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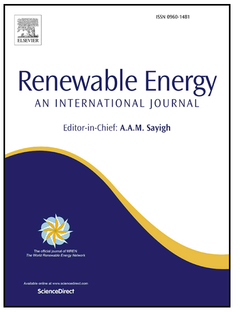

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Performance analysis of a large geothermal heating and cooling system

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Abstract

Ground Source Heat Pump systems can play an important role in reducing carbon emissions associated with building heating and cooling. The efficiencies and carbon emission savings achieved, partly depend on the optimization of the design, the control of the system and its reliability during extended operation. This paper reports the detailed investigation of the performance of a large system that includes fifty-six vertical borehole heat exchangers and four large heat pumps that provide both heating and cooling. High frequency data have been collected during the initial three years of operation that allow seasonal performance factors to be derived and detailed analysis of system operation. Annual performance has been found to be satisfactory overall but is highly variable depending on operating conditions and control system actions. A series of analyses have been carried out to investigate the roles of circulating pump energy, control system operation and dynamic behavior. A series of recommendations concerned with better design for part-load operation, reduction in pump energy demands and more robust control systems, are made with a view to improved system design and operation. Data from the study are being made available for further work on performance analysis and model validation studies.

Keywords: Geothermal, Ground Source Heat Pump, System Performance Factor, Borehole Heat Exchanger, Performance Analysis

Geothermal heating and cooling systems or ground source heat pump (GSHP) systems are seen as a low-carbon technology that can play a role in mitigating energy demands and carbon emissions from both residential and non-residential buildings. For example, in the UK, residential heat pumps and large-scale ground source heat pumps in heat networks have been proposed in the future of heating strategic framework [1, 2]. A recent worldwide survey [3] suggested there were more than 4.5 million systems in operation—mostly in residential systems but also a number of high capacity non-residential systems in Europe [4], China and North America [5].

For many countries, the effectiveness with which carbon emissions can be reduced relative to conventional heating and cooling technologies, is firstly dependent on the carbon emission factor of grid electricity and, secondly, on the efficiencies of alternative technologies in realistic operating conditions [6]. This is particularly the case for countries with relatively high usage of fossil fuels in the primary energy mix. Consequently, evidence-

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based policy and industry practice can benefit from long-term field monitoring and analysis of geothermal heating and cooling systems[7]. One of our aims has been to collect data that can contribute to this evidence in the case of larger systems used for both heating and cooling.

1. Geothermal heating and cooling system performance

The reported work on GSHP system monitoring and performance analysis varies in the scale of the systems studied, the prevailing climatic conditions, whether there is both heating and cooling, the level of monitoring, the performance metrics derived and level of technical analysis. Several field surveys of residential scale systems have been published [8] but relatively few concerned with non-residential systems and not of the scale reported here.

Following the introduction of incentives to promote residential heat pump technology (heating only) in the UK, a number of field surveys were carried out. The first of these monitored 83 residential heat pumps that included 43 GSHPs installed by fifteen manufacturers in the period July 2007 to June 2010. Poor performance in terms of seasonal heating efficiency in these trials prompted a number of follow-up studies. Dunbabin et al. [9] have presented the detailed technical analysis of the initial trial data. Subsequently a number of interventions were made and a further technical analysis carried out [10]. The studies identified a number of issues that had contributed to excessive use of auxiliary electrical energy such as excessive use of supplementary resistance heating and excessive pump energy demands. Other important issues relating to poor performance were questionable sizing of the systems and excessive cycling [6] as well as non-technical issues [11]. Although supplementary resistance heating is not relevant to large non-residential systems such as that studied here, some of the other issues will be shown to be so.

Gleeson and Lowe [8] conducted a meta-analysis of the most significant trials of residential heat pumps, reporting a total of 622 systems monitored in the collective results. These data mostly came from German, Swiss, Swedish and UK studies of systems used for space heating and domestic hot water production but not cooling. One of their main observations was that (at that time) a range of efficiency metrics were used in the various studies and this presented problems when trying to make comparisons between reported results. Since the time of their study, some consensus has emerged regrading definitions of seasonal efficiency metrics so that the Seasonal Performance Factors, SPF_{H2} and SPF_{H4} defined by SEPEMO [12] have become commonly used in reporting and regulation. After making some corrections, Gleeson and Lowe presented results for GSHP systems in the various studies in terms of SPF_{H4}. Mean values in these studies ranged from 2.5 to 3.75 with a mean value of 3.2 for the whole sample. They suggested better values were associated with houses with low temperature heating and relatively low domestic hot water demands. We address the definitions of SPF and how we have used them in this work below.

The most recent residential GSHP system field trials in the UK resulted in detailed data for 92 installations

[13]. The conclusions of this study were that efficiencies were better than the earlier UK trials with ranges and mean SPF_{H2} and SPF_{H4} values closer to the values reported in other European trials. The mean value of SPF_{H2} was reported as 2.99 and SPF_{H4} as 2.77 for the GSHP systems. The number of systems performing in the range of SPF_{H2} between 3 and 4 was also closer to other European studies.

Of the smaller non-residential systems that have been monitored and analysed, the system at Valencia Polytechnic University, Spain (the GeoCool system) is particularly noteworthy. The system is used for both heating (18kW capacity) and cooling (14kW capacity), provides energy to 12 fan-coil units and uses 6 borehole heat exchangers (BHE). Data has been used to evaluate the system and the performance of system models [14], to develop improved circulating pump controls [15] and to derive reference data sets that will allow borehole heat exchanger models to be evaluated [16]. They have also commented on performance improvement modifications such as the placement of storage tank and replacement of the one compressor with two smaller compressors.

Michopoulos at al. [17, 18] have reported the performance of the GSHP system at Municipality Hall of Pylaia in Greece. The GSHP system consists of 11 water-to-water heat pumps and the BHE that includes 21 boreholes of 80 m depth. The operation of the GSHP system was monitored for eight years starting in January 2003. Kim et al. [19] have presented the operating performance of a GSHP system in a school building in South Korea. The GSHP system includes ten heat pumps of capacity 29 kW and 24 boreholes of 175 m depth. The monitoring system recorded data over two years. It has been shown that the inverter controlled compressor was able to keep power consumption low during partial load operation and improve the Coefficient of Performance (COP).

There have been a very limited number of publications available on the performance analysis of large-scale GSHP systems. One of the first reports is by Witte and van Gelder [20] who monitored a GSHP system of a UK office building in Croydon, UK. This was the largest system in the UK at the time and had an installed heating capacity of 225 kW and cooling capacity of 285 kW. The BHE includes 30 boreholes each with a depth of 100 m. The work aimed to compare the actual operating conditions with the original design and the actual energy load and average borehole fluid temperature were considered in the detailed analysis. Zhai and Yang [21] have presented monitoring results for a GSHP system in Shanghai that includes 2 500 kW heat pumps and 280 boreholes. The system works continuously meeting the building heating and cooling loads in a cooling dominated archives building with a floor area of 8000 m^2. They report a summer COP of 4.7 and 4.6 in the winter season and estimate a 56% reduction in running costs compared to a conventional air-source cooling system. Although it is not explicitly stated in their paper, it is assumed these COP are defined in a similar way to SPF_{H1}, i.e. do not consider circulating pump energy.

We have concluded that, of existing reports on monitoring of large systems, no data regarding the behaviour of large borehole heat exchanger arrays have been provided and little comment has been made on the reasons why performance could not be improved in large systems. It has been our aim in this work to develop evidence for the operating efficiency of large non-residential geothermal heating and cooling systems and to provide analysis that identifies the technical barriers to achieving good efficiencies and effective system designs. It has furthermore been our aim to develop high frequency data sets starting at the beginning of system operation that can be used as reference data for model validation purposes. Data from our study has already been used in an analysis of borehole heat exchanger design procedures [22] and is openly available from an Institutional Data Repository [23].

2. Performance monitoring

In describing the work, we provide details of the building, it's heating and cooling systems and instrumentation used for monitoring. Further details are available in the corresponding doctoral thesis [24].

2.1. The building

The Hugh Aston building has a net floor area of 16,467 m^2 and comprises three linked wings of between seven and five storeys formed around the open central courtyard (Figs. 1 and 2). The building is relatively well insulated and built to airtightness standards applicable at the time of opening in 2010. The building includes a variety of accommodation including classrooms, academic offices, a mock courtroom, large lecture theatres, meeting rooms, library, IT facilities, and retail outlets. The building uses a hybrid ventilation approach so that only spaces with high occupancy or high internal gains are actively cooled. The GSHP system provides all Fan Coil Unit (FCU) and Air Handling Unit (AHU) chilled water demands of the building and part of the building space heating demand: all the underfloor heating demands. The remaining proportion of heating requirement is met by a natural gas fired boiler system. In principle, the heating and cooling demands are relatively well balanced: we comment on the monitored heat exchange in detail later.

Figure 1: The Hugh Aston building at De Montfort University, South elevation.

2.2. The ground heat exchanger array

The source side of the system is served by 56 borehole heat exchangers, each with a diameter of 125 mm and depth of 100 m. Fig. 3 shows the arrangement of borehole arrays in relation to the building. The average distance between adjacent boreholes is 5 m, and the boreholes are in 2 arrays with 19 located to the south of the building

Figure 2: The Hugh Aston building at De Montfort University, West elevation.

and the remainder installed below the central courtyard. Each borehole has a single U-tube inserted that consists of high density polyethylene (HDPE) pipe with an outer diameter of 32 mm. The borehole is partly backfilled with drill cuttings and grouted over the top 25 m. The geology is predominantly Marl. Thermal Response Test (TRT) data provided by the contractor has been analysed using parameter estimation methods [24] so that the best estimate is an effective ground thermal conductivity of 3.17 W $m^{-1}K^{-1}$. The undisturbed ground temperature was also derived from the TRT data and estimated to be 12.3 °C. Particular attention was paid in this work to ensure the ground heat exchanger temperatures were recorded from the very start of system operation.

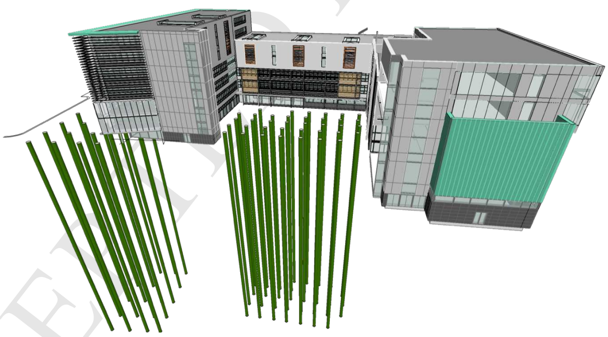

Figure 3: The Hugh Aston building at De Montfort University showing the two borehole arrays of 19 and 37 boreholes.

2.3. *The heat pump system*

The GSHP system has four water-to-water heat pumps [25] each with a heating capacity of 110 kW and cooling capacity of 120 kW that use refrigerant R410A. These are located in a basement plant room adjacent the courtyard as illustrated in Fig. 4. The heat pumps each have two single-speed scroll compressors and are fully reversible. Accordingly, they are configured to allow two-stage operation in either heating or cooling mode (nominally 55 kW of heating per stage and 60 kW of cooling per stage). The schematic arrangement of the heat pumps, circulating pumps and instrumentation is shown in Fig. 5. The system is designed so that each heat pump can add heat or cooling to the chilled water header or heating header system respectively depending on the valve positions and reversing mode. It was intended that no more than three heat pumps would be required to provide a design heating capacity of 330 kW and similarly no more than three heat pumps would be required to provide a design cooling capacity of 360 kW (the coincidence of peak heating and cooling not being expected). The combination of AHU and FCU equipment being used has resulted in design flow temperatures of 55 °C for heating and 6 °C for cooling. We comment on the monitored heating and cooling load balances later.

Figure 4: The heat pump installation in the basement plant room.

The source-side of the system consists of a borehole heat exchanger (BHE) array and header pipes to which the heat pumps add or extract heat as fluid is circulated by a variable speed circulating pump. Heat can be added or extracted concurrently by any heat pump so that the temperature of the fluid entering the ground loop is dependent on the balance of heating and cooling being demanded at a particular time. Each heat pump has its own micro-controller that determines compressor and reversing valve operation, safety interlocking and alarm monitoring. The circulating pumps and valves external to the heat pumps as well as the heat pump operating mode and sequence are controlled by a central Building Management System (BMS).

There are three sets of circulating pumps associated with the system, and these are for the ground loop, heating header, and chilled water header as indicated in the schematic (Fig. 5). The circulating pump installation is shown in Fig. 6. These variable speed pumps are controlled to vary the flow in steps according to how many heat pumps are operating. The maximum heat pumps operating for cooling operation at one particular time can be three and for the heating operation can be two. The maximum heat pumps operating at any particular time can be four. Consequently, the ground loop pump is controlled with four-speed steps: 53%, 69%, 85%, and 99%. The circulating pump on the cooling loop is operated in three speed steps: 70%, 80%, and 90%. The circulating pump on the heating loop is operated in two speed steps: 80% and 90%. These percentage speeds were determined during commissioning of the system to achieve the corresponding flow rates required. The nominal flow rates were 7.5 L/s, 15 L/s, 21.5 L/s for the ground loop, 6 L/s, 13 L/s, and 18 L/s for the cooling loop and, 9 L/s and 15 L/s for the heating loop.

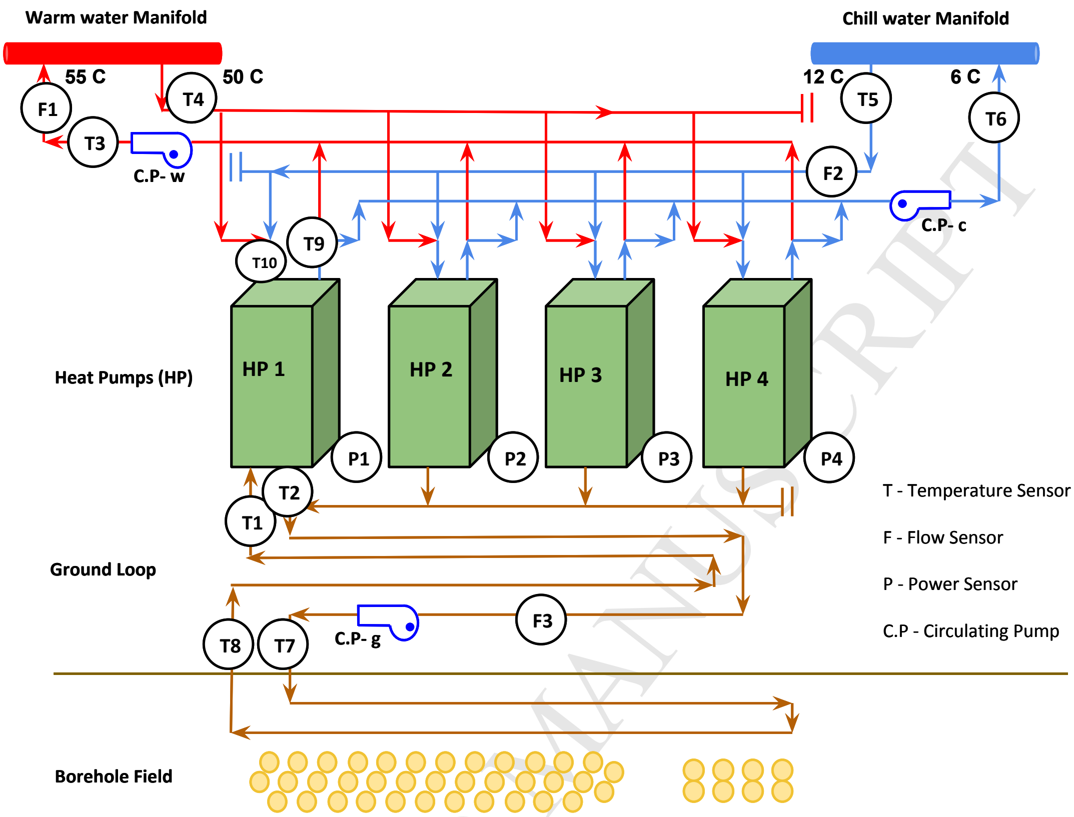

Figure 5: A schematic of the heat pump system showing the heat pumps, circulating pumps, circuits and sensor locations.

2.4. *Instrumentation and data collection*

Performance evaluation has primarily been made on the basis of heat transfer and power data. One aim has been to generate a reference data set and so measurements have been recorded at high frequency (one-minute intervals) for later sampling and integration. The monitoring of Borehole Heat Exchanger (BHE) started in January 2010 and continued up to February 2013. The primary temperature and flow rate measurements have been made with calibrated instruments working independently of the control systems. Further data such as the heat pump operating mode, compressor status, circulating pump speed, and flow switch positions have been collected from Building Management System (BMS) between January 2011 and July 2012. These data have provided additional information for the analysis of the dynamic operation of the GSHP system.

The fluid temperature measurements have been measured at the inlet and outlet of the borehole field, chilled water and warm water headers (see the schematic in Fig.5). Resistance Temperature Detector (RTD) sensors have high accuracy and repeatability and were chosen for the primary temperature measurements. Robust industrial pattern Pt100 RTD sensors are inserted into pockets in the plant room pipe work for all temperature

Figure 6: The system circulating pumps installation. Heating and chilled water header circuit pumps are in the foreground and the ground loop pumps in the background.

measurements. Each pair of sensors, including the cable and data logger channel, was carefully calibrated prior to installation in the pipes to minimise the error in measurements of temperature difference. A four-wire system is used for sensor connection with data loggers throughout the installation.

Three clamp-on ultrasonic flow meters are used to measure volumetric flow rates on the ground, heating, and cooling loops. As the heat transfer calculation requires mass flow rate, the volumetric flow rates on each loop (ground, chilled, warm) is multiplied with respective fluid's density value (ρ) to derive mass flow rate according to the glycol fluid properties data. The ultrasonic flow meters used have the advantage of being non-invasive and also of high accuracy ($\pm 0.5\,\%$ measured value). The calibration of the flow meter and correct installation has been verified by the manufacturer. Circulating pump power has been derived by using the known percentage speed setting and the measured flow rate. A typical sensor installation is illustrated in Fig.7. By using the manufacturer's pump curve information it was then possible to correlate flow rate and power demand to derive the required data.

Heat pump performance evaluation requires simultaneous monitoring of heat pump electrical demand data (compressor and parasitic electrical demands). This has been done by making use of the buildings integrated power meters and energy data collection system. The meters are class 1 pulsed power meters and have their data recorded every half-hour. Further data processing is required to align the data time signals so that the heat transfer data can be integrated to align with the corresponding half-hourly electrical demand data.

The total uncertainty of the heat transfer rates has been evaluated by combining all the uncertainties arising from each source of error in quadrature. Heat transfer depends on the volume flow rate ($\dot{v}$), density (ρ), specific heat (C_p), and temperature difference between flow and return points ($\Delta T = T_f - T_r$). Although the uncertainty in each measurement has been evaluated (see [24] for details) overall uncertainty is dependent on the size of the

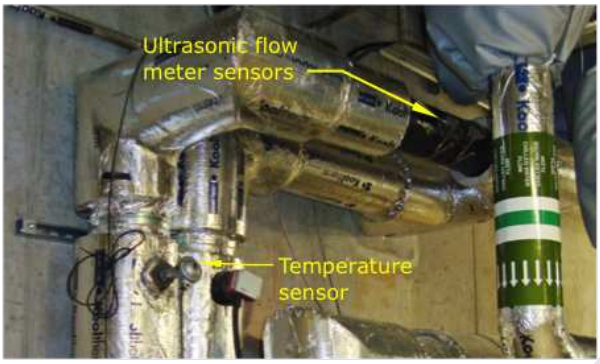

Figure 7: Temperature sensors and ultrasonic flow meter sensors installed in the ground loop pipework.

temperature differences over a particular period in relation to the uncertainty in the temperature measurements. The uncertainty in individual temperature difference measurements has been estimated based on repeated calibration tests. Evaluation of overall uncertainty has been estimated by considering the frequency of occurrence of temperature differences in the monitoring data. For both heating and cooling system operation, the uncertainty for heat transfer rate calculation is estimated to be no more than ±9.0 %. However, it can be said that more than 55% of occurrence the uncertainty is less than ±4.5 %. The uncertainty in the SPF data is estimated to be ±4.5 %. The monitoring data sequence was disrupted for a few short periods and in these cases the records were filled by making use of the available BMS temperature data or inferring flow rates from knowledge of the pump control status and temperature profiles.

2.5. Performance metrics

In analysing performance in this work, we have primarily used the Seasonal Performance Factor (SPF) metric and the particular definitions derived in the SEPEMO project [12]. The different definitions of SPF are dependent on the scope of the electrical demands included in the consumption data—increasing indexes indicating larger scope. The metric SPF_1 is associated with the heat pump compressor (and possibly parasitic demands from related controls and valves) but specifically excludes supplementary electrical heating or any circulating pump demands so that,

$$SPF_1 = \frac{Q}{(E_{HP1} + E_{HP2} + E_{HP3} + E_{HP4})}, \tag{1}$$

where Q is the heat transfer (i.e. delivered to the header system) in the period of consideration (kWh) and E_{HPn} are the heat pump electrical demands in the same period (kWh). SPF_1 can be expected to be closest to the manufacturers values of SPF measured in standard lab tests. SPF_2 also includes the ground loop circulating pump demands ($E_{CP,g}$) so that,

$$SPF_2 = \frac{Q}{(E_{HP1} + E_{HP2} + E_{HP3} + E_{HP4}) + E_{CP,g}}. \tag{2}$$

This metric is often used to compare GSHP efficiency with alternative heating or cooling sources (e.g., boilers or chillers). SPF_4 further includes the heating and cooling system circulating pumps ($E_{CP,h}$ and $E_{CP,c}$). In this work, we include the circulating pumps associated with the header circuits so that,

$$SPF_4 = \frac{Q}{(E_{HP1} + E_{HP2} + E_{HP3} + E_{HP4}) + (E_{CP,g} + E_{cp,h} + E_{CP,c})}. \tag{3}$$

SPF_{H3} is irrelevant here as there is no supplementary electrical heating to consider. In this work we further derive the SPF data by segregating between heating and cooling modes (heat transfer Q_H and Q_C) and designating these metrics SPF_{H2} SPF_{C2}, SPF_{H4} and, SPF_{C4}. The usual period of consideration in SPF metrics are seasons of operation (e.g. one annual cycle). Here we use both longer and shorter periods using the same definitions but with integration periods between one hour, one month, one year and the whole monitoring period as noted in the following sections and Figures.

3. Performance analysis

Our approach to the analysis of performance has been to evaluate the long-term behaviour of the ground heat exchanger array and the monthly demands of the heating and cooling systems. We present the SPF data in different integration periods before attempting further analysis to identify what operating conditions correspond to high and low periods of performance. We further attempt to evaluate the performance of a single heat pump to compare its performance with that stated by the manufacturer.

3.1. Ground heat exchanger responses

Hourly mean values of the ground loop flow and return temperatures, and circulating fluid flow rates in the 38-month period between January 2010 and February 2013 are shown in Fig. 8. Apart from a few months, the flow temperatures are higher than return temperatures, indicating a net effect of heat being rejected to ground. The ground loop return temperature varies between 11 °C and 19 °C and flow temperature changes between 9 °C and 27 °C. It is noticeable in the high frequency data that the response in the return temperature lags behind changes in the ground loop inlet temperature. This reflects the large thermal mass of the fluid in the circuit and the long nominal transit time.

The ground loop flow rate is activated in four nominal levels depending on the number of heat pumps running; as noted above. The flow rate data in Fig. 8 indicates the ground loop operates the majority of the time at the lowest flow rate. It can be observed that some days there are faults in setting the flow rate, particularly the end of July 2011 and start of August 2011 where the flow rate is around 3 L/s. The ground loop flow temperature attains peak values during abnormally low flow rates such as these. There is a period in August and part of September 2012 where the flow rate is zero. This is not a gap in the data, but rather a period where the system had to be shut down due to a leak being discovered in the ground loop (individual boreholes cannot be isolated in this particular array hydraulic design).

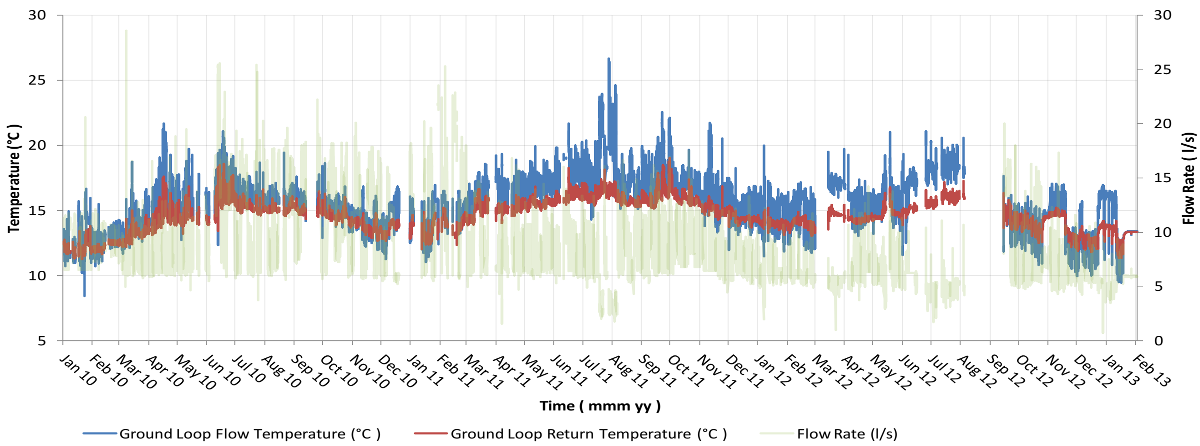

Figure 8: Variations in daily mean ground loop flow and return temperatures and flow rates during the monitoring period.

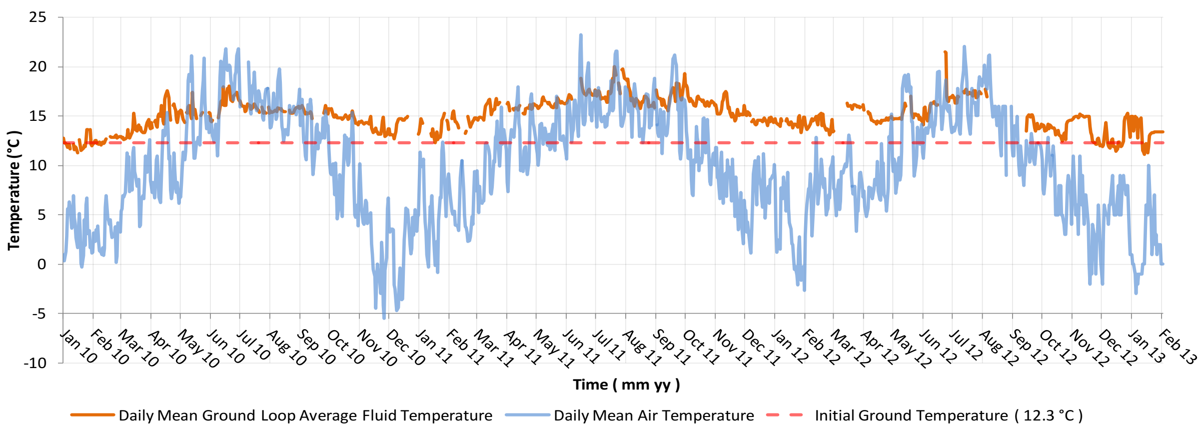

Figure 9: Variations in daily mean borehole and external air temperature during the monitoring period. The dashed line indicates the initial ground temperature.

The trends in ground heat exchanger behaviour are also shown in Fig. 9 where daily mean fluid temperatures are shown alongside the corresponding external drybulb temperatures and the initial ground temperature. If the peak summer temperatures are compared in 2010 and 2011, a slight increase is observed relative to the undisturbed ground temperature. This modest year-to-year increase can also be seen by comparing the minimum temperatures each January. In the year 2012, the ground loop return temperature drops to the level of the year 2010 due to less summer use of the system due to the interuptions noted above. The seasonal trends and relationship to climatic conditions are evident, with months with the highest fluid temperatures corresponding to high drybulb temperatures. In this system, it can be seen that the fluid temperatures are both noticeably higher than the coincident drybulb temperature in winter and lower than the coincident drybulb temperature in summer. This illustrates the advantages of ground-coupling rather than air-coupling of heat pumps.

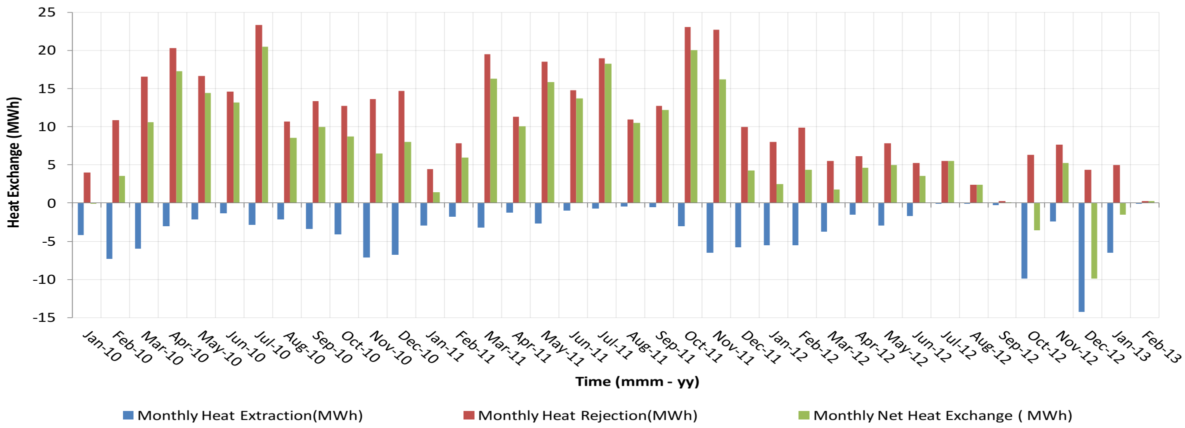

Figure 10: Monthly ground heat exchanges.

Monthly heat extraction, heat rejection, and net heat exchange for the BHE over three years is shown in Fig. 10 and is derived from monthly integration of the minutely heat transfer. The lowest monthly heat rejection is 0.27 MWh during September 2012, and the highest is 23 MWh during July 2010. The maximum monthly heat extraction is 14 MWh during December 2012, and the lowest is 0.003 MWh during July 2012. There is persistent heat rejection during the winter as some of the cooling loads are related to student labs with high densities of computer equipment. The trends in monthly heat exchange correspond to the trends in system temperatures noted above.

The study of ground behaviour consequently reveals that the BHE is predominantly used for heat rejection rather than heat extraction. When heat exchange is expressed as mean hourly heat transfer rate, the maximum heat extraction was found to be 53 kW and the maximum heat rejection to be 73 kW. These rates are substantially less than the heat pump design capacity. Similarly, the flow rate data indicates that the number of times more than one heat pump is required to meet heating and cooling loads is relatively small. For heat rejection, the maximum rate of heat exchange per unit depth of borehole is 13.3 W m^{-1}K and for heat extraction it is 9.4W m^{-1}K. These W/m values are very low compared with other installations and it can be said that the BHE is underused. No data is available to say what was assumed regarding the extremes of fluid temperature or heat exchange rates at the design stage. We conclude that the ground heat exchange system is within its capability and is behaving well. We further conclude that any poor system performance is not related to the limitations of the ground heat exchanger system.

3.2. Energy demands

The analysis of both energy demands and system efficiencies covers the period from February 2010 to July 2012: the last period of disrupted summer operation is excluded. In this work, system heating and cooling demands are measured at the supply headers. Consequently, heat added by circulating pumps as well as the

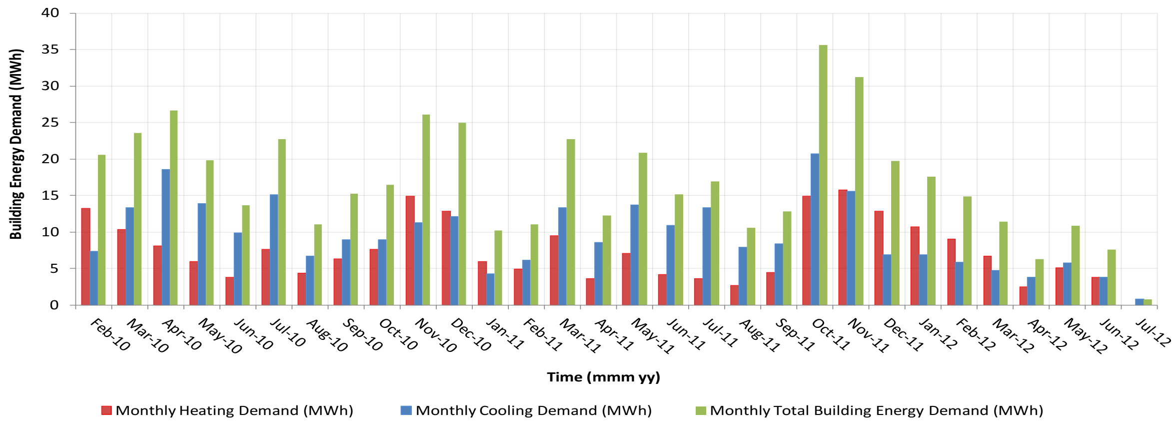

Figure 11: Monthly building system heat exchanges.

compressors is taken into account. Monthly heating, cooling, and total system heat exchange is shown in Fig. 11. The monthly system demand varies between 6 MWh and 36 MWh. The monthly heating demand varies between 3 MWh and 16 MWh, and monthly cooling demand is found to be between 4 MWh and 21 MWh. The peak demands are consistent with the corresponding ground loop heat exchange (noted above and in Fig. 10) after taking account of the compressor heat being added or subtracted in cooling and heating modes respectively: maximum monthly heat extraction was 14 MWh and heat rejection was 23 MWh respectively.

3.3. System efficiencies

Figure 12 summarises overall monthly Seasonal Performance Factor SPF_1 and segregated between heating and cooling modes, SPF_{H1} SPF_{C1} respectively. It can be seen that in all months SPF_{C1} is greater than SPF_{H1}. This is expected as the temperature lift (difference between source-and load-side fluid temperatures) is, by design, greater for heating than cooling operation. The monthly SPF_{C1} values are consistently higher during the first year than the second year. Correspondingly, monthly SPF_{H1} are lower during the first year than the second year. There are wide ranges in the values: SPF_{C1} varies between 3.24 and 5.61; SPF_{H1} fluctuates between 1.68 and 4.07. The heat pump catalogue data for specific operating conditions suggest higher efficiencies than these might be expected. Even though there are occasionally small values like SPF_{H1} for the month June 2010, the median values for SPF_1, SPF_{C1}, SPF_{H1} are calculated over the monitoring period as 3.54, 3.98, 3.3 respectively.

Figure 13 compares the monthly SPF_1, SPF_2, SPF_4 values fluctuating over the monitoring period. The general relationship between these metrics is that values are smaller with higher indices as more circulating pump demands are included. However, the ratios of the metrics is noticeably variable. For example, the difference between SPF_1 and SPF_2 is greater in months like July 2012 and difference between SPF_2 and SPF_4 is larger in some months like April 2011 and smaller in some months like June 2010. We investigate the role of variations in pump demands and operating patterns in detail below.

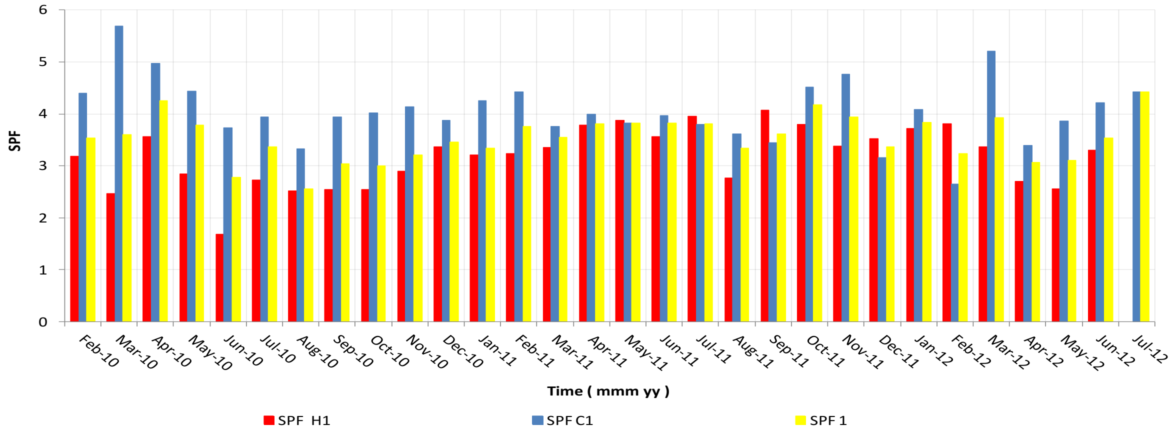

Figure 12: Monthly variations of overall SPF_1 and for heating and cooling modes, SPF_{H1} SPF_{C1}. These SPF metrics exclude consideration of circulating pump energy.

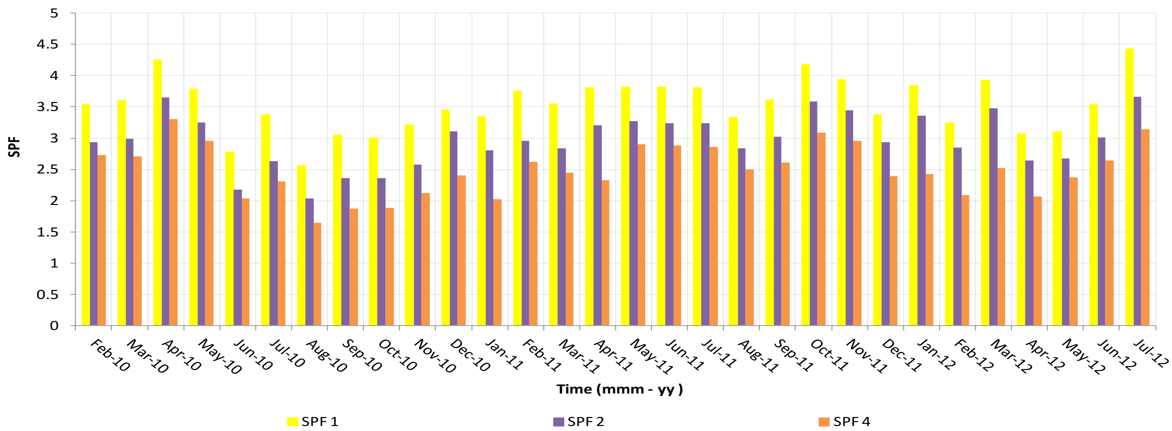

Figure 13: Monthly variations of SPF_1, SPF_2, SPF_4.

The seasonal values of the SPF metrics that have been derived from the monitoring data are summarised in Table 1. The first two cases, May 2010 to April 2011 and May 2011 to April 2012 represent the first and second whole years respectively. The third case gives the value for the whole two and half years monitoring period from Feb 2010 to July 2012. The SPF_{H1} increases from 2.89 in the first year to 3.55 in the second year, but SPF_{C1} decreases from 3.99 in the first year to 3.87 in the second year. This could be expected due to some seasonal increases in array temperature in favour of heating operation. However, pump operation and control operation between each year were known to vary and so are considered further below.

For the whole period, Feb-2010 to July-2012, the derived SPF_2 value, which includes ground loop circulating pump energy, is 2.97, and SPF_4 which further includes all circulating pump energy is calculated to be 2.49. These values can be considered in comparing this type of system with alternate technologies and relative energy costs[10]. The primary questions that arise from these findings are: (i) why are the values much lower than the 'headline' figures given by the manufacturers, (ii) what accounts for the large difference between the different SPF levels, and; (iii) what accounts for the variability between different operating periods? We investigate these issues in the following sections.

Period	SPF_{H1}	SPF_{C1}	SPF_1	SPF_2	SPF_4
May 2010 to April 2011	2.89	3.99	3.31	2.69	2.22
May 2011 to April 2012	3.55	3.87	3.67	3.16	2.61
February 2010 to July 2012	3.19	4.06	3.54	2.97	2.49

Table 1: A summary of SPF metrics over different periods of operation.

4. Detrimental effects on performance

4.1. Heat pump performance

One of the heat pumps was instrumented with additional inlet and outlet temperature sensors (HP1 in Fig. 5). Although it did not have a dedicated flow meter, by selecting the data when it is known that this was the only heat pump in operation it was possible to analyse its performance in further detail. We have sought to evaluate whether the heat pump was performing as the manufacturer's data suggests. We have constructed a parametric model that predicts heat transfer rates and power demand given the inlet temperatures and flow rates on the source and load sides. This is the same form of 'curve-fit' model with model coefficients derived from the manufacturer's tabulated data as that used in many building simulation tools: in this case the details are the same as EnergyPlus [26, 27]. The results of fitting the linear model to the catalogue data show that the data fit well—less than 2% RMSE between catalogue data and the model. The data is derived from steady test conditions and so the model is naturally steady-state. Given that the experimental fluid inlet temperatures have been measured, it is straightforward to use the model to predict the outlet temperatures on source and load side and compare these with the equivalent measured data.

The first thing we point out from the catalogue data is that there is a linear relationship between lift and SPF—as expected. The data are shown in Fig 14 and Fig 15 for heating and cooling respectively. The range of lift conditions during monitoring are also shown (essentially as the design requires). The heating design temperature of 55°C results in lift temperature differences of more than 30 K and leads to a noticeable detrimental effect on efficiency so that SPF_{H1} of 3.3–4.3 could be expected. The chilled water design temperature is in the range 6–12°C leading to lift temperature differences of 8–15 K and suggesting that SPF_{C1} values in the range 6–7.5 could be expected.

An example of a comparison between measured and modelled results is shown in Fig. 16 that shows hourly load-side heat transfer plotted for one day with predominant cooling operation. The deviation between measured

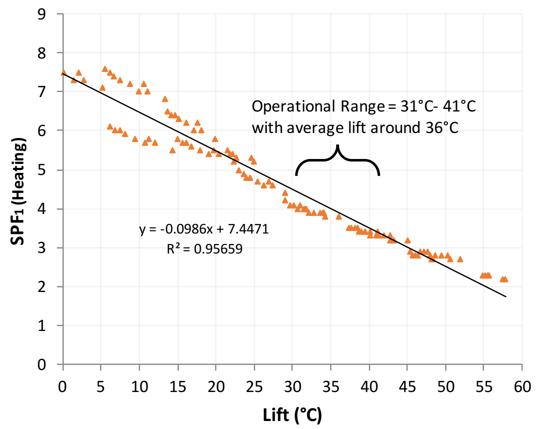

Figure 14: The relationship between SPF_1 and the temperature lift in heating mode shown in the heat pump catalogue data.

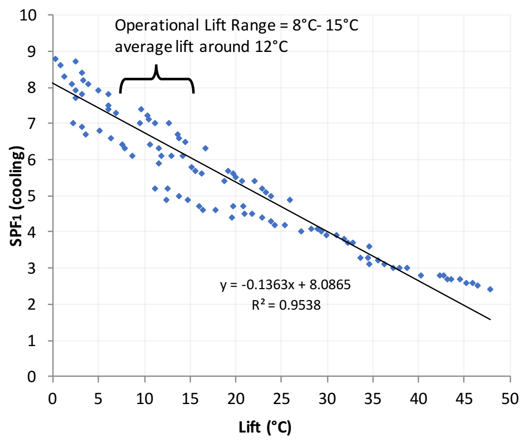

Figure 15: The relationship between SPF_1 and the temperature lift in cooling mode shown in the heat pump catalogue data.

heat transfer and those predicted with the steady-state model show a consistent under-performance of 7%. Similar deviation was found on most days of heating operation. This is a relatively modest level of under-performance in-situ compared to the predictions that were based on catalogue data. Some variation of this level could be expected due to variations in hydraulic conditions, fluid properties, heat exchanger fouling, variation in refrigerant charge or lubrication. However, this modest level of under-performance would not fully explain the low levels of SPF when considered over longer monitoring periods (Table 1).

4.2. Part-load operation

In this installation, the way in which heat pumps are controlled to match the required demands is by switching on or off individual compressors and stepping up or down the pump flow rates accordingly. The change in state between on and off is also dependent on the fluid temperature set points and dead-bands. At high loads a number of compressors may be operating and the off portion of the operating cycles relatively short with few cycles per

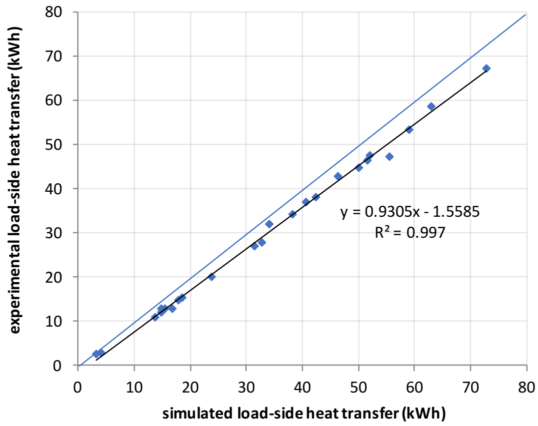

Figure 16: Measured hourly load-side heat transfer plotted against predictions for cooling operation.

hour. At low loads, it may only be necessary to operate a single compressor with the off portion of the cycle relatively long or the cycle time short. Some systems are configured so that cycle durations are not allowed to become of short duration or a certain period must pass before a heat pump can be restarted. This is not the case here. Consideration of the minutely temperature and flow rate records showed noticeable variation in operating cycle duration - some cycles being only a few minutes at certain periods. The impact of this behaviour has been analysed in a number of ways as follows.

When comparisons are made between modelled and measured heat transfer using data that includes a range of operating cycle durations, further variation in performance for a particular heat pump is revealed: such data is shown in Fig. 17. It can be seen that at short cycle durations the level of under-performance relative to the steady-state model increases to approximately 25% and there is a clear negative trend between differences in load-side heat transfer and cycle time. One would not expect the model to accurately predict the performance of the heat pump during the start-up and shut-down parts of the cycle when pressures and distribution of refrigerant is rather different to normal conditions. The data in Fig. 17 is consistent with this expectation but also shows actual performance is particularly sensitive to such effects.

The impact of the dynamics of the cyclic operation of the heat pumps on seasonal performance also depends on the frequency of occurrence of shorter or longer cycles during the period. Monitored SPF values have been segregated according to heating and cooling modes and the duration of each cycle. These SPF_{C1} and SPF_{H1} data are shown in Figs. 18 and 19 respectively. Considering the shortest cycle time data (0-10 minutes), these account for much of the SPF values found to be less than 3. The data corresponding to the longest cycle time (51-60 minutes) can be seen to be limited to SPF values greater than 3. In general, as cycle time increases, the occurrence of operating conditions with higher SPF also increases.

Part load operation has also been analysed by considering the frequencies of occurrence of system demands. The mean hourly load (heating or cooling delivered by the system) has been segregated into 10 kW bands in

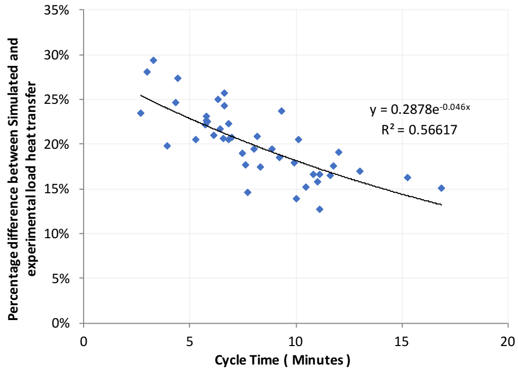

Figure 17: Variation in predicted heat transfer differences according to heat pump cycle time.

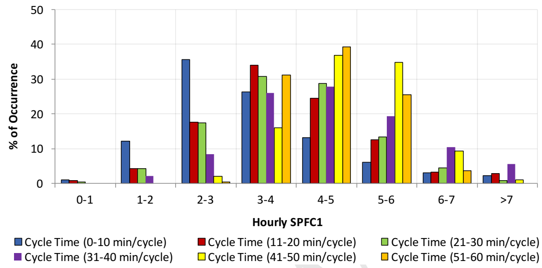

Figure 18: Occurrences of SPF_{C1} according to cycle time.

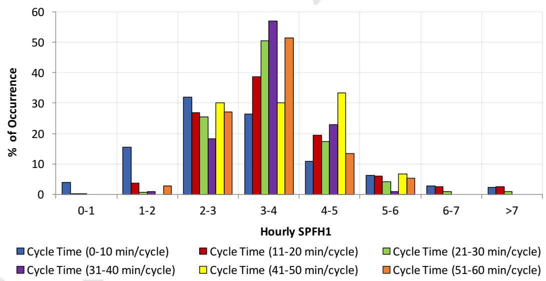

Figure 19: Occurrences of SPF_{H1} according to cycle time.

Fig. 20. Recalling that the nominal capacity of a single compressor stage is 60kW it can be seen that there is a high frequency of low loads such that for a large proportion of the time it was only necessary to use a single compressor stage and often using multiple cycles per hour.

The potential relationship between part-load conditions and efficiencies has also been analysed by calculating daily SPF values and considering variation with the daily heat exchange (both heating and cooling). This data is summarized in Fig. 21. One feature of the data is that there is larger variation (scatter) in the daily SPF_1 when the loads are low, with values as low as 2 and as high as 7. There are also a larger number of days at lower loads and relatively few at the upper end of the scale where multiple compressors are required for extended periods

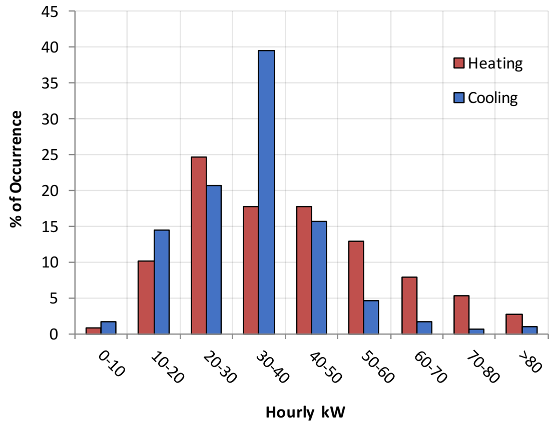

Figure 20: Occurrences of mean hourly load over the monitoring period.

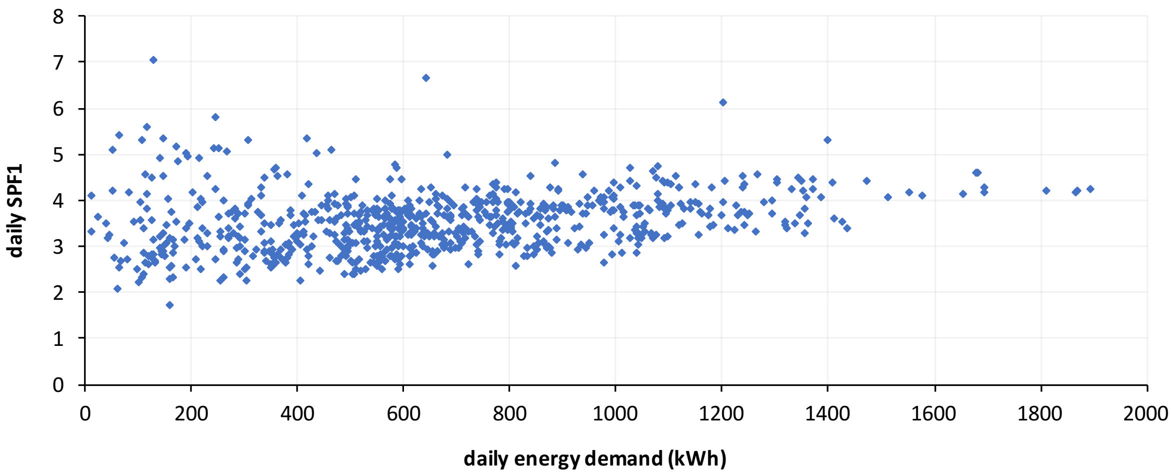

Figure 21: Variation of daily SPF_1 and daily system energy demand.

of the day. A second feature is that, as loads increase the variation in SPF diminishes. In other words, at higher loads, SPF_1 values are consistently higher: approximately 4.

4.3. Circulating pump demands

Seasonal performance factors of interest in comparing technologies or costs are SPF_2 and SPF_4 and include energy demands from ground-loop and heating/cooling circuits. The significance of including these pump demands in the SPF metrics has been shown in Table 1. The relationship between metrics is not linear and depends on dynamic conditions, however. The system pump energy demands in the circuits are functions of the hydraulic design and also the dynamics of the operating cycle and control system operation.

To evaluate the significance of the circulating pump energy demands the ratios of circulating pump energy (Wp) to compressor energy (W_c) have been calculated. These ratios are $Wp_{(\mathrm{SPF2})}/W_c$ and $Wp_{(\mathrm{SPF4})}/W_c$ and have been calculated on a monthly basis and shown in Figs. 22 and 23 respectively. The monthly $Wp_{(\mathrm{SPF2})}/W_c$ data show that the proportion of energy associated with the ground-loop circulating pumps vary between approximately 10% and 30%. This has a significant effect on the SPF_2 metric being reduced compared to SPF_1.

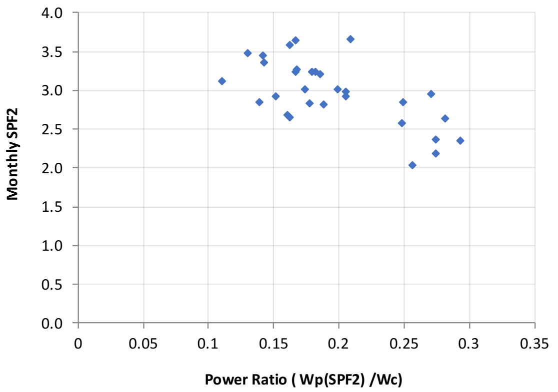

Figure 22: The relationship between SPF_2 and the circulating pump demand ratio.

Similarly, the monthly $Wp_{(\mathrm{SPF4})}/W_c$ data (Fig. 23) show that the proportion of energy associated with both the ground-loop and heating/cooling header circulating pumps vary between approximately 25% and 60%. Accordingly, this has a significant effect on the SPF_4 metric being reduced compared to SPF_1. As noted above, the pump

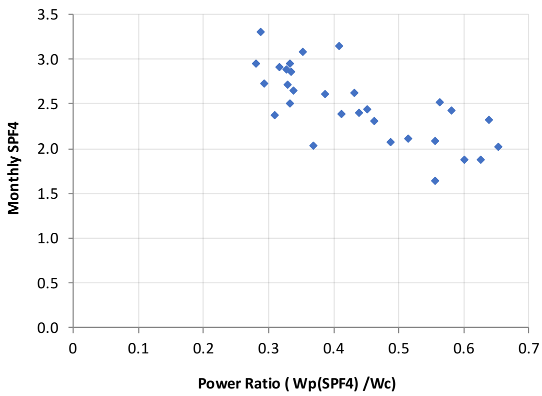

Figure 23: The relationship between SPF_4 and the circulating pump demand ratio.

energy demands depend on control system operation. Actual pump operation has been compared with expected demands based on measured flow rates and knowledge of the intended operation. In general, the circulating pumps are expected to run for more hours than the compressors as the control system is designed to wait approximately 180 seconds between switching on a pump and starting a compressor and similarly after shutting down a compressor. The control system also checks the status of flow switches after opening a valve before starting a compressor. This means that circulating pump operating time is always longer than compressor operating time.

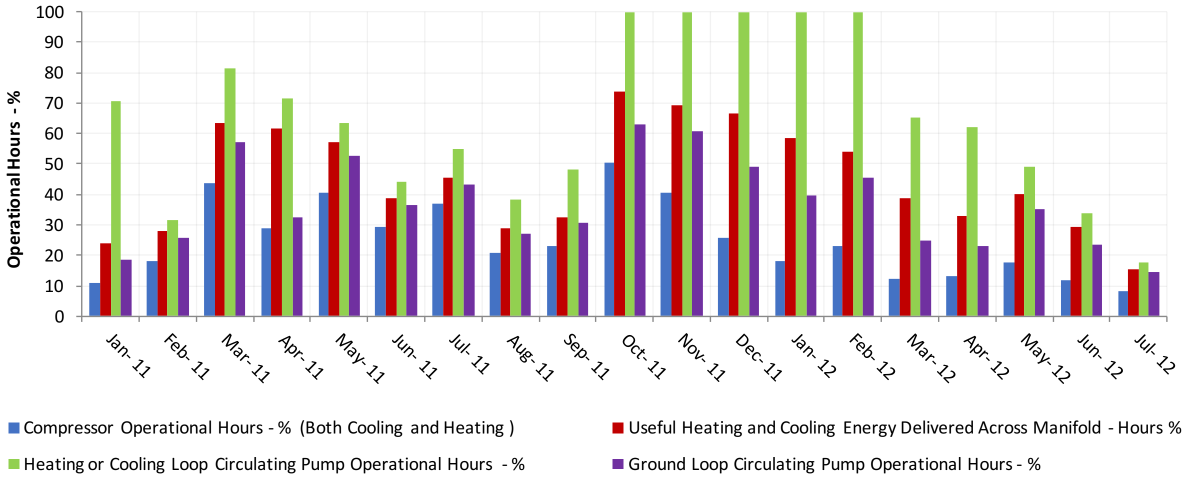

Figure 24: Proportion of monthly operating hours of circulating pump operation.

A further consequence of this is that any periods with short/frequent on-off cycles also increases the proportion of time when the circulating pumps run without any useful heating or cooling being achieved.

Fig. 24 compares monthly operational hours (as a percentage) for the circulating pumps and heat pump compressor when useful cooling is measured across the chilled water manifold. It can be clearly seen that from Oct 2011 to Feb 2012 the circulating pump operates continuously, which is far higher than the compressor operational hours. This is almost certainly a fault. In all months, the circulating pump operating hours are greater than the compressor operational hours but there are also differences between months. For example, comparing between months March 2012 and May 2012 for the approximately same level of compressor operating time there is a considerable difference in circulating pump operating hours. This could be expected to some degree as the number of on/off cycles depends on the heating and cooling demands. In some months, there appear to be faults and at other months the built-in time delay leads to excessive pump energy demand. Pump operation has accordingly been investigated further.

Analysis of the flow rate data and the control system data logged from the BMS system revealed a number of pumping related faults. These include:

- continuous operation of the ground-loop pump in some months;
- a higher pump speed/flow rate being selected than required by the number of compressors in operation;
- flow being divided between heat pumps but only one having the compressors operating.

Some of these faults seem to have arisen due to faulty detection of flow and/or faulty valve operation. The latter fault arises where only one heat pump is called for but two valves are open: one being stuck open in error. In this case the flow through the operating heat pump falls from approximately 7 L/s to 3.5 L/s which is somewhat below that recommended by the manufacturer and so the heat transfer rate and SPF is reduced. These latter classes of faults were found on a number of occasions later in the monitoring period but were not persistent; valve actuators being replaced or the control system being reset after a number of weeks.

5. Discussion

Although the seasonal performance (Table 1) can be regarded as satisfactory, consideration of the much higher levels demonstrated in the most favourable conditions suggests higher performance could be expected. This raises questions as to what design factors could have been improved, and can potentially be improved during operation?

The two issues that have been found to effect performance levels that are related to design features are the operating temperatures and circulating pump demands. Consideration of the heat pump characteristics shown in Fig 14 shows that there could have been benefit in designing for lower heating circuit temperatures than the values of 55°C and 50°C flow and return temperatures used in this design. Whether this is feasible depends on the heat emitter or heating coil designs elsewhere in the system. If the system only supplied underfloor heating, certainly lower temperatures could have been adopted. In this building design fan-coil-units were also used. Lower temperature operation could have been adopted for these but would have required larger heat transfer surfaces. In any case, what could be done to improve seasonal performance would have been to adopt a floating heating circuit temperature so that it was reduced in favourable conditions i.e. a form of weather compensation. Higher cooling temperatures could have been adopted although Fig 15 indicates this would have made less difference to seasonal performance.

The relatively high circulating pump demands suggest that the hydraulic design of the system could have been improved (Figs. 22 and 23). This suggestion is supported by consideration of the design velocities in the header and main distribution pipes. In the case of the ground-loop circuit, at the maximum design flow rate (all heat pumps operating), the velocity is 2.44 m/s in the header and 1.8 m/s in the main horizontal distribution pipes of the BHE array. In the case of the chilled water and heating water header systems, the design flow velocities are calculated to be 2.96 m/s and 1.98 m/s respectively. These velocities are at the upper end of the range recommended for larger pipes in most building services design purposes [28, Table 4.6]. It is not known what contribution the heat pump heat exchanger pressure drops made to the overall pressure drop. Evidently more generous pipe sizes would have been reasonable and improved the pump energy demands and consequently the SPF_2 and SPF_4 metrics.

From our analysis of part load operation over the monitoring period it is clear that there is something of a mismatch between the total heat pump capacity installed and the size of the loads occurring in the system. We have also pointed out that the ground loop temperatures are very modest so that the ground heat exchanger in fact should allow good levels of efficiency. Both these observations suggest that the heating and cooling the loads the system has been designed to deal with are higher than have occurred in practice. This raises questions as to the quality of the design information (about system heating/cooling loads and their frequency of occurrence) early in the design/procurement process and uncertainty in this information relative to operating conditions. Unfortunately, in this study we did not have access to the original design information to investigate this further. We know that the building was analysed using dynamic thermal modelling methods but cannot comment on the type of analysis or level of detail in the information generated. It is well known that the dynamics of the loads need to be considered carefully in ground heat exchanger design as borehole lengths can be sensitive to temporal dynamics as well as maximum/minimum values [29]. The sometimes high levels of uncertainty in building simulation results (the so-called 'performance gap') and variabilities in operating conditions dependent on user and management practice have also been commented on a number of times in recent years [30, 31, 32] so that the issue is familiar and frequently occurring. In this case the lower loads during operation should have improved performance by virtue of more modest ground temperatures but it is unfortunate that the heat pump system design has meant that it has not been able to respond well to low loads.

There are a number of approaches in system design that could have been taken to implement a system that responds to part loads besides the one adopted here. These alternative approaches that may give better results than on-off control of compressor stages include:

- Incorporation of buffer tanks. Adding to the volume of water in the circulation system between the heat pumps and the building systems could help decouple them. In conditions of low load it would have taken longer for the cooling temperature (for example) to change in response to imposed demands. This would have increased the heat pump cycle times and reduced their frequency. This has been commented on in other monitoring projects with smaller buildings [33, 34, 9] and recent design guidance [35].

- Incorporation of a smaller capacity lead heat pump. In this design, the heat pumps were of identical size such that operation of a single compressor was often more than enough to meet demands. Use of a smaller heat pump (e.g. one half the capacity of the others, or less) would have enabled periods of low load to be dealt with by the smaller heat pump running alone but with longer cycle times.

- Use of variable speed compressors. The system monitored uses fixed speed compressors. Technology has advanced to the point where variable speed scroll compressors are readily available and often used in heat pumps. Although low speed operation may not give the same efficiency as full speed operation, the dynamic losses and excessive circulating pump operation associated with on/off cycling could have been avoided.

Some of the issues with control system operation and fault conditions suggest that a more robust design would be beneficial. Essentially the system was vulnerable to failures of control valves and flow switches. Such events were not detected automatically and such hardware faults have meant that operation continued in an inefficient manner (e.g. pump continuing to run or the incorrect flow rate). To improve this situation would have required a more robust control algorithm or a control system enhanced by flow or heat measurement devices and a more sophisticated approach to fault detection and long-term performance monitoring to be adopted.

6. Conclusions

A large-scale non-residential ground source heat pump system has been monitored over a period of approximately three years. High frequency fluid temperature and flow rate data have been collected so that the resulting data set should be useful to researchers interested in validating models of ground heat exchange systems. The system provides both heating and cooling functions for a University building in the UK. The efficiency of the heating and cooling operations have been analysed in terms of Seasonal Performance Factors. Overall levels of efficiency have found to be satisfactory (combined SPF_2=2.97 and SPF_4=2.49) but not as high as expected. The ground heat exchanger system data shows that system demands are cooling dominated and that swings in temperature—both over particular days and over each season—have been relatively modest. We conclude that this is due to lower loads occurring in practice than were expected at the design stage.

The performance of the heat pumps (by themselves) in steady conditions has been found to be only slightly lower than that indicated by the manufacturers data. However, overall system seasonal efficiencies have been found to be particularly sensitive to the dynamics of the operating conditions. There has been a relatively high frequency of occurrence of small heating and cooling loads during the monitoring period. The simple on-off control system adopted in this system has meant that there have accordingly been significant periods of operation with frequent short cycles of heat pump operation. By comparing monitored operation with steady-state modelling results it has been shown that this behaviour introduces significant losses in efficiency.

Some features of the system design such as the relatively high pipe velocities have contributed to the large differences between the metrics SPF_1 and SPF_4. Lower heating temperatures could also have improved the efficiency of the system. Both the short/frequent cycles of operation and some faults that were found in the control of the system at times, have contributed to the circulating pump energy demands being higher than they could have been.

The issues of low loads in relation to those expected highlights the importance of having accurate design information that is sufficiently resolved to enable proper design of ground source heat pump systems. The problems highlighted in this study also indicate the importance, in practice, of having system configurations and controls that are able to respond to part load conditions without compromising overall efficiency. A number of potential improvements to the system configuration have been proposed that could have helped improve performance in this regard. We hope that this data will prove useful to those wishing to validate both system and ground heat exchanger models in future.

Acknowledgements

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Data Statement

The data collected in this work has been made publicly available at the University of Leeds Research Data Archive https://doi.org/10.5518/255 [23]. This archive includes the high frequency temperature and flow rate data for each loop. Data definitions and error protocols are documented with this data.

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HIGHLIGHTS

Paper title: Performance analysis of a large geothermal heating and cooling system

- A large scale geothermal heating and cooling system has been monitored
- A detailed data set has been developed
- Detailed performance analysis is presented
- Diagnosis of a number of faults and discussion of performance levels is presented
- A number of suggestions are made as how to improve performance