Energy Optimization for Transcritical CO2 Heat Pumps for Combined Heating and Cooling and Thermal Storage Applications

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ABSTRACT

A transcritical heat pump (THP) cycle using carbon dioxide (CO_2) as the refrigerant is known to feature an excellent coefficient of performance (COP) as a thermodynamic system. Using this feature, we are designing and building a system that combines a water-to-water CO_2 heat pump with both hot and cold thermal storages know as Thermal Battery (TB) (Blarke, 2012). Smart and effective use of intermittent renewable energy resources (for example solar and wind power) is obtained supplying water heating (>70 ^{o}C) and cooling services (<10 ^{o}C) for residential and commercial buildings.

Our fundamental hypothesis is that if electricity generated by intermittent sources is destined for thermal end-uses an efficient conversion of electricity to thermal energy and storage enables a flexible power supply. Thermal storage is more cost-effective than any electro-chemical or mechanical storage technology. The usability and the cost effectiveness are critical for smart grid policies on large-scale integration of intermittent renewables. In this paper, we present an analytic thermodynamic model that predicts the effect of temperature and flow rate of hot and cold water circulation on system COP. The analytical predictions are consistent with the experimental results (Sarkar, 2010).

1. INTRODUCTION

1.1 Background

Limited fuel sources, dependency on external suppliers and global warming encourage societies to push for higher penetration levels of renewable energy generation technologies. Renewables are categorized as either intermittent or dispatchable sources of electrical energy. Geothermal and hydro (in most cases) are dispatchable. While dispatchable is easier to manage, it is not well distributed or readily accessible for most countries. On the other hand, solar and wind are available everywhere but clouds passing over solar systems or temporary drops in wind create power fluctuations, resulting in intermittent supply. As they become a larger fraction of electricity and thermal energy generation, electrical grid integration must be resolved to smooth fluctuations.

An electrical grid is an interconnected network for delivering electricity from suppliers to consumers. Not only power generation systems but also the demand profile unbalances the grid operated at a constant frequency (60Hz North America and 50Hz in Europe and Asia). If demand exceeds supply, grid frequency drops. When supply exceeds demand, frequency rises. Major frequency distortions can disrupt the grid, in extreme cases causing blackouts. With the aim of maintaining a constant frequency and supply and demand balance, system operators require generators to regulate supply by increasing or decreasing output from their power stations (A.Woyte, 2006). To balance intermittent renewable suppliers and consumers characteristics for utility-scale generation it is generally recognized that there will be a need for energy storage technology. Energy storage allows ‘smoothing’ in power delivery, absorbing production peaks and filling troughs, hence reaching higher renewable penetration levels. This strategy may involve balancing both electrical grid suppliers and consumers in an intelligent manner, often referred to as the Smart Grid concept (Katz, 2011).

In addition, space heating and cooling represent a major part of end-use in electrical power demand, approximately 40% (e.g. Swedish Energy Agency, 2010).

This paper suggests that thermal energy storage has a great potential as energy storage solution for grid balance. The excess power from the renewable sources is converted to heat (hot and cold) and stored. This stored heat energy provides a balance between the demand and the supply. Combining hot and cold thermal storage with a transcritical CO_2 heat pump for combined production of useful heat and cooling, the presented work further investigates the novel Thermal Battery (TB) concept (Blarke et al. , 2011).

1.2 Thermal battery (TB) concept

Over the last decades, HVAC&R(Heat, Ventilation, Air Conditioning and Refrigeration) systems have evolved in order to respond to human necessities in a safer, healthier, greener and more energy and cost-efficient way (McDowall, 2007).

Among modern HVAC, heat pumps offer an energy-effective and cost-efficient alternative to combustion-based or electrical resistance heating systems. Heat pumps use input energy to transfer heat from one zone to another but do not intentionally produce the heat from the input energy. They have proven to operate with COP in the range of 3 to 4 for water heating applications alone (Kim, 2004). In this paper combining water heating and cooling capacities is proposed to enhance system COP.

Figure 1 illustrates TB concept, consisting of a transcritical heat pump (THP) and two water tanks. The TB produces heated water stored in the tank (hot reservoir) that can be used for space heating and/or hot tap water, while stored cooled water can be used for space cooling. Because the heating and cooling capacity combination uses the heat pump to full potential, this paper analyzes an appropriate operation mode for the specific application, allowing an increase of renewable penetration levels in the electrical grid.

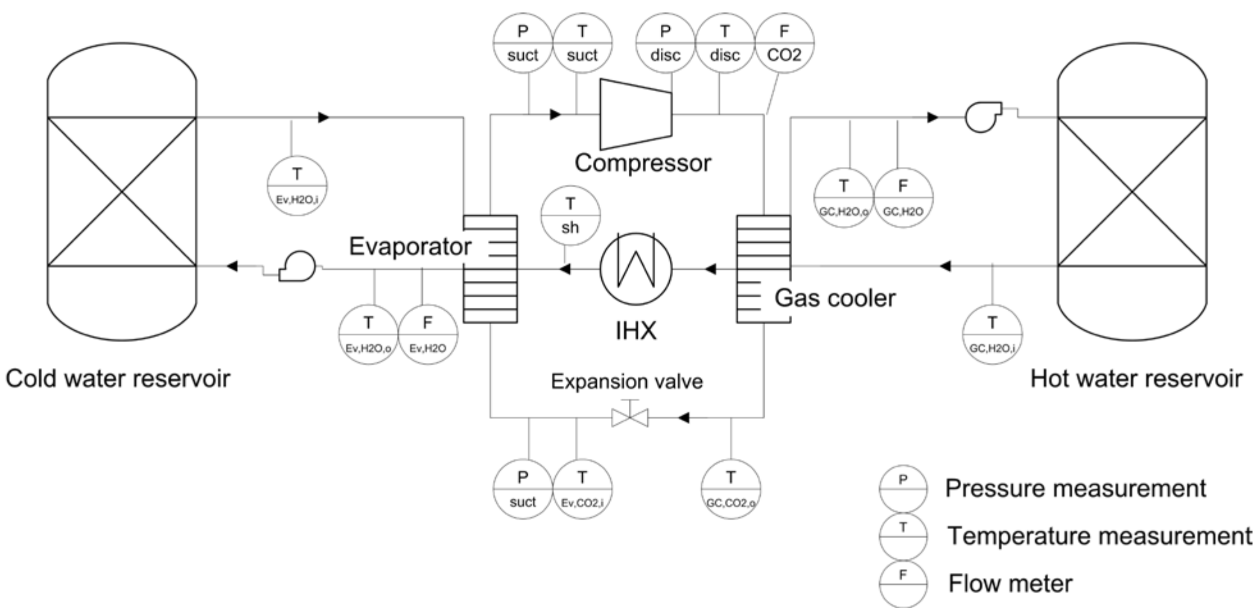


Figure 1 Schematic layout of the TB concept

1.3 CO_2 as refrigerant

A desire to utilize low global warming potential (GWP) fluids in HVAC&R equipment generates much interest in carbon dioxide (CO_2) as a refrigerant. CO_2 is not toxic, flammable or corrosive, inexpensive and has competitive unique thermal properties with GWP =1 (Nekså, 2002). CO_2 becomes a supercritical fluid at 31,1°C at 73,7bar.At supercritical region, temperature and pressure are uncoupled and critical temperature, T_{crit} no longer limits the heat delivery temperatures. The low T_{crit} requires high working pressure but because heat transfer occurs by sensible cooling, the difference between the CO_2 temperature at the gas cooler inlet and outlet is typically larger than during heat rejection by condensation. This temperature difference is known as the refrigerant temperature glide. The independent gliding temperature profile of CO_2 can be more closely matched to the gliding temperature profile of the secondary fluid, which improves heat exchanger effectiveness (Brian T. Austin, 2011). Thus, the performance of the transcritical heat pump (THP) increases when the temperature difference between CO_2 and the heat recovery fluid (air or water) at any point in the gas cooler is reduced.

In addition, with new compressor technology, the high pressure hurdle is no longer a technical obstacle, and comes with additional benefits: lower pressure ratios so higher compressor efficiencies are achieved, and the relatively high vapor density of CO_2 corresponds to a high volumetric heating capacity ($C = \dot{m}c_p$). This allows a smaller volume of CO_2 to be cycled resulting in smaller components and a more compact system.

The idea of using CO_2 heat pumps for simultaneously producing useful heat and cooling has been dealt with by Adriansyah (2004) and more recently by Sarkar (2010) and Chen (2010). Our novel focus is the integration of both hot and cold thermal storage to buffer electrical grid fluctuations in a Smart Grid perspective to balance power supply and consumer thermal demand.

1.4 Methodology

Based on the proposed model of a heat pump for simultaneous water heating and cooling with hot and cold thermal storages, we analyze the system COP. The set temperatures of the water reservoirs were imposed to residential house applications, $T_{hot} = 70 - 80℃$ and $T_{cold} = 0 - 15℃$. The purpose of the work developed was to systematically investigate the possibilities for increased energy efficiency of TB systems when applying different demand conditions. The first step was developing a theoretical operation model of the THP on EES (Klein, 2002) to obtain the impact of various operating parameters. Most of the various conditions tested on the model are validated against experimental results of other published works. While the water temperature in the outlet of the gas cooler and evaporator COP variation have not been experimentally validated to date.

2. THEORY

In this section the governing thermodynamic equations are presented. Thermodynamic properties of CO_2 provided in EES software are based on the equations of state developed by Span and Wagner (1996).

2.1 Process analysis

Modeling involves two parts the thermodynamic analysis of the system and the transport characteristics of the refrigerant and water.

Following assumptions are used for simplification:

- Steady-state system operating mode
- Changes in kinetic and potential energy are negligible
- Compressor operates adiabatically and at constant conditions of pressure and temperatures
- Heat losses in connecting piping is negligible
- The expansion process is isenthalpic
- Steady refrigerant flow, equal for all components, inlet and outlet.
- Effect of internal heat exchanger are marginal (Sarkar, 2010)

Energy balance equations are defined for the heat flux in gas cooler $\dot{Q}_{GC}$ and evaporator $\dot{Q}_{Ev}$ for both refrigerant and water side as shown in Eqs. (1)-(4):

$$\dot{Q}_{GC} = \dot{m}_{H_2O,GC} \times C_{p,H_2O} \times \left(T_{H_2O,GC,o} - T_{H_2O,GC,i}\right) \text{ [kW]} \quad (1)$$
$$\dot{Q}_{GC} = \dot{m}_{CO_2} \times C_{p,CO_2} \times \left(T_{CO_2,GC,i} - T_{CO_2,GC,o}\right) \text{ [kW]} \quad (2)$$
$$\dot{Q}_{Ev} = \dot{m}_{H_2O,Ev} \times C_{p,H_2O} \times \left(T_{H_2O,Ev,i} - T_{H_2O,Ev,o}\right) \text{ [kW]} \quad (3)$$
$$\dot{Q}_{Ev} = \dot{m}_{CO_2} \times (h_{Ev,i} - h_{Ev,o}) \text{ [kW]} \quad (4)$$

Where $\dot{m}$ is mass flow rate in kg/s, $C_{p,fluid}$ is the fluid specific heat of the fluid in kJ/kgK at the arithmetic mean temperature in each heat exchanger, T is temperature in ℃ and h is the enthalpy in kJ/kg.

Eq. (4) has to be determined with enthalpy values since it is a ‘change of state’ process, i.e. evaporation.

Moreover, the heat flux can be defined based on the overall heat transfer coefficient and the temperature difference between two fluids, using the convection heat transfer equations.

$$\dot{Q}_{GC} = U \times A_{GC} \times LMTD_{GC} = U \times A_{GC} \times \frac{(T_{CO2,GC,i} - T_{H2O,GC,o}) - (T_{CO2,GC,o} - T_{H2O,GC,i})}{ln\left[(T_{CO2,GC,i} - T_{H2O,GC,o})/(T_{CO2,GC,o} - T_{H2O,GC,i})\right]} \text{ [kW]} \quad (5)$$

$$\dot{Q}_{Ev} = U \times A_{Ev} \times LMTD_{Ev} = U \times A_{Ev} \times \frac{(T_{H2O,Ev,i} - T_{CO2,Ev,o}) - (T_{H2O,Ev,o} - T_{CO2,Ev,i})}{ln\left[(T_{H2O,Ev,i} - T_{CO2,Ev,o})/(T_{H2O,Ev,o} - T_{CO2,Ev,i})\right]} \text{ [kW]} \quad (6)$$

Where U is the overall heat transfer coefficient in W/m^2K and A_{GC} and A_{Ev} are the heat transfer areas of the gas cooler and evaporator in m^2.

The compressor work can be found based on energy conservation law with adiabatic boundaries as shown in eq. (7).

$$\dot{W}_{comp} = \frac{\dot{m_{CO2}}}{\eta_{is}} \times (h_{discharge,is} - h_{suction}) \text{ [kW]} \quad (7)$$

Where η_{is} is the compressor isentropic efficiency and $\dot{W}_{comp}$ the compressor power in kW.

The energy balance for the entire system is expressed:

$$\dot{Q}_{Ev} + \dot{W}_{comp} = \dot{Q}_{GC} \text{ [kW]} \quad (8)$$

Finally, the heat pump system performance is discussed in terms of heating and cooling capacity and COP. Heating and cooling capacity are defined as the heat delivered and removed, respectively and they are calculated as shown in eqs. (1) to (6). COP can be defined in terms of heating or cooling. For heating, in this paper, it determined by the ratio of the heat output or removed to the sum of the compressor and water pump work. The cooling COP is the ratio of heat removed over compressor and water pumps work.

$$COP_{heating} = \frac{\dot{Q}_{GC}}{[\dot{W}_{comp} + \dot{W}_{pump}]} \text{ [-]} \quad (9)$$

$$COP_{cooling} = \frac{\dot{Q}_{Ev}}{[\dot{W}_{comp} + \dot{W}_{pump}]} \text{ [-]} \quad (10)$$

Since in this paper the THP is used for simultaneous heating and cooling production, performance is discussed in terms of system COP, defined as:

$$COP_{sys} = \frac{\dot{Q}_{GC} + \dot{Q}_{Ev}}{[\dot{W}_{comp} + \dot{W}_{pump}]} = COP_{heating} + COP_{cooling} \text{ [-]} \quad (11)$$

Where the water pumps work for both, gas cooler and evaporator water supply, is defined as:

$$\dot{W}_{pump} = \frac{\dot{m_{H2O}}}{\eta_{is,pump}} \times \Delta P \text{ [W]} \quad (12)$$

Where, η_{is}, is the electric-fluid power conversion efficiency and ΔP the pressure drop of water for horizontal pipes and fully developed flow (Kays, 1993).

2.2 Possibilities for improved system energy efficiency

Key design parameters for the CO_2 heat pump are the temperature and pressure of the refrigerant (CO_2), secondary fluid (water) temperature at hot and cold heat exchanger ends, and mass flow rate. Manipulating these parameters yields different coefficient of performance as shown in previous published papers (Sarkar J., 2010), (Sarkar J., 2004) and (Yokoyama R, 2006).

Furthermore, in the TB concept the system can be operated in two different modes: a) independent charging and tapping mode or b) simultaneous charging and tapping mode. The independent operation mode support bigger volumes, as all the heat capacity will be generated during a longer period (electricity off peak demand) and supplied gradually along the day. In contrast the simultaneous mode leads to smaller volumes as the heat capacity is shortly dispatched. This study is primarily focused on the influence of temperature and mass flow rates in THP performance. Sizing of the two heat storage water tanks directly connected to the THP in the above modes is only qualitatively discussed.

3. ANALYSIS AND DISCUSSIONS

In this section the THP performance at different operational conditions is analyzed to select an appropriate operational mode for simultaneous household heating and cooling applications.

3.1 Calculations conditions

The optimization of the CO_2 heat pump on simultaneous heating and cooling system is complex as it depends on several parameters such as compressor speed and efficiency, water and refrigerant inlet/outlet temperature and pressures, flow rates and heat exchanger dimensions. However constraining some of the parameters based on the application requirements make the task easier.

Based on Sarkar et al. (2004), two parameters, $T_{CO2,Ev}$ and $T_{CO2,GC,o}$, have a stronger influence on optimum discharge pressure than the others (see section 2.1 for all parameter dependency). The optimum discharge pressure increases with the increase in gas cooler exit temperature and decrease in evaporator temperature. Liao et al.(2000), Sarkar et al. (2004), Kauf (1999) and Chenand Gu (2005) theoretical studies have proposed correlations on discharge pressure optimization. However, the Sarkar et al. (2004) correlation has seemed best compared to other proposed correlations based on experimental studies (Cabello R, 2008) and is expressed by:

$$p_{opt} = 4{,}9 + 2{,}256T_{CO2,GC,o} - 0{,}17T_{CO2,Ev} + 0{,}002T^2_{CO2,GC,o}[bar] \quad (13)$$

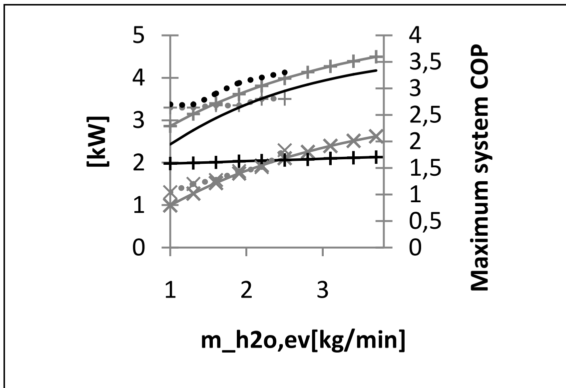
The compressor discharge pressure $P_{discharge}$ is set to give the maximum COP value for $T_{CO2,GC,o} = 35°C$. Lower temperature approach on the gas cooler cold side (difference between CO_2 gas cooler outlet temperature and water gas cooler inlet temperature) reduces the optimum pressure and increases the system COP (Fronk BM, 2011a). The evaporation temperature ($T_{CO2,Ev}$) was set to 5°C in all analysis studies and the degree of superheat 20°C. At this condition the evaporation pressure of refrigerant ($P_{evaporation}$) is 40 bar. These constraints consider the application requirements, water heating and cooling, considering gas cooling process. They also match the experiment condition values from previous work.

Heat exchangers are defined by their geometrical parameters and the heat transfer coefficients. In the analysis, we start with giving appropriate values for the heat transfer coefficients.

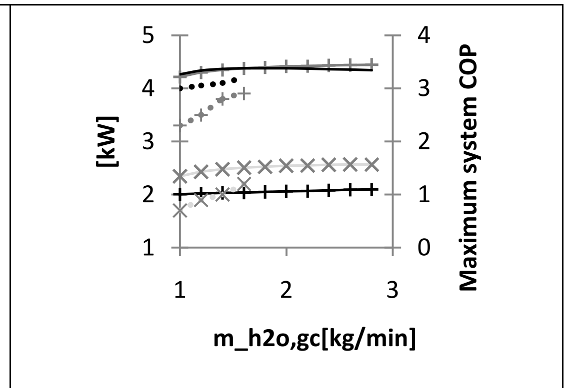
Within the heat exchangers the thermo physical properties of the fluids were calculated on the basis of the arithmetic mean temperature of outlet and inlet on the respective components.

3.2 Results and discussion

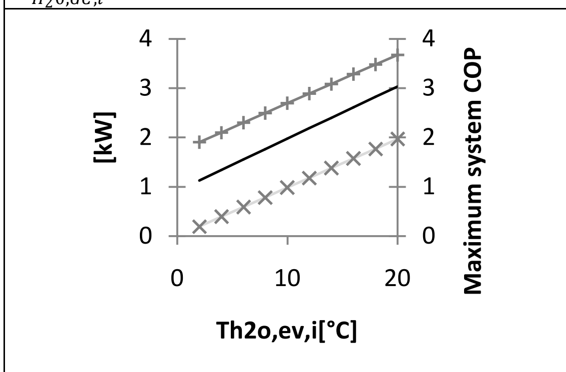
The COP of the CO_2 heat pump being studied for simultaneous heating and cooling applications is evaluated on the basis of system COP (eq. 11).



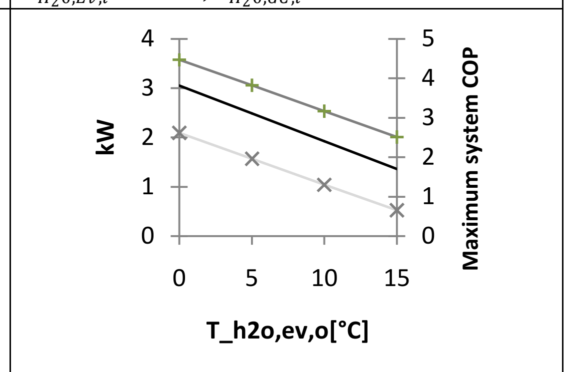
A. $\dot{m}_{GC,H_2O} = 1\ kg/min$, $T_{H_20,Ev,i} = 20°C$, $T_{H_20,GC,i} = 20°C$



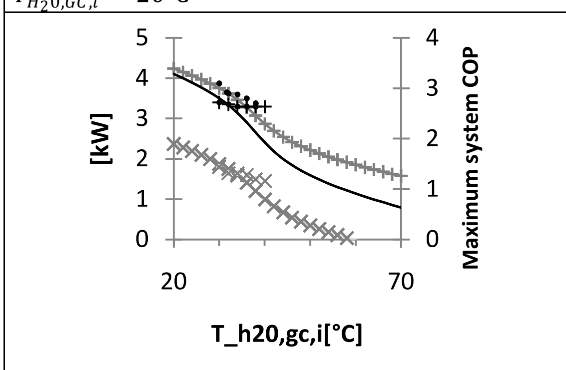
B. $\dot{m}_{Ev,H_2O} = 1{,}5\ kg/min$, $T_{H_20,Ev,i} = 20°C$, $T_{H_20,GC,i} = 20°C$



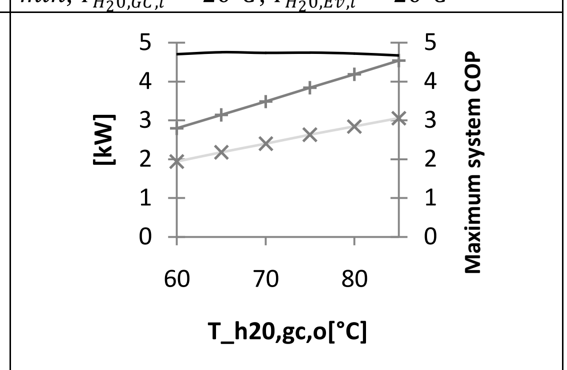
C. $\dot{m}_{GC,H_2O} = 1\ kg/min$, $\dot{m}_{Ev,H_2O} = 1{,}5\ kg/min$, $T_{H_20,GC,i} = 20°C$



D. $\dot{m}_{GC,H_2O} = 1\ kg/min$, $\dot{m}_{Ev,H_2O} = 1{,}5\ kg/min$, $T_{H_20,GC,i} = 20°C$, $T_{H_20,Ev,i} = 20°C$



E. $\dot{m}_{GC,H_2O} = 1\ kg/min$, $\dot{m}_{Ev,H_2O} = 1{,}5\ kg/min$, $T_{H_20,Ev,i} = 20°C$



F. $\dot{m}_{GC,H_2O} = 1\ kg/min$, $\dot{m}_{Ev,H_2O} = 1{,}5\ kg/min$, $T_{H_20,GC,i} = 20°C$, $T_{H_20,Ev,i} = 20°C$

q_gci q_evi W_total q_evi exp q_gc exp COP COP exp

Figure 2 Influences of water mass flow rate and inlet/outlet temperatures

Our model simulates the heat pump performance by changing the inlet and outlet water temperatures and the water mass flow rates considered both for the gas cooler and the evaporator. The numerical test on this analytic model includes stepwise variable change in the condition cases A through F as shown in Figure 2.

First, in *case 2.A* with the increase in water mass flow rate in the evaporator, the cooling capacity increases due to the increase of heat transfer rate in water. Thus, the heating output increases. Compressor work is maintained constant and pump work increases with water mass flow rate. Both water outlet temperatures increase with the increase of water mass flow rate in the evaporator.

Then**,** in *case 2.B*, heating output increases in the water side due to the heat transfer rate which also enhances the cooling capacity. Also, water outlet temperature decreases due to the constant evaporator water mass flow rate (from eq. (3) and (6)). The COP increase is given by the increasing of heating and cooling capacities and constant compressor work even for an increase work from the water pumps.

In *case 2.C* with the increase of the water inlet temperature to evaporator, the water outlet temperature of the evaporator increases and enhances the cooling capacity which for a constant compressor work enhances the heating capacity and so water outlet temperature of the gas cooler also increases. Hence the system COP increases.

Case 2.D shows that the COP decreases with the increase of the evaporator outlet water temperature.

As the gas cooler water inlet temperature increases, in *case 2.E*, the heating capacity decreases due to deterioration in heat transfer properties of CO_2 in gas cooler and the refrigerant exit temperature increases, which increases the vapor quality in evaporator inlet and hence cooling capacity also decreases. As the compressor work is maintain constant the system COP decreases significantly. Combining the results from cases 2.*C* and 2.*E* it can be concluded that admitting the coil exchanges directly heat between the water coming from the gas cooler and evaporator with the water contained in the tanks continuously lead to system COP decrease. This is because the water temperature in the tanks becomes hotter or colder in the hot and cold tanks, respectively.

Finally, *case 2.F* shows the effect of increasing the water outlet gas cooler temperature is COP small decrease. Results from case *2.F* follow the trends of previous studies (Yokoyama R, 2006) but should need future experimental as the information for the different system parameters from previous work were not sufficient.

In order to validate the model with Sarkar (2004) pressure and temperature in the compressor are maintained constant. These specific conditions limit the COP enhancement since they are based on the constraints that are not necessarily optimum conditions. The differences between the absolute values of numerical model and experimental results did not match perfectly due to different heat exchanger dimensions and compressor specification as they were not clearly specified on the respective papers.

Comparisons with solar technology studies claim comparable results. Presented solutions consist in lower inlet water temperature in thermal solar systems and thermal storage stratification for both thermal solar and photovoltaic systems are the future for system improvement (Koppen, 1979), (Hollands, 1989) and (Andersen, 2007). For our future work, combination of the solar system technology state-of-the-art and simultaneous heat and cooling THP technology water storage solutions seem to have a potential as smart grid management optimization.

4. CONCLUSIONS

The steady-state performance of a CO_2 transcritical heat pump for simultaneous heating and cooling is presented. The results are obtained by varying operating parameters, such as inlet/outlet temperatures and water mass flow rates, of a system suitable for household applications. The predicted performance is in reasonable agreement with available experimental data in literature. Heat exchanger specific characteristics and the compressor performance approximation may lead to the small difference between experimental and analytical results.

Results indicate that the effect of evaporator water mass flow rate (COP increases 0,8 for 1 kg/min versus 0,2 in the gas cooler) is more pronounced compared to the water mass flow rate in the gas cooler. They similarly show that the effect of the gas cooler water inlet temperature is less significant (COP decreases 0,9 for the range between 30° C and 40° C) compared to the evaporator water inlet temperature (COP increases 1 for the range between 5° C and 15° C). And also the effects of the water outlet temperature are more pronounced for the evaporator outlet temperature (COP decreases with the temperatures increase in both cases).

In sum, it is concluded that dynamic modeling and system components integration should be considered as it directly affects the system COP. Based on the analysis high $T_{H_20,Ev,i}$, low $T_{H_20,GC,i}$ and large $\dot{m}_{GC,H_2O}$ and $\dot{m}_{Ev,H_2O}$ yields highest COP with a fixed performance of the compressor. Attention should be brought to the evaporator side and compressor operational limitations. More importantly, the results show that COP values greater than 4 are feasible with the presented system. Therefore, TB concept shows to have potential as a smart grid technology solution.

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NOMENCLATURE

A	Heat transfer area	(m^2)
C_p	Specific heat	(kJ/kg K)
h	Specific enthalpy	(kJ/kg)
m	Mass flow rate	(kg/s)
P	Pressure	(Pa)
$\dot{Q}$	Heat transfer rate	(kW)
T	Temperature	(̊C)
U	Heat transfer coefficient	(kW/m^2K)
W	compressor/pump work	(kW)
ε	Heat exchanger effectiveness	(-)
η	Efficiency	(-)
ρ	Density	(kg/m^3)

Subscripts

GC	gas cooler
Ev	evaporator
i	inlet
o	outlet
is	isentropic
exp	experimental

REFERENCES

Adriansyah , 2004, Combined air conditioning and tap water heat ing plant using CO2 as refrigerant, *Energy and Building*, no. 36: pp. 690-695.

Andersen E., Furbo,S. and Fan,J., 2007, Multilayer fabric stratification pipes for solar tanks, *Solar Energy*. Vol.81, pp. 1219-1226.

Austin B. T., and Sumathy,K., 2011,Transcritical carbon dioxide heat pump systems: A review Renewable and *Sustainable Energy Reviews*, 15 pp. 4013-4029.

Blarke Morten , 2011,Smart intermittency-friendly cogeneration: Technoeconomic performance of innovative double storage concept, *6th International Renewable Energy Storage Conference*, Berlin , 2011.

Blarke M., Yazawa, K., Shakouri, A. and Carmo,C. , 2012,Thermal Battery with CO2 compression heat pump: Techno-economic optimization of a high-efficiency Smart Grid option for buildings, *Energy and Buildings*, 50:pp.128-138

Cabello R., Sánchez, D., Llopis, R.and Torrella, E. , 2008,Experimental evaluation of the energy efficiency of a CO2 refrigerating plant working in transcritical conditions *Applied thermal engineering* 28 : pp. 1596-1604.

Chen H. and Lee, W.L. , 2010,Combined space cooling and water theating system for Hong Kong residences,*Energy Builiding*,no.42:pp. 243-250.

Chen Y. and Gu, J. , 2005, The optimum high pressure for CO2 transcritical refrigeration systems with internal heat exchangers , *Int.J.Refrig.*, Vol. 28: p.1238-1249.

Fronk B.M. and Garimella, S. , 2011a,Water-coupled carbon dioxide microchannel gas cooler for heat pump water heaters: part I—experiments , *International Journal of Refrigeration* 34, pp. 7–16.

Fronk B.M., Garimella, S. , 2011b, Water-coupled carbon dioxide microchannel gas cooler for heat pump water heaters: part II—model development and validation, *International Journal of Refrigeration* 34.: p. 17–28.

Hollands K.G.T. and Lightstone, M.F. ,1989, A review of low- flow, stratified-tank solar water heating systems *Solar Enegy*, Vol.43, No. 2, pp.97-105.

Katz R.H., Culler, D.E., Sanders, S., Alspaugh, S., Chen, Y., Dawson-Haggerty, S., et al., 2011, An information-centric energy infrastructure: The Berkeley view *Sustainable Computing: Informatics and Systems*, 1, p. 7-22..

Kauf F. , 1999, Determination of the optimum high pressure for transcritical CO2 refrigeration cycles *Int.J.Therm.Sci*., Vol. 38: p.325-330.

Kays W.M. and Crawford, M.E. , 1993, *Convective Heat and Mass Transfer*, McGraw-Hill, 1993.

Kim et al. , 2004, Fundamental process and system design issues in CO2 vapor ,*Progress in Energy and Combustion Science* 30. p. 119-174.

Kim M.H., Pettersen, J. and Bullard, C.W. , 2004, Fundamental process and System design issues in CO2 Vapor compression systems,*Progress in Energy and Combustion Science* Vol.30, no.2: p. 119-174.

Klein S. A.2002 ,Engineering Equation Solver (EES) for Microsoft Windows Operating System: Commercial and Professional Versions [http://www.fchart.com]. - Madison : WI: F-Chart Software.

Koppen C.W.J. van, Simon Thomas,J.P. and Veltkamp,W.B. , 1979, The actual benefits of thermally stratified storage in a small and a medium size solar systems, *Proc. ISES World Congress Atlanta. - USA*, p. 576-580.

Liao S.M., Zhao, T.S. and Jakobsen, A. , 2000, A correlation of optimal heat rejection pressures in transcritical carbon dioxide cycles,*App.Therm. Eng.*, Vol. 20: p. 831-841.

McDowall Robert , 2007, *Fundamentals of HVAC systems: SI edition*, Academic Press, ISBN 0123739985, 9780123739988.

Nekså P. ,2002, CO2 heat pump systems, *International Journal of Refrigeration* 25: p. 421-7.

Nekså P., Pettersen, J. and Skaugen, G. , 2002, CO2 Refrigeration, Air Conditioning and Heat Pump Technology

Sarkar J. ,2010, Review on Cycle Modifications of Transcritical CO2 Refrigeration and Heat Pump Systems,*Journal of Advanced Research in Mechanical Engineering* Vol.1. p. 22-29.

Sarkar J., Bhattacharyya, S. and Ramgopal, M. , 2004, Optimization of transcritical CO2 heat pump cycle for simultaneous coolind and heating applications, *International Journal of Refrigeration* 27: p. 830-838.

Sarkar J., Bhattacharyya, S. and Ramgopal, M. , 2010, Experimental investigation of transcritical CO2 heat pump for simultaneous water cooling and heating,*Thermal science* 42:p. 57-64.

Swedish Energy Agency , 2010 Energy in Sweden 2010: Facts and figures p. 48-49.

Wagner and Span R. , 1996, A new equation of state for carbon dioxide covering the fluid region from the triple-point temperature to 1100K at pressure up to 800 MPa, *Journal of Physical and Chemical Reference* Data 26. p. 1509-1596.

Woyte A., Thong, V.V., Belmans R. and Nijs, J. , 2006, Voltage fluctuations on distrution level introduced by photovoltaic systems, *IEEE Trans. Energy Conservation*, vol.21 no.1: p. 1652-1657.

Yokoyama R., Shimizu, T., Takemura, K. and Ito, K. , 2006, Performance analysis of hot water supply system with a CO2 heat pump by numerical simulation *JSME International Journal* 49: p. 541-548.