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Long-term evaluation of an office building with large-scale heat pump and aquifer system in southern Sweden

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Abstract

IEA Annex 52 is the first long-term evaluation of large-scale heat pump systems. The Annex aims to survey and create a library of quality long-term measurements of GSHP system performance for commercial, institutional and multi-family buildings. The IEA Annex 52 also aims to provide a set of benchmarks for comparisons of such GSHP systems around the world.

The GSHP system in this report consists of two serial coupled heat pumps connected to a four-well aquifer and produces both heating and cooling. Total heating capacity is 0.3 MW. Due to a complex system with poor control system the Seasonal Performance Factor, SPF_{H4} is as low as 2.5 for the first year. Several control errors have been identified, including a potential short circuit of the serial DHW heat pump, erroneous operation of it for heating purpose and probable unnecessarily high condensation temperatures of the main heat pump.

The cooling system has high Seasonal Performance Factor, SPF_{C4} is 5.7 the first year. The system mainly uses free cooling, but performance decreases significantly at low load, possibly due to poor pump control. All system boundaries used are the IEA Annex 52 proposal

Keywords: System performance; System boundaries; Large-scale heat pump system; Annex 52 long-term evaluation; Performance Factor, Coefficeint of Performance; Office; GSHP; Sweden;

1. Introduction

1.1 Background

Measured long-term performance data for ground source heat pump systems serving commercial, institutional and multi-family buildings are rarely reported in the literature [1].

IEA HPT Annex 52 [2] is focused on long term performance measurement of GSHP systems serving commercial, institutional and multi-family buildings. Performance varies between different plants, and there is a need of more knowledge of the underlying causes. An important part of Annex 52 is to develop a methodology for measurement strategies and common system boundaries for larger heat pump systems. To achieve this 40 GSHP monitoring case are studied, covering a range of applications, located in seven countries.

It is important to make analyses based on a large amount of data and to identify key performance indicators when comparing different heat pumps.

More information with focus on factors that influence the performance is needed, identifying causes of low performance and unnecessary errors that can be avoided in the future. One example that causes low performance of a GSHP system is for example dimensioning issues and suboptimal control systems. Reasons of low performance needs to become visible, in order to get knowledge and understanding.

The over-all performance of a GSHP system is affected by the performance of the source side ground circuit, as well as the heat pump unit performance and the load side circuit performance, including supplementary heating and cooling. The varying design and complexity of GSHP systems poses challenges in comparable performance factors.

Detailed long-term analyses of large GSHP systems for commercial, institutional and multifamily buildings are rare. This article describes one of the 40 GSHP monitoring case studies and focuses on evaluating an aquifer heat pump system. The system design is described, and Seasonal Performance Factors (SPF) are analyzed from 2 years of measurement data.

1.1.1 System boundaries

The IEA Annex 52 project has identified the need of and proposed system boundaries for large-scale GSHP system [3] based on the 2012 SEPEMO work [4]. The system boundaries are shown in Figure 1 below. The system boundaries used in this report are 0, 1 and 4. System boundary 0 is the aquifer system, including pumps and are discussed in chapter 3.1, system 1 is inside the heat pump cabinet and is estimated in chapter 3.2. The project has good measurements for system boundary 4, this is examined in chapter 3.3

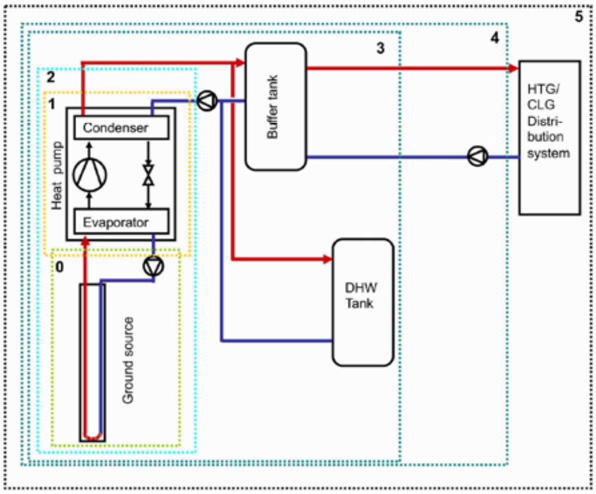

Figure 1 System boundaries according to the IEA Annex 52 project proposal (preliminary)

2 Methodology

One aim of the project is to aid the Annex 52 project in finding relevant system boundaries and test them. Another is to evaluate the performance of the GSHP system and find Performance Factors on the different system boundaries, whenever possible to calculate. This project has historic data from 2012-2014, data that previously was analyzed by RISE in an earlier, different project. No newer measurements have been used, but few changes have been made to the system until today, meaning the data is still valid for the system. The data set consist on electrical energy metering on each heat pump and summed on all circulation pumps and energy metering on aquifer, cooling, heating, subcooler heat and Domestic Hot Water, DHW, all on hourly basis. Temperatures were only measured at the four wells of the aquifer; no additional temperature data is available. Outdoor temperatures were taken from the SMHI [5] weather station at the nearby airport, corrected due to the site’s much closer distance to a large lake and the elevation difference (140 m). The temperature was corrected +0,8°C, which was the annual temperature difference between the now closed city weather site and the still operating airport weather site, when comparing historic data. This approach is not perfect, as the variation is great and the temperature difference is slightly higher in the winter, but no better correction was found.

To understand the project a site visit was performed in October 2019.

2.1 Object

The heat pump and aquifer system serving an office building is located in southern Sweden, with undisturbed ground temperature of about 6 °C. One advantage offered by aquifers is the seasonal storage and another that different temperatures effectively could be stored separately, by using cold and warm wells. Essential for good performance is to optimise temperature-levels, for optimal use of the advantageous temperatures of aquifers. The system is described in Figure 2.

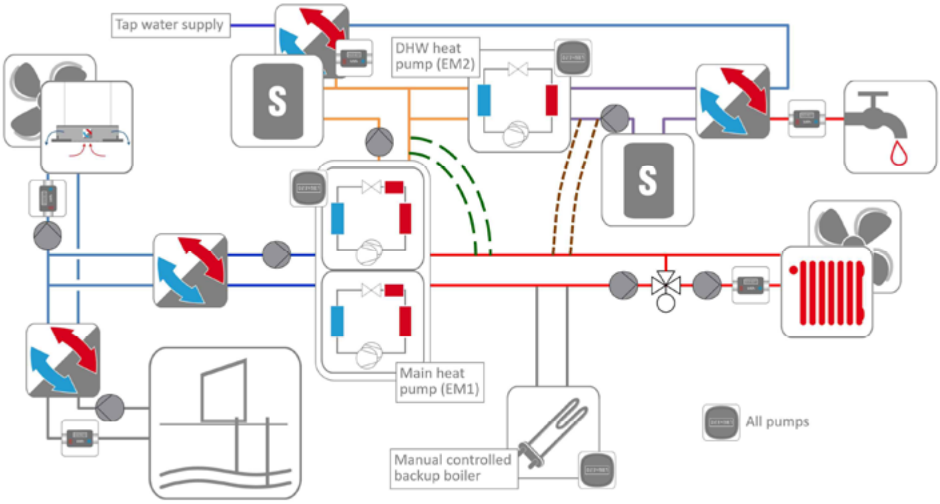

Figure 2 The heat pump and aquifer system. The Main heat pump (EM1) is using the aquifer as its heat source, while the DHW heat pump (EM2) is using the subcooler of the Main heat pump as its heat source. Dashed green and brown lines show possible heat flows, flows that are not intended according to the documentation, but are fully possible. All functionality is not described and especially the dashed lines are symbolically drawn. Pictograms by TU Braunschweig IGS, used with permission within the course of IEA HPT Annex 52

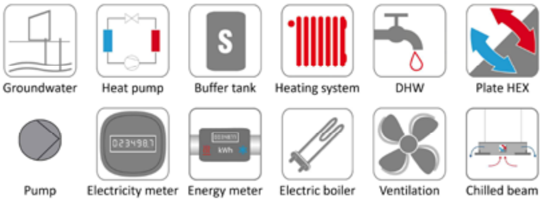

Figure 3 Explanation of pictograms used in Figure 2. Note that the heat pumps used are all dual compressor heat pumps and the main heat pump has subcooler heat exchangers. Pictograms by TU Braunschweig IGS, used with permission within the course of IEA HPT Annex 52

The main heat pump (EM1) has a heating capacity of about 300 kW and has four compressors in two refrigerant circuits, see Figure 2. The main heat pump (EM1) uses the aquifer as the heat source, while the Domestic Hot Water (DHW) heat pump uses the subcooler of the main heat pump as the heat source. The condenser of the main heat pump (EM1) heats the building, but with a change of two valves, the DHW heat pump can also heat the building. At times with to low heating capacity of the subcooler, the condensor has the possibility to heat the DHW as well, see dashed curved lines of Figure 2. The cooling system in the building is, via two heat exchangers, connected to the aquifer and the evaporator of the main heat pump. According to design documentation the main heat pump is used for heating, with temperature aid from the DHW heat pump during winter conditions with need of high heating system temperatures. The subcooler has a setpoint of 20°C, giving the DHW heat pump the possibility of high Coefficient of Performance (COP) when producing DHW.

The setpoints of the heating system is 35°C at an ambient temperature of 0°C and 50°C at dimensioning ambient winter temperature (-18°C), meaning a relatively low temperature system.

The strategy of the control system is to keep the condensing temperature of the Main heat pump (EM1) as low as possible. The main heat pump is used for heating only, the control system can not start it for cooling purpose, regardless of cooling demand. The cooling capacity is regulated by the flow from the aquifer, it has no dependency control wise to the heat pump. There is a need of cooling all year round. The cold wells of the aquifer are used at ambient temperatures above 12 °C, while the warm wells are used below 10 °C. There is no other hysteresis in changing the well of the aquifer, meaning the wells will be changed many days of the year. According to the site visit the wells are now (2019) changed manually instead.

2.2 Method

2.2.1 Calculation method

An estimated Performance Factor for heating on system boundary 1, PF_{H1}, for the main heat pump was calculated using the following formula:

$$PF_{H1} \approx \frac{Q_{Heating\ system} + Q_{Subcooler\ DHW} + Q_{DHW} - P_{EM2}}{P_{EM1}} \tag{1}$$

The only measurement done on the system boundary 1 for the main heat pump (EM1) is its own power consumption, P_{EM1}. The heat from the subcooler to preheat the DHW, $Q_{Subcooler\ DHW}$ and the heat to the heating system, $Q_{Heating\ system}$, are measured on system boundary 4. All energy supplied by the system originates from the main heat pump, the DHW heat pump is only increasing the temperatures. This means that the power consumption of the DHW heat pump, P_{EM2}, should be subtracted from the total heat supplied to calculate the performance factor of the main heat pump (EM1), PF_{H1}. As losses in the system are not stringently taken into account this can only be considered an estimate.

To be able to calculate Performance Factors on system boundary 4 the power consumption of the circulation pumps and the heat pump compressors must be allocated to the heating and the cooling side. This is done based on the heat and cooling produced for each hour of the year, according to the consensus in the Annex 52 project group. Power consumption allocated for cooling is calculated according to:

$$P_{Cooling,\ allocated} = \left[\frac{Q_{Cooling}}{Q_{Cooling} + Q_{Heating\ system}}\right] \cdot (P_{EM1} + P_{pumps}) \tag{2}$$

While power consumption allocated for heating is calculated according to:

$$P_{Heating,\ allocated} = \left[\frac{Q_{Heating\ system}}{Q_{Cooling} + Q_{heating}}\right] \cdot (P_{EM1} + P_{pumps}) + P_{EM2} \tag{3}$$

All circulation pumps are measured combined with one electrical energy meter; thus, the dedicated cooling or heating system pumps could not be allocated directly to their side of the system. All circulation pump power consumption is thus allocated according to the formulas, as this was the best possible method. The compressor power consumption of the EM2 is only used to produce DHW and does not cool the aquifer, all that power consumption is allocated to the heating side.

The calculation of the Performance Factor for heating on system boundary 4, PF_{H4}, is calculated with the following formula:

$$PF_{H4} = \frac{Q_{Heating\ system} + Q_{Subcooler\ DHW} + Q_{DHW}}{P_{Heating,\ allocated}} \tag{4}$$

The calculation of the Performance Factor for cooling on system boundary 4, PF_{C4}, is calculated with the following formula:

$$PF_{C4} = \frac{Q_{Cooling}}{P_{Cooling,allocated}} \quad (5)$$

In this paper performance factor SPF and DPF are used, were SPF is calculated over one year and DPF is calculated over 24 hours.

3 Results (Discussion)

The office building has a heating and cooling load according to Figure 4 below, the diagram has 24h average values due to low resolution of energy data.

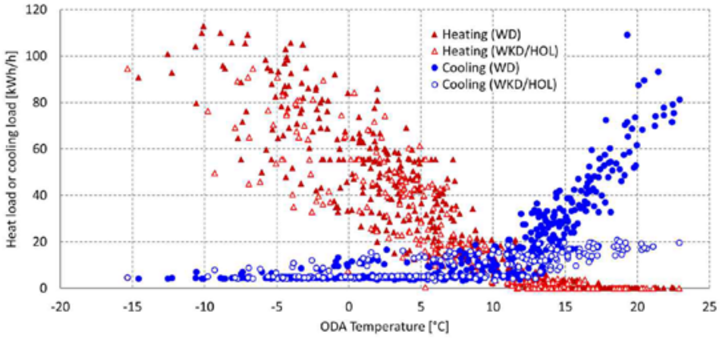

Figure 4 Heat and cooling load as a function of outdoor temperatures, average over 24h

As seen in Figure 4 the cooling load is highly dependent on the activities of the employees, during weekends and holidays the cooling load is a fraction of the workdays. The same is not seen in heating, but the variations at the same outdoor temperatures is very large. This could be due to the weather (sun/cloud) as the building has very few shadowing buildings surrounding it, but no data could verify this.

A thermal balance of the entire system was done, showing good results, see Figure 5. The negative side is sources of heat: The warm well of the aquifer, the compressors of the two heat pumps and the pumps in the system. The positive side is the sink of heat: The cold wells of the aquifer, the heating system and the Domestic Hot Water (DHW)

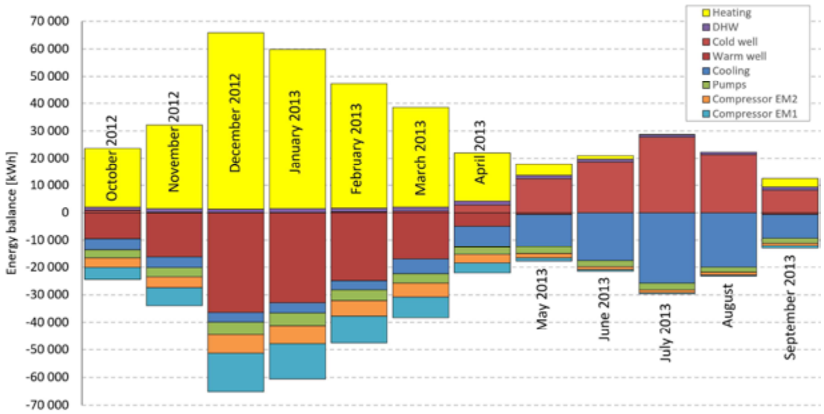

Figure 5 Thermal balance of the GSHP system, same months shown as in Figure 6

Total pump power consumption is high at low load conditions, especially compared to compressor power consumption.

3.1 IEA Annex 52 system boundary 0

The energy extracted and injected into the aquifer was measured, but as seen Figure 6, the resolution is low (100 kWh per pulse). As the pump of the aquifer were not measured separately no Performance Factor could be derived. In order to express the energy in kWh/h a 24-hour average value was calculated.

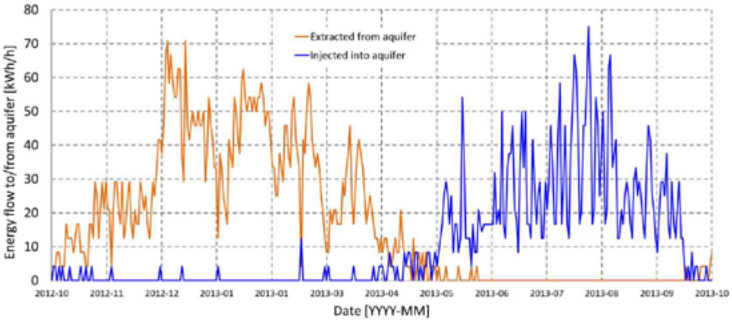

Figure 6 Heat extracted from and injected into the aquifer during one full year, mainly the first year in the time series, 24h average.

As seen in the diagram the pumps are at several occasions reversed, especially in April, May, September and October. The16th of February, a cold day, the pumps reverse, and the cold well was used, clearly seen in Figure 6. The control system didn't handle the aquifer optimal, as mention in chapter 2.1 the flow direction of the aquifer was change just according to ambient temperature, meaning the flow direction changes often in the spring and autumn. According to a site visit (2019) this was now handled manually instead.

3.2 IEA Annex 52 system Boundary 1

3.2.1 Heating

The measured data of the office building is not enough to calculate the performance on individual heat pump level but it could be estimated by subtracting the power consumption of the DHW heat pump (EM2) from the total heat produced, see equation 1 in chapter 2.2.1. The only function of the DHW heat pump is to increase the temperature from the main heat pump (EM1), EM2 has the main heat pump (EM1) as its only heat source, see Figure 2. The heating Seasonal Performance Factor on system boundary 1, SPF_{H1} is estimated to 4.2 the first year and 3.7 the second year, when the DHW heat pump has a better control strategy. See chapter 3.4 for further details and discussion.

3.3 IEA Annex 52 System Boundary 4

On system boundary 4, see to Figure 1 for definition, meaning the entire heat pump installation, excluding the heating, cooling and ventilation system of the building itself, all necessary data has been measured over the period September 2012 to August 2014. The allocation of pump and compressor power consumption is seen in equations 2 and 3, while the Performance Factors are calculated according to equations 4 and 5, all in chapter 2.2.1

3.3.1 Heating

The following diagram over Daily Performance Factor, DPF, see Figure 7, has been developed to fully understand the performance of a heat pump system at a glance. It directly shows that the heat pump system, with four compressors in the main heat pump (EM1) and a total heating capacity of 300 kW, is oversized and only uses one compressor in on/off mode most of the year. The maximum capacity used is 110 kW or 37 % of installed capacity, despite both winters during the period having low temperatures. It also directly shows that the heat pump system has almost the same performance factor regardless of heat output and that it at very low output mainly collapses in performance.

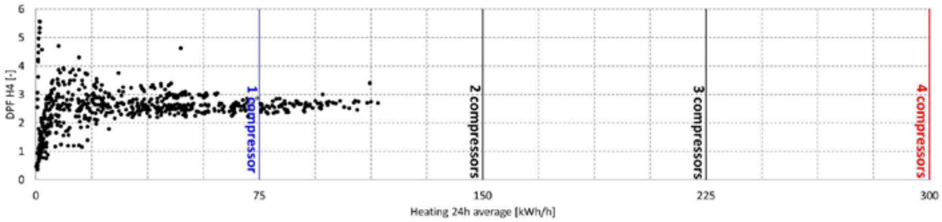

Figure 7 Overview of heat pump system performance factor H4 as a function of heat provided to the office building, blue line shows lowest capacity without ON/OFF operation, red line maximum heating capacity at highest heating system temperatures (winter)

The heat pump system has been compared to a state-of-the-art heat pump system produced by a major Swedish manufacturer in order to show the expected performance for a well-designed system, see Figure 8 below.

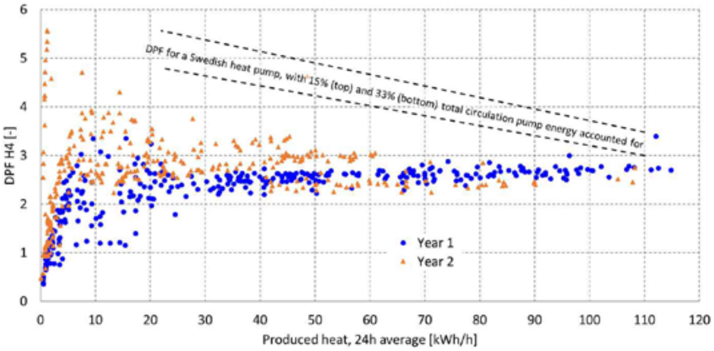

Figure 8 The heat pump system heating Daily Performance Factor, DPF_{H4}, compared a state-of-the-art heat pump system

The second year the control system is changed September 16th, 2013, it is noted as a significant decrease of power consumption of the DHW heat pump (EM2), but exact implication of the change is not known. This clearly increases performance in the medium output region, but the change is reversed between January 15th 2014 and March 4th 2014, giving poor performance in the high output region again. The change is likely done for the system to be able to deliver high enough temperature to the heating system, but that has not been possible to verify, as temperature data is not available.

The heat pump system is within the range of the total pump energy seen in Figure 8 above (dashed lines), 15-33 % of total compressor power consumption, above 40 kWh/h average produced heat. Below 40 kWh/h pump energy increases dramatically and is 90% at lowest heating load. This indicates a systematic error in the control of circulation pumps, which could not be explained by the nature of the aquifer system.

It is difficult to understand that the DPF_{H4} is below one (1) at low capacity, worse than resistive heating, but the control of the aquifer pumps is likely a major cause. The exaggerated pump power then gives very little heat to the brine system, as little is being needed, the losses in the water-cooled pumps basically heats the aquifer instead.

The Seasonal Performance Factor, SPF_{H4}, is calculated to 2.5 for the first year and due to the change in the control system the SPF_{H4} increased to 2.7 the second year. For a well-designed heat pump system, the SPF_{H4} is estimated to be able to reach 4 in the office building, around 50 % higher than the system used.

3.3.2 Cooling

The heat pump system is used to cool the building, it has a fairly constant equipment cooling load of 4-6 kWh/h clearly seen during heating season, but the main cooling load is during the short Swedish summer in June, July and August.

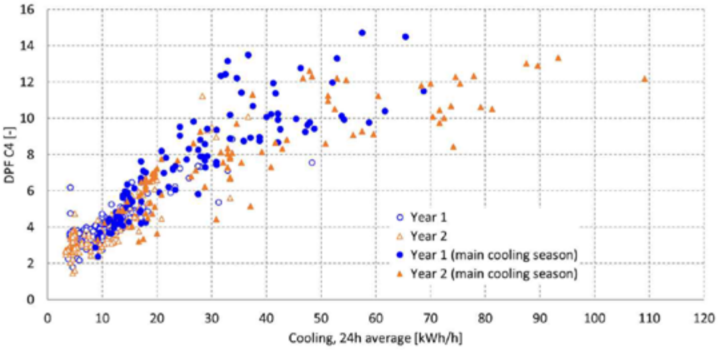

Figure 9 The heat pump system cooling Daily Performance Factor, DPF_{C4}

As seen in Figure 9 the Daily Performance Factor, DPF_{C4}, is high or very high during most of the year, but at low load the performance is in parity with conventional chiller performance. It is clear from the data that the cooling is mainly produced directly from the aquifer, by means of free cooling, else this high performance would be impossible. The low performance at low load is likely due to poor control of the aquifer pumps, note that the performance is low at low load even at the main cooling season, with very low heating demand, see filled markers in Figure 9. The performance is thus not mainly an effect of the method used for allocating the pump electricity. Note that the power consumption of the DHW heat pump (EM2) is not affecting the cooling Performance Factor, it is accounted on the heating side only, see chapter 2.2.1.

A well performing aquifer system, with good pump management at lower load, could perform at the top performance seen in Figure 9 most or all of the cooling season.

The Seasonal Performance Factor, SPF_{C4}, is calculated to 5.7 for the first year and slightly higher 5.8 the second year

3.4 The serial DHW heat pump (EM2)

The DHW heat pump (EM2) has the subcooler of the main heat pump (EM1) as its main heat source, it can not use the aquifer directly as a heat source. The DHW heat pump (EM2) is designed to mainly produce Domestic Hot Water, DHW, but can aid the main heat pump (EM1), only producing heat, to keep high enough temperature in the heating system. This means that the DHW heat pump (EM2), at DHW production, should use significantly less power than the main heat pump (EM1), else the thermal balance is not fulfilled. The reason being the subcooler of the main heat pump produce a fraction of the total condenser heat and varying depending on condensing temperature. The DHW heat pump has temperature wise good possibility to perform well, meaning with low power consumption.

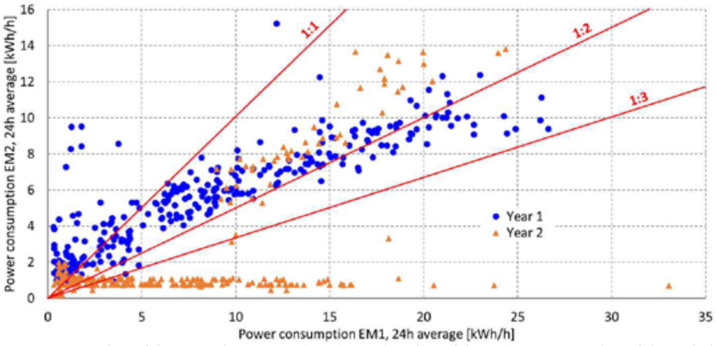

Figure 10 Power consumption of the DHW heat pump (EM2) as a function of the power consumption of the main heat pump (EM1). Only compressor power consumption measured, no pumps included.

Looking at the power consumption of the DHW heat pump (EM2) compared to the power consumption of the main heat pump, see Figure 10, the DHW heat pump (EM2) has too high power consumption. With all heat coming from the subcooler of the main heat pump, in a well-designed system, the ratio would be below 1:3 or at least 1:2 at all time, but mostly significantly lower. This is not the case, the main part of operation is at higher ratios, especially the first year and during the winter the second year. This likely means that the EM2 is heated by the condenser of EM1, causing the lower than necessary Performance Factor seen in Figure 7 and Figure 8. This is possible, circuit and control wise, see green long-dashed lines of Figure 2. Noteworthy is the extreme low ratio in the left side of Figure 10. It is identified that the condenser of the DHW heat pump (EM2) has the possibility, control and circuit wise, to heat its own evaporator, this possible short circuit could be the reason for this extreme operation, see green long-dashed lines in combination with brown dashed lines of Figure 2.

The line of orange rectangles along the x-axis in Figure 10 is likely when the EM2 is focused on producing DHW, meaning the control system is operating correctly. In Figure 11 the DHW production as a function of power consumption of the same heat pump is seen.

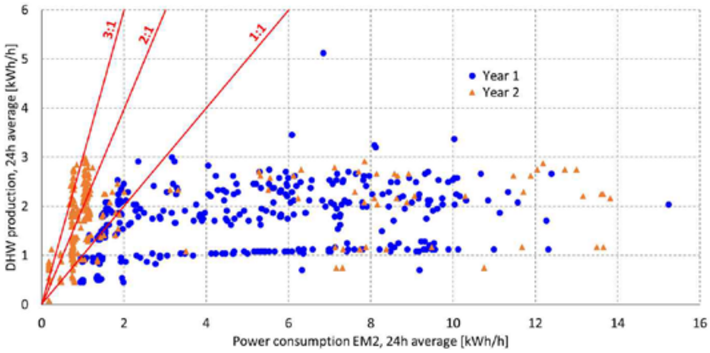

Figure 11 Domestic Hot Water production (DHW) as a function of power consumption of the DHW heat pump (EM2)

It is clear that the DHW heat pump (EM2) is operating in heating mode most of the first year and during the winter of the second year, as the ratio is well below 1:1, meaning extensive unaccounted losses in the DHW system as the only other possible solution. As far from full heating capacity of the main heat pump is not used, see Figure 7, this is not optimal. One possible answer to this poor operation could be found on the nameplate of the main heat pump, it is a chiller with a maximum return temperature of 45°C, meaning the DHW heat pump possibly must aid it at higher heating system temperatures.

According to the nameplate of the DHW heat pump it is also a chiller, not a dedicated heat pump. It has the same maximum return temperature (45°C), meaning about 5-10°C lower than what is necessary to handle legionella temperatures according to Swedish legislation. This means the DHW heat pump continuously is operating outside the envelope stated by the manufacturer, to handle the legionella temperatures, if it does. At the site visit the return DHW circulation temperature was 8°C lower than the legislative requirement, and the cover of the heat pump was removed for simple reset when the machine tripped a high pressure side alarm.

The building used about 15 MWh of DHW per year, including DHW circulation losses, a fraction of the heat supplied (210-290 MWh). The DHW heat pump (EM2) is largely oversized for the purpose, giving it low possibility to handle DHW temperatures accurately.

3.5 The three-port rotary control valve

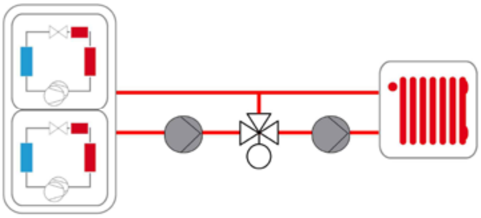

Figure 12 The three-port rotary control valve solution within the heating system, simplification of Figure 2. Pictograms by TU Braunschweig IGS, used with permission within the course of IEA HPT Annex 52

The three-port rotary control valve used between the main heat pump and the heating system, see Figure 12, leads to elevated temperature for the condenser, if not fully open at all time, causing lower Performance Factor, especially at lower load operation, seen in Figure 7 and Figure 8. At the site visit the temperature increased rapidly over the condenser, causing compressor stops after five minutes. Repeated short compressor operating time can cause premature compressor failures and will lead to transient losses [1] that, especially in combination with elevated condenser temperature, will lead to poor performance. The reason for the short compressor operating time is likely that the three-port rotary control valve decreases the working fluid volume for the heat pump to a minimum, by effectively cutting of the heating systems large fluid volume.

This three-port rotary control valve solution is not to be used in a well-designed heat pump system, except for decreasing temperatures to sub heating systems with lower operating temperatures.

3.6 Discussion

The heat pump solution underperforms according to expected performance for a heat pump with an aquifer heat source, heating a state-of-the-art Swedish office building with a low temperature heating system. The unnecessary complexity of the system, with a DHW heat pump in series with the main heat pump and unintuitive circuit layout means it is very difficult to understand when in operation. This was clearly stated by the personnel met at the site visit.

Due to lack of temperature data no relevant uncertainty analysis for the performance factors could be calculated

4 Conclusions

The GSHP solution in this report is complex, with a control system that most likely would need improvements to perform well. The 0.3 MW total heating capacity is almost three-fold the maximum heating demand seen during the two-year measurement period 2012-2014, despite having very cold winter days these years. This means only two of the four compressors in the main heat pump will be used, most of the year there is surplus heating capacity with only one compressor.

The DHW heat pump is seen operating many hours in heating mode, despite its purpose is for DHW. With an unknown change to the control system the second year, made twice, the DHW heat pump heated the heating system significantly less, meaning an overall higher performance factor for the system. Due to system design the DHW heat pump has the ability to heat its own evaporator, it has been identified that it likely does that during the first year. Moreover, the DHW was at times heated by the condenser of the main heat pump, causing lower performance factor, instead of the intention of using the subcooler heat.

A three-port rotary control valve is restricting the flow to the heating system and thus likely elevating the condensing temperatures of the main heat pump. This causes lower performance factors, but also leads to shorter than necessary run time for the compressors, being observed during the site visit. This solution should be avoided in heat pump heating systems. The Seasonal Performance Factor, SPF_{H4} is as low as 2.5 the first year and 2.7 the second year.

The compressors of the heat pump system are not controlled to run actively at cooling operation, the aquifer is then passively supplying the cooling to the building. This means a good potential for high performance factors, but it deteriorated with lower load, suggesting pump management is poor. The pump power consumption is significantly higher than needed at low cooling demand. The cooling system has regardless high Seasonal Performance Factor, SPF_{C4} was 5.7 the first year and 5.8 the second year.

No temperature data was available; thus no scientifically valid uncertainty analysis was possible to perform. All system boundaries used in the report are the IEA Annex 52 proposal.

Acknowledgements

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