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Kelly, N.J. and Cockroft, J. (2011) *Analysis of retrofit air source heat pump performance : results from detailed simulations and comparison to field trial data.* Energy and Buildings, 43 (1). pp. 239-245. ISSN 0378-7788

ANALYSIS OF RETROFIT AIR SOURCE HEAT PUMP PERFORMANCE: RESULTS FROM DETAILED SIMULATIONS AND COMPARISON TO FIELD TRIAL DATA

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ABSTRACT

In the UK, gas boilers are the predominant energy source for heating in housing, due primarily to the ready availability of natural gas. The take-up of heat pumps has lagged far behind Europe and North America. However, with the development of standards for low and zero-carbon housing, gas price rises and the depletion of the UK's natural gas reserves, interest in heat pump technology is growing. Heat pumps, particularly air source heat pumps (ASHP), have the potential to be a direct, low-carbon replacement for gas boiler systems in housing.

In this paper, monitored data and simulations were used to assess the performance of ASHP when retro-fitted into a dwelling. This required the development and calibration of a model of an ASHP device and its integration into a whole-building, dynamic simulation environment. The predictions of the whole-building model were compared to field trial data, indicating that it provided a suitable test bed for energy performance assessment. Annual simulations indicated that the ASHP produced 12% less carbon that an equivalent condensing gas boiler system, but was around 10% more expensive to run. However, the proposed UK renewable heat incentive transforms this situation, with income from ASHP heat generation exceeding the fuel costs.

INTRODUCTION

In the UK, the legally-binding requirement for an 80% reduction in greenhouse gas emissions as set out in the UK's climate change act [1] and commitments made under the Kyoto protocol are driving increasingly rigorous requirements for building energy efficiency such as the net-zero-carbon target in England and Wales set out in the Code for Sustainable Homes [2] and a whole-life zero-carbon homes target in Scotland by 2030 [3]. In parallel, increasing fuel prices [4], particularly in the price of natural gas is forcing a re-think in the provision of domestic sector space and water heating. Approximately 80% of the UK's domestic water and space heating demands are met using gas boilers [5] and it is highly unlikely that the UK's ambitious emissions reduction targets could be achieved if this state-of-affairs persists. Hence, there is increasing interest in meeting the energy of buildings using low or zero-carbon (LZC) alternatives such as micro-renewables, biomass-based heating, and heat pumps. Heat pumps are belatedly attracting increasing interest in the UK in that they (particularly ASHP) offer a direct alternative to existing boiler installations. They can also operate in areas of high-density housing such as flats and terraced dwellings, where the installation of ground source heat pumps would be infeasible. High-density housing comprises approximately 40% of the UK housing stock [5], so the potential for retro-fitting of the technology is significant. Additionally, in an attempt to boost the installation of low-carbon technologies in the domestic sector, the UK government has recently launched a renewable heat incentive (RHI), which offers householders payments for heat produced from low or zero-carbon sources. Heat pumps are one of the qualifying technologies [6]. The effect of the RHI on the economics of a heat pump installation is illustrated later.

Heat pumps also complement the major changes occurring in large-scale electricity production: the UK is currently embarking on the development of huge quantities of onshore and offshore wind generation. For example, offshore sites with a potential generating capacity of 25 GW have recently been leased by the UK Government [7]. Heat pumps offer the potential for low or zero carbon heating as the carbon content of grid electricity reduces into the future with increasing quantities of renewable electrical generation.

Whilst there is an extensive literature on the performance of ground source heat pumps in the domestic sector (e.g. Healy and Ugursal [8], Kummert and Bernier [9] in Canada; and Underwood and Spitler [10], Jenkins et. al. [11] in the UK) the literature on ASHP performance is more sparse. Most modelling work focuses on specific aspects of device performance (e.g. Lui et al [12], Yao [13]) rather than integrated performance. In those performance studies that exist, Cockroft and Kelly [14] used a low-resolution model to determine that in a UK context ASHP could achieve carbon savings in comparison to other domestic heating technologies, including condensing gas boilers. Jenkins et al. [15] looked at the carbon savings potential of ASHP in office buildings and concluded that the technology did not guarantee emissions savings under all circumstances; this study used a performance map model of the ASHP and hourly predictions of heating and cooling from a simulation tool.

This paper builds on previous performance studies, specifically attempting to quantify the magnitude of the carbon savings when ASHPs are retro-fitted into existing housing. To this end, the following tasks were undertaken.

- ASHP performance data was collected from an existing field trial;
- an ASHP device model was developed and calibrated using laboratory test data;
- a whole-building simulation model of one of the field trial houses was developed and integrated with the ASHP model;
- the predictions of the whole building model were compared to the field trial data; and
- following some refinement, the model was used to predict the annual energy and economic performance of the ASHP and alternative heating systems.

The following sections describe these activities in more detail.

FIELD TRIAL DATA COLLECTION

Energy supplier SSE Ltd. and West Lothian District Council commissioned a field trial of ASHP heating systems in the village of Westfield in central Scotland with the objective of determining their effectiveness in meeting the space heating needs of the Council's existing housing stock. The climate and location of Westfield contribute to a significant requirement for heating: with the heating season typically running from early September until the end of May. So, in addition to being carbon intensive, the dwelling's existing heating systems (solid-fuel stoves) were extremely expensive to run often leaving tenants in fuel poverty[1]. Hence, one of the key objectives of the subsequent simulation exercise was to investigate the likely energy expenditure through a switch to ASHP for space heating and how this compared to the alternative retrofit options.

A total of 10 houses in the village were retrofitted with an ASHP system; the houses were all of a similar size (Table 1). Note that data was collected from only 8 of the houses as 2 withdrew from the study.

A typical terraced house encountered in the study is shown in Figure 1. The dwelling used in the simulation study had external walls comprising two-leaf, 100mm brick, with a 120mm insulated cavity (the building fabric was upgraded prior to installation of the heat pumps); the walls were rendered externally and wet-plastered internally. The pitched roof comprised concrete tiles lying on top of a bituminous felt and plywood skin supported by wooden trusses. The ceiling below the loft space was had 200mm of insulation installed. The floors in the dwelling were suspended timber, with a ventilated crawl space under the ground floor.

The retrofitted ASHP heating system comprised the ASHP unit directly feeding hot water radiators via insulated pipes running under the flooring. The ASHP used had a nominal coefficient of performance of 3.0 with a rated thermal capacity of 8 kW. Note that in these trials the ASHP served the space heating load only, with the hot water for each house being supplied using an existing 3kW direct electric heating coil, which heated a hot water storage tank; for cost reasons this was not integrated with the ASHP system. The radiators were sized for the nominal flow temperature of the ASHP device of 55^{o}C. The flow into each radiator (except a by-pass radiator, typically located in the hall of each dwelling) was controlled by a thermostatic radiator valve (TRV).

Figure 1: *terraced dwellings at Westfield.*

The ASHP was controlled based on the temperature reading from a wall-mounted thermostat located in the living room of each dwelling. The set point temperature could be selected by the occupant. During the trial, the occupants were also free to select the set point temperature of the heating system, its operation time, select TRV settings, open doors and windows, and to occupy the property as they normally would.

A radio frequency telemetry system, with wireless sensors/transmitters, was installed in each house, with one central receiver/logger. Data was recovered and transmitted wirelessly via the mobile phone network. The performance parameters recorded were as follows:

- the electrical consumption of each heat pump was measured using a current transformer clamp on the supply to the heat pump (the instantaneous apparent power consumption was derived by multiplying the recorded current draw in amps by the recorded site mains voltage);
- the total electrical consumption in each house was measured along with voltage (as voltage variations affect the performance of the heat pump system);

[1] In the UK the definition of fuel poverty is a household spending more than 10% of its income on fuel bills.

- the heat output of the heat pump was measured using a heat meter with integrating electronics and a pulse output per kWh; and
- outside air temperature and relative humidity were recorded.

Temperatures were measured at three locations (typically the living room, hall and a bedroom) in each house. The transmission interval for all sensors except electrical current clamps was set at 2.5 minutes. The current clamp transmitters were set at a 30 second transmission interval.

Monitoring started in February 2008, and stopped in July 2008. There were some interruptions due to equipment malfunctions so that a total of 91 days of data was collected for all houses over this time period covering winter, spring and early summer weather.

DEVELOPMENT AND CALIBRATION OF THE ASHP DEVICE MODEL

In order to simulate the performance of the ASHP heating system, an explicit device model was developed that could be fully integrated into a larger building and plant model (described later). From a performance analysis perspective, the main requirement for this model was to accurately predict the time-varying electrical demand and thermal output of the device and its explicit interaction the building's envelope, thermal plant, and control systems. The model had to be capable of predicting the key variables that coupled the ASHP device to the other constituents of a building simulation model, specifically the hot water output temperature and flow rate. Other outputs of interest included the device's operational status (e.g. on/off cycling, defrost status, temperature compensation, etc.) performance efficiencies, fuel consumption and resulting carbon emissions.

Ferguson et al. [16] developed a model that fulfilled similar criteria, but that was employed in the modelling of micro-cogeneration devices; this model was adapted for the purposes of this study. The model is best described as 'grey-box', where the structure reflects the underlying physical features of the device; however, its key performance characteristics are approximated using empirically-derived expressions.

Figure 2: *diagrammatic representation of the ASHP model*

The model structure comprises three control volumes as shown in Figure 2. The first (functional) volume is employed to calculate the performance of the refrigerant circuit and auxiliary components. The algorithm underpinning the volume calculates the time-varying COP and the nominal heat output ($\dot{Q}'_{th}$) from the refrigeration cycle using 2nd-order polynomial functions of the heating system return water temperature and the external ambient temperature difference (ΔT) (Equations 1 and 2).

$$COP(t) = f_1(\Delta T) \qquad (1)$$

$$\dot{Q}'_{th}(t) = f_2(\Delta T) \qquad (2)$$

where,

$$\Delta T = T_{return}(t) - T_{\infty}(t) \tag{3}$$

and

$$Q'_{th}(t) = Q_{th}(t) + UA[T_A(t) - T_{\infty}(t)] \tag{4}$$

Where $\dot{Q}_{th}$ is the heat supplied (W) to the water in the heating circuit; T_{return} is the temperature of the water flowing into volume B; T_A is the temperature (^oC) of volume A; UA is the heat loss coefficient (W/K) to the environmental temperature (T_{∞}).

The total instantaneous electrical demand of the device compressor, $\dot{Q}_e$ (W) is calculated as follows:

$$\dot{Q}_e(t) = \frac{\dot{Q}_{th}(t)}{COP(t)} \tag{5}$$

The performance map equations were augmented (1)-(5) with algorithms to calculate the defrost status of the device and to modulate the return water temperature set point based on outside temperature.

The heat output from equation (2) is passed to a lumped-capacitance volume (volume A) that essentially represents the thermal capacitance of the device on the refrigerant-side of the condenser. Heat is transferred between this volume and volume B representing the water-side of the condenser. Volume B is also the connecting point to the heating system. Together, these lumped mass volumes enable the transient thermal response of the device to be encapsulated in the model output. The general form of the energy balance for these volumes A and B is as follows

For volume A:

$$M_A c_A \frac{dT(t)_A}{dt} = \dot{Q}'_{th}(t) - UA[T_A(t) - T_{\infty}(t)] - K_{AB}[T_A(t) - T_B(t)] \tag{5}$$

For volume B:

$$M_B c_B \frac{dT_B(t)}{dt} = K_{AB}[T_A(t) - T_B(t)] - \dot{m}(t)c[T_w(t) - T_B(t)] \tag{6}$$

Where M_i is the mass (kg) of volume A or B; T_B is the temperature (^oC) of volume B; K_{AB} is the heat exchange coefficient (W/K) between the volumes A and B; and $\dot{m}$ is the flow rate of coolant (kg/s) to volume B; c is the specific heat of water (J/kgK);

The parameters for the polynomials used in equations 1 and 2 were calibrated using data from laboratory performance tests conducted on the ASHP; these calibrated equations enable the performance characteristics of the device to be simulated over the range of ambient and flow temperatures experienced during normal operation. The parameters of the capacitance volumes A and B were calibrated using an iterative parametric identification technique described by Ferguson et al [16] using high resolution start-up data from the field trial data.

A comparison of the predictions of COP from the calibrated model are shown against measured data (independent of the calibration data set) in Figure 3.

Figure 3: *calibrated COP characteristics of the ASHP device*.

INTEGRATED MODEL AND SIMULATION

The ASHP device model was integrated into a larger building and systems model of one of the test houses developed on the ESP-r building simulation tool [17]. The model comprised a detailed representation of the dwelling, the heat pump and heating distribution system. Each individual room was explicitly represented in the model and the constructions used were identical to those in the actual building. The building model was also augmented with a zonal airflow model; this calculated air-exchange between the different spaces within the building and between the building and the exterior according to prevailing wind-driven pressure differences and temperature differences. The model was calibrated to give the same external air leakage rate as that was determined from a blower door test of the actual building (approximately 0.5 air changes per hour).

The heating system was modelled in detail, where all of the heating system's components (radiators, piping, valves, etc.) and their inter-connections are modelled explicitly. A comprehensive description of the basic heating system component models used in this work is given by Hensen [18] and Clarke [19].

As no monitored occupancy data was available, characteristic internal gains profiles were developed for weekdays and weekend periods. Profiles for the lounge and bedroom are shown in figures 5a and 5b. The profiles are intermittent during the week and more continuous at weekends. Compatible heating system control regimes were developed, with the heating system operating 07:00-09:00 and 17:00-22:00 during weekdays and 08:00-23:00 at weekends. The assumption was made that the system was switched off at night rather than reverting to a set back-temperature (this is common practice in the UK). For the purposes of the heating system comparison, the heat gain profiles were normalized to be consistent with the sensible internal heat gains employed in the UK's standard assessment procedure (SAP [20]) of some 55.7kWh/week.

Figure 4a: weekday and weekend *heat gain profiles for the lounge*.

Figure 4b: weekday and weekend *heat gain profiles for the bedroom*.

SIMULATIONS

The integrated ASHP/dwelling model was simulated at a time resolution of 1-minute intervals; this fine temporal resolution was employed to adequately capture the action of control on the operation of the ASHP (e.g. on/off cycling) and other pertinent operations such as defrost and any overheat shutdowns. This temporal resolution has been employed by other researchers, notably Hawkes and Leach [21] who indicated that high resolution modelling was required in order to adequately characterise performance. Finally, as no long-term climate data existed for the Westfield site, the simulations were run with an equivalent Scottish climate data set that was representative of the field trial location.

The simulation of the model was done in two stages. First, the model's predictions for heat pump performance were compared to the data collected from the field trials, with the model's parameters being adjusted where necessary. Second, the adjusted model was simulated over a full-year period and aggregate annual performance data was extracted from the high-resolution time series output.

COMPARISON WITH FIELD TRIAL DATA

Figure 5a shows the model's predictions of the coefficient of performance (COP) vs. ambient air temperature against averaged COP data from the field trials; this average relationship was derived from the measured COP values from the 8 monitored houses and represents typical ASHP performance under varying conditions of occupancy, climate and internal conditions. As the climate and assumed occupancy for the simulation differed from the exact conditions experienced during the field trials (this data was not captured in the monitoring data) a temporal comparison of simulation and monitored results would not be appropriate. Instead, the comparison looks at whether the range and trend of the COP emerging from the simulations is similar to the averaged trend from the actual houses: this would indicate that the simulation is providing as a suitable test bed for the device in that the operating conditions and resulting performance are similar to those encountered in reality.

The initial simulation results clearly diverged from the monitored data at higher ambient air temperatures (Figure 5a). Closer investigation revealed that the temperature compensation facility on the heat pumps used in the Westfield houses had not been enabled due to an installer error. Consequently, a further simulation was conducted with the temperature compensation disabled in the model. Figure 5b shows the results from this simulation. It is evident that the low-density scatter in the ASHP model output is predominantly above the average COP line: these points are indicative of the dynamic nature of the model and represent periods when the heating system and building are warming up and where the resulting low heating system flow and return temperatures produce a temporarily high COP. However, the highest density of points is close to the average performance line; corresponding with periods in which the heat pump was operating with the return water temperature close to the set point (45^{o}C); this demonstrates that the ASHP device in the ESP-r model is operating in under similar conditions to those experienced in the actual houses.

Figure 5a: *predicted COP vs Westfield average COP*.

Figure 5b: *predicted COP (no temperature compensation) vs Westfield average COP*.

ANNUAL SIMULATIONS

Following the comparison against the field trial data, a full annual simulation data (with temperature compensation enabled) was initiated and to give an indication of likely annual ASHP performance when retro-fitted into the Westfield dwelling model.

Two further simulations were undertaken, again at a time-resolution of 1-minute. First, the ASHP heating system model was replaced with a model of a gas-condensing boiler heating system. The boiler had a rated thermal output of 10.5 kW at full load. Other than the re-sizing of the radiator components to account for a higher supply temperature of 75^{o}C, the system details are the same as those for the ASHP-based system shown in Figure 2. The nominal efficiency of the boiler device model used was 91%, which is typical of a modern boiler being installed in the UK at present. Note that the resulting simulated *system* efficiency was lower due to parasitic losses from piping, control inefficiencies and on/off cycling.

In the final simulation, space heating was provided by means of a simple, direct-electric system that consisted of an electric heater in each room of the house. Each heater was assumed to be individually thermostatically controlled and with an assumed electric-to-heat conversion efficiency of 100%.

RESULTS ANALYSIS

The simulations of the dwelling with the three different heating options produced large volumes of performance data in the form of time series temperatures, flows, heat fluxes and derived quantities such as efficiency and fuel consumption. Figure 6 is a typical example of the high resolution data extracted from the simulations and shows the ambient air temperature, ASHP heat output, ASHP power consumption and return water temperature during part of a typical winter weekday when the heating system cools down overnight, restarts in the morning and then cycles to maintain internal temperatures.

Figure 6: *typical winter day results from the ASHP simulation.*

Annual performance metrics were extracted from the high-resolution data and are shown in Tables 2a-2c. Note that the entries for June, July and August are greyed-out on each table as the heating system was not active during these months.

Looking first at the ASHP system, the average annual COP for the device was approximately 2.7 (in comparison to its nominal COP of 3.0), while the annual electricity consumption to meet the space heating load was some 5.87 GJ (1631 kWh). The distribution system efficiency was approximately 92%

Turning to the condensing gas boiler, the total fuel consumption was 18.99 GJ or 485 m^3 of gas. The average device efficiency was 88%. However, the overall system efficiency (including parasitic losses) was approximately 81%. The distribution system losses are slightly higher than with the ASHP as the supply water temperature to the heating system is higher (70-80^{o}C). On/off cycling of the boiler is also greater than the ASHP, particularly during the winter months, due primarily to the higher heating output of the gas boiler and lower thermal capacitance of the device in comparison to the ASHP.

The overall energy consumption/heat output of the direct-electric heating system is 3640 kWh. The heat output of this system is approximately 12 and 17% lower than the ASHP and gas condensing boiler systems respectively. This energy saving was predominantly due to the fact that each electric space heater in the dwelling was separately controlled in comparison to the other two systems which were controlled from a central thermostat; this resulted in more accurate temperature control in each room. For the other two systems, the central thermostat resulted in accurate temperature control in the living room and some over-heating on other spaces.

Based on these energy consumption figures the CO_2 emissions were calculated as 881, 1002 and 1966 kg for the ASHP, gas boiler and all-electric systems, respectively; these CO_2 emissions are based on CO_2 intensities of 0.19 kg/KWh for natural gas and 0.54 kg/KWh for grid electricity [22]. The ASHP system produces approximately 12% less CO_2 than the gas condensing boiler and 55% less CO_2 than the all-electric heating system. The reduction in emissions in comparison to the gas unit is particularly sensitive to electricity CO_2 emission factors, deteriorating as these increase and vice-versa. It should be noted that that the trend for UK grid electricity CO_2 intensity has been upwards in recent years rising from 0.52 kg/kWh in 2001 to the current value of 0.54 [22].

Again, based on the calculated energy consumption figures and using typical UK gas and electricity prices seen in late 2009 of £0.121 and £0.034 for electricity and gas respectively [23] it is evident that there is a running cost penalty associated with the ASHP system, which is around 10% more expensive to run than the condensing gas unit per annum However, the ASHP system is 55% less expensive to operate than the all-electric system. There will be an additional capital cost penalty in that (at least initially) ASHP devices will have a higher capital cost than the more conventional gas condensing boiler and the electric systems. As with CO_2 savings, these figures will be sensitive to future variations in gas, electricity and technology prices.

ASHPs will benefit from the introduction of the UK Renewable Heat Incentive (RHI), which will begin to operate over the whole of the UK in early 2011. The projected incentive for ASHP heat is £0.075 per kWh of thermal output (for units under 45kW thermal output) [6]. Accounting for the RHI (last entry in Table 3) results in a turnaround in the running costs of the system. The expenditure on fuel remains the same, however the income from heat generation amounts to some £330, with the result that there is a net *income* from operating the ASHP of £132.71. The application of the RHI therefore transforms the economic viability of ASHP heating systems in comparison to the conventional alternatives.

CONCLUSIONS

A combination of simulation and field trial data has been used to assess the annual performance of a domestic ASHP heating system when retro-fitted into social housing in Westfield, Scotland.

A detailed ASHP device model has been developed on the ESP-r platform, based on the work of Ferguson et al. [15]. The model was calibrated using laboratory data and then integrated into a larger whole-building model of one of the Westfield dwellings.

The predictions of COP from the model were compared to data emerging from the Westfield field trial; these initially indicated a divergence between monitored and simulated values. Investigation revealed some shortcomings in the ASHP installation at Westfield, where outside air temperature compensation was not enabled on the installed units. Adjusting the simulation model to account for this flaw and re-simulating indicated that the model suitably replicated the ASHP operating conditions observed in the field trial.

Simulations were undertaken to estimate the annual energy performance of the ASHP device and an equivalent gas condensing boiler system when retro-fitted into a typical Westfield dwelling. These indicated that: the provision of space heating using the ASHP resulted in approximately 12% CO_2 savings in comparison to a gas condensing boiler system and 55% CO_2 savings compared to the all-electric system (using 2009 UK CO_2 emissions coefficients for electricity and gas).

ASHP running costs were 10% higher than the gas-condensing boiler system, but 55% lower than direct-electric heating (using 2009 UK average electricity and fuel prices). However, the introduction of a renewable heat incentive (RHI) in the UK in 2011 will completely alter this situation, with the income from payments for heat from the ASHP exceeding the fuel (electricity) costs.

ACKNOWLEDGEMENTS

The authors would like to thank SSE plc. who funded the field trial and performance study; and Mr Chris Chisman of TEV Ltd. who provided some heat pump technical data. The development of the heat pump device model was part-funded through SUPERGEN Highly Distributed Energy Research Consortium, which is part of the Research Council’s Energy Programme.

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Tables

Table 1: *westfield house types.*

Storeys	Type	Date of construction	Bedrooms	Roof	Windows	Number of houses
1	Semi-detached	1969	1	Concrete tile	double glazed	1
1	4-in-a-block flat	1938	3	Natural slate	double glazed	2
2	End terrace	1938	3	Natural slate	double glazed	1
2	End-terrace	1967	3	Concrete tile	double glazed	2
2	Mid terrace	1967-1969	3	Concrete tile	double glazed	4

Table 2a: *ASHP detailed simulation output.*

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Year
Actual COP	2.58	2.55	2.71	2.77	2.90				3.10	2.91	2.78	2.67	2.77
Electrical(kWh)	308	261	219	113	57				37	143	221	272	1631
ASHP Heat Out (kWh)	793	667	593	314	164				113	418	613	727	4403
System Heat Loss (kWh)	46	44	41	35	30				22	37	39	42	335
Total System Heat Out (kWh)	747	623	552	279	134				91	381	574	685	4067
ON/OFF cycles (-)	459	456	477	301	166				116	388	478	470	3311

Table 2b: *boiler detailed simulation results.*

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Year
Efficiency	0.88	0.88	0.88	0.88	0.89				0.89	0.88	0.88	0.88	0.88
Gas (kWh)	937	788	701	385	203				144	521	732	865	5275
Boiler Heat Out (kWh)	821	691	617	341	180				128	459	643	759	4638
System Heat Loss (kWh)	34	40	45	47	41				38	56	42	36	379
Total System Heat Out (kWh)	787	652	571	294	139				91	403	601	722	4259
ON/OFF cycles (-)	1087	932	911	528	282				206	758	964	1044	6712

Table 2c: *direct electrical heating simulation results.*

	Jan	Feb	Mar	Apr	May	Jun	July	Aug	Sept	Oct	Nov	Dec	Year
Total System Heat Out (kWh)	692	572	496	250	116				69	320	505	620	3640

Table 3: *comparison of ASHP and gas condensing boiler system (space heating only)*.

	Fuel Price p/kWh	Annual Energy use kWh	Annual Cost £	Annual CO_2 emissions kg
ASHP heating	12.11	1,631	197.51	881
Gas condensing boiler heating	3.41	5,275	179.88	1,002
All-electric heating	12.11	3,640	440.80	1966
ASHP heating (with RHI)	12.11	1,631	-132.71**	881

** A renewable heat incentive payment of £0.075 per kWh of heat produced (4403kWh) amounts to a payment to the householder of £330.22; the net income is therefore £132.71.

Figure1
Click here to download high resolution image

Figure2

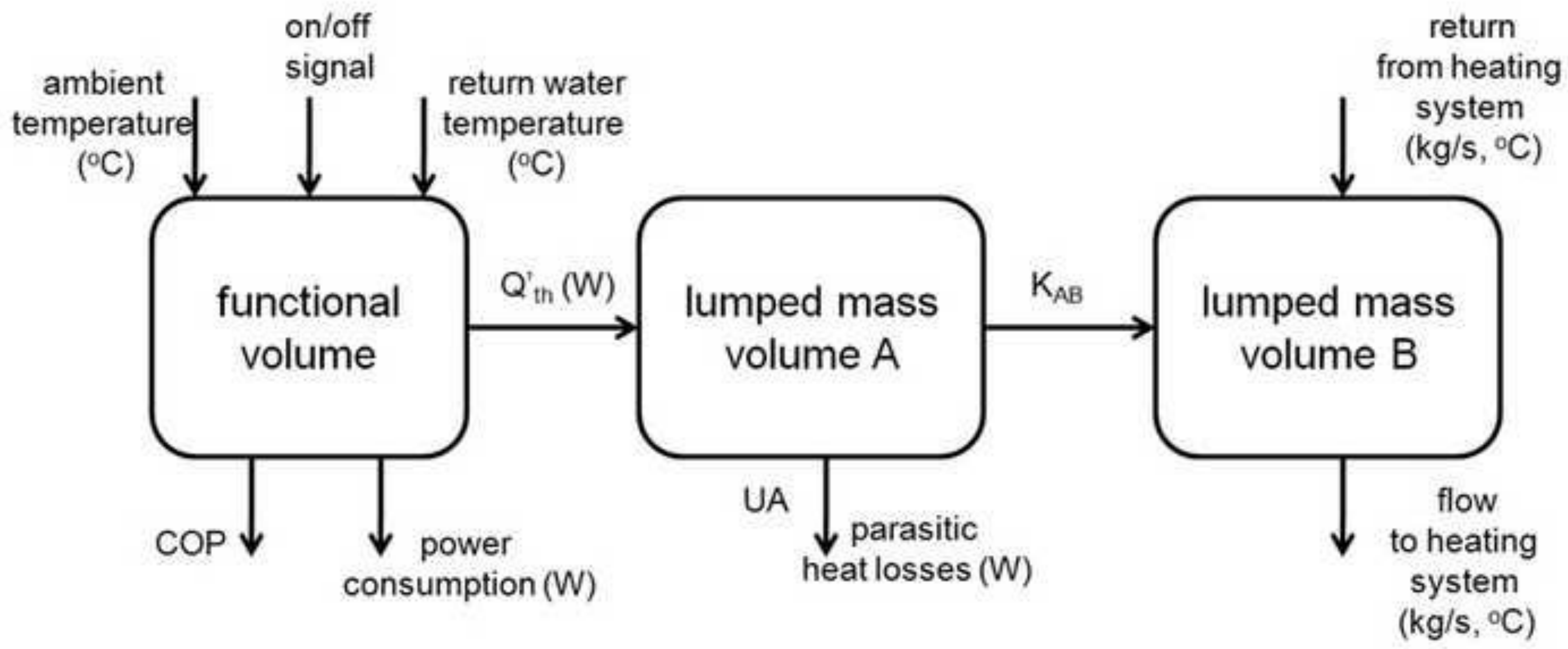
ambient temperature (°C)
on/off signal
return water temperature (°C)
functional volume
COP
power consumption (W)
Q'th (W)
lumped mass volume A
UA
parasitic heat losses (W)
KAB
lumped mass volume B
return from heating system (kg/s, °C)
flow to heating system (kg/s, °C)

Figure3

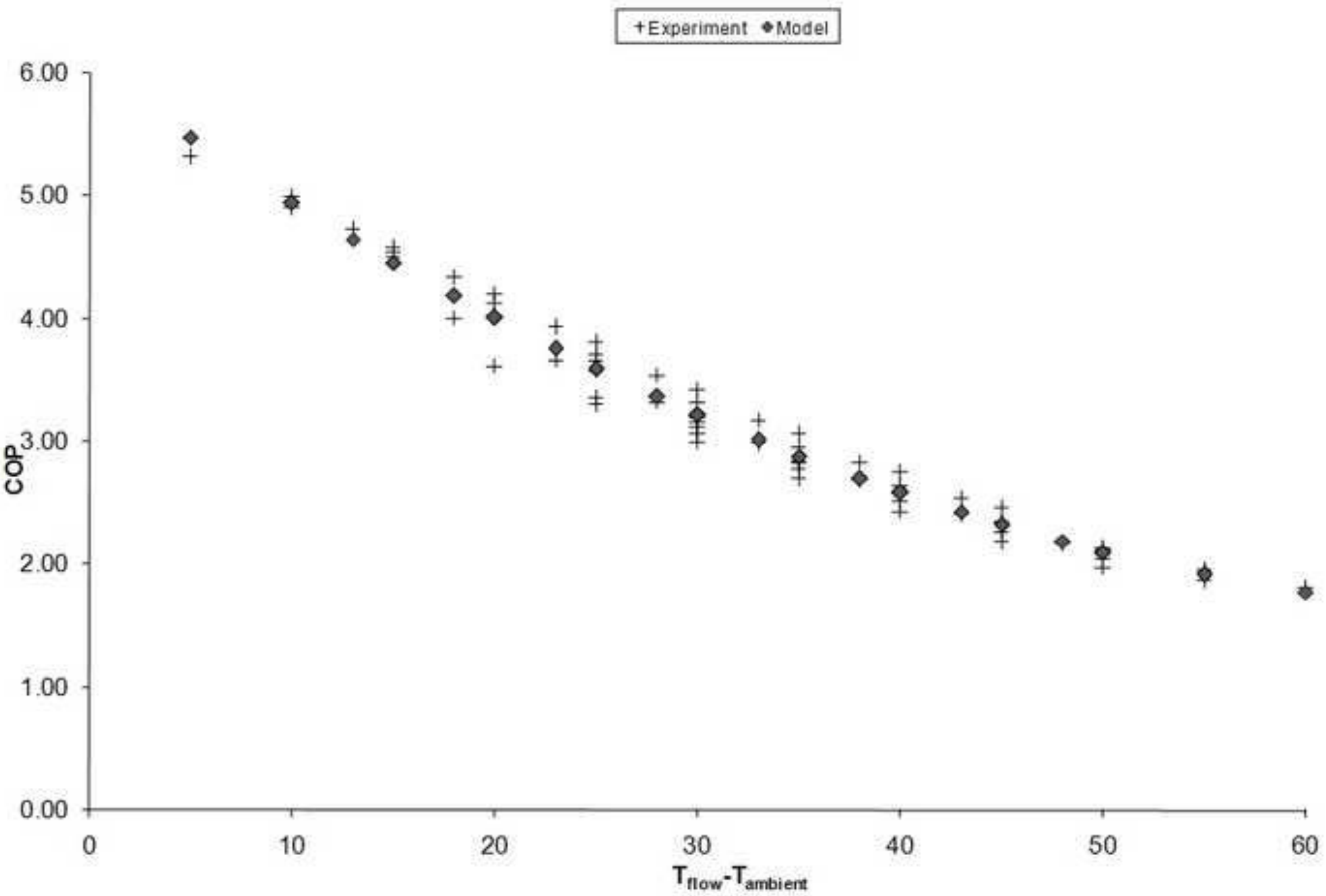
+Experiment ◆Model
6.00
5.00
4.00
3.00
2.00
1.00
0.00
COP
0
10
20
30
40
50
60
T_{flow}-$T_{ambient}$

Figure4a
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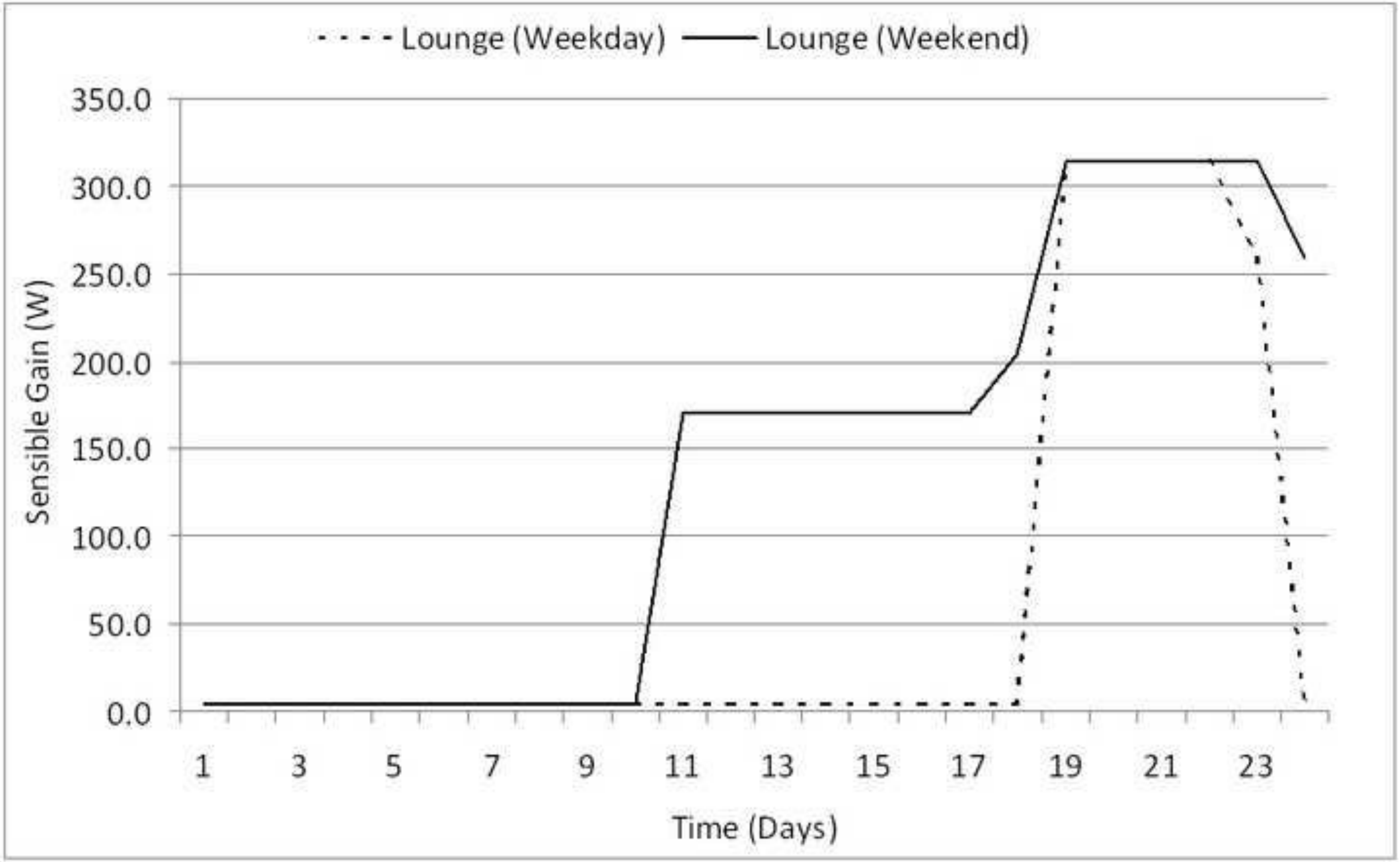
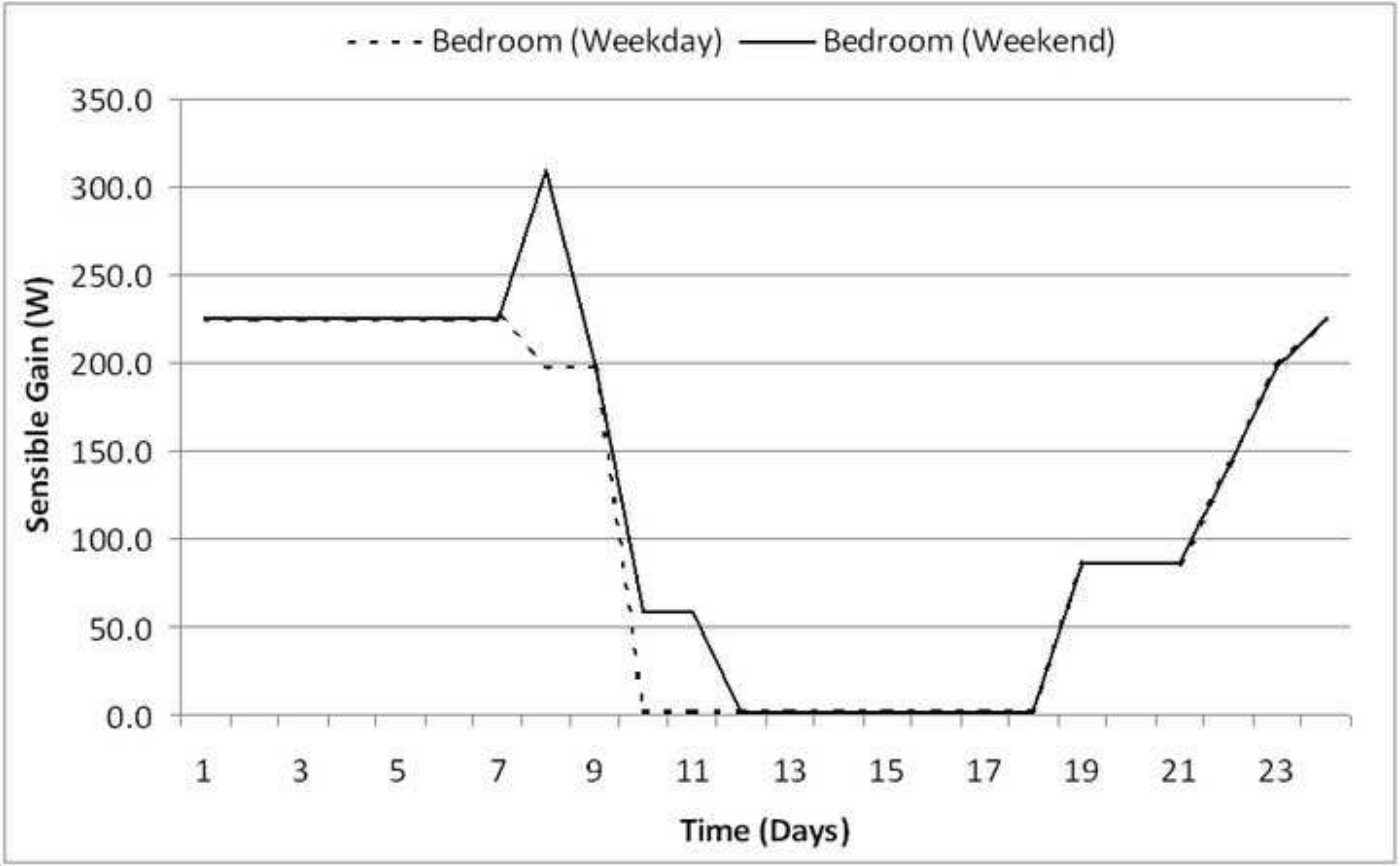


Figure4b



- - - - Bedroom (Weekday)
—— Bedroom (Weekend)
Sensible Gain (W)
350.0
300.0
250.0
200.0
150.0
100.0
50.0
0.0
1
3
5
7
9
11
13
15
17
19
21
23
Time (Days)

Figure5a

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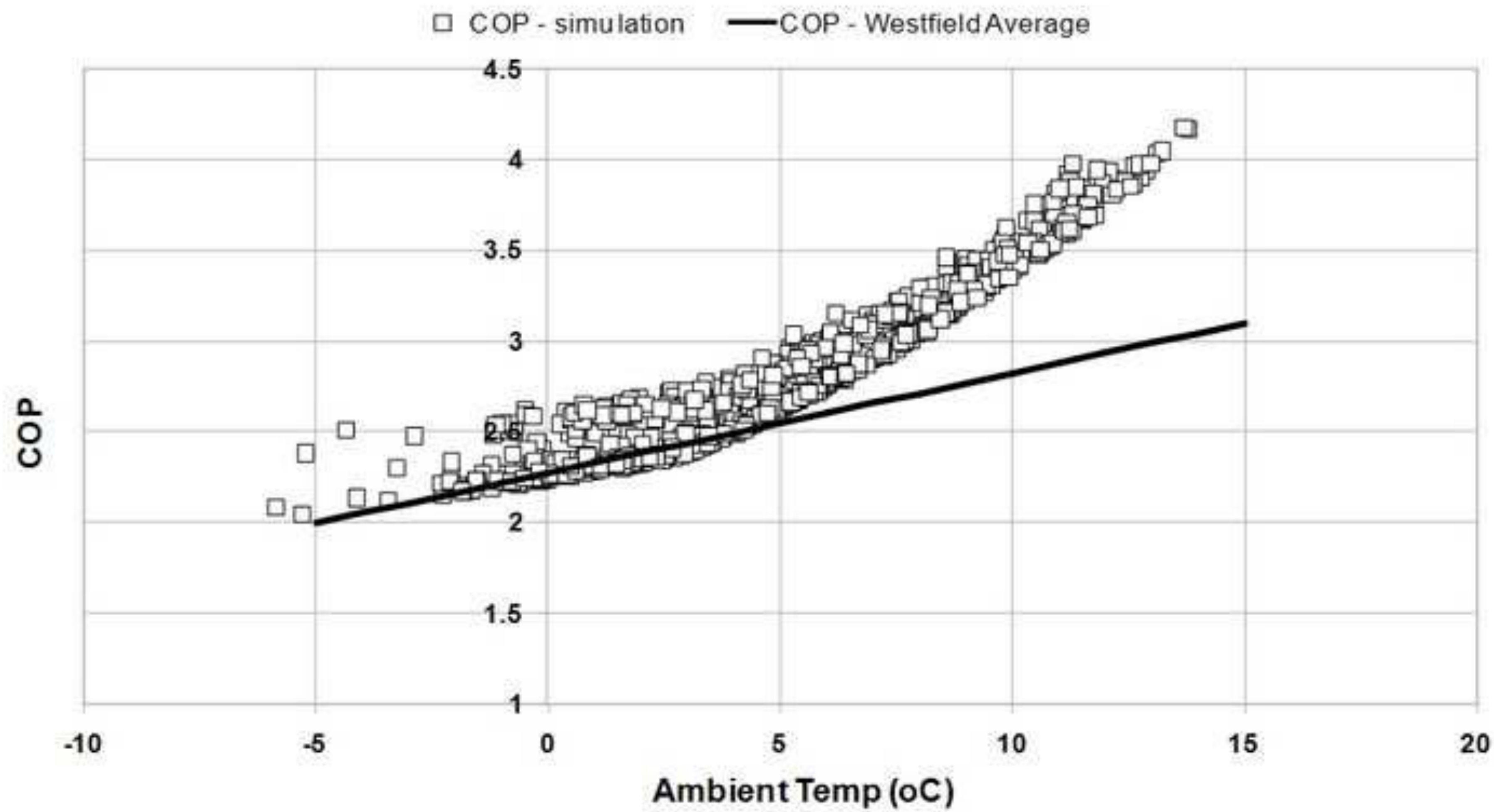

Figure5b

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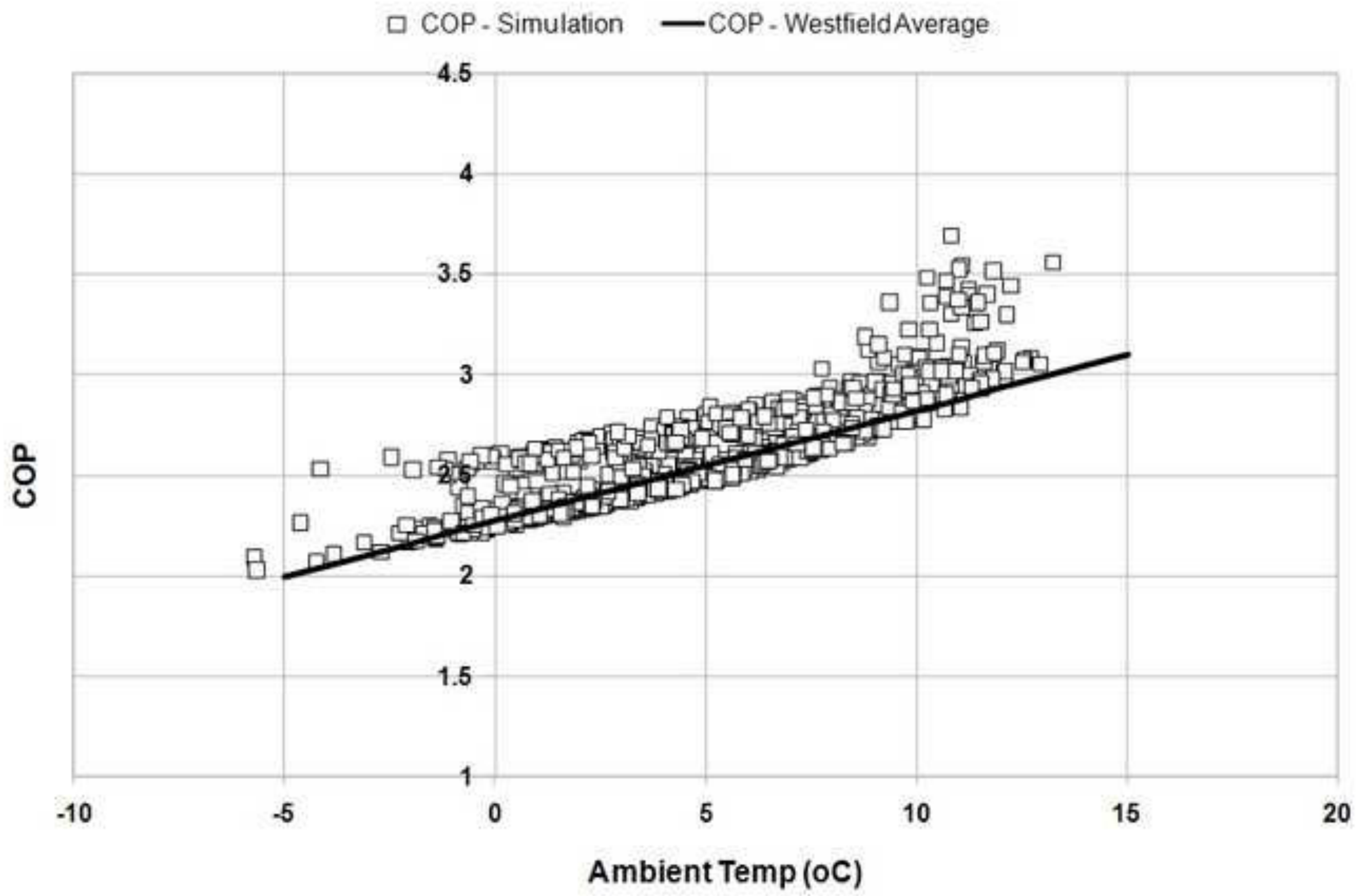
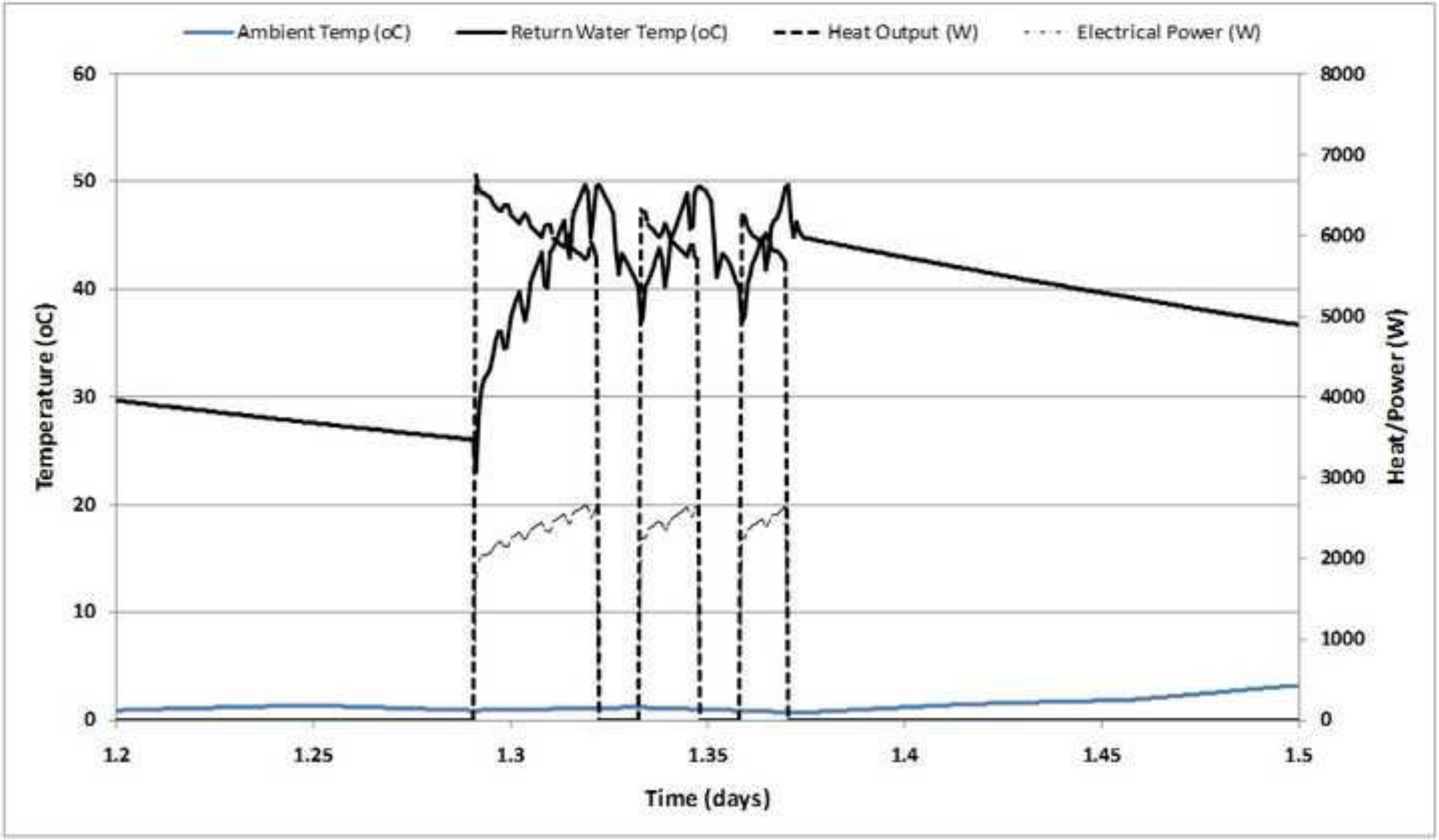


Figure6


Ambient Temp (oC)
Return Water Temp (oC)
Heat Output (W)
Electrical Power (W)
60
50
40
30
20
10
0
Temperature (oC)
8000
7000
6000
5000
4000
3000
2000
1000
0
Heat/Power (W)
1.2
1.25
1.3
1.35
1.4
1.45
1.5
Time (days)