected. This process took approximately 15 minutes until the optimal pressure was determined.

3.3 ExV opening

The ExV opening determines the refrigerant quantity that enters the evaporator as well as the superheat. In this experiment, the ExV opening is set by turning counts, where 0 turn is the closed position. The ExV openings are tested at 1, 1 ¼, 1 ¾, 2 ½, 4, and 5 turns. During the experiment, the ExV was kept in the same opening position until the system reached the quasi steady-state.

3.4 Optimum system performance

In order to achieve optimum performance, two different strategies were investigated. The first control strategy was to use a larger water flow rate to enhance the heat exchanger capacity, and the second control strategy was to use a smaller water flow rate to maintain the thermal profile in the water tanks. The performance was determined by the time required to reach an average hot water tank temperature of 50ºC (122ºF). The quicker the average hot water temperature is reached, the better is the heat pump performance.

4. RESULTS AND DISCUSSION

The combined system COP is determined by using Equation 1:

$$COP = \frac{\dot{Q}_{evap}+\dot{Q}_{gas}}{\dot{W}_{comp}+\dot{W}_{pimp}} \tag{1}$$

Where the cooling capacity, $\dot{Q}_{evap}$ is given by Equation 2:

$$\dot{Q}_{evap} = \dot{m}_R(h_1 - h_4) \tag{2}$$

And the heating capacity, $\dot{Q}_{gas}$ is defined by Equation 3:

$$\dot{Q}_{gas} = \dot{m}_R(h_2 - h_3) \tag{3}$$

4.1 Impact of water flow rate

Two sets of experimental results are presented in Table 1 and Table 2. Table 1 shows results for ExV opening of 1¼ and Table 2 shows results for ExV opening of 2 ½. It is evident from both tables that increasing the hot water flow rate leads to a decrease in discharge and suction pressures. For instance, in Table 1, when the hot water flow rate increases from 1 L/min (0.26 gpm) to 3 L/min (0.79 gpm) while cold water flow rate is kept constant at 2 L/min (0.53 gpm), the discharge pressure decreases from 92.68 bar (1344.2 psia) to 84.49 bar (1225.4 psi) and the suction pressure decreases from 38.48 bar (558.1 psia) to 36.72 bar (532.6 psia). On the other hand, the increase of the cold water flow rate shows an opposite trend. For instance, in Table 1, when the cold water flow rate increases from 1 L/min (0.26 gpm) to 3 L/min (0.79 gpm) while hot water flow rate is kept constant at 1 L/min (0.26 gpm), the discharge pressure increases from 84.95 bar to 95.63 bar and the suction pressure increases from 33.84 bar to 41.13 bar.

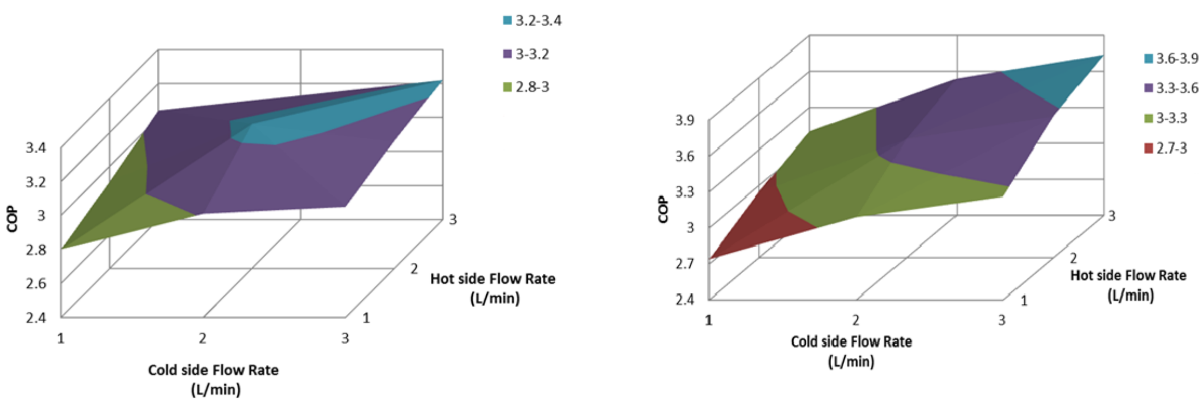
However, the overall system COP does not change with the increase of water flow rate. Table 1 shows that the COP drops from 5.68 to 5.56 when the hot water flow rate is increased from 2 L/min to 3 L/min. Figures 3(a) and 3(b) presents a three-dimensional plot of the COP as a function of the cold-side water flow rate and the hot-side water flow rate. Figure 3(a) shows that the peak value in COP is achieved when both hot water and cold water flow rates are 2 L/min (0.53 gpm). Figure 3(b) shows the results of the ExV opening at 2 ½. The COP peak trends to shift to a larger water flow rate, where both hot-side and cold-side water flow rates are 3 L/min (0.79 gpm). Thus, there is a maximum value for the COP where the capacity of the heat exchanger reaches a maximum limit for a certain ExV opening and a certain flow rate. After the maximum value, the COP gradually decreases as the water flow rates increases.

Table 1: Impact of water flow rate at ExV 1 ¼ opening

V_{cold}=1 L/min			
V_{hot} [L/min]	1	2	3
P_{dis} [bar]	84.95	78.46	78.59
P_{suc} [bar]	33.84	32.89	32.88
COP_{total}	4.66	5.16	5.07
V_{cold}=2 L/min			
V_{hot} [L/min]	1	2	3
P_{dis} [bar]	92.68	86.52	84.49
P_{suc} [bar]	38.48	38.29	36.72
COP_{total}	5.14	5.68	5.56
V_{cold}=3 L/min			
V_{hot} [L/min]	1	2	3
P_{dis} [bar]	95.63	89.06	87.63
P_{suc} [bar]	41.13	38.72	39.01
COP_{total}	5.34	5.56	5.63

Table 2: Impact of water flow at ExV 2 ½ opening

V_{cold}=1 L/min			
V_{hot} [L/min]	1	2	3
P_{dis} [bar]	82.49	75.33	73.21
P_{suc} [bar]	35.78	34.16	34.45
COP_{total}	4.58	5.19	5.5
V_{cold}=2 L/min			
V_{hot} [L/min]	1	2	3
P_{dis} [bar]	86.35	77.51	74.47
P_{suc} [bar]	41.17	38.13	39.01
COP_{total}	5.15	5.96	6.36
V_{cold}=3 L/min			
V_{hot} [L/min]	1	2	3
P_{dis} [bar]	88.15	78.54	78.11
P_{suc} [bar]	43.77	40.68	40.84
COP_{total}	5.42	6.35	6.59



(a) (b)

Figure 3: (a) COP vs hot and cold water flow rates, ExV at 1 ¼ opening
(b) COP vs hot and cold water flow rates, ExV at 2 ½ opening

4.2 Refrigerant Charge

The experimental results of the test while changing the refrigerant charge are presented in this section. Based on previous experimental results, this test was conducted at an ExV opening of 2 ½, water flow rates for both sides of 3 L/min (0.79 gpm), and initial tank water temperatures of 23ºC (73.4ºF) for both tanks. The results presented in Figure 4 shows the discharge pressure as a function of time. Refrigerant is released from the system while the compressor is running to find the optimum charge. As the refrigerant is released, both the discharge and suction pressures decrease. The transient COP starts to increase and reaches a maximum value at a discharge pressure of approximately 83 bars (1200 psia). A further release of refrigerant results in a decrease in the COP. Thus, the optimum charge was found at a discharge pressure of 82.5 bar (1196 psia), which is highlighted in Figure 4.

4.3 Impact of ExV opening

The impact of the ExV opening was investigated with two different temperature conditions. The first set of readings is collected at the cold initial condition. In the cold initial condition, the tank water temperature was set to 23 ºC (73.4ºF) for both hot and cold inlets. The results are shown in Figure 5. The ExV position starts from an opening of 1 turn. While the ExV is opened, the COP value reaches a maximum value of 6 at the ExV opening of 2 ½ turns. At 4 turns of the ExV opening and further, the COP drops significantly. The test result shows that the best COP value makes a peak in between the ExV opening of 1 ¾ opening turns and 2 ½ opening turns.

The next set of readings was collected after the experiments have been run for some time and the two tank temperatures differ substantially. Figure 6 represents the results when the hot water inlet is 47ºC (116.6ºF) and cold water inlet is 10ºC (50ºF). The COP shows a relatively flat trend from 1 to 1 ¾ opening turns of the ExV, and then it begins to drop at 2 ½ opening turns of the ExV. Compared to the initial temperature experiment, the COP is lower during the second test and the optimum COP moves to the smaller ExV opening.

Figure 7 shows the cycle state points in a T-S diagram for the cold initial condition experiment. It can be seen from the figure that as the expansion valve opening increases, the cooling capacity decreases and so does the compressor work. However, after an ExV opening of 4, the decrease in cooling capacity is much greater than the decrease in the compressor work. This makes the system inefficient beyond an ExV opening of 4.

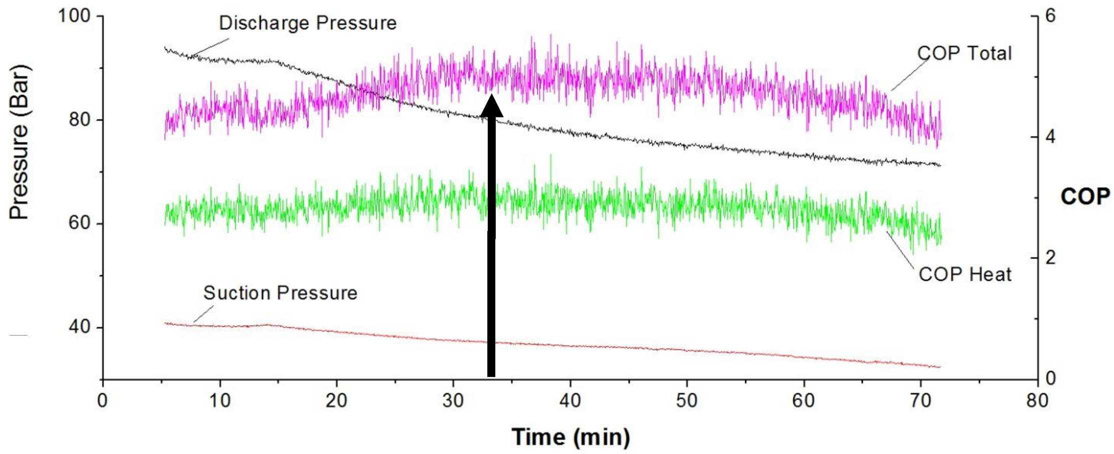


Figure 4: Impact of pressure charge on COP

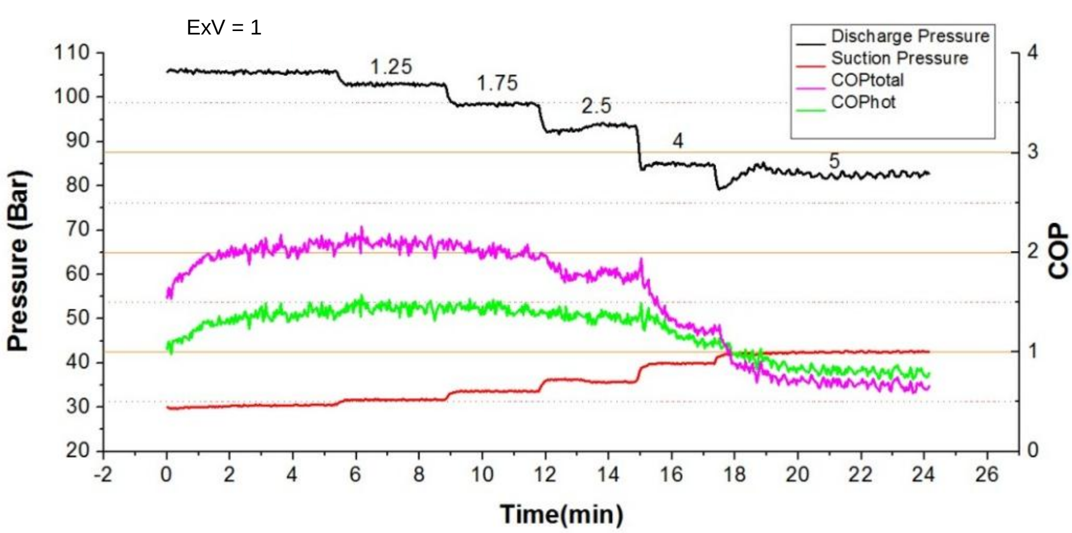


Figure 5: ExV stage opening at 23ºC (73.4ºF) for both cold and hot storage tank

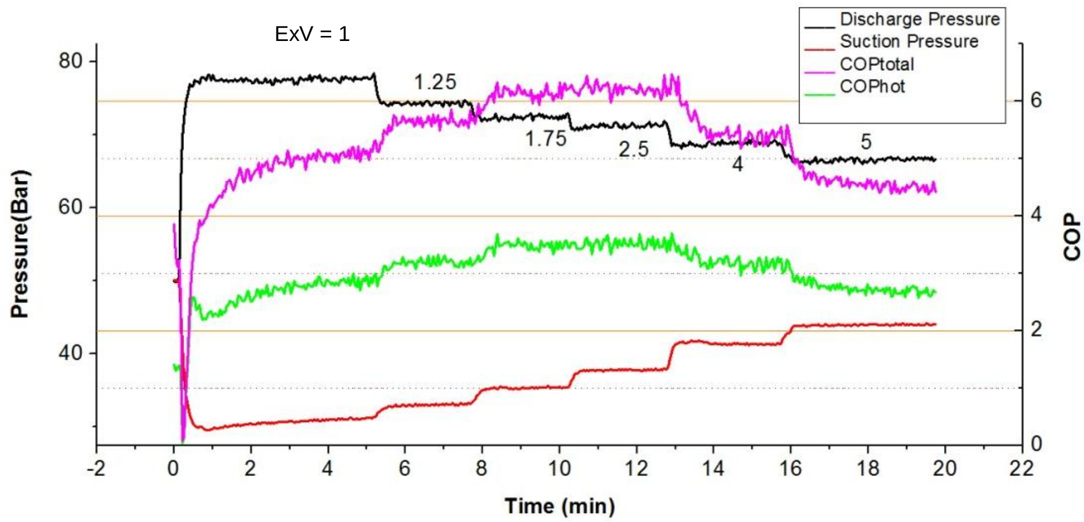


Figure 6: ExV stage opening at 47ºC (116.6oF) for hot water and 10 ºC (50oF) for cold water

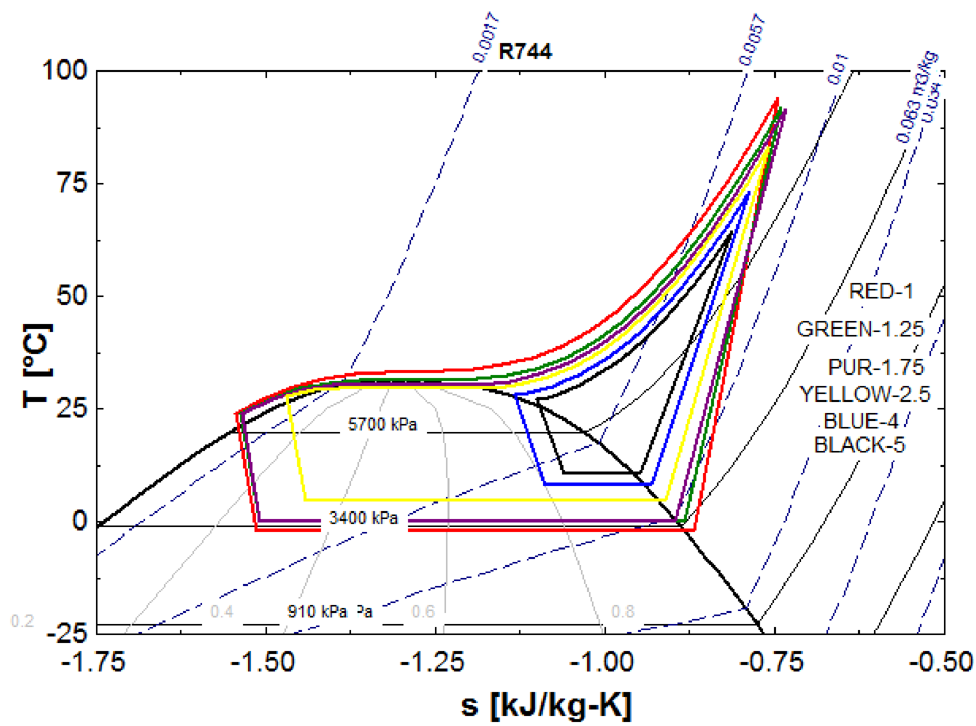
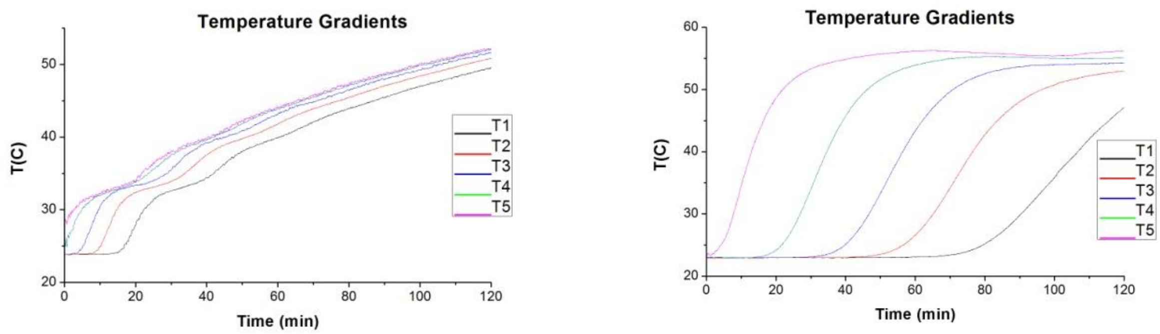


Figure 7: T-s Diagram of ExV stage opening at 23ºC (73.4ºF) for hot and cold storage tank

4.4 Optimum System Performance

It is observed from the experimental results that the CO_2 heat pump efficiency is especially sensitive to the temperature of the inlet water for gas cooler. Thus, two different strategies are investigated to achieve optimum performance. The time required to heat the water from 23 ºC (73.4ºF) to an average of 50ºC (122ºF) in the hot water tank is used to determine the better control strategy. The first strategy (Case 1) involves maximizing the water flow rate to enhance the heat exchanger capacity and system COP and the second strategy (Case 2) is controlling the flow rate to maintain the thermal profile of the water tanks. Figure 8 (a) and Figure 8 (b) shows the water temperature gradient in the hot tank for Case 1 and Case 2, respectively. From Figure 8 (a), it can be seen that the bottom layer water temperature (T1) begins to increase after 18 min for Case 1, while in Case 2 (Figure 8 (b)), T1 starts to increase after 70 min. This delayed thermal profile helps to achieve a stable discharge pressure in Case 2 and consequently stabilizes the COP. As shown in Figure 9, the discharge pressure in Case 2 remains nearly constant at 95 bars (1378 psia) up until 70 min. The discharge pressure is controlled under the range of 95 bars (1378 psia) to 110 bars (1595 psia). The system transient COP is stable at the value of 4 until the discharge pressure starts to increase. For Case 1, as shown in Figure 10, the discharge pressure starts to increase from 75 bars (1088 psia) at 18 min and keeps increasing to 110 bars (1595 psi), which results in a sharp decrease of COP. The average COP and average gas cooler capacity are calculated and presented in Table 3. It is evident from Table 3 that the average COP in Case 2 (4.00) is higher than the average COP in Case 1, which is 3.88. Case 2 can achieve 20% more average gas cooler capacity than Case 1. Case 2 takes 10 minutes less than case 2 to achieve the average 50 ºC hot water temperature.



a) b)

Figure 8: a) Hot tank temperature gradients vs time with fixed flow control rate of 3 L/min.
b) Hot tank temperature gradients vs time with controlled flow rate starting from 1 L/min to 3 L/min

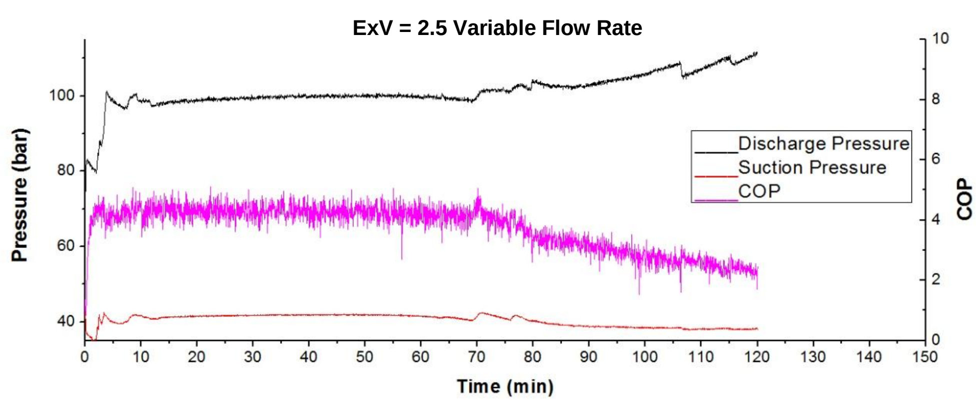


Figure 9: Heating with controlled water circulation rate (50ºC at 99min)

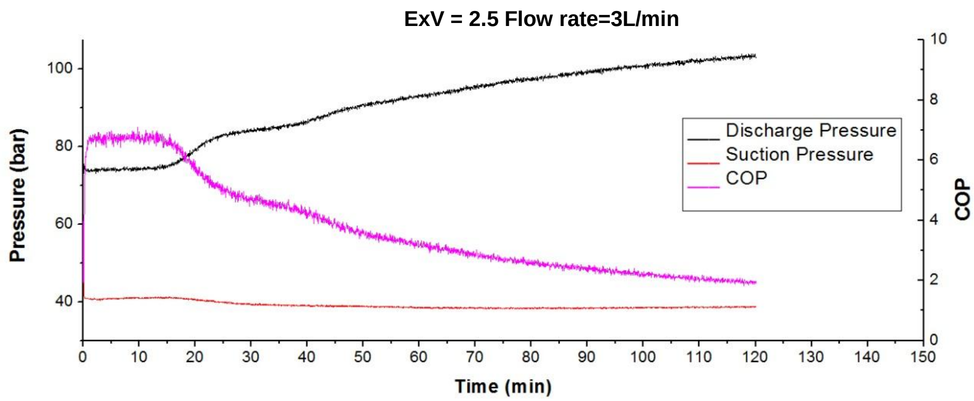


Figure 10: Heating with fixed water circulation rate (50ºC at 109min)

Table 3: Impact of Water Circulation Flow at ExV=1 ¼

	COP Total [-]	**Gas Cooler Capacity [W]**	**Time [min]**
Case 1	3.88	1598.00	109.0
Case 2	4.00	1989.10	99.0

5. CONCLUSIONS

Transcritical CO_2 water source heat pump system with the hot and cold thermal storage tanks were experimentally investigated. The tests were carried out by controlling the water temperature at heat exchanger inlets, ExV opening, and the refrigerant charge level. In addition, two cases of a fixed and a variable water flow rate control in sequence are tested to find the best operating COP. The findings throughout the investigations are:

1. The increase of the water inlet temperature at the gas cooler raises the discharge pressure significantly and drops both the heating and cooling COP, while the increase of the water temperature at the evaporator raises the discharge pressure moderately.
2. Although a larger water flow rate enhances the heat exchanger capacity and the total COP, a smaller water flow rate seems better to maintain the thermal profile of the water tanks.
3. COP tends to increase while closing the expansion valve at a constant water circulation rate of 3 L/min and at 50ºC (122ºF) gas cooler water inlet temperature. In this particular setup, the best COP is found to be approximately 7.0 at a specific ExV opening of 2 ½ opening and a discharge pressure between 75 – 83 bars (1088 psia – 1204 psia) (Figure 10).
4. Variable flow rate control strategy case proves to be 20% more efficient in terms of gas cooler capacity when compared to fixed flow rates.
5. Based on experimental results, a matched tank capacity is suggested between two storage tanks.

NOMENCLATURE

Q_{evap}	Evaporator capacity	[kW]
Q_{gas}	Gas cooler capacity	[kW]
W_{comp}	Compressor power	[kW]
W_{pump}	Water pump	[kW]
m_R	Refrigerant mass flow rate	[kg/s]

$h_{1\text{-}4}$	Refrigerant enthalpy, subscript indicates the position at 1,2,3,4	[kJ/kg]
V_{cold}	Water flow rate at evaporator side	[L/min]
V_{hot}	Water flow rate at gas cooler side	[L/min]
P_{dis}	Pressure at compressor outlet	[Bar]
P_{suc}	Pressure at compressor inlet	[Bar]
COP_{tot}	Overall coefficient of performance including heating and cooling	

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ACKNOWLEDGEMENT

The authors acknowledge the support for this work by Dr. Ali Shakouri of the Birck Nanotechnology Center, Purdue University in preparation of the experimental setup.