EVALUATION OF A BUILDING INTEGRATED GROUND SOURCE HEAT PUMP USING SYSTEM PERFORMANCE FACTORS

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Abstract; A mathematical model of a ground source heat pump system has been developed and validated using experimental data from a full system installation. This system includes a heat pump with a fixed speed compressor, variable speed internal and external circulation pumps and provides space heating or cooling using fan coil units. In this study, the performance of the system, subject to varied boundary conditions, is evaluated using different SPF (system performance factor) definitions. The results show that the auxiliary components can consume a large proportion of the total system energy consumption. Depending on the definition of SPF, different system performances were observed. The internal circulation pump and fan coil units, which remain in constant operation, act to degrade the SPF, particularly as the building load factor is reduced. This degradation is present in both heating and cooling. In heating mode, the SPF increases for decreasing return water set-point temperature, which is primarily attributed to an improved heat pump COP. However, for decreasing return water set-point temperature, the fan coil unit heating capacity is reduced, which suggests an optimal return water temperature for maximising SPF, while matching the required building demand is possible. For cooling mode, a higher return water set-point temperature increases the SPF, but again the heat transfer across the fan coils is reduced, leading to a similar conclusion.

Key Words: heat pumps, SPF, model, simulation.

1 INTRODUCTION

The need to reduce global CO_2 emissions has become a concern of governments in recent years. For building heating and cooling, heat pumps are potentially one of the most efficient heating and cooling technologies available, particularly when low carbon electricity is utilised. The maximum possible coefficient of performance (COP) of a heat pump system is constrained by the temperature lift between the system source and sink. COP is further degraded by any additional heat exchangers, power consumption of auxiliary units such as circulation pumps and fan coils and losses in the compressor. Improvements in efficiency and correct sizing of components, along with improved system design and better system integration can reduce this degradation. A large amount of the literature focuses on design improvements to the efficiency of compressors, heat exchangers and other heat pump components. However, control optimisation at a systems level can also help improve system COP and system Seasonal Performance Factor (SPF). Air source heat pumps (ASHPs) are currently the most utilised type of heat pump, particularly for cooling applications, while ground source heat pumps (GSHPs) have predominantly been implemented in space heating applications (Urchueguía *et al*). Variable speed control of circulation pumps have been studied by Hepbasli *et al* (2003) and Fahlén and Karlsson (2005) for a GSHP system. Variable speed compressor control has also been compared to fixed speed by Da Silva *et al* (2010) and by Karlsson and Fahlen (2005). Adaptive water set-point control has been examined for small air-source heat pump systems with the use of heuristic control related to current building load along with night set-back and free cooling strategies (Guo and Nutter 2010), (Malmberg and Mattsson 2008). The motivation for the current study is to develop control strategies aimed at system level optimisation that considers the performance of all system components, from source to sink. While the COP is a measure of the instantaneous performance of the heat pump itself, the seasonal performance factor (SPF) refers to the performance of system components over an extended heating or cooling period and therefore gives a more realistic indication of overall system performance.

2 HEAT PUMP SYSTEM

2.1 Heat Pump System Description

In this study, a general mathematical model of a GSHP system was developed, based on independent models of the heat pump, circulation pumps, heat emitters, cooling devices and a coupling hydronic circuit (Corberan and Finn 2011). Validation of the model was carried out by reference to an installed water-to-water reversible ground source heat pump test rig with propane as its primary refrigerant. The heat pump system, a schematic of which is shown in Figure 1, is installed at the Universidad Politécnica de Valencia (UPV), Spain and has a rated capacity of 18.5 kW (35/40°C building, 15/10°C ground) for heating and 15 kW for cooling mode (15/10°C building, 30/35°C ground). The heat pump has a fixed speed compressor and variable speed circulation pumps in the internal and external hydronic circuits. The external circuit contains six boreholes and the building circuit is connected to twelve fan-coil units. The on/off cycling of the heat pump compressor is controlled by the return water temperature set-point and bandwidth, while the fan coil units bypass valves are controlled by the room air temperature set-point. The ground circulation pump is coupled electrically with the heat pump compressor and cycles in tandem with the compressor action. The rating of the internal and external circulation pumps is 840 W at a rated flow of 5 $m^3.hr^{-1}$ and 480 W at a rated flow of 2.5 $m^3.hr^{-1}$ respectively.

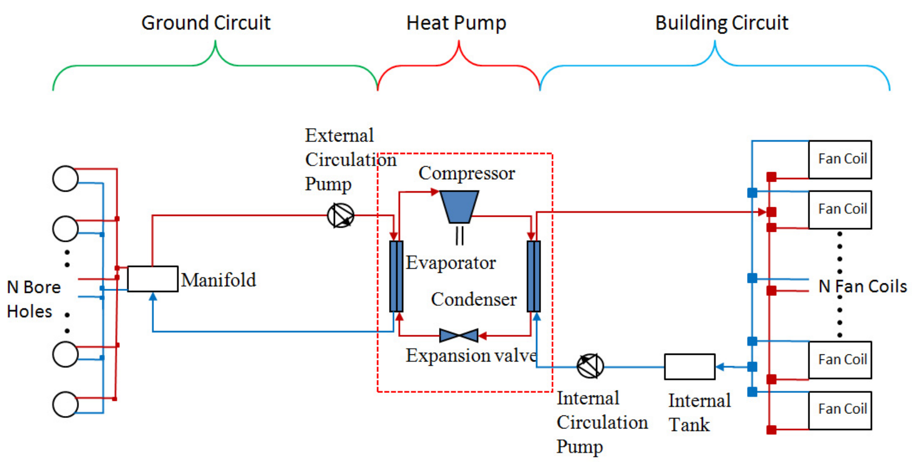

Figure 1: Schematic representation of ground source heat pump system.

2.2 Heat Pump System Model Development and Validation

The simulation model was developed with reference to experimental data gathered from the installed GSHP. The majority of system components, including the circulation pumps and fan coil units, were empirically characterised using curve fitting techniques from experimental data and manufacturer specifications. The heat pump model was developed using IMST-ART, a program for modelling steady state vapour compression cycles (Corberan *et al* 2002). The heat pump power consumption and capacities was characterised as a function of circulation pump speeds and return water temperatures using curve fitting methods. This was done for both heating and cooling and the component models were combined in MATLAB to create a quasi-steady state system model (MathWorks 2007). The complete GSHP system model was validated against collected data from the test rig in terms of heating capacity, cooling capacity and power consumption. Figure 2 compares simulation and experimental results for cooling over a daily cycle in terms of cooling capacity and compressor power consumption. Further details of the model are available in Corberan and Finn (2011).

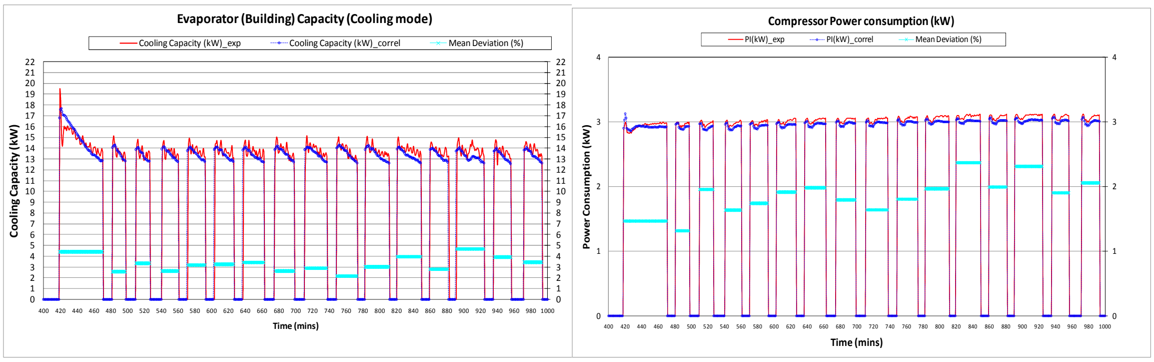

Figure 2: (a) Evaporator capacity and (b) compressor power consumption (cooling): simulated performance and experimental data.

3 SYSTEM PERFORMANCE FACTOR

Different definitions of seasonal performance factor (SPF) exist depending on the heat pump type and associated application domain. Nordman defines SPF_1 on the basis of the energy consumption of the compressor; therefore this is equal to the average COP of the heat pump over a given period of time, whereas SPF_2 considers the compressor and external circulation pump or external fan energy consumption. SPF_3 is utilised where any back up heaters used and finally SPF_4 includes all auxiliary drives in the system (Nordman et al. 2010).

For this study, a similar method is used to define a System Performance Factor (SPF) which is adapted for ground source heat pumps. The SPF as shown in Figure 3, is defined as the system performance over a specified interval of time, where the heat pump operates under quasi-steady state conditions. SPF_1 considers the heat pump alone and SPF_2 includes both the heat pump and external circulation pump. SPF_3 also considers the internal circulation pump and SPF_4 includes the fan coils units. Equations (1) to (4) summarise these definitions.

$$SPF_1 = \int_0^t \frac{\dot{Q}_{HP}}{P_{HP}} dt \qquad \textbf{(1)}$$

$$SPF_2 = \int_0^t \frac{\dot{Q}_{HP}}{P_{HP} + P_{S\_pump}} dt \quad (2)$$

$$SPF_3 = \int_0^t \frac{\dot{Q}_{HP}}{P_{HP} + P_{S\_pump} + P_{B\_pump}} dt \quad (3)$$

$$SPF_4 = \int_0^t \frac{\dot{Q}_{HP}}{P_{HP} + P_{S\_pump} + P_{B\_pump} + P_{B\_fans}} dt \quad (4)$$

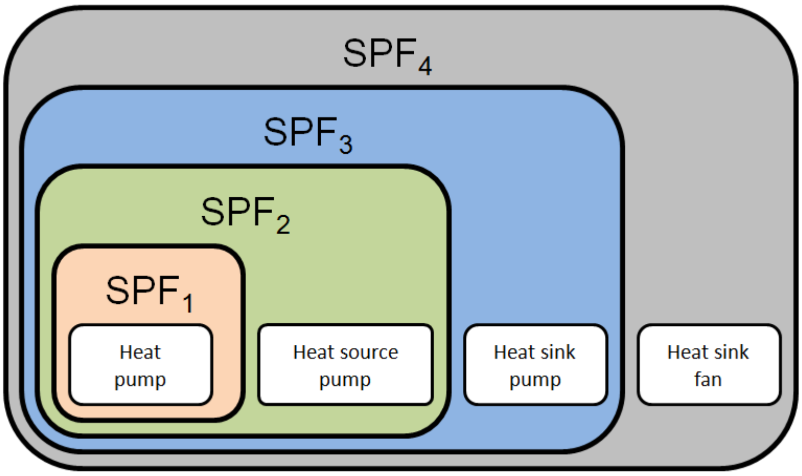

Figure 3: System boundaries for calculations of SPF.

4 RESULTS

Using the GSHP model, a series of sensitivity studies were completed to evaluate the system performance for a range of boundary conditions. The variables being controlled in this study include the building return water and the space air temperature set-points, where the external ambient temperature is assumed constant. All studies were carried out where quasi-steady state conditions prevail over a fixed time period. The performance of the system is evaluated in terms of the various SPF definitions as defined in Equations (1) to (4). Simulations were run with external ambient temperatures ranging from 6°C to 17°C in heating mode and 25°C to 35°C in cooling. This range was specified to demonstrate the system performance when operating marginally above capacity, i.e., with a load factor above 1.0 for the 6°C heating and 35°C cooling cases, and at a low load factor where heating/cooling may not actually be required in practice (at 17°C heating and 25°C cooling). The load factor is defined as the ratio of the heat pump capacity, as measured at the condenser (heating) or evaporator (cooling), to the building load.

The average energy consumption, for the simulation time period normalised over a one hour period, of the individual system components for a range of external ambient temperatures in heating and cooling mode is shown in Figure 4. In cooling mode, the set-point of the return water and space temperature is 12°C and 23°C respectively. In heating mode, the return water temperature set-point is 40°C and the space temperature is fixed at 22°C. In all cases the set-point bandwidth is ±1°C. Similar energy consumption results are observed at external temperatures of 9°C and 6°C in heating mode and 32.5°C and 35°C in cooling mode. At 9°C and 32.5°C the system is operating just under 100% capacity and is operating above capacity at 6°C and 35°C, for heating and cooling modes, respectively. Therefore the heat pump system is in constant operation for either mode and similar energy consumption values are observed. For heating mode, the compressor is the largest consumer of energy followed by the fan coil units. As the external temperature is increased, the energy consumption of the compressor decreases as an increase in external temperature reduces the building heating load. As the external circulation pump switches on and off with the compressor, the power consumption associated with its operation is proportional to that of the compressor. As the fan coil units and internal circulation pump are all the time in operation, a constant power consumption is observed for those components. Therefore at lower loads, these components comprise a large proportion of the total energy consumption than is observed at higher loads.

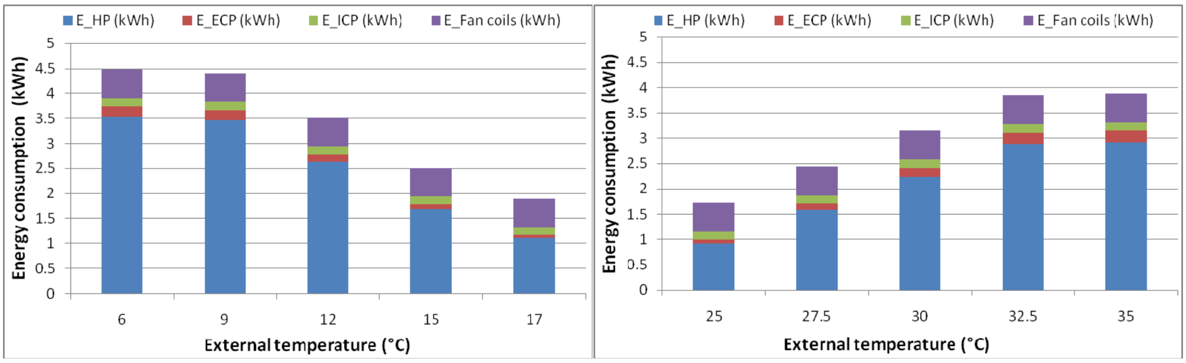

Figure 4: Average energy consumption normalised for a one hour period of the heat pump, internal and external circulation pumps and fan coils in quasi-steady state for different external temperatures in (a) heating and (b) cooling mode.

Figure 5 shows the variation of different SPF values for the range of external ambient temperatures examined in Figure 4, for both heating and cooling mode. Considering heating mode, a decrease in the external temperature is observed to result in an increase in the building load. As the external circulation pump is switched on and off with the heat pump, an increase in heat pump power consumption results in a proportional increase in external pump power consumption. As SPF_2 includes the external circulation pump energy consumption SPF_1 and SPF_2 track each other for all external temperatures. Slight variations in SPF_1 values for different external ambient temperatures (9°C, 12°C, 15°C, 17°C) are attributed to differences in the state of the system at the end of simulation. For the 6°C external condition in heating and 35°C in cooling, a load factor greater than one exists, which results in a reduction of building water return temperature set-point, hence the increased SPF_1 value for this case. Therefore these values of SPF must be considered in this context, as the heat pump is operating above capacity as the required load is not delivered. SPF_4 values are lower than the corresponding SPF_1 and SPF_2 values, as it considers the energy consumption of the internal circulation pump and fan coil units. SPF_4 shows a considerable increase for increasing building loads. As the fan coil units and internal circulation pump are constantly in operation, their associated energy consumption constitutes a larger proportion of the total system energy consumption at low loads, which results in a reduced SPF.

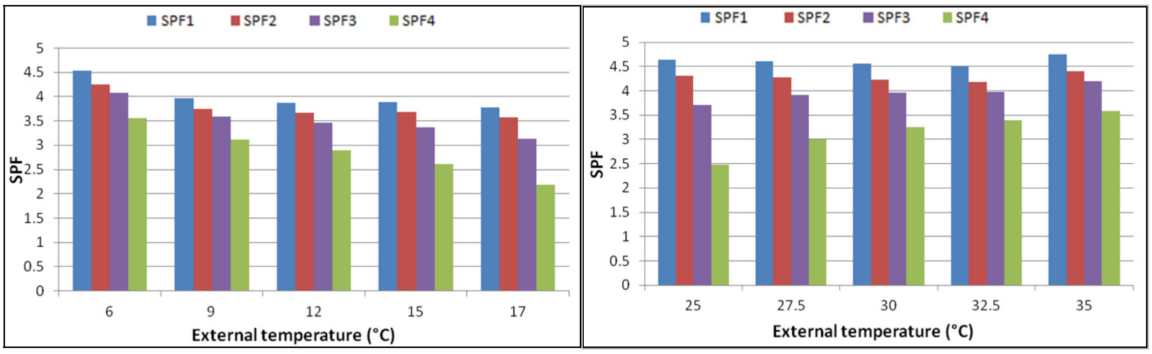

Figure 5: SPF_1, SPF_2, SPF_3 and SPF_4 in quasi-steady state for different external temperatures in (a) heating and (b) cooling mode.

In cooling mode, all SPF values are higher than in heating mode. This is because, in cooling mode, the average ground loop water temperature is closer to the average building water temperature. Comparing SPF_1, SPF_2, SPF_3 and SPF_4, it can be concluded that for this system, the fan coil units power consumption have a greater influence on the overall performance than the internal and external circulation pumps. SPF_1 is noted to be relatively constant for different building loads, however an increase in SPF_1 at 6°C external temperature in heating and 35°C in cooling can be observed. This occurs because the system is operating above capacity and the return water temperature is no longer capable of being controlled. In cooling this means that the return water temperature operates above its set-point, whereas in heating the water return temperature operates below its set-point. In both cases, this leads to an improvement in the respective SPF_1 values.

Figure 6 demonstrates the variation in SPF_2 and SPF_4 and load factor for a range of external and return water set-point temperature. In heating mode, as the return water temperature set-point is increased, the temperature difference between the heat pump condenser and evaporator is increased. This results in a decrease in heating capacity and an increase in the heat pump power consumption. Therefore, a decrease in SPF_2 and SPF_4 is observed for increased return water set-point temperatures. As mentioned earlier, the SPF_2 values do not decrease significantly for varying external temperatures, whereas the SPF_4 values decrease for increasing external temperature. The heat pump capacity is observed to exhibit a weakly increasing relationship for decreasing return water temperatures. At an external temperature of 6°C, the load factor is greater than one indicating that the system is operating at full capacity. Because of this, the SPF values converge as the return water temperature is no longer controllable at its set-point, resulting in a decrease in water set-point, which results in an increased SPF.

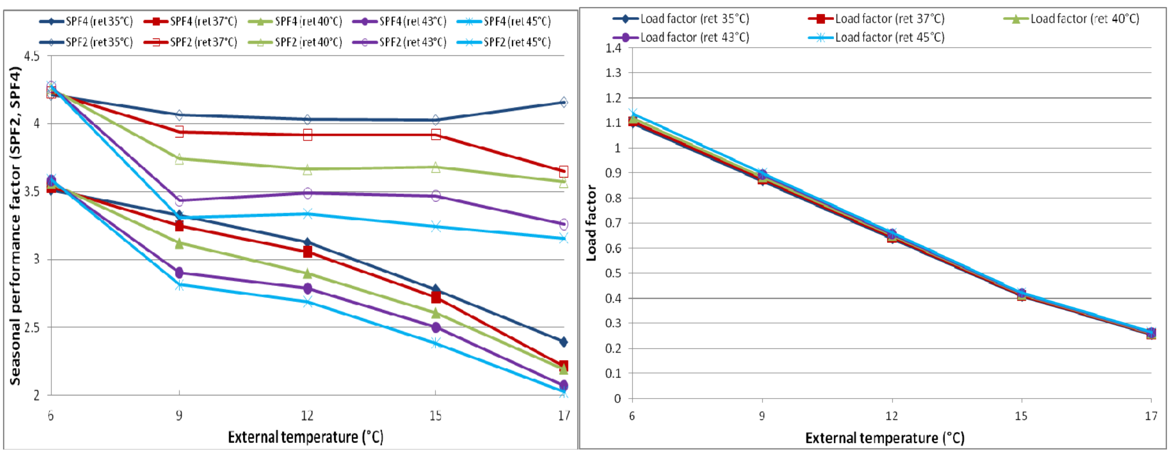

Figure 6: Quasi-steady state heating mode conditions: (a) SPF_4, SPF_2 and (b) load factor for different external temperature and return water temperature set-points.

Considering cooling mode as shown in Figure 7, a greater increase in SPF_2 and SPF_4 is observed for reducing temperature difference across the heat pump when compared to heating mode. In cooling mode, the refrigerant mass flow-rate increases for higher evaporator saturation pressures resulting in an increased cooling capacity. Because of this, the load factor is lower for increased return water set-point temperatures. As with heating mode, once the load factor rises above unity, the SPF values converge. These results suggest that raising the return water temperature in cooling mode and lowering the temperature in heating mode will maximise the system SPF. It should be noted however, that in both heating and cooling mode, reducing the heat pump temperature lift, results in a reduced temperature difference between the supply water temperature and the building space set-point, thus resulting in a reduced fan coil heat transfer to the building space. Therefore the heat pump system capacity is reduced. This may also result in the heat pump return water temperature bandwidth being exceeded before that of the space set-point. In that case the heat pump would cycle independently of the state of the fan coil units and the building space temperature would no longer be controlled.

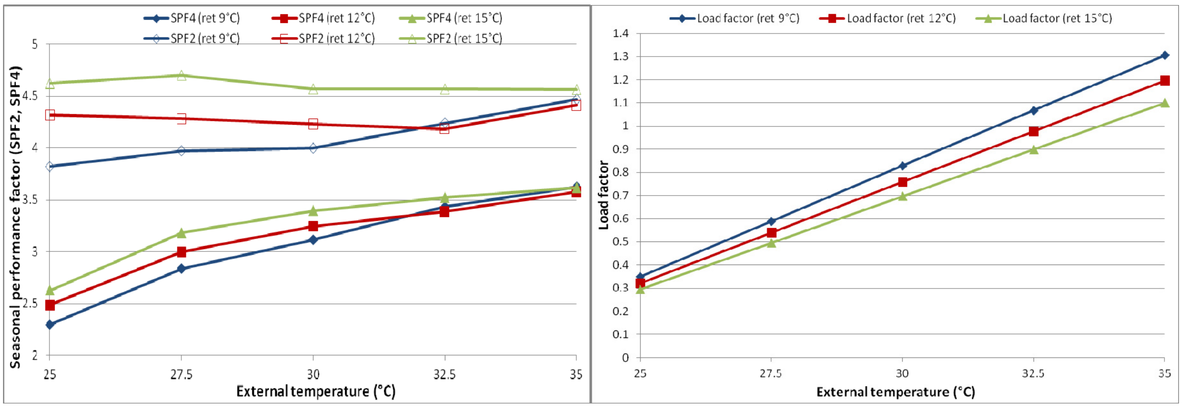

Figure 7: Quasi-steady state cooling mode: (a) SPF_4, SPF_1 and (b) load factor for different external temperature and return water temperature set-points.

Figure 8 shows the heat transfer across one of the fan coil units for different return water set-point temperatures to the heat pump. As noted in Figure 6 for heating, the SPF is observed to increase for decreasing return water temperatures. However for decreasing return water temperatures, it is noted that the fan coil heat transfer to the space is reduced. This is as a result of the reduced temperature difference between the water and the air in the fan coil units. Therefore, at low water return set-points and where large building loads exist, the fan coils may not be capable of matching the required building space demand. This would result in the building space temperature deviating from the specified space set-point. In cooling mode, a similar trend is observed. Although the heat pump capacity is increased for increasing return water temperature, the capacity of the fan coil units is reduced. Again if the capacity across the fan coils in not sufficient to match building load, control of the space temperature will not be possible. As the load factor only considers the capacity output from the heat pump, any change in the fan coil heat transfer is not represented in the load factor calculations as defined.

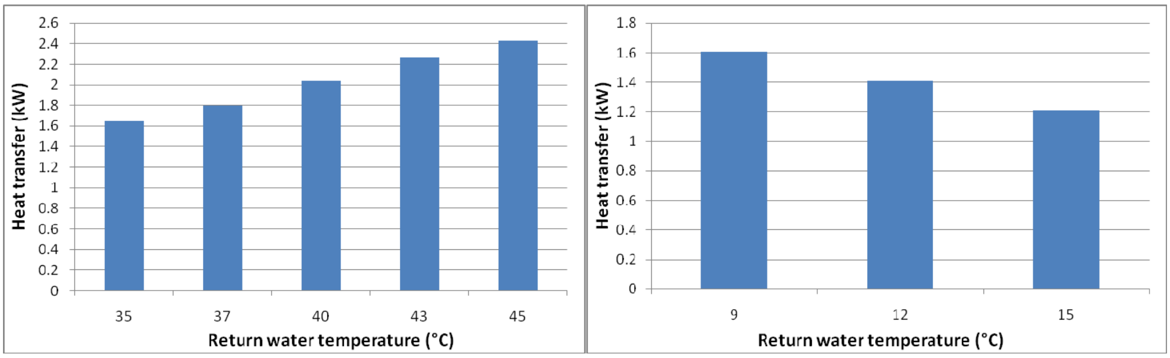

Figure 8: Fan coil capacity (single unit) for varying return water temperature for (a) heating mode with space set-point of 22C and (b) cooling mode with space set-point of 23C.

The variation in SPF_4 for a range of external temperatures and space set-point temperatures is shown in Figure 9. It can be seen that higher SPF_4 values are observed for higher space set-point temperatures in heating mode and for lower temperatures in cooling mode. This is due primarily to the increased building load evident at higher space set-point temperatures, where the relative penalty of the fan coils is mitigated under increased load factor resulting in an increased SPF_4. However the increased SPF does not reflect the penalty of increased energy consumption due to the altered building load. Therefore the energy consumption rather than the SPF of the system should be considered when adjusting the space temperature set-point. Similar results are observed for cooling. In cooling mode at external temperatures above 33°C, the system operates above its capacity.

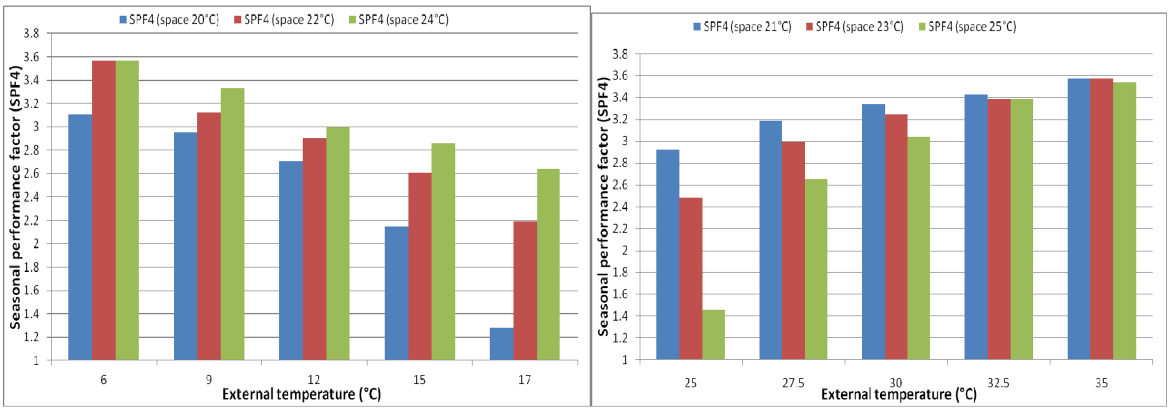

Figure 9: SPF_4 for different external temperatures and space temperature set-points for (a) heating and (b) cooling mode.

Figure 10 shows the total energy consumption of the heat pump system normalised for one hour for varying external temperatures and space temperature set-points. For higher space set-points in heating and lower set-points in cooling, an increase in energy consumption is observed. This is due to the increased building load. At an external temperature of 6°C for heating, the system is operating above capacity at the space set-point temperatures of 22°C and 24°C but not 20°C due to the reduced building load. Changing the space temperature by 1°C has nearly a three times greater effect on the system energy consumption than a similar change in the return water temperature in both heating and cooling. Therefore, the space air temperature should be controlled where possible to reduce the energy consumption load.

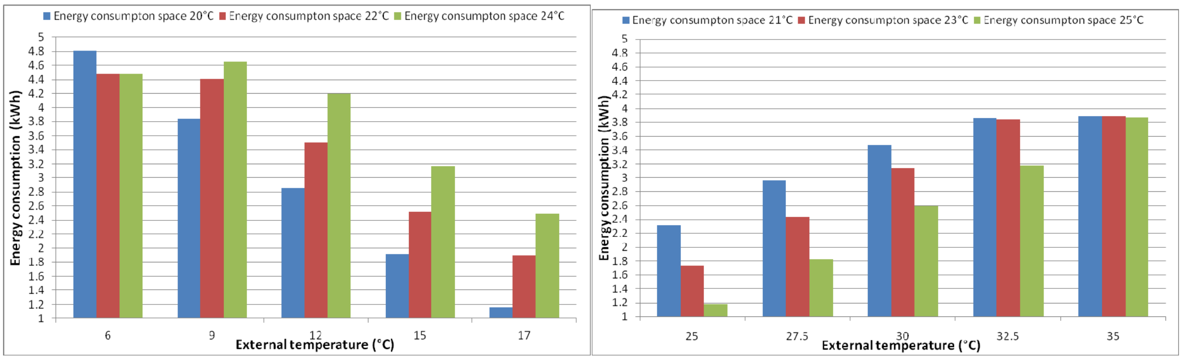

Figure 10: Total energy consumption over one hour for different external temperature and space temperature set-point and for a steady state simulation in (a) heating and (b) cooling mode.

5 CONCLUSIONS

As can be observed from the results auxiliary components consume a large proportion of the total energy consumption. At low building loads components such as the internal circulation pump and fan coils degrade the SPF to a larger extent than for high building loads. This degradation is present in both heating and cooling. An increase in SPF_4 is observed for lower return water temperature set-points in heating mode and higher set-points in cooling mode due to the improved heat pump performance. However, for these conditions the heat transfer across the fan coil units is reduced. By controlling the space temperature set-point the building load is effectively changed which can reduce the energy consumption of the system.

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Nomenclature

P	Power consumption [W]
$\dot{Q}$	Heat pump heating or cooling capacity [W]

Subscripts

B	Building	NFC	No fan coils
HP	Heat pump	S	Source
ECP	External circulation pump		
ICP	Internal circulation pump		