Control Strategy of Novel Dry Air Ground Source (DAGS) System

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Abstract

Based on a number of studies carried out; it has been identified that, Ground Source Heat Pump (GSHP) systems are widely used as one of the preferred low carbon technologies in the UK. The use of these systems is due to their economic advantages and potential reduction of carbon footprint. However, a number of the studies have highlighted that the systems are either installed incorrectly or operated and controlled improperly and therefore result in poor performance. GSHP performance is affected by the temperature of the ground and when thermally saturated its efficiency reduces significantly.

This paper investigates the potential to reduce the level of thermal saturation by rejecting heat via a Dry Air Cooler (DAC) when the ground and ambient temperatures favour this. DACs are often fitted to GSHP systems to reject heat during extreme conditions to protect the system, rather than improve performance. In this investigation, an empirical Transient System Simulation (TRNSYS) model has been developed and used to investigate the control algorithms so as to identify the optimal operation and control strategies for DAGS system for enhancing the system efficiency.

Specifically, the paper investigates the effect of using a DAC in conjunction with a GSHP system. This includes investigating the (i) energy input into the GSHP system, (ii) annual ground temperature variation and (iii) Coefficient of Performance (COP). The results show significant savings can be achieved using optimal operation and control strategies for DAGS system.

Keywords:
Control Strategy, Dry Air Cooler, Ground Source Heat Pump, TRNSYS, Ground Temperature variation.

1.0 Introduction

It is widely accepted that global climate change is predominantly due to the emissions of greenhouse gases (GHG), 75% of which are attributable to CO_2 [1, 2]. In the UK, 47% of CO_2 emissions are due to the production of heat with a significant contributor to the total emissions from heat generation [3]. The heat pump stock in 2013 contributed to 20 Mt of greenhouse gas emission savings. The current European installed base of heat pumps (HPs) produces 35 TWh of renewable energy from the air, water and the ground and is responsible for the abatement of 8 Mt of CO_2 per annum [4].

The performance of GSHP systems is intrinsically related to the ground and load temperatures. It is unavoidable that ground temperatures will change to some degree in response to extraction of heat from, or rejection of heat to, the ground. However, it is important to recognise that the ground is not an infinite source or sink of energy, and that excessively large net rates of extraction or rejection of heat to the ground must be avoided. If excessive rates of heat extraction from, or rejection to, the ground are allowed for prolonged periods, then it is likely that significant changes in ground temperature will occur; such ground temperature changes can have significant detrimental impact on overall system efficiency or Coefficient of Performance (COP), as well as large environmental impact. Zoi and Constantinos [5] proposed a GSHP system controlled with only cooling tower and investigated three control strategies to minimise this significant change in ground temperature by using simpler heat rejection or 'free cooling'. The first one determines the set point at which a cooling tower starts its operation according to the fluid temperature exiting HP and ambient air wet bulb temperature exceeds a given set point. The second one activates the cooling tower when the fluid temperature exiting the ground heat exchanger (GHX) is greater than a certain value. The third one sets the cooling tower on when the fluid temperature exiting the HP is greater than a given value.

This paper investigates the use of a DAC rather than a cooling tower to reduce the level of ground temperature saturation by rejecting heat via a DAC when the ground and ambient temperatures favour this. DACs are often fitted to GSHP systems to reject heat during extreme conditions to protect the system, rather than improve performance. Opportunities exist to control the performance of the GSHP using a DAC and also the predicted seasonal or daily ground temperature as well as predicted/available energy demand of the building. However these control systems are not reported in the literature.

This paper includes the description of the existing system and its operation, the simulation setup, the investigation of new control strategies using a DAC and the conclusions.

2.0 Description of the New System and Its Operation

The proposed system employs the existing London South Bank University's (LSBU) GSHP installation and components but operates it differently to how it was originally configured. The GSHP system within the K2 building at LSBU uses four WaterFurnace EKW130 reversible HP units. Each has a nominal capacity of 120 kW for heating and 125 kW for cooling. The heat is transferred from and to the ground through a closed loop system with the aid of 159 vertical energy piles which are built into the foundations of the structure and bored into the London clay. The building's heating and cooling generation is fully provided for by the GSHP system. The source-side of the system consists of energy piles and header pipes to which the HPs add or extract heat using a heat transfer fluid which is pumped and exchanges energy between the building and the ground.

The original system utilised a DAC designed to operate when the heat sink temperatures were either too high or too low. The DAC was therefore employed as a safety device to protect the heat pump from operating outside its safe envelope. In the proposed system the DAC was used tactically to improve the efficiency and performance of the heat pump and therefore system. The system simulated is shown in Figure 1 below. This shows the system controlled to provide heat rejection via the DAC rather than the ground loop to achieve the best COP. This relies on the principle that heat pump efficiency or COP is affected significantly by its temperature lift with a 1K reduction giving typically a 3% rise in COP. The DAC can therefore be employed selectively when it will produce more favourable heat sink temperatures (and therefore higher COP) compared to those generated by the ground sink. The proposed system has the potential to save energy, however should not require additional components compared to the existing system, although it will be controlled differently. The performance improvement of the proposed system is investigated in the following sections.

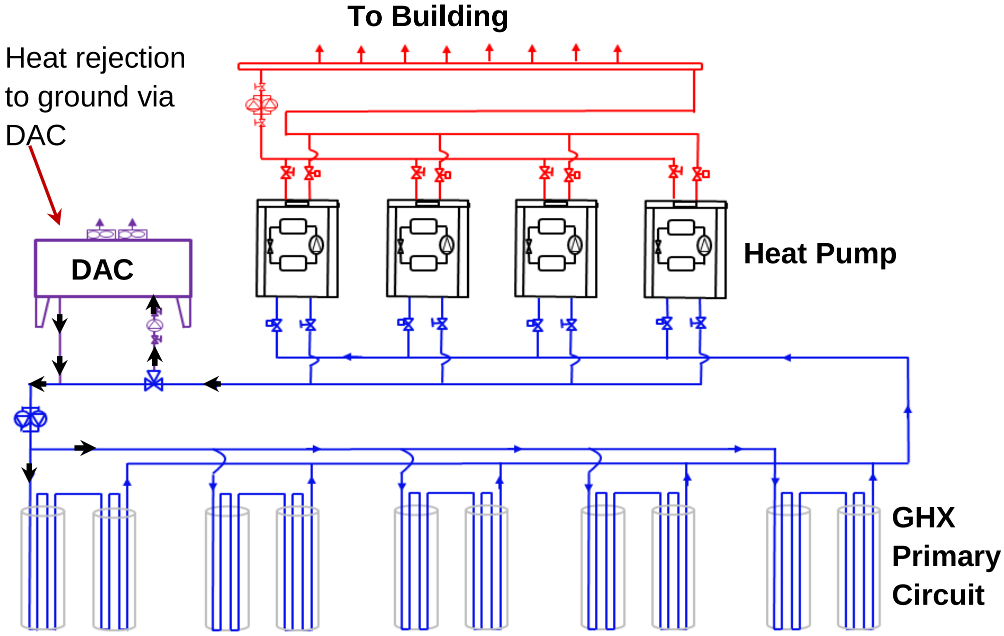

Figure 1 Schematic of the System Simulated.

3.0 The Simulation Setup

The following section presents a description of the operation of the different components of the system which is replicated by interconnecting a set of models. In order to simulate the experimental observations, a model has been built using the TRNSYS 17 simulation software [6]. This allows the construction of a GSHP system simulator that closely resembles and simulates the actual HP installation. The main parts of the GSHP system that have been used in building the simulation model are: the ground heat exchanger (Type 557), the HP model (Type 668), the circulating pumps, flow stream loads (Type 682), DAC (Type 511), tempering valve (Type 11) and tee piece (Type 511).

3.1 Ground Heat Exchanger (Type 557a)

The ground heat exchanger component (Type 557a) was set up with the appropriate geometrical configuration and relevant ground thermal properties some of which were derived from the thermal response testing carried out in the GSHP design stage. In the current work 159 energy piles are used to exploit the ground's heating and cooling capacity. Type 557a models a set of equal vertical U-tube heat exchangers which thermally interact with the ground. This ground heat exchanger model is most commonly used in GSHP applications. A heat carrier fluid is circulated through the ground heat exchanger and either rejects heat to, or absorbs heat from the ground depending on the temperatures of the heat carrier fluid and the ground.

3.2 Heat Pump Model (Type 688)

The HP model (Type668) relies upon catalogue data readily available from HP manufacturers for the performance measurement related to the HP that is being simulated. At the heart of the component are two data files: a file containing cooling performance data, and a file containing heating performance data. Both data files provide capacity and power draw of the HP (whether in heating or cooling mode) as functions of entering source fluid temperature and entering load fluid temperature these establish the performance envelope of the HP over a range of ground source side temperatures and a range of load side temperatures.

The data used to build this HP model were obtained from the manufacturer WaterFurnace. The Type668 HP is equipped with two control signals, one for heating and one for cooling. However, heating mode takes precedence over cooling mode. If the heating and cooling control signals are both ON, the model will ignore the cooling control signal and will operate in heating mode.

The HP's COP in heating is given by equation 1.

$$\mathrm{COP} = \frac{\mathrm{Q_{HP}}\ [\mathrm{kW}]}{\mathrm{W_{HP}}\ [\mathrm{kW}]} \qquad \text{(Eq.1)}$$

The amount of energy absorbed from the source fluid stream in heating is given by equation 2.

$$Q_{absorbed} = Cap_{heating} - P_{heating} \qquad \text{(Eq.2)}$$

The outlet temperatures of the two liquid streams can then be calculated using equations 3 and 4.

$$T_{Source,out} = T_{source,in} - \frac{Q_{absorbed}}{m_{source}Cp_{source}} \qquad \text{(Eq.3)}$$

$$T_{load,out} = T_{load,in} - \frac{Cap_{heating}}{m_{load}Cp_{load}} \qquad \text{(Eq.4)}$$

The HP's COP in cooling mode is given by equation 5.

$$\mathrm{COP} = \frac{\mathrm{Q_{HP}}\ [\mathrm{kW}]}{\mathrm{W_{HP}}\ [\mathrm{kW}]} \qquad \text{(Eq.5)}$$

The amount of energy rejected by the source fluid stream in cooling mode is given by equation 6

$$Q_{absorbed} = Cap_{cooling} + P_{cooling} \qquad \text{(Eq.6)}$$

The outlet temperatures of the two liquid streams can then be calculated using equations 7 and 8.

$$T_{Source,out} = T_{source,in} + \frac{Q_{rejected}}{m_{source}Cp_{source}} \qquad \text{(Eq.7)}$$

$$T_{load,out} = T_{load,in} + \frac{Cap_{cooling}}{m_{load}Cp_{load}} \qquad \text{(Eq.8)}$$

3.3 Circulation Pumps (Type 741)

There are two circulation pumps. In reality each pump represents a series of pumps; Type741 models a variable speed pump that is able to produce any mass flow rate between zero and its rated flow rate. The pump's power draw is calculated from pressure rise, overall pump efficiency, motor efficiency and fluid characteristics. As with most pumps and fans in TRNSYS, Type741 takes mass flow rate as an input but ignores the value except in order to perform mass balance checks. Type741 sets the downstream flow rate based on its rated flow rate parameter and the current value of its control signal input.

3.4 DAC (Type 511)

Type511 models a dry air cooler; a device used to cool a liquid stream by blowing air across coils containing the liquid. This model assumes that the device can be modeled as a single-pass, cross-flow heat exchanger.

3.5 Tempering valve (Type 11b)

The use of pipe or duct 'tee-pieces', mixers, and diverters, which are subject to external control, is often necessary in thermal systems. This valve allows the system to be controlled in response to temperature of the fluid leaving the heat pump.

3.6 Tee piece (Type 511h)

This instance of the Type11 model uses modes 1 and 6 to simulate the function of a tee-piece that completely mixes two inlet streams of the same fluid at different temperatures and or humidities.

Figure 2 shows a schematic of the empirical GSHP system model used for investigating a range of control strategies. A list of assumptions used to simulate the TRNSYS system model is provided in table 1.

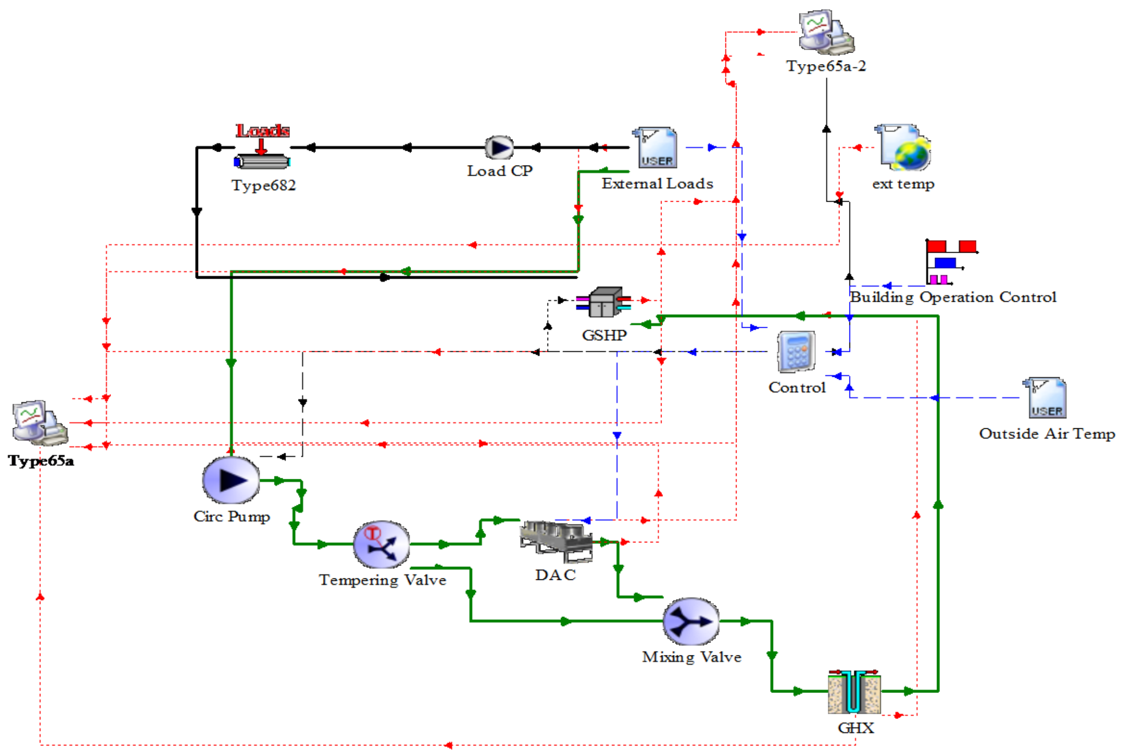

Figure 2 Schematics of the DAC simulation setup connected to GSHP system

List of Model Inputs and Assumptions
Occupancy period 13 hours every day except weekends
Historical outside air temperature (OAT)
Flow and return fluid temperatures on both source and load side of the system
Heating and cooling performance data of the heat pump model
Heat pump and Circulation pumps to operate in heating mode if the OAT>18°C
Heat pump and Circulation pumps to operate in heating mode if the OAT<14°C

Table 1 list of model inputs and assumptions

4.0 Model Validation

The empirical TRNSYS model developed was validated using experimental data from LSBU's actual GSHP system installation. For validation of the model, several tests have been conducted, the various physical components of the system have been kept as close to reality as possible. A comparison between model predicted and independently determined COP values for both the actual and predicted test shows a very reasonable agreement. Figure 3 shows that a maximum deviation of about $\pm$7 % is observed.

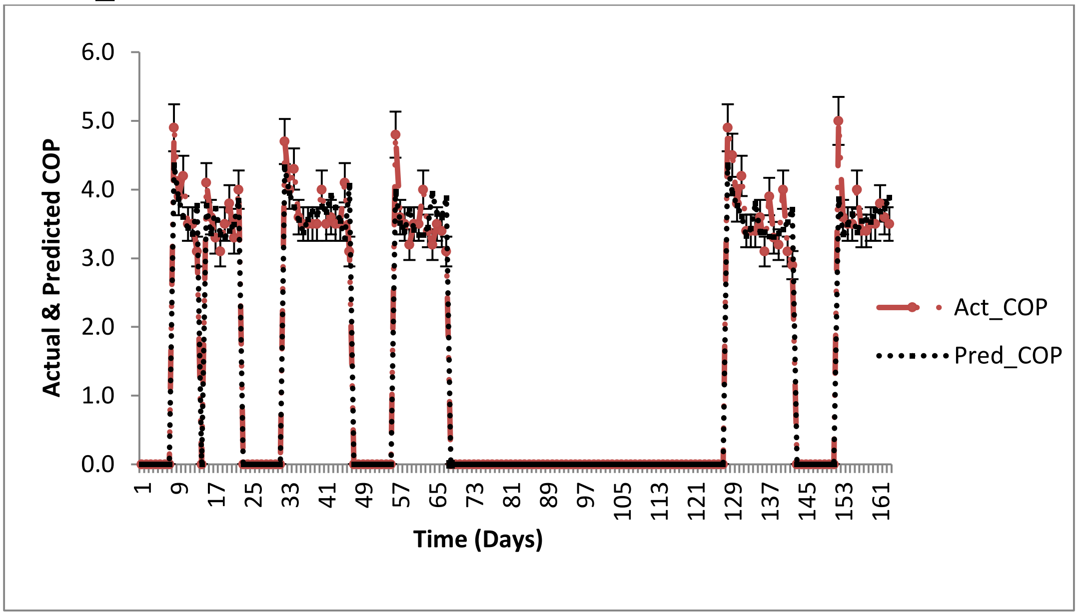

Figure 3 Comparison of actual and predicted COP

5.0 Investigating New Control Strategies Using DAC

The current control strategy of K2 is designed to operate the DAC only in the event that the temperature of the water returning from the ground loop exceeds 38 °C. The control system enables the dry air cooler shunt pump which is positioned in the loop supplying the DAC and circulates the water to the already enabled DAC. The DAC has its own internal PID based control system which controls the temperature of the water leaving the dry coolers to 22 °C.

The existing system model was reconfigured to reject heat into the ground to ease ground saturation which has consequences on the performance of the system. Therefore having built and established a validated empirical TRNSYS system model the opportunity was taken to investigate the impact of different control strategy approaches using the DAC on HP performance and ground temperature variation.

Control strategies utilized in this study define when and how the DAC circuit, circulation pumps and the HP should be turned on or off. The system's electrical power consumption is the sum of three terms: HP power, power of each circulating pump and DAC fan power.

The control strategies turn the DAC on when the fluid temperature exiting HP is greater than a given value. Different desired outlet fluid temperatures of 22, 24, 26 and 28 °C are examined and hence the normal operating condition of the system has been compared to these four different scenarios to investigate the impact of running the DAC at different temperature set points. Scenarios CS1, CS2, CS3 and CS4 are being used instead of the four desired control temperatures. In these comparisons the following parameters have been investigated:

- COP of the system.
- Ground temperature variation
- Heat Pump Energy Consumption

These results are presented on the following sections 5.1 to 5.4 and compared against the normal operating control scenario.

5.1Effect of DAC on COP

The cooling COP value of the GSHP system under different temperature set points is illustrated in Figure 4. It can be noted that at the beginning of the season between April and June 2013 the COP values for all the four set temperature scenarios were very close to each other. This is when the DAC is not running and this is labelled Time Period A. In Time Period B the DAC is operating partially or fully between the periods of June to October 2013 and there is a variation in COP between the options. This COP differences is purely because the DAC has been utilised in lowering the leaving fluid temperature from the HP by rejecting the heat back to the ground at a lower temperature compared to the normal operating leaving fluid temperature.

It is clear that the GSHPs COP value decreases continuously with increasing set point temperature. For the first year's operation of the GSHP system, the COP values for cooling are highest for the lowest set point temperature.

Compared to the normal operating scenario in which the COP value for cooling is decreased to 5.2 while setting the temperature set point control to 22 °C achieves a cooling COP of 6.2 which is 19.2 % higher than the normal operating scenario.

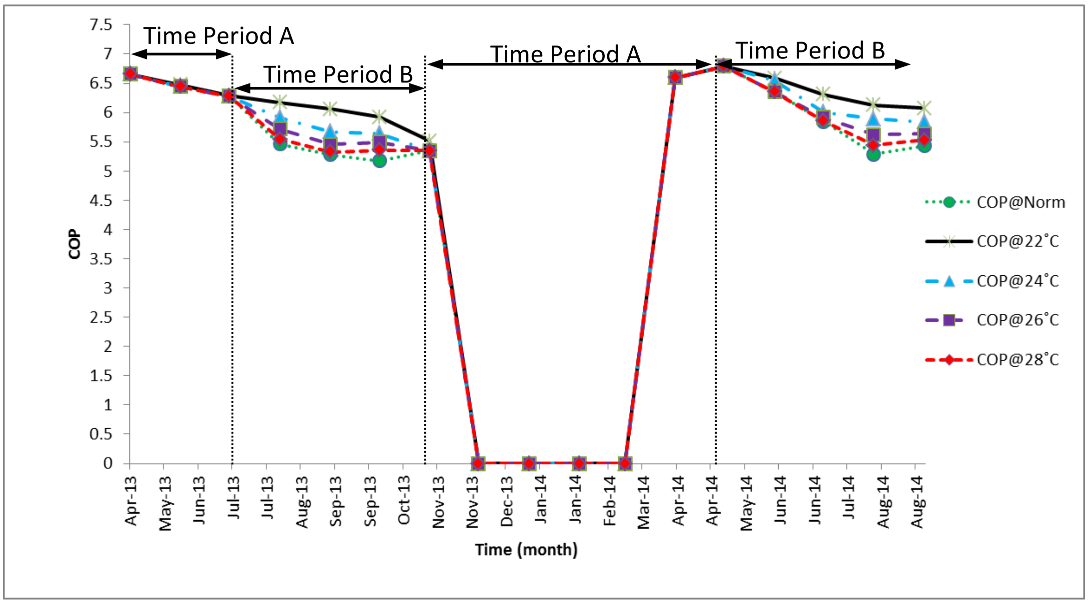

Figure 4 Average Monthly system COP

5.2 Effect of DAC on Ground Temperature

As well as investigating the effect of operating period of the DAC on the performance of the system, the effect on the ground temperature is also investigated. The simulation results of ground temperature for 1 year’s operation are presented in Figure 5. The results show that the four different set point operation temperatures of the DAC leads to differences in the ground temperature variation when the set point temperature varies. In Time Period A when the DAC is not operating at the beginning of the cooling season the ground temperature is similar and hence varying the ground temperature set points has no effect at all. Therefore the ground temperature for all scenarios remain constant. However in Time Period B the temperature variation between the scenarios becomes clear that the more the DAC is running the lower the ground temperature variation is. The highest ground temperature after 1 year’s operation in cooling mode is 23 °C for normal operation period, compared to 20 °C for the lowest set point temperature which is 15 % lower than the normal operation. This impacted on COP and therefore by reducing the ground temperature from 23 to 20 °C, the overall system performance has improved by approximately 9 % and this can be seen clearly in Figure 4 above.

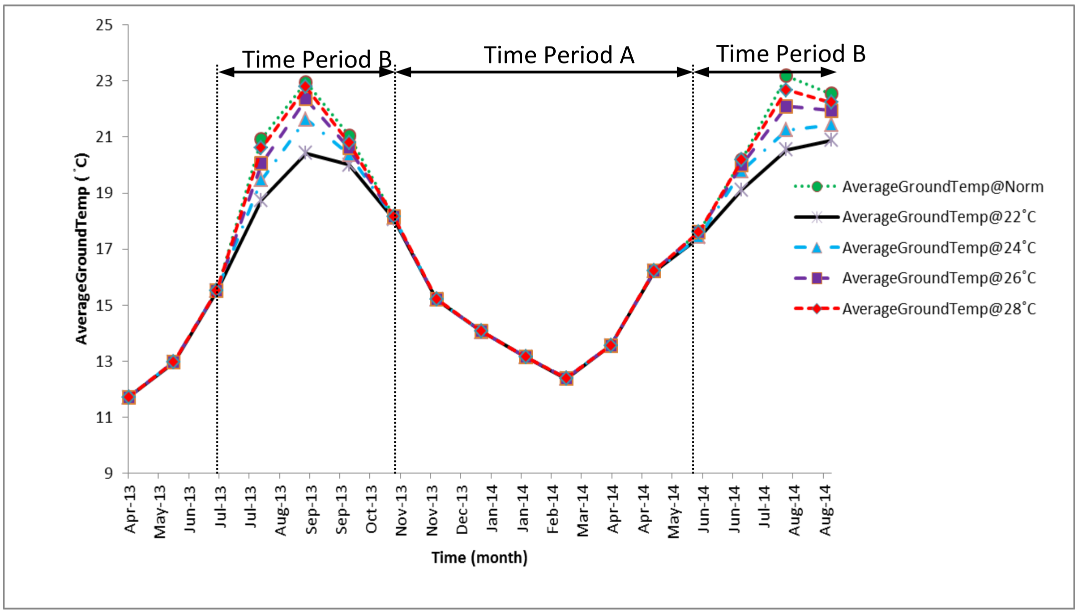

Figure 5 Monthly ground temperature variations

With the decrease of average ground temperature around the ground heat exchanger, the temperature difference between the ground and the circulated heat carrier fluid decreases, this phenomena has both advantages and disadvantages on the system. Although this incremental temperature change improves the COP value during cooling season, however it also reduces the COP value of the system in heating season.

5.3 Effect of DAC on Heat Pump Energy Consumption

The total monthly HP energy consumption for scenarios CS1, CS2, CS3 and CS4 is presented in Figure 6. The operation cycle of the DAC can determine whether the GSHP system consumes more energy, compared to a GSHP system without DAC. Figure 6 shows that between the periods of April 2013 and August 2014 the highest energy consumption of the HP was 7923 kWh and 7669 kWh respectively and this has occurred at both when the system was running without the aid of the DAC and also when the DAC was controlled at the highest set point temperature of 28 °C, this dynamics can be shown clearly in Figure 6 during Time Period B. Time Period A shows that when the DAC is off the output of the four set points remain unchanged. Time Period B shows that when the DAC was operating at different set points, this zone also highlights the benefits of reducing the ground temperature as shown in Figure 5 to the energy input of the system. Reducing the ground temperature helps to reduce the temperature difference between the ground loop heat exchanger leaving temperature and HP leaving temperature and hence increasing COP of the system.

Similarly the lowest energy consumption of the system during the transition period to heating mode was 267 kWh. In Time Period B between the periods of July and September 2013, comparisons of the four scenarios revealed that the energy consumption of the system has decreased by 8 % when the lowest control set point of 22 °C (CS4) was compared to the normal operating conditions of the system.

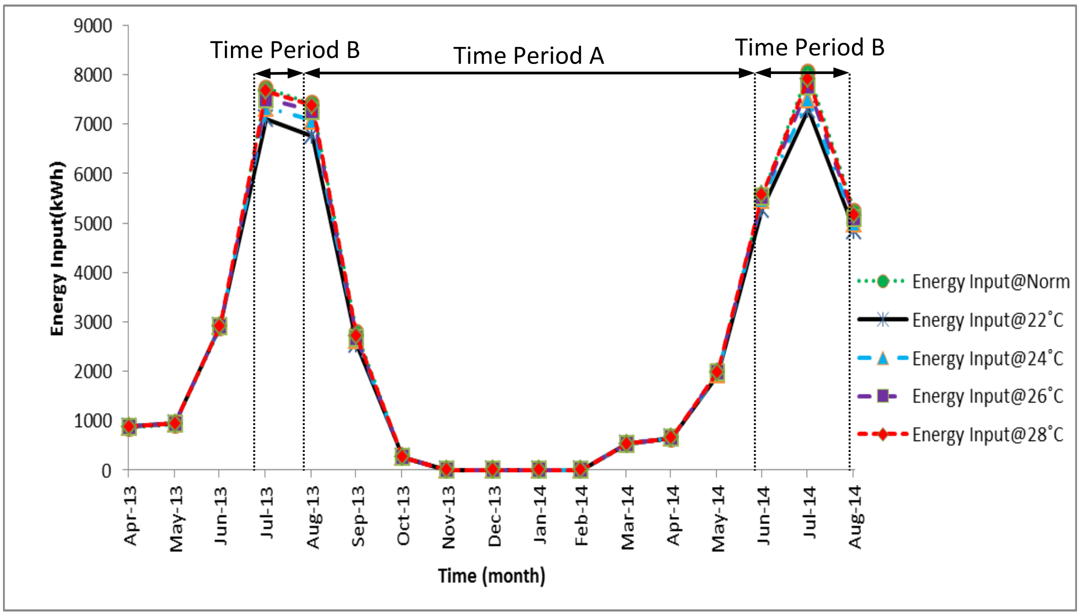

Figure 6 Monthly HP energy consumption

5.4 Economics of The Control Strategy

In this section the financial and CO_2 emission savings of each temperature set point have been compared to the normal operation of the system. Figure 7 shows the additional monthly CO_2 emission savings ($kgCO_2$) that can be achieved by implementing the different temperature control set points. The graph shows that in July 2014 a maximum CO_2 emission saving of 420 $kgCO_2$ was achieved. Relating this to Figures 4 and 5 this is also the point at which the highest COP and highest ground temperature was recorded. Notably this has occurred at the lowest temperature set point of 22 °C. Moreover the lowest CO_2 emission saving was approximately 20 $kgCO_2$ and this has occurred at the highest temperature set point of 28 °C. As can be seen the lowest temperature set point can yield a saving of approximately 17 % compared to the highest temperature set point. In addition the lowest temperature set point can achieve cost savings of approximately 18 % compared to the highest temperature set point.

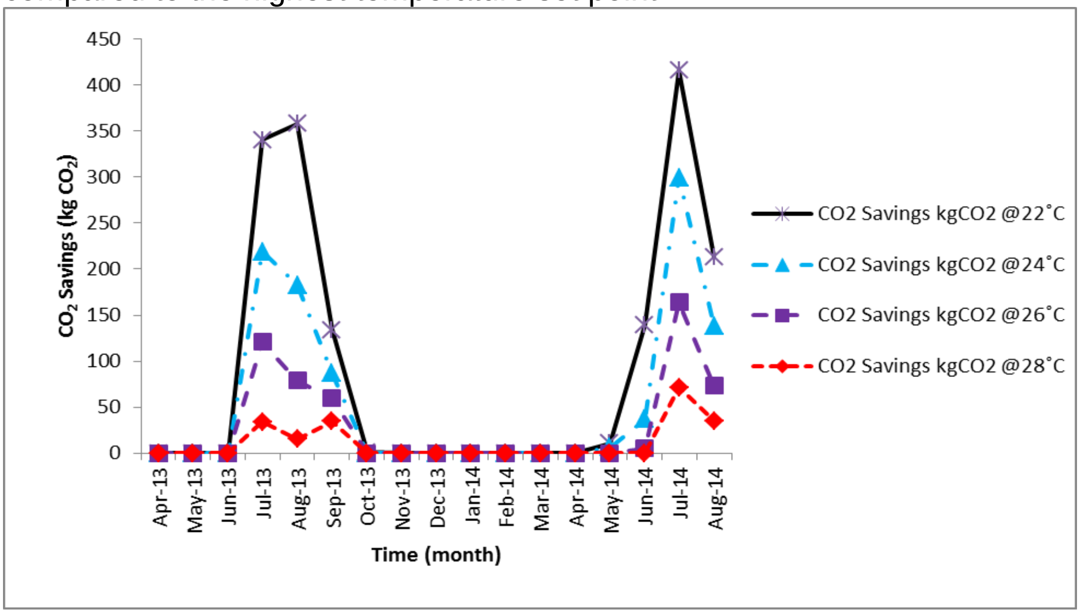

Figure 7 Monthly CO2 emission savings from different control strategy

6.0 Conclusion

This paper has described different control strategies for DAGS system optimization during the net cooling period to a university building. The investigation has focused on the effect of DAC on heat rejection to the ground, COP of the system, ground temperature variation, minimization of electric power consumption of the compressor and circulation pump, assuming certain values for the building load, the HP and DAC maximum cooling capacity. Therefore, it might not be considered as a full system optimization but it still could be considered as a determining improvement in system's operation.

This paper has shown that by utilising and controlling a DAC using different temperature set points, a significant reduction to GSHP operating cost, the electric power consumption and an improvement to performance of the system could be achieved. However, it is difficult to claim that this is the most economically beneficial scenario, not only because the heating period is not examined but, also because the investment and maintaining cost have not been considered in unit selection. This is subject of further work.

This paper has identified that the lowest temperature set point control is the best of the examined so as to regulate DAC's operation in the GSHPs. A comparison of these four scenarios illustrates that there are significant cost and carbon savings that can be made and all new control strategies achieve a better regulation to system operation which leads to an extra reduction in the energy consumption, carbon and cost savings. These remarks can be used as guidance to future GSHP designers.

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