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Experimental Analysis of a Hight Temperature Heat Pump Using Stored Heat from a Solar Thermal System

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Abstract

Thermal storage systems use for district heating can improve their efficiency by decreasing their storage temperature and therefore minimizing heat losses and cost. A high temperature heat pump (HTHP) operating with heat source temperature between 30 °C to 50 °C can further optimize the performance of the system and deliver hot water above 70 °C at demand. In this study a HTHP that uses thermal energy produced by a solar thermal system and stored in a seasonal thermal storage has been developed, installed and experimentally investigated under real conditions. An additional challenge of the present work was to build the heat pump using off-the shelf components without further modifications. During two years of operation the water source temperatures measured were from 31 °C to 40 °C and the heating capacity resulted from 35 kW to 43.7 kW. The COP calculated resulted in values from 5.3 to 6 for pressure ratios between 2.3 to 2.8. The results demonstrated that HTHPs are an attractive approach to use low grade heat sources specially when produced by renewable energy sources.

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Keywords: Seasonal Thermal Storage; High Temperature Heat Pump; Vapour Compression; Space Heating; Hot Water Supply;

1. Introduction

Seasonal thermal energy storage systems charged by solar thermal energy present a potential to provide heating for domestic applications. Large solar collector surfaces and large volume of storage accumulators can balance the seasonal mismatch during the year [1]. For thermal storage specially during the season, low temperature storage is preferable to minimize the heat losses during the year. Heat pumps are systems that can use low temperature heat source to produce high temperature water for space heating applications. In addition, heat pumps help the stratification of the storage tank by returning the source water colder [2]. In previous studies on heat pumps combined with seasonal thermal storage and gas boilers, results showed that heat pumps can provide between 49 to 67 % of the heating demand, while the direct discharge of the storage can cover between 33 to 39 % [3].

The performance of a heat pump as part of a combination of heating systems is maximized when is correctly integrated considering the thermal behavior of the system. The correct integration of the heat pump in the system has a greater impact on its performance than the COP of the heat pump itself [4]. Systems such as seasonal thermal storage charged by a solar thermal system present high temperature fluctuations during the year. For high fluctuation profiles between heat source and heat demand temperatures the capacity of the heat pump is recommended to be 60 % of the maximum hourly peak demand [5].

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Nomenclature

COP	Coefficient of performance, [-]	*ev*	Evaporator
cp	Specific heat capacity, [kJ/kg°C]	*i*	Inlet
f	Frequency, (Hz)	*is*	Isentropic
h	Enthalpy, [kJ/kg]	*out*	Output
m	Mass flow rate, [kg/s]	*r*	Refrigerant
n	Number of pistons	*su*	Suction
Q	Heat capacity, [kW]	*vol*	Volumetric
T	Temperature, [°C]	*w*	Water
W_{in}	Electric power consumption, [kW]		
		Abbreviations	
Greek letters		CAES	Compressed Air Energy Storage
Δ	Difference, [-]	EEV	Electronic Expansion Valve
η	Efficiency, [%]	GWP	Global Warming Potential
		HFC	Hydrofluoro-carbon
Subscripts		HTHP	High Temperature Heat Pump
cd	Condenser	STES	Seasonal Thermal Energy Storage
dis	Discharge	SEER	Seasonal Energy Efficiency Ratio
displ	Displacement		

2. Description of the site

The heat pump prototype of this study is part of a seasonal thermal storage demonstration plant installed in Warsaw, Poland. The installation consists of a solar thermal collector field, a seasonal thermal energy storage tank (STES), a gas boiler, the heat pump and the heating load (building) as presented in Fig. 1. The solar thermal collector field's total area is 151 m^2 of flat plate collectors positioned on the ground. The thermal storage tank consists of a heavily insulated water tank with a total internal volume of 800 m^3. The building is part of a hospital with a total surface area of 793 m^2 in two-floors. The building´s heating system consists of a gas boiler of 92 kW heating capacity. Finally, the high temperature heat pump (HTHP) is a prototype that has been developed for this specific application. It is designed for nominal operating conditions of 30 °C evaporation and 75 °C condensing temperatures and a maximum heating capacity of 63 kW.

The solar collectors' field is located near the storage tank and heat is transferred using glycol as a medium. The HTHP is located in the underground floor of the building to heat. Heat from the STES tank is transferred to the building via underground insulated water pipes.

The operation of the system starts when the STES is charged with thermal energy via the solar collectors during high solar radiation periods. Thermal energy can then be stored in the seasonal thermal energy storage for long periods during the year, in order to be used to heat the building during the heating period (April – September).

In the heating season heat from the storage tank is directed to the building heating system by demand. Hot water from the storage tank is directed to the building via the underground piping and heat is transferred to the building´s heating water loop via heat exchangers. When the temperature of the hot water coming from the STES is below the temperature set by the heating system the water is directed to the heat pump. If the supply water temperature is below 45 °C then the HTHP is activated. The hot water produced by the heat pump is driven into a 1m^3 buffer to moderate its cycle periods. When the hot water temperature coming from the storage tank is not high enough for space heating neither for the heat pump operation (< 30 °C), then the gas boiler is activated to cover the heating demand.

The heating water supply temperature and flow conditions are set by the control algorithm of the building´s heating system and gas boiler. The mass flow rate in the sink side is set at 1.08 lt/s, which is the maximum flow rate that the gas boiler can operate. The building heating requirements is 82.8 kW with a maximum supply and return water temperatures of 75 °C and 56 °C.

The operation of the heat pump prototype during the testing campaign is fully dependent on the environmental conditions, the heat stored in the storage tank and the heating demand. The water supply temperature is defined by the control algorithm of the building´s heating system. Therefore, the ON/OFF cycles of the heat pump are conditioned to the heating capacity stored into the buffer tank and the heating demand.

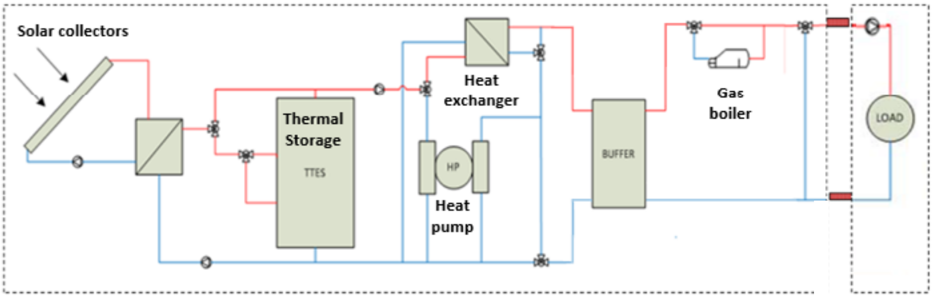

Fig. 1. Schematic diagram of the demo site in which the HTHP is installed, showing the configuration between the solar thermal field, the STES, the HTHP, the buffer tank, the gas boiler and the load.

3. Heat pump description

The HTHP prototype is a single loop water-to-water heat pump equipped a reciprocating compressor. The main components of the refrigeration cycle consist of the compressor, the condenser, a liquid receiver, the expansion device, and the evaporator as shown in Fig. 2. The safety equipment and auxiliary components consists of a suction accumulator, an oil separator, a filter/drier, two sight-glasses and block valves. The suction accumulator and the oil separator are positioned in the suction and discharge line respectively. The liquid receiver is installed between the condenser and the expansion device.

The compressor used in this prototype is the semi-hermetic reciprocating compressor 6MU-40X made by Copeland. It is a six cylinders compressor with bore 80.6mm, stroke 57.5 mm and 153 m^3/h volumetric displacement. The maximum suction and discharge pressures are 22.5 bar-g and 32.5 bar-g. The compressor is charged with 3.3 lt of POE ISO46 lubricant. The expansion device installed is an electronic expansion valve (EEV) ETS25 made by Danfoss. It is equipped with a stepper motor of 160 steps and it is controlled via an electronic control (EKD316). The evaporator is the brazed plate heat exchanger B120Tx90 made by SWEP and it has a heat transfer surface area of 6.19 m^2. The condenser is the V200Tx50 also a brazed plate heat exchanger made by SWEP with a surface area of 11.6 m^2.

The working fluid used in this prototype is the 1,1,1,3,3-pentafluoropropane (R-245fa). This fluid is a non-flammable HFC normally used for organic Rankine cycle applications. It has a boiling point of 15.3 °C and critical point of 153,8 °C [6]. It belongs to B1 category of ASHRAE 34-2013 [7]. It has a global warming potential (GWP) of 1030. [8]

3.1. Methodology and data monitoring

The data collected and used for this work belongs to a period of 12 months of operation. During the datalogging period the HTHP operated for a total of 9934 minutes which consisted in 209 operation cycles. The duration of the cycles ranged from 13 minutes to 10 hours and 40 minutes of continuous operation. All monitored values are measured and recorded every 30 seconds for 24 hours per day.

For the data analysis the data logged is filtered and only operation cycles with steady source and supply water temperatures are considered. Data belonging to the start-up period of approximately 5 minutes and the last 1 minute of every cycle is not considered in this analysis. In addition, cycles lower than 20 minutes of duration are not considered in this study due to their unstable performance. Average values of temperature, pressure, power and flow measurements are calculated and analyzed.

Fig. 2 shows the location of the sensors in both the refrigerant and water cycles. Temperature in both the refrigerant and water circuits are measured using 4-wire PT100 type “A” sensors. The temperature sensors in the water side are intrusive type inserted within the water pipes at the inlet and outlet ports of the heat exchangers. The temperature sensors of the refrigerant cycle are mounted on the surface of the copper pipeline at the inlet and outlet of the main components. The refrigerant pressure is measured by pressure transducers mounted at the inlet and outlet of the main components via Schrader access ports. The sensors output signal is 4-20 mA and require a supply voltage of 24 Vdc. There are two models of sensors mounted on the circuit of different measurement range, the PT5-07 and PT5-18 both made by ALCO Emerson, both have a measurement error of ≤±1 % FS.

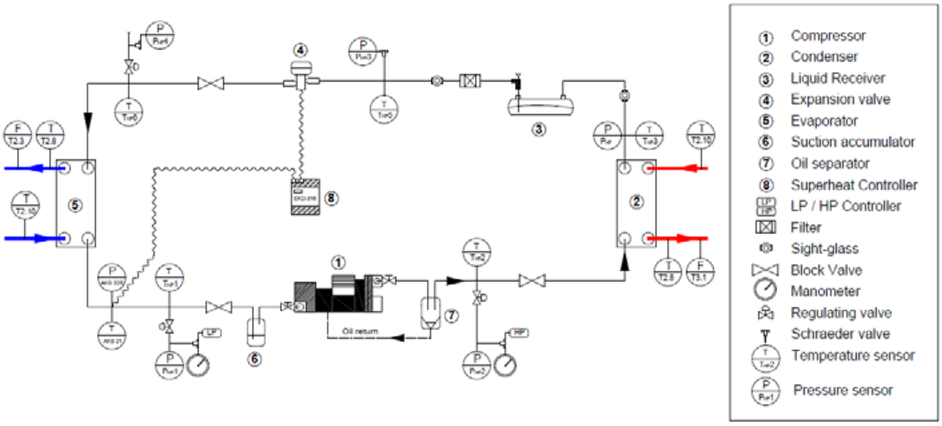

Fig. 2. Schematic diagram of the heat pump prototype presenting the configuration, the main components and the data measuring points.

The water flow rate in both the heat source and heat sink is measured by two electromagnetic ENDRESS HAUSER flow meters. The output signal is 4-20 mA and the maximum operating temperature is 225 °C with an accuracy of ±0.5 % of reading. The energy consumed by the compressor is measured by an energy meter connected to the power supply within the electrical power box. In addition, voltage and current of the compressor power supply are also measured in all three phases.

4. Experimental results

During the 12 months of monitoring the seasonal storage tank temperature ranged from 60 °C at the beginning of September to 25 °C in January. From January to April the temperature of the water in the tank remained low and was not used. After April the solar radiation increased and the charging of the storage tank continued.

In Fig. 1 are presented the water source and sink temperatures during the heat pump operation. The water source temperature variated between 31.2 °C to 39.9 °C and the water outlet in the condenser from 46.2 °C to 54.5 °C. The temperature difference between the water inlet and outlet of the evaporator and condenser was calculated from 5.2 K to 7.1 K and 5.4 K to 6.8 K respectively. Both water pumps in the evaporator and condenser loop were maintained at steady speed. However, the flow rate fluctuated slightly during the monitoring period, from 1.25 to 1.33 kg/s in the evaporator and from 1.52 to 1.56 kg/s in the condenser.

The cooling and heating capacity were calculated from the measurements of mass flow rate and temperature in the water cycles as presented in Equation 1.

$$Q_w = m_w \cdot c_{p,w} \cdot \Delta T_w \qquad (1)$$

In the evaporator the capacity transferred from the storage tank water to the refrigerant ranged between 29 up to 37.1 kW as presented in Fig 3. On the condenser side the minimum and maximum heating capacity was calculated 34.9 and 43.7 kW respectively.

The variation of the water source and sink side temperatures was approximately between 8.3-8.7 K, which resulted in a small range of operation conditions. Three points were selected to present the lowest and highest performance values. In Table 1 are presented three operating cycles that correspond to minimum and maximum source temperatures recorded (31.2 and 39.9 °C). The data from the operation during the maximum temperature lift (31.3 K) is also presented, which is calculated from the difference between the condensing and evaporating temperature.

Table 1. Experimental results from selected operating cycles.

			Tw,ev,i		ΔT_lift
			Min	Max	Max
Evaporator	Water inlet	°C	31,24	39,93	36,03
	Water outlet	°C	26,02	32,81	30,12
	Water flow	kg/s	1,33	1,25	1,31
	Evaporating temp.	°C	18,94	25,82	22,07
	Evaporating press.	bar-a	1,18	1,53	1,33
Condenser	Water outlet	°C	47,60	54,53	52,39
	Water inlet	°C	42,24	47,75	46,42
	Water flow	Kg/s	1,56	1,54	1,52
	Condensing temp.	°C	48,6	55,64	54,44
	Condensing press.	bar-a	3,19	3,9	3,7
Temperature lift		K	29,7	29,9	31,4

The refrigerant evaporating and condensing temperatures were calculated via REFPROP software [9] using as input values the low and high side pressures. The evaporating temperature and pressure ranged from 18.9 °C to 25.8 °C and 1.18 bar-a to 1.53 bar-a respectively. The condensing temperature and pressure resulted from 47.3 °C to 55.64 °C and 3.05 bar-a to 3.9 bar-a respectively. The pressure ratio calculated from the ratio of the absolute evaporating and condensing pressures resulted between 2.3 to 2.8.

The superheating degree at the suction line was controlled by the superheat controller of the EEV. It was set at higher value than in conventional systems due to the risk of saturation in the compressor´s suction port. The superheating and subcooling degree values ranged from 7.7 to 13 K and 7 to 9 K respectively.

Figure 4, shows the pressure enthalpy diagram of the cycles presented in Table 1. The green and red lines correspond to the operation during minimum and maximum source temperature. The blue line corresponds to the highest temperature lift recorded.

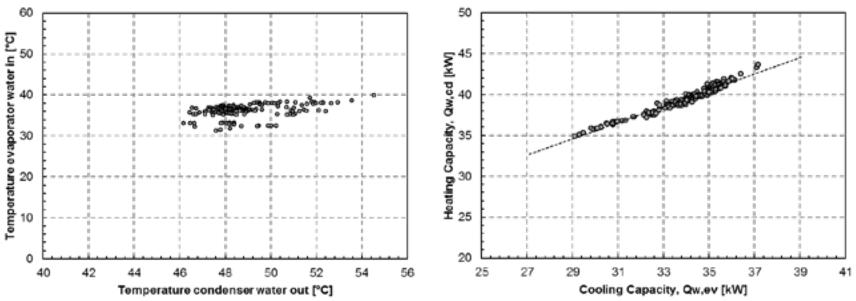

Fig. 3. Graphs of the temperatures of heat source and heat sink (left) and the heating capacity over the cooling capacity (right).

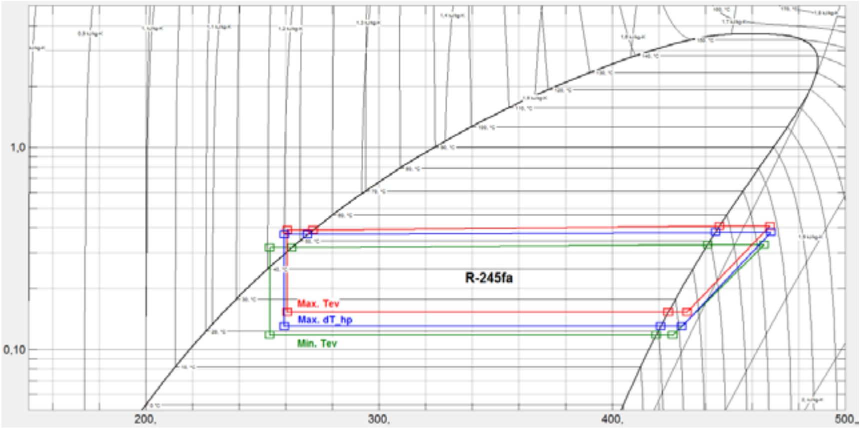

Fig. 4. Pressure and enthalpy diagram of R-245fa showing the operation of the HTHP under the conditions presented in Table 1.

During the monitoring campaign the speed of the compressor was maintained steady at 1500 rpm. The minimum and maximum suction and discharge temperatures recorded ranged from 26.8 °C and 35.3 °C to 72.2 °C to 77.3 °C respectively. The suction and discharge pressure ranged from 1.2 bar-a to 1.5 bar-a and 3.2 bar-a to 4.1 bar-a respectively. The power consumed by the compressor was measured in the range from 6.3 kW to 7.8 kW.

To evaluate the compressor performance the isentropic and volumetric efficiencies were calculated using Equations 1 and 2. The isentropic efficiency presents the ireversibilities during compression. The volumetric efficiency is the ratio of the volumetric flow rate over the volumetric displacement of the compressor. The volume in the suction (V_{su}) is calculated by REFPROP. The mass flow rate (m_r) is calculated from the ratio of the heating capacity at the condenser over the enthalpy difference between the refrigerant inlet and outlet in the condenser (Equation 3). The volumetric displacement (V_{displ}) is provided by the compressor manufacturer, the frequency (f) is 1500 rpm and the number of pistons (n) is six.

The minimum and maximum isentropic efficiency resulted from 45 % to 51 %. The volumetric efficiency was almost stable and ranged from 56 % to 58 % as shown in Figure 4 (left) in which are presented both efficiencies over the pressure ratio.

$$\eta_{is} = \frac{h_{dis,is} - h_{su}}{h_{dis} - h_{su}} \tag{1}$$

$$\eta_{vol} = \frac{m_r . V_{su}}{V_{displ} \cdot (f/60) \cdot n} \tag{2}$$

$$\dot{m}_r = \frac{Q_{w,cd}}{\Delta h_{cd}} \tag{3}$$

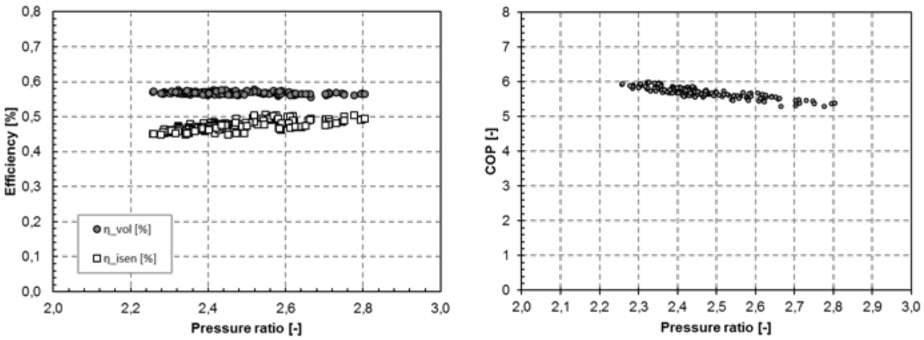

Fig. 4. Diagram of the isentropic and volumetric efficiencies over the pressure ratio (left) and the COP values of the HTHP over the pressure ratio (right).

The low isentropic efficiency values show that ireversibilities and losses are considerable under these operating conditions. Low temperature lift and overheating of the compressor could also be the causes of this inefficiencies. It is also observed that the volumetric efficiency is low. The low evaporating temperature and pressure conditions reduce the density of the fluid, reducing also the volumetric flow rate.

The COP of the heat pump was calculated by the ratio of the heating capacity by the power consumed by the heat pump (Equation 4). The results present a short range of values due to the also small range of operating conditions. The minimum and maximum calculated COP values resulted in 5.3 and 6 respectively. The power consumed by the compressor resulted from 6.3 kW to 7.8 kW. In Figure 4 (right) are presented the COP values over the pressure ratio of the heat pump.

$$COP = \frac{Heating\ Cap}{Power\ input} = \frac{Q_{out}}{W_{in}} \tag{4}$$

The total amount of energy stored in the seasonal storage during the datalogging period was 55.44MWh from which 34 % were lost through heat losses. The total amount of thermal energy delivered from the storage to the HTHP was 18.23 MWh and to the building 18.46 MWh. The total amount of energy consumed by the building for heating was 112.32 MWh in one year, from which 16.4 % was covered by the heat directly supplied from the seasonal thermal storage, 19 % was covered by the HTHP and 64.6 % was produced by the gas boiler. The total amount of electricity consumed by the compressor resulted in 3.73 MWh.

From the ratio of the total energy supplied to the building to the total energy consumption the Seasonal Energy Efficiency Ratio (SEER) was calculated 5.73.

According to the European Environment Agency from data of 2016, the emissions related to electricity generation are equivalent to 0.773 grams of CO_2eq per kWh [10]. The heat pump´s electricity consumption is equivalent to 2.88 tonnes CO_2eq per year. The emissions that have been saved by the heat pump considering a gas boiler instead would be 4.3 tCO_2eq per year. Therefore, total emissions of 1.42 tCO_2eq are saved by the HTHP during the year of operation. On the other hand, the emissions that saved from the heat provided by the seasonal thermal storage to the building are 3.72 tCO_2eq considering that gas boiler would be the heat source. In total the saved emissions of the full installation are 5.14 tCO_2eq in one year of operation.

5. Conclusions

The experimental data of one year of operation of a water-to-water high temperature heat pump which used heat from a STES tank to provide heat to a hospital has been presented. The HTHP was designed and manufactured using off-the-shelf components. The design operating temperatures are between 30-45 °C in the heat source side and up to 75 °C in the sink side. The working fluid used is R-245fa and the secondary fluid in both loops, the evaporator and condenser is water. The hot water produced by the heat pump is directed to a buffer tank of 1 m^3 internal volume and then delivered to the heating water grid.

Data was collected for a period of 12 months of the system´s operation. During this period the heat pump operated for a total of 9934 minutes which consisted in 209 operation cycles. The monitored values were measured and recorded every 30 seconds for 24 hours per day.

The water source temperature variated between 31.2°C to 39.9°C and the water outlet in the condenser from 46.2°C to 54.5°C. Both water pumps in the evaporator and condenser loop were maintained at steady speed and presented flow rates between 1.25 to 1.33 kg/s in the evaporator and from 1.52 to 1.56 kg/s in the condenser. The heating capacity of the heat pump variated between 34.9 kW and 43.7 kW.

The evaporating temperature and pressure ranged from 18.9°C to 25.8°C and 1.18 bar-a to 1.53 bar-a respectively. The condensing temperature and pressure resulted from 47.3°C to 55.64°C and 3.05 bar-a to 3.9 bar-a respectively. The pressure ratio calculated variated between 2.3 to 2.8.

The compressor´s isentropic efficiency resulted in values between 45 % and 51 %. The volumetric efficiency presented stable values that ranged from 56 % to 58 %. The compressor presented low performance due to the losses related to the high temperature operation. In addition, the fluid conditions at the suction line present low density which could be improved by increasing the evaporation temperature. The power consumption of he compressor resulted from 6.3 to 7.8 kW and the COP of the heat pump variated between 5.3 and 6.

In terms of the demo system, the total amount of energy stored in the seasonal storage during resulted in 55.44 MWh from which 34 % were lost through heat losses. The heat delivered by the seasonal thermal storage to the building corresponded to the 50.3 % and to the heat pump the 49.7 % of the total heat delivered. The total heat demand of the building was 112.32 MWh in one year, from which 16.4 % was covered by the heat directly supplied from the seasonal thermal storage, 19 % by the HTHP and 64.6 % was delivered by the gas boiler.

The total saved emissions from the heat pump operation comparing to a gas boiler operation in Poland are 1.42 tCO_2eq. The combined emissions saved from the heat delivered by both the heat pump and the thermal storage are 5.14 tCO_2eq in one year.

It is concluded that improvements on the control algorithm will certainly increase the seasonal performance of the heat pump and of the installation. During the operation cycles the heating water outlet temperature was controlled by the building´s control algorithm. The temperature of the buffer tank could not exceed the temperature set by the control causing short cycling of the heat pump.

The experimental results have demonstrated that HTHP systems are an attractive approach to use when low temperature heat source is available. In cases where the heat source is produced by renewable energy sources it could further improve the efficiency and the carbon footprint of the system.

Acknowledgements

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