Monitoring and Performance Analysis of Large Non-domestic Ground Source Heat Pump Installation

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Abstract

Application of GSHP systems to provide heating and cooling for non-domestic buildings is seen as a viable and effective way of reducing carbon emissions and achieving design renewable energy targets. The application of GSHP systems and their optimal design can be improved through use of reliable system design and simulation models. To assess the validity of design models, availability of high quality field data is critical. Many experimental and monitoring studies have been concerned with domestic scale GSHP installations. In this research work a monitoring system has been implemented to collect high quality dataset from functioning GSHP heating and cooling system in a large educational building at De Montfort University. Operational data have been logged for every minute since the system was commissioned and will provide high quality heat transfer and energy data that will be used for GSHP system model development and validation. Preliminary performance analysis has been carried out and this paper describes the GSHP installation and the monitoring system. Operational data collected over the first year is presented along with a detailed analysis of system performance. The daily average COP of Heat Pumps varied between 3 and 6. The seasonal COP of the system was found to be 4.13. When the ground loop circulating pump was taken into consideration the seasonal COP was found to be 3.41. The reasons for performance variation over different periods are discussed.

Keywords Heating, Cooling, Ground Source Heat Pump (GSHP), Borehole Heat Exchanger (BHE), Monitoring

1.0 Introduction

The advantages of Ground Source Heat Pump with regard to carbon emissions reduction and life cycle cost make Ground Source Heat Pump (GSHP) systems a very attractive option in the design of UK non-domestic buildings. The UK has lagged behind other EU countries in applying such technology but recent years have seen the completion of a number of large heating and cooling systems in non-domestic buildings. The Hugh Aston building at De Montfort University (DMU) is a significant example. A domestic scale trial and monitoring project has recently been completed in the UK but data from non-domestic buildings (both in the UK and internationally) is scarce. Some experimental data collection and validation activity have been carried out at a single and three-borehole facility in Oklahoma State University [1, 2].

The application of GSHP systems and their optimal design can be improved through use of reliable system design and simulation models. To assess the validity of design models, availability of high quality field data is critical[3]. This paper describes the system setup, instrumentation, data collection from a monitoring project of a large scale GSHP system at the Hugh Aston Building, De Montfort University, Leicester. The data is intended to be used for,

- Large scale system performance evaluation
- Borehole Heat Exchanger and Heat Pump model validation
- Study of system operation and control strategies

The detailed performance data analysis has been carried out for the period between March 2010 to May 2011 and key findings are presented in this paper.

2.0 The Building

The Hugh Aston building has opened in spring 2010 and is the home of the DMU Faculty of Business and Law. The 15,607m^2 building includes a variety of accommodation including classrooms, offices, library, retail outlets and large lecture halls (Figure 1). The building has been designed to be the University's foremost low energy building and has been awarded a BREEAM rating of excellent. The building includes a number of sustainable design features including grey water recycling, solar hot water generation as well as the geothermal heating and cooling system described here. The classrooms have been designed to operate using mixed-mode ventilation that incorporates night ventilation control. The GSHP system provides all the cooling for the building and a proportion of the heating system output.

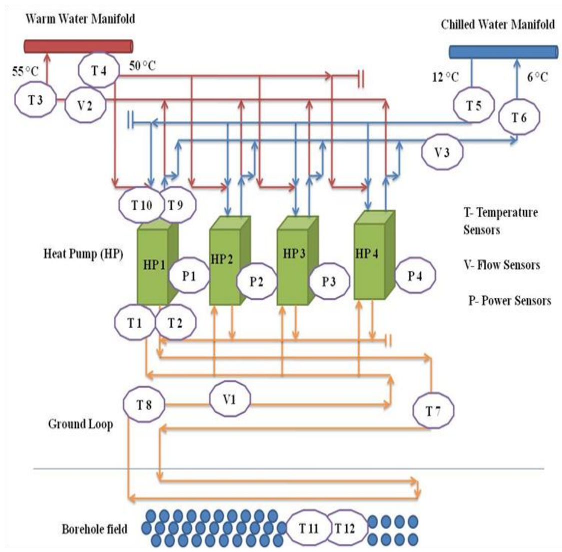


Figure 1 – The Hugh Aston Building **Figure 2 – GSHP System Schematic**

2.1 The GSHP System

The Ground Source Heat Pump system consists of four reversible heat pumps that supply all of the cooling needs and a portion of the buildings heating demand. A schematic of the main components and monitoring equipment is shown in Figure-2. 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 it is circulated. Heat can be added or extracted concurrently by any heat pump so that the temperature of the fluid entering the ground loop is dependent of the balance of heating and cooling being demanded at a particular time. The 'load side' of the system is divided between warm and chilled water headers each with their own circulating pump. Each heat pump can switch between adding heating water or chilled water to the respective headers. The manifolds are, in turn, connected to the building's heating and chilled water distribution systems. Chilled water is supplied to the building's central air conditioning equipment and local Fan Coil Units. Warm heating water is circulated to a number of zones that have under floor heating. The general heat pump arrangement and ground loop circulation pump is shown in Figure.3 and 4 respectively.

Figure 3 – The Heat Pump Plant Room Installation

Figure 4 – The Ground Loop Circulating Pump

The System has four Water Furnace EKW130 heat pumps that are two stage reversible devices with two scroll compressors and plate heat exchangers. The system is designed to meet cooling load of 360 kW and heating load of 330 kW. The source side of the system is served by 56 boreholes, each with a diameter of 125 mm and depth of 100 meters. The boreholes are in two arrays with 19 located outside the building and the remainder installed below the central courtyard of the building. Each borehole has a single U-tube that consists of SDR11 HDPE pipe with a nominal diameter of 32 mm. The borehole is partly backfilled towards the bottom and grouted near the top. Grout thermal conductivity has been specified to be 2.0 W/m.K. The ground thermal conductivity has been estimated from Thermal Response Test results[4] to be 3.1-3.3 W/m.K. The borehole heat exchanger is served by a variable speed circulation pump with flow rate capacity of 30 l/s. The flow is varied in four speed steps depending on how many heat pumps are operating.

3.0 The Instrumentation and Monitoring System

The principle measurements of interest are the heat transfer rates in the ground loop and heating and cooling header systems. These are determined by primary measurements of fluid flow and return temperature and simultaneous flow rate. Measuring this data at relatively high frequency also allows the characteristics of the heat pumps and the control system operation to be monitored. Two temperature sensors have also been incorporated down one of the boreholes.

Figure 5 – The data logger system

3.1 Data Acquisition system

Data loggers are used to record the primary temperature and flow measurements. The Data Acquisition Units Fluke 2640 NetDAQ Networked unit and Fluke 2635 Data Bucket (Figure 5) are chosen to collect Data over annual periods. Each data logger provides 20 channels for measurement and is controlled by a local PC. Remote access to this PC allows the logger equipment to be managed and data transferred at convenient intervals.

3.2 Temperature and Flow Sensors

Figure 6 - Pt100 Temperature Sensor **Figure 7 - Ultrasonic Flow Sensor**

The temperature measurements are required at inlet and outlet of the borehole field, chilled water header, and warm water header (see schematic in Figure 2). Resistance Temperature Detector (RTD) sensors have high accuracy and repeatability, they are chosen for the primary temperature measurements. Robust industrial pattern Pt100 sensors of 250mm length and 6mm diameter are inserted into pockets in the plant room pipework (Figure 6). Obtaining accurate heat transfer data requires that the temperature differences are accurate. Hence each pair of sensors, including the cable and data logger input, was carefully calibrated prior to installation in the pipes. The borehole temperatures at 5m and 100m depth are measured with thermistors attached to the U-tube during installation. A four wire system is used for sensor connection with data loggers throughout the installation.

Three ultrasonic flow meters are used to measure volumetric flow rates on the ground loop, heating and cooling headers. In this type of flow meter, ultrasonic waves are transmitted through the wall of the pipe and through the fluid from a clamp-on emitter. The ultrasonic pulses pass through the fluid, are reflected by the pipe and received at a second clamp-on sensor. The motion of the fluid causes a shift in the transit time of the ultrasonic pulses that is correlated with fluid bulk velocity. The ultrasonic flow meter has the advantage of being non invasive but also of high accuracy. The flow meter sensors are positioned on carefully chosen positions of ground, cooling and heating Loops in accordance with the manufacturer's guidelines. Each flow meter is configured for the appropriate pipe thickness, pipe material, and fluid properties. Figure 7 shows the sensor placement on the heating loop prior to insulation. The measurements are logged into internal memory of the flow meter and periodically downloaded. As the dynamic nature of the GSHP operation is of interest flow rate data is logged every minute.

4.0 Performance Parameters

The overall efficiency of the GSHP system can be determined by comparing the heat exchanged with the ground with that delivered to the building – the difference between the two is provided by the refrigeration system compressors.

The heat transfer across three loops is calculated from a heat balance on the fluid using the flow and return temperatures
(T_o and T_i) and flow rates ($\dot{m}$) as follows:

$$Q_g = \dot{m}_g \, C_{aw} \left(T_{go} - T_{gi}\right)$$
$$Q_w = \dot{m}_w \, C_w \,(T_{wo} - T_{wi})$$
$$Q_c = \dot{m}_c \, C_w \,(T_{co} - T_{ci})$$

Where the subscripts g,w and c indicate ground, chilled water and warm water respectively. The specific heat (C) is calculated according to the properties of water or the water-antifreeze mixture in the ground loop.

The Coefficient Of Performance has been used as the primary result of interest in the analysis reported here. Compressor work is determined from the half-hourly sub-meter data. We distinguish between heating and cooling mode data and combined heat transfer performance as follows:

The COP of the Heat pump; $COP\ hp = \frac{Total\ Building\ Load}{Total\ Compressor\ work\ input}$

COP for heating mode; $COP\ heating = \frac{Heating\ Load}{Compressor\ work\ input\ for\ heating}$

COP for cooling mode; $COP\ cooling = \frac{Cooling\ Load}{Compressor\ work\ input\ for\ cooling}$

The System COP takes account of the ground loop circulating pump energy and is calculated as,

$$COP\ system = \frac{Total\ Building\ Load}{Total\ Compressor\ work\ input + Total\ circulating\ Pump\ power}$$

5.0 Analysis and Discussion

In this section, the performance analysis of GSHP system is presented for the period March 2010 to May 2011. Though monitoring started in December 2009, the full data required to make performance analysis is available from first week of March 2010.

5.1 System Operation

There are three circulating pumps associated with the system and these are for the ground loop, heating header and chilled water header. These are variable speed pumps that vary the flow in steps according to how many heat pumps are operating. Although the heat pumps are two-stage devices, and so in principle there are eight stages altogether, the flow through each heat pump is constant whenever it operates. The ground loop pump is consequently controlled with four speed steps.

The heat pumps are operated in sequence, much like many multiple boiler and chiller installations, so that the total number of compressor stages is increased according to the header temperature difference compared to the respective set point temperature. When the set point is satisfied the heat pump and circulating pumps are switched off. This results in cycles of operation as, in the case of the chilled water, the supply temperature swings through a two degree deadband. This pattern of operation is well illustrated in Figure 8. The inlet temperature for the cooling loop varies between 5 to 7.5 °C, outlet temperature between 6 to 8.5 °C, and flow rate between 7 to 17 l/s. The length of the operating cycles can be seen to increase in the central part of the day as the building cooling load increases. This control strategy is conventional but probably not optimal.

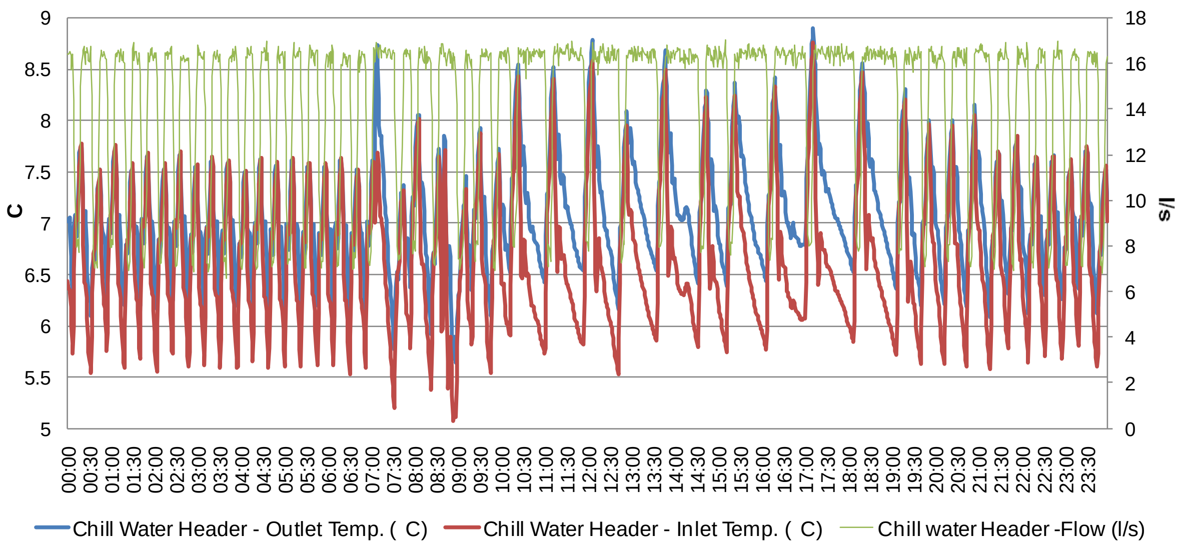


Figure 8 - The minutely averaged chilled water loop operating data during a typical day.

5.2 Ground Temperatures

Ground temperature variation is of great importance in GSHP design and operation and so it has been carefully monitored in this study. The hourly variation in mean ground loop temperature over the whole monitoring period to date is shown in Figure 9 along with an indication of the initial (undisturbed) ground temperature which is 13.2°C. The fluctuation of mean ground loop temperature is from 11.4°C in the winter period to 19.4°C in summer 2010. The daily fluctuations in temperature can be seen to be superimposed on an annual cyclic variation in mean temperature.

The winter heating period ground temperature dips only slightly below the undisturbed ground temperature. This is expected from a system dominated by cooling rather than heating loads. Some of the summer time peaks are very noticeable. It is interesting to note that something of a super-annual variation is emerging in that temperature in April and May of the second year are higher than those in the first year of operation. The overall 8K swing in ground loop temperature and peak of 19.4 °C can be considered very modest when compared to the design conditions. Condensing temperatures would be noticeably lower in the summer period compared to air-cooled systems and higher in winter than an equivalent air-source heat pump system.

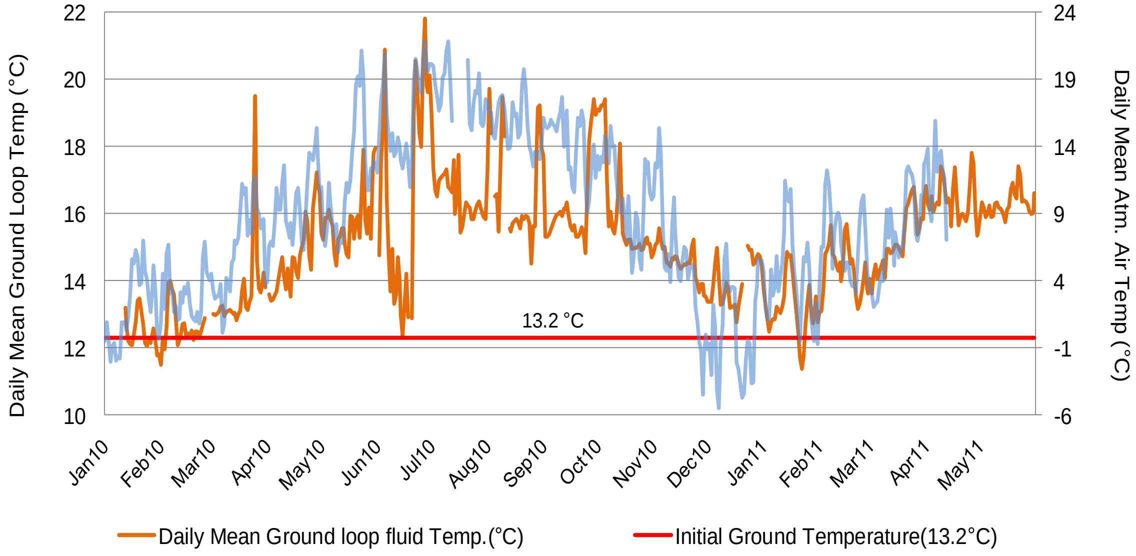


Figure 9 - Variation in mean ground loop circulating fluid temperature (gaps in the data indicate maintenance periods). The initial ground temperature is 13.2°C.

It is interesting to consider the relationship between ground loop heat transfer and the loop temperature and this is illustrated in Figure 10. Heat extraction varies from 0.075 to 0.2 MWh/day and heat rejection ranges between 0.15 and 0.85 MWh/day. The heat rejection to the ground is always higher than heat extraction for every month but the difference between this two varies significantly between months. This difference is between 0.5 and 0.85 MWh/day during the months March to July and between 0.2 to 0.6 MWh/day during the months August to February. The reduction in heat rejection over the monitoring period can be seen to correspond with the seasonal variation in monthly mean atmospheric temperature. The ground loop temperature changes in a corresponding manner but with a noticeably lower swing than the air temperature.

5.3 Annual Performance

Variation in operating efficiency can be examined by considering the variation in mean COPs from month to month. This data is presented in Figure 11 which shows the system COP by month in addition to the heating, cooling and combined heat COPs. This monthly mean COP data does not show a wide variation from month to month but there are some characteristics to note. Some of the highest cooling efficiencies occur in the first months of operation and there is an improvement in heating COP in the March to May period of the second year compared to the first. This can be explained by the fact that ground temperatures are at their lowest in the first months of operation and higher in these months in the second year. Heating COP is generally lower than that in cooling mode. This may be explained by the fact that the difference between the ground and the heating supply temperature (temperature lift) is much greater than in the case of the chilled water system.

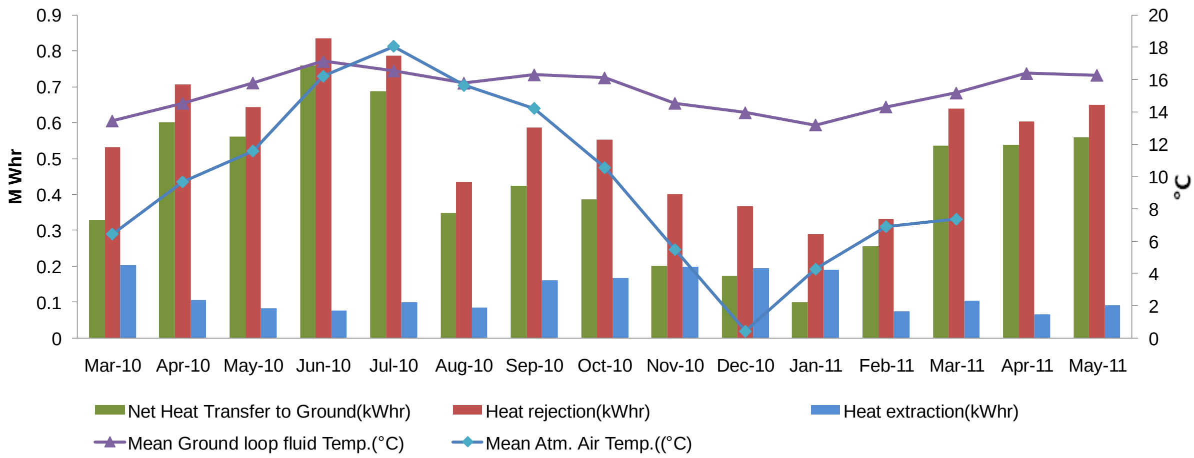


Figure 10 - Monthly heat exchange and operating temperatures.

When the ground loop circulating pump energy is included in the electrical energy demand, the COP is reduced by approximately 0.5. The efficiencies when calculated over the whole operating period are shown in Table 1. It is interesting to see that the circulating pump energy is significant compared with the compressor power.

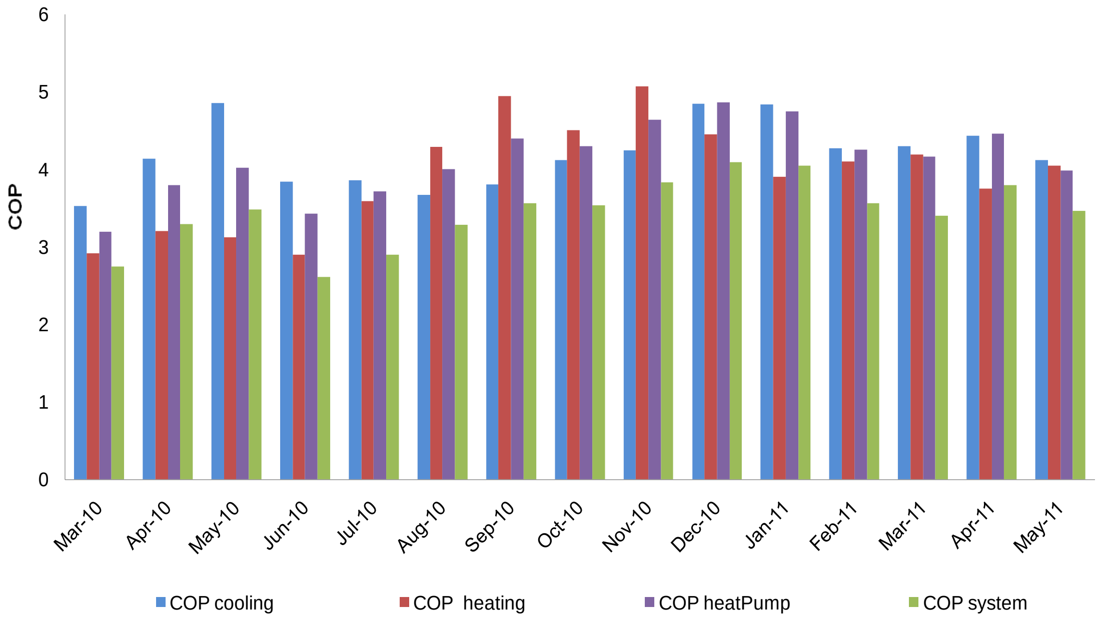


Figure 11 - Monthly daily average Heat pump performance over the period March 2010 to May 2011

Operation	COP
Heating COP	3.93
Cooling COP	4.19
Heat pump COP	4.13
System Seasonal COP	3.41

Table 1 – System efficiencies calculated over the whole monitoring period

Performance is further illustrated in Figure 12 which shows the daily mean cooling, heating and ground loop operational data over the month of November 2010. The daily average efficiencies have been shown along with the daily mean heat transfer rates. In this month the heating load varies between 0.3 and 1.5MWh and cooling load varies between 0.2 and 0.6 MWh/day. The ground heat rejection rate is 0.6 MWh/day at the start of the month and decreases as the month progresses so that there is net heat extraction between 24th and 26th November. This trend corresponds to the falling outside air temperatures and ground loop temperatures as shown in Figure 13. The temperature of ground loop mean fluid temperature varies between 13°C to 15.5 °C in this period and the outside air temperature is in the range -4°C to 15.4°C. There is noticeable variation in efficiencies during this month and the heating mode COP approaches 6 at times.

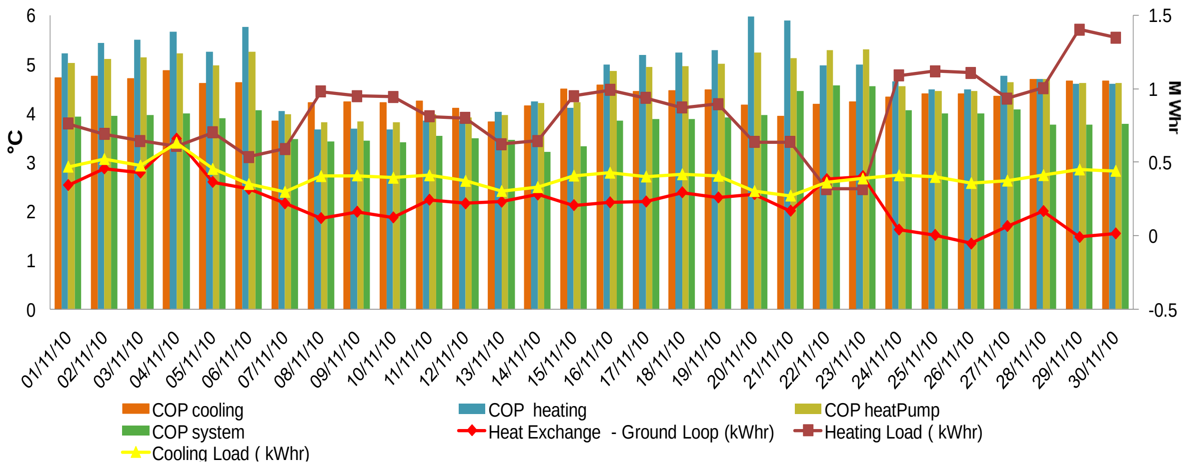


Figure 12 - Daily mean building heat demands and efficiencies in the month of November 2010.

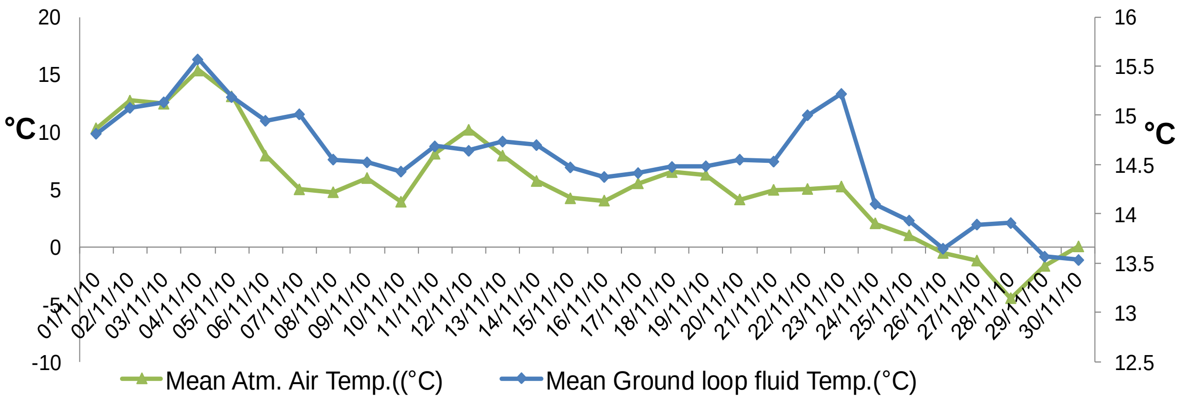


Figure 13 - Daily average outside air and ground loop temperatures in the month of November 2010.

The building heating load does vary in accordance with outdoor air temperature in this month whereas the cooling demand varies little. This might be expected as the heating system serves foyer and other circulating spaces but the cooling system serves parts of the building with loads more dependent on internal heat gains such as computer labs. The heating demands are of the order expected at the design stage but the cooling loads are rather lower. During this period the demand is mostly met with only one heat pump in operation providing heat and one providing cooling. The frequency of the heating and cooling loads in this period are illustrated in Figure 14 and are, in fact, mostly within the limits of a single compressor stage operating.

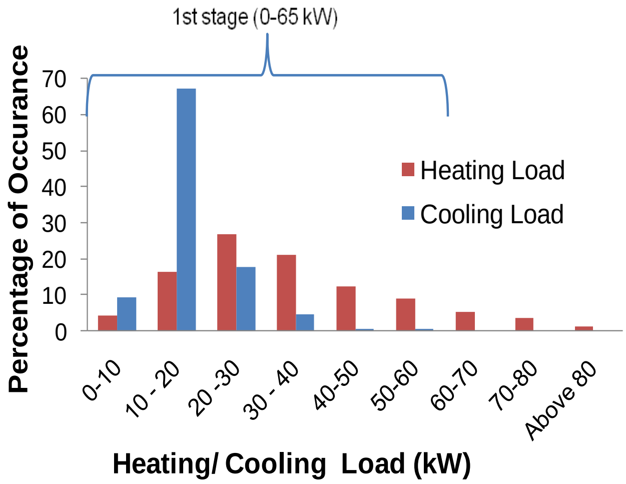


Figure 14 – Frequency of occurrence of the building heating and cooling loads.

5.4 Part Load Performance

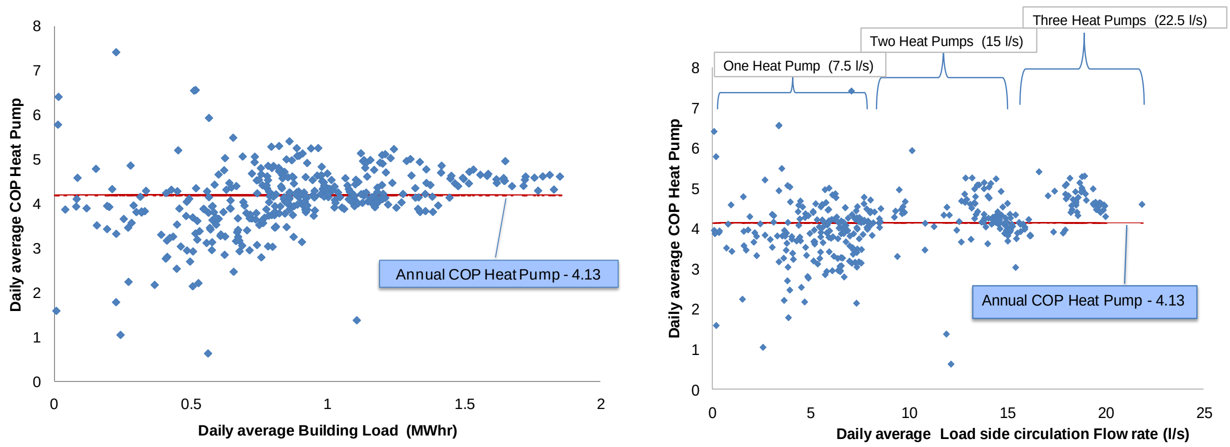
It is apparent that with COPs found to be in the range 3 to 6 there is noticeable variation about the mean annual value between different days – even in the same month. One would expect that efficiencies would vary as the ground loop temperature varied over the season. However, the variation in ground loop temperature is very modest during this monitoring period and so other effects must be considered. Similarly, there is noticeable variation in daily efficiencies when there is little variation in heating or chilled water supply temperatures.

What is noticeable from higher frequency data, such as that shown in Figure 8, is that operation is very dynamic and the system changes state relatively frequently, particularly when loads are relatively small. To investigate whether there is a reduction in efficiency due to reduced performance during the start-up and shut-down phases the correlation between efficiency and cycle frequency has been examined.

Quantifying cyclic operating frequency is difficult and so daily building heating demands and average flow rates have been used as proxy measures. Mean flow rates and loads lower than nominal values indicate the heat pump being off for part of the day. Low values are taken to indicate frequent cycling of the system.

The daily mean heat pump (combined heating and cooling) COP is correlated with mean daily building load and mean daily flow rate in Figure 15 (a) and (b) respectively. It is apparent that at mean daily loads less than approximately 1.0 MWh/day a wide range of COP values occur – some noticeably above and some noticeably below the annual mean. As mean daily loads increase the range of values diminishes so that the highest values are in a narrow band slightly above the annual mean.

This is a clear indication of the fact that better efficiencies are achieved where the heat pumps run for longer periods. Part load conditions can result in daily efficiencies much lower than the nominal values expected from manufacturers' data. The correlation between efficiency and mean daily flow rate shows a similar trend. Data is noticeably grouped around mean flow rates of approximately 14 and 19 l/s (Figure 15(b)). This discrete feature of the data is due to the fact that flow rates change in this system according to pump speed in four steps. Data in these groups can be associated with the second and third steps of circulating pump operation. Again operation in these conditions is associated with longer periods of operation at higher loads.



(a) **(b)**

Figure 15 – Heat pump COP correlated with daily mean loads and flow rates

The cyclic nature of the system operation under part load conditions suggests that efficiencies could be improved, in this building, by developing better control strategies. This might include allowing wider temperature deadbands or applying limits to the time between cycles. More generally, slightly different system configurations might improve overall efficiency, for example, by using one additional smaller heat pump to deal with low load conditions. Other configurations that might reduce cycling could include heating and chilled water buffer tanks.

6.0 Conclusions and Future Work

Operational data has been collected from a large Ground Source Heat Pump system over a 15 month period. This high frequency heat transfer, power demand and borehole field temperature data can be used to (i) characterise system operation and quantify efficiency in a range of conditions (ii) calibrate GSHP system component simulation models (iii) analysed to identify improved control strategies.

The fluid temperatures in the Borehole Heat Exchanger field fluctuated on a daily and seasonal basis in response to the next heat exchange rate and showed a slight increase at the start of the second year of operation. On the whole the heating and cooling loads are better balanced than expected and the temperature swings relatively modest. Condensing temperatures in winter and summer have been better than could be expected with an equivalent air cooled system.

Mean daily Coefficients of Performance in operating conditions were found to be in the range 3.0 to 5.5. Seasonal COP for the heat pump system was found to be 4.13 but reduced to 3.41 when the ground loop circulating pump power is taken into consideration. Operational COP was found to correlate with heating and cooling loads in that better than average performance was demonstrated at higher load conditions. In part load conditions the system cycles off more frequently and this seems to account for the poorer mean daily efficiencies measured.

The data continues to be collected from this system and will be used: (i) to validate Borehole Heat Exchanger models; (ii) develop a dynamic heat pump model; (iii) investigate alternative control strategies and system configurations through system simulation studies. Improving the operating efficiency of the real system through improvements to the control system will be investigated.

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