Monitoring of Non-Domestic Renewable Heat Incentive Ground-Source & Water-Source Heat Pumps

Final Report

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This report has not considered the detailed design of the heat source or ground loop, sizing of the buffer vessel nor sizing (or nature) of the heat emitters. All of these factors are known to be critical to the overall performance of a heat pump. The data presented in this document should therefore not be taken as necessarily representing the performance of other installations using the conceptual designs described. No information has been presented on the suitability of the systems as to their intended purpose. Caution should be taken when extrapolating the data presented in this report to other applications.

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David Hughes.

Glossary

Term	Explanation
Ancillary (equipment)	Equipment including, but not limited to, controls, circulating pumps and heaters that are installed along with a heat pump to create a complete installation.
Auxiliary (heat)	Heat that is provided (e.g. by an electric immersion heater or gas-fired boiler) to supplement the heat provided by a heat pump.
BEIS	Department for Business, Energy & Industrial Strategy.
Bivalent	A term used to refer to a heating system that uses two types of heat generator that may operate simultaneously – e.g. a heat pump and a boiler or a heat pump and a solar thermal collector.
Brine	A water-glycol (antifreeze) mixture used for transferring heat at low temperature (typically in the range -15 to +5 °C).
Carnot effectiveness	Ratio of measured COP to Carnot COP.
Coefficient of determination (R^2)	The R^2 coefficient of determination (value between 0 and 1) is a statistical measure of how well a regression line approximates the real data points. R^2 = 1 indicates that the regression line perfectly fits the data. R^2 = 0 indicates that the regression line does not fit the data. R^2 > 0.5 is taken in this report to indicate a good fit. R^2 > 0.25 is taken in this report to indicate a reasonable fit.
CO_2	Carbon dioxide.
CO_2e	Carbon dioxide equivalent. A measure of greenhouse gas emissions expressed as the equivalent quantity of carbon dioxide.
COP_H	Coefficient of performance of the heat pump for heating (ratio of output thermal power to input electrical power).
°C	Degree Celsius. A unit of temperature. Equal in magnitude to the kelvin and defined such that the ice point of water is 0 °C.
DECC	Department of Energy & Climate Change (became part of BEIS in July 2016).
Desuperheater	A heat exchanger that removes heat from superheated gas discharged from a compressor. This provides a small amount of heat at a temperature which is higher than that of the main condensation process.
DHW	Domestic hot water.
Energy Fence	A proprietary design of heat collector that combines ground-source and air-source.
EPC	Energy performance certificate.
FTP	File transfer protocol.
GSHP	Ground-source heat pump.
HTTPS	Secure hypertext transfer protocol.
Isentropic efficiency	A measure of the power used by a compressor compared to the power theoretically needed.
Kelvin	The Thermodynamic Kelvin Temperature Scale, named after the Belfast-born engineer William Thompson, Lord Kelvin.

kelvin (K)	Unit of thermodynamic temperature. The kelvin (not capitalised) and its symbol K are used to express the value of a temperature interval or a temperature difference.
kW	Kilowatt. A unit of power = 1000 W.
kWh	Kilowatt-hour. A unit of energy = 1000 Wh = 3.6 megajoules.
kWh_{TH}	Kilowatt-hour (thermal).
LPG	Liquefied petroleum gas.
M-Bus	A European standard for remote reading of heat meters and other types of consumption meter, sensors and actuators. See www.m-bus.com .
MCS	Microgeneration Certification Scheme.
Monovalent	A term used to refer to a heating system that uses only one type of heat generator.
Pulse	An electrical pulse, typically the momentary closing of a circuit, used by an energy or flow meter as an output signal to indicate that a certain quantity of energy or fluid has been measured.
PV	Photovoltaic.
RHI	Renewable Heat Incentive scheme.
SEPEMO	SEasonal PErformance factor and MOnitoring for heat pump systems in the building sector. See http://sepemo.ehpa.org/ .
SH	Space heating.
SI	The International System of Units (Système International d'Unités, with the international abbreviation SI). See www.bipm.org .
SIM	Subscriber Identity Module (used for cellular modems).
SPF	Seasonal performance factor: the ratio of [thermal energy delivered] to [electrical energy used], calculated over a year.
SPF_{H1}	SPF (in heating mode) of the heat pump, excluding the energy used by the heat source pump(s), auxiliary heaters and the pumps needed to deliver the heat to the sink).
SPF_{H2}	SPF (in heating mode) of the heat pump, taking into account the energy used by the heat source pump(s) (but excluding auxiliary heaters and the pumps needed to deliver the heat to the sink).
SPF_{H4}	SPF (in heating mode) of the total heat pump system, taking into account all ancillary pumps and heaters.
SQL	Structured Query Language: a programming language used for managing and manipulating data held in a relational database management system.
T-Test	A statistical method used to assess whether the means of two groups of values are statistically different from each other.
Temperature lift	The difference between the source (input) and the sink (output) temperatures.
TRV	Thermostatic radiator valve.
UFH	Underfloor heating.
UTC	Coordinated Universal Time (≈ Greenwich Mean Time).
V	Volt. The SI unit of voltage. See www.bipm.org .

VPN	Virtual Private Network (an encrypted communications tunnel via the Internet).
W	Watt. The SI unit of power. See www.bipm.org .
Weather compensation	The technique used in heating system control whereby the temperature of the water supplied to the heat emitters is reduced as the outdoor air temperature increases.
Wh	Watt-hour. A unit of energy. Equal to 3.6 kilojoules.
WSHP	Water-source heat pump.
ZigBee	An open, global wireless standard that provides the foundation for the Internet of Things by enabling simple and smart objects to work together. See www.zigbee.org .

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Key Messages

This report presents results from monitoring a sample of 28 ground- and water-source non-domestic Renewable Heat Incentive (RHI) heat pump installations, with a combined capacity of 1600 kW_{TH}. 21 sites were monitored from mid-2014 to June 2016 with an additional 7 sites monitored from March 2015 to June 2016. As of April 2017, there were 769 ground- and water-source heat pump installations accredited under the non-domestic RHI with a total combined capacity of 84 800kW_{TH}. This sample therefore represents a very small proportion of the total number of systems installed.

The Seasonal Performance Factor (SPF) is a measure of average heat pump performance over time as external temperature and heat demand change. It represents the ratio of total heat output to total electrical input at different system boundaries denoted by SPF_{H1}, SPF_{H2}, etc., (with the higher the number indicating the wider system boundary and the more ancillary electrical equipment taken into account). Heat pumps achieving an SPF_{H2} ≥2.5 are considered as producing renewable energy under the EU's Renewable Energy Directive.

Of the 19 heat pumps for which performance results can be reported[1], 15 demonstrated levels of performance better than an SPF_{H2} ≥2.5. This means that for each unit of electricity they delivered at least 2.5 units of heat, whereas an electric heater would deliver only one unit of heat for each unit of electricity consumed. Six heat pumps achieved an SPF_{H2} ≥3.0, while four heat pumps demonstrated performance SPF_{H2} of less than 2.5.

When the 18 monovalent[2] heat pump systems are compared to oil-fired heating, all would have lower operating CO_2 emissions (the breakeven SPF_{H4} was 1.35) but only one of the sample would have delivered fuel bill savings (the breakeven SPF_{H4} was 2.16). The high breakeven SPF_{H4} was a consequence of relatively low oil prices. The average price of oil during the monitoring period was 33.8 p/litre. If a more representative oil price of 48.8 p/litre (the mean price for the 10 years from December 2006 to November 2016) is used for comparison, then the breakeven SPF_{H4} is 1.50 and eight of the sample would cost less to run than oil-fired heating.

When the monovalent heat pump systems are compared to natural gas-fired heating, 16 systems would have lower operating CO_2 emissions (the breakeven SPF_{H4} was 1.81), but only five systems would deliver fuel bill savings (the breakeven SPF_{H4} was 2.75).

A properly designed, installed and operated ground-source or water-source heat pump should be able to achieve SPF_{H2} ≥3.0 or above in typical UK climatic conditions. The monitored systems that did not achieve this had factors present that should not have occurred.

Numerous factors influence heat pump performance. There is not one overriding factor that needs to be addressed, but more careful design, installation, commissioning and operation are all required to ensure a high-performance system. It is always important to pay particular attention to maximising the source temperature at the heat pump evaporator inlet, to minimising the temperature at the heat pump condenser outlet, to minimising the energy used by ancillary equipment, to avoiding exceptional heat losses (for example from underground heat distribution pipes) and to use the correct configuration of controls. Each application has its own particular characteristics and each system therefore needs to be designed and optimised to suit that application.

The observed sample performance should not be taken as representative of the Non-Domestic RHI ground- and water-source heat pump population (or the wider heat pump population) due to the sampling method and site selection process employed. The findings present a range of seasonal performance factors found on a sample of Non-Domestic RHI ground- and water-source heat pumps and examines factors that may be affecting their performance.

[1] Performance data have been omitted for 7 systems where the heat metering arrangement had unacceptably high uncertainty of measurement, for one system where exceptional operational difficulties led to insufficient useful data being available and for one system where the heat provided to domestic hot water could not be measured.

[2] One of the systems monitored is bivalent (heat pump + oil-fired boiler) and has not been included in this comparison.

Executive Summary

The Non-Domestic Renewable Heat Incentive (RHI) is a government scheme designed to incentivise organisations in Great Britain to install heating systems that use renewable energy. The main aim of this project was to measure the in-service performance of a sample of ground-source and water-source[3] heat pumps installed under the RHI scheme.

Only those systems that were considered to have characteristics representative of mainstream non-domestic heating installations were included in the monitoring programme. Following survey of 51 sites, 21 installations were monitored from mid-2014 until June 2016. A further 7 installations were monitored from March 2015 to June 2016. However, the sample monitored is not statistically representative of the non-domestic RHI heat pump population, which has increased considerably since this project started.

Of the 28 installations monitored, 17 provide both space heating and domestic hot water. The others provide space heating only.

The type of buildings heated varies considerably: public halls, offices, residential houses, apartment blocks, rental accommodation, agricultural buildings, healthcare buildings. Heating is required 24 hours per day on some sites, but only during Monday-Friday office hours on others. Table 2 and Table 3 present summary details for the sites monitored.

The thermal capacity of the installations ranges from 10 to 268 kWTH. 16 of the installations have a design output not greater than 45 kWTH and were therefore classified as microgeneration systems[4] that required MCS (Microgeneration Certification Scheme [1]) accreditation.

The monitoring equipment installed on each site provided detailed measurement of:

- electrical energy used by the heat pumps and ancillary equipment
- thermal energy output measured by the heat meter(s) already installed for the RHI
- temperatures at key points: outdoor and indoor air, ground or source water, heat pump input and output, buffer tank input and output, domestic hot water.

Recorded data was sent from each site via the cellular wireless network to a secure data server for subsequent analysis.

This report sets out the results of the analysis carried out on the data collected from the installations.

Research on heat metering accuracy [2] carried out independently of this project identified very large measurement errors with some heat metering arrangements – especially those that use temperature sensors strapped to the outside of pipes or otherwise incorrectly mounted. These findings became available during the course of this project and indicated that the uncertainty of measurement of the heat metering on 7 of the systems being monitored was too great for the measured performance data to be reliable[5]. In addition, on one system where the buffer tank incorporated into the heat pump provides both space heating and domestic hot water tank, the heat output to domestic hot water could not be measured. One other system suffered operational difficulties during much of the monitoring period, resulting in insufficient performance data being obtained.

Seasonal performance factors were calculated for 19 systems and a range of factors were investigated to gain insights into their impact on performance.

[3] Air-source heat pumps were not included in the Non-Domestic RHI scheme at the time this project commenced.

[4] This definition of a microgeneration system is as per MCS Microgeneration Installation Standard MIS 3005 (Issue 3.1 or 3.2) that was pertinent at the dates of installation of the monitored systems.

[5] It is understood that the heat meter installations have subsequently been amended to rectify this problem on at least some of the affected systems, but not within the timeframe of this project.

Observations on system performance

The measured heat pump performance[6] SPFH2 ranged from 2.24 to 4.49, while the overall system performance SPFH4 ranged from 1.21 to 4.12. Table 1 shows the performance statistics for the 19 installations that could be measured, during the period from 1st July 2015 to 30th June 2016. The median value is that of the middle of the ranked values. The mean value is calculated as the total heat delivered divided by the total electricity used by all 19 systems.

	SPF_{H2}	SPF_{H4}
25th percentile	2.58	2.07
Median	2.73	2.39
75th percentile	3.10	2.79
Mean	2.71	2.04

Table 1 – Summary of performance statistics for the period 1st July 2015 to 30th June 2016

The SPFH4 results are presented in Figure 1 as a histogram. Each system is represented by a coloured rectangle, with the SPFH4 value shown in each rectangle.

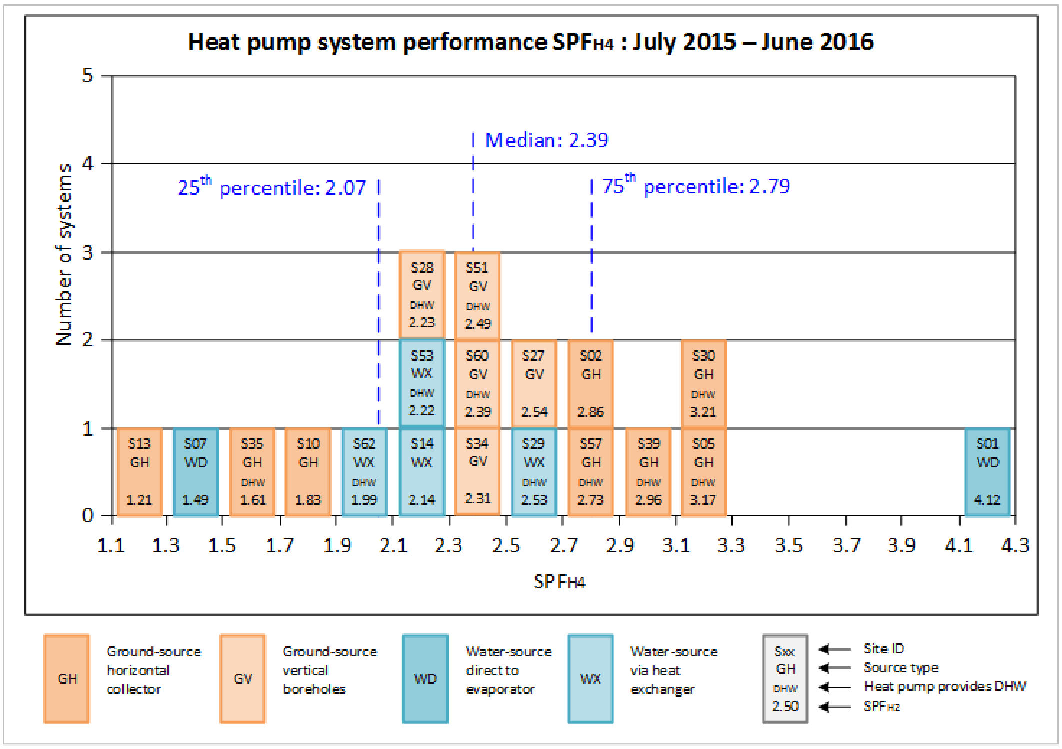

Figure 1 – Histogram showing system performance of the 19 systems monitored for which performance results can be presented, for the 12 months from July 2015 to June 2016

Key points

Key points from the project can be summarised as:

- The systems studied vary widely in application, design and complexity.
- 21 heat pumps have been monitored for 24 months (July 2014 to June 2016) and a further 7 for 12 months (July 2015 to June 2016). With a combined installed capacity of 1601 kW_{TH}, this sample

[6] SPFH (seasonal performance factor for heating) is the ratio of heat output to electrical energy input.
SPFH2 represents the performance of the heat pump, taking into account the heat source pump.
SPFH4 represents the performance of the complete system, and takes into account the heat pump and all pumps for the heat source and heat sink as well as any auxiliary heaters.

was equivalent to 10.5% of the total non-domestic RHI ground- and water-source heat pump capacity accredited as of 1st June 2015 (the start of the monitoring period considered in this report).

- Of the 19 heat pumps for which performance data can be reported, four demonstrated levels of performance below that required to be considered "renewable" (SPF_{H2} ≥2.5) under the Renewable Energy Directive[7] and when wider system energy use was taken into account, the number operating with an SPF_{H4} <2.5 increased to 11. Only six of the sample achieved an SPF_{H2} >3.0 and three achieved an SPF_{H4} >3.0.
- When the 18 monovalent heat pump systems are compared to oil-fired heating, all would have lower operating CO_2 emissions (the breakeven SPF_{H4} was 1.35) but only one of the sample would have delivered fuel bill savings (the breakeven SPF_{H4} was 2.16). The high breakeven SPF_{H4} was a consequence of relatively low oil prices. The average price of oil during the monitoring period was 33.8 p/litre. If a more representative oil price of 48.8 p/litre (the mean price for the 10 years from December 2006 to November 2016) is used for comparison, then the breakeven SPF_{H4} is 1.50 and eight of the sample would cost less to run than oil-fired heating.
- When the monovalent heat pump systems are compared to natural gas-fired heating, 16 systems would have lower operating CO_2 emissions (the breakeven SPFH4 was 1.81), but only five systems would deliver fuel bill savings (the breakeven SPFH4 was 2.75).
- When the monovalent heat pump systems are compared to natural gas-fired heating systems, 14 (78%) have lower CO_2 emissions but only three of the sample cost less to run.
- The heat metering arrangements on eight installations, whilst likely to have been in line with RHI requirements at the time (the requirements were updated in 2014) were not considered by this study to be of sufficient standard for performance analysis. Some of the heat meter installations are understood by the author to have been improved subsequent to the end of the data collection period.
- Heat pump systems using underfloor heating were not found to have significantly higher system performance than those using radiators – not even those with radiators that had been installed and sized for use with oil-fired boilers, and were not increased in size for use with heat pumps. Indeed, two of the five systems in the monitored sample that used radiators not upsized for the heat pump had SPF_{H4} values in the upper quartile of those of the sample monitored, and four of those five had SPF_{H4} values above the median.
- The hours of operation of the heating system (i.e. times of heat demand – ranging from weekday office hours to 24/7 operation) were not shown to have a significant influence on system performance. However, longer-than necessary heat pump operating hours potentially caused energy wastage which is undesirable.
- There was no significant difference in performance between heat pumps from different manufacturers.
- Eight systems do not appear to be being operated in line with current best practice guidance with regard to Legionella control in domestic hot water systems.
- The mean temperature lift (heat pump output temperature minus source temperature at the heat pump inlet) during the monitoring period ranged from 21 °C (for the system with the highest performance) to 55 °C. Systems with low temperature lift tended to have higher performance.

[7] Heat pump installations accredited onto the RHI are all required to meet minimum quality standards. All of the monitored installations were accredited before May 2014, and hence were required to demonstrate that the heat pump units achieved a COP of at least 2.9. Since May 2014, newly accredited RHI installations have also been required to demonstrate a minimum design SPF of 2.5. As 16 of the monitored systems have a capacity below $45kW_{TH}$, they will also have been required to achieve MCS certification standards. Further information on scheme eligibility and minimum standard requirements is available from Ofgem: https://www.ofgem.gov.uk/environmental-programmes/non-domestic-renewable-heat-incentive-rhi/eligibility-non-domestic-rhi .

- On two of the water-source systems, the source water is pumped directly to the heat pump – whereby the absence of an intermediate heat exchanger provides the highest possible temperature at the heat pump evaporator. One of these systems had the highest performance of all systems monitored. (The other system had unrelated operational problems that caused overall low performance.)
- Systems that provide heat at a continuously high temperature to a combined space heating and domestic hot water system were found to have performance lower than most other systems[8].
- High heat pump output temperatures, with mixing valves used to reduce the temperature supplied to the heat emitters, were observed on some systems. These systems could not benefit from lower output temperatures that would have given higher heat pump performance.
- The energy used by ancillary equipment (pumps and immersion heaters) varied from 8% to 55% of total electricity used by the heat pump system. The median value was 24%. Systems with lower energy use by ancillary equipment typically had higher performance.
- Various control issues were identified:
    - auxiliary heaters being used when apparently not needed
    - a smart controller intended for use with a boiler causing cycling of a heat pump
    - heat pumps being used during periods when the building was not occupied
    - circulating pumps running when not needed, or – more seriously – not running when needed
    - heat pump output temperatures higher than needed (e.g. weather compensation not working)
    - lack of temperature controls in individual rooms sometimes leading to wasted energy
    - heating and cooling occurring during the same day
    - multiple heat pumps starting at the same time with consequent short run times, when starting one at a time would have been more efficient.
- Weather compensation (reduction of the heat pump output temperature to space heating when the outdoor temperature rises) was not proven to improve system performance on the sample monitored.
- The mean outdoor air temperature at the sites monitored varied from 8.3 °C to 12.7 °C, but was not found to have a significant influence on system performance.
- Provision of small quantities of domestic hot water by a heat pump, as identified on two sites, may not be justified. Point-of-use water heaters could be more efficient for some applications.
- Short cycling of the heat pump (known to be a possible cause of reduced performance and excessive equipment wear [3]) was found to occur on only two of the systems monitored.
- Monitored systems used to heat buildings older than 50 years (and with probably poor energy performance compared to modern or refurbished buildings) were not found to have lower performance.
- System proprietors all stated at the start of the monitoring project that they were satisfied or very satisfied with their heat pump installation. One proprietor of a system that uses a common high-temperature output to provide both space heating and domestic hot water has subsequently commented that he would consider a different system design if he had the opportunity of starting again. Proprietors of the systems monitored in this project have not been asked about satisfaction since monitoring was completed. However, in a separate research project carried out as part of the Renewable Heat Incentive Evaluation, the findings of a survey of other non-domestic RHI applicants has been published [4].

[8] The apparently lower performance of the small sample (three) of these systems was not proven with statistical significance.

The observed sample performance should not be taken as representative of the Non-Domestic RHI ground- and water-source heat pump population (or the wider heat pump population) due to the sampling method and site selection process employed. The findings present a range of seasonal performance factors found on RHI ground- and water-source heat pumps and examines factors that may be affecting their performance.

Conclusions

The project shows that it is possible to design, install and operate a heat pump system to provide a high seasonal performance factor, but that this high level of performance is not being realised on some of the monitored installations.

From the available data obtained from the small sample of systems, it is difficult to draw general conclusions about the performance of the wider population of non-domestic heat pump installations. This is a consequence mainly of the wide variation in the application and design of the systems monitored. A larger sample would have been useful to yield better statistical significance of the analyses performed, but at the time the project was started, the available sample represented 21% of the total non-domestic RHI heat pump population[9]. It is also worth noting that working with a larger sample may have precluded carrying out some of the detailed analyses of individual systems that has been undertaken in this project.

Numerous factors influence heat pump performance. There is not one overriding factor that needs to be addressed, but more careful design, installation, commissioning and operation are all required to ensure a high-performance system. It is always important to pay particular attention to maximising the source temperature at the heat pump evaporator inlet, to minimising the temperature at the heat pump condenser outlet, to minimising the energy used by ancillary equipment, to avoiding exceptional heat losses (for example from underground heat distribution pipes) and to the use and correct configuration of controls that are appropriate for heat pumps. Each application has its own particular characteristics and each system therefore needs to be designed and optimised to suit that application.

[9] As of December 2013, there were 100 accredited Non-Domestic RHI ground-source and water-source heat pump installations with a total thermal capacity of 4900 kWTH. [23]

1 Project Profile

This report presents results from monitoring of 28 non-domestic ground-source and water-source heat pump systems accredited onto the Non-Domestic Renewable Heat Incentive (RHI). The RHI is a government scheme designed to incentivise organisations in Great Britain to install heating systems that use renewable energy.

Heat pump installations that had characteristics representative of mainstream non-domestic heating installations were selected for monitoring in order to measure in-service performance and investigate factors influencing their performance.

1.1 Aims

The aims of the project are:

- To determine the range of in-service performance of a sample of the RHI heat pump population.
- To understand causes of performance variations, including factors related to system and equipment design, commissioning, control and operation.

1.2 Approach

In order to achieve the aims of the project, it was necessary to identify a suitable sample of the RHI heat pump population, and then to install monitoring equipment on the selected installations to permit analysis of heat pump operation and performance.

Prior to installation of monitoring equipment, each heat pump installation was surveyed to assess its suitability for monitoring. This survey usually presented an opportunity to meet the proprietor and to learn about the nature of the application; the rationale for the heat pump having been installed; other relevant information such as how the system is managed and controlled; about any problems encountered with installation or operation; about the proprietor's opinions of owning and using a heat pump.

An important aspect of the monitoring process was the need to utilise the heat meters that were already installed for RHI purposes, as it was considered impractical to carry out the invasive work that would be necessary at each installation to install new heat meters. The monitoring systems were therefore designed around this constraint.

1.3 Work to date

The project started in January 2014. An interim report [5] was published in February 2016.

The key messages from that report were:

- Of the 21 heat pumps monitored for the 12 months to mid-2015, twelve (57%) demonstrated levels of performance better than an $SPF_{H2} \geq 2.5$, with seven (33%) achieving an $SPF_{H2} \geq 3.0$. [10]
- When compared to typical oil-fired heating systems, all heat pump installations would have lower operating CO_2 emissions and 16 (76%) would deliver fuel bill savings[11].
- When compared to typical gas-fired heating system, 19 (90%) heat pump installations would have lower operating CO_2 emissions, but only two systems would deliver fuel bill savings[8].

Case studies [6] have been prepared for each system monitored. These provide details of the individual systems and contain analyses of performance, operational behaviour and notes about particular factors that influence performance. The case studies are due to be published at the same time as this report.

[10] It should be noted that the results for five systems were presented in the interim report with very high levels of uncertainty due to their heat metering arrangements. A subsequent review of heat metering on all monitored systems has led to the measurements for nine systems being considered as having high uncertainty and results for these systems have therefore not been included in this final report.

[11] The fuel prices (44 p/litre = 4.12 p/kWh for oil and 2.50 p/kWh for gas) and CO_2 emission factors used for comparison in the Interim Report were not the same as those used in this report or in the case studies, as each report relates to a different monitoring period. The most relevant prices and factors were used at the time of publication of each report.

2 Site Selection and Installations Monitored

2.1 Phase 1 : 2014

The proprietors of all of the heat pump installations in the RHI database at the time of starting the project were contacted by the Department of Energy & Climate Change (DECC) – now part of the Department for Business, Energy and Industrial Strategy (BEIS) – to seek their cooperation for the monitoring project. The proprietors of 51 sites responded positively.

49 sites were surveyed to assess suitability for monitoring. One site was assessed as being essentially domestic in nature and was not surveyed; the other site was not surveyed for logistical reasons.

From the information collected during the surveys, 22 sites were selected by DECC as suitable for monitoring. The other sites were rejected as being unsuitable either because they were essentially domestic[12], or because the application was considered to be unusual (e.g. an unusual building design or unusual heat collector arrangement) and would therefore be unlikely to be representative of the wider population of non-domestic applications.

Monitoring equipment was subsequently installed on 21 sites during May – July 2014. One of the selected sites was abandoned because of access issues.

2.2 Phase 2 : 2015

The sample of monitored sites was expanded as additional installations became RHI-accredited. A further nine sites were surveyed, and one site previously surveyed during phase 1 was reconsidered.

Monitoring equipment was installed on seven sites during March 2015. One of the sites was rejected because the heat metering installed for RHI did not facilitate SPF_{H2} and SPF_{H4} performance monitoring. The other two sites were abandoned because of access issues.

Monitoring of all systems was completed at the end of June 2016.

2.3 Sites monitored

Descriptions of the sites are given in the case studies, available as separate documents [6].

Summaries of the sites being monitored are shown in Table 2 (phase 1) and Table 3 (phase 2).

[12] Some essentially domestic installations are classified for RHI purposes as non-domestic, because of some commercial activity at the site.

Site ID	Monitoring start date	Building type	Capacity kW_{TH}	No. of heat pumps	Source type	Heat source	Heat emitter	DHW	Auxiliary heat
01	10/07/2014	Offices	26	1	WD **	Ground water from borehole; returned to river	Underfloor heating	No	None
02	27/06/2014	Large house	93	1	GH	Horizontal ground loops: 12 x 200m	Radiators	No	Oil-fired boiler (backup only)
04	23/06/2014	Large house	57	2	GH	Horizontal ground loops: unknown size	Radiators	Yes	4 x 3 kW immersion heaters in DHW cyls. (manual control)
05	09/06/2014	Public hall	21	1	GH	Horizontal ground loops: 6 x 200m	Radiators	Yes	4.5 kW immersion heater in buffer tank (emergency use); 4 kW immersion heater in DHW cyl.
10	09/06/2014	Offices	22	1	GH	Horizontal ground loops: 8 x 100m	Radiators	No	None
13	27/05/2014	Greenhouse	144	3	GH	Horizontal ground loops: 4000m	Pipes at high and low level.	No	Oil-fired boiler
14	09/07/2014	Healthcare clinic	60	2	WX **	Ground water from 2 x vertical boreholes	Underfloor heating	No	21.6 kW electric boiler; 6 kW immersion heater in buffer tank (backup only)
17	08/07/2014	Public hall	30	1	GV	Vertical boreholes: 1 x 65m, 6 x 75m	Underfloor heating (part of building); radiators.	Yes (top-up of solar heat)	Solar thermal, 3 kW immersion heater in DHW cylinder
18	12/06/2014	19 apartments in 2 adjacent buildings	79	2	GV	Vertical boreholes: 12 x 100m	Underfloor heating	Yes	3 x 9 kW immersion heaters in DHW cylinders
27	26/06/2014	Accommodation building	54	1	GV	Vertical boreholes: 10 x 150m	Underfloor heating	No	None
28	11/07/2014	Hotel	71	2	GV	Vertical boreholes: 12 x 125m	Radiators	Yes	4 x 6 kW immersion heaters in DHW cylinders; 7.5 kW immersion heater in buffer tank; oil-fired boiler (backup)
29	06/06/2014	Large house	126	1	WX	Coils in river	Radiators	Yes	9 kW immersion heater in buffer tank; 9 kW immersion heater in DHW cylinder
30	09/07/2014	Public hall	14	1	GH	Horizontal ground loops	Underfloor heating	Yes	9 kW immersion heater in heat pump

Site ID	Monitoring start date	Building type	Capacity kW_{TH}	No. of heat pumps	Source type	Heat source	Heat emitter	DHW	Auxiliary heat
33	09/07/2014	Healthcare clinic	10	1	GH	Horizontal ground loops: 500 m	Underfloor heating	Yes	4 kW immersion heater in heat pump
34	14/07/2014	Healthcare clinic	64	1	GV	Vertical boreholes	Underfloor heating	No	Gas boilers
35	15/07/2014	3 dwelling houses	20	2	GV	Vertical boreholes: 5 x 90 - 140 m	Underfloor heating (ground floor); radiators (first floor).	Yes	3 kW immersion heater in DHW cylinder in each house
37	29/05/2014	Public hall / sports pavilion	17	1	GH	Horizontal ground loops: 880 m	Underfloor heating (ground floor); radiators (first floor).	Yes	7 kW immersion heater in heat pump
39	25/06/2014	3 dwelling houses and first floor offices	23	1	GH	Horizontal ground loops: 3 x 400 m	Radiators	Yes	9 kW immersion heater in DHW cylinder
40	11/07/2014	Short-term rental apartments	31	1	GH	Horizontal ground loops: 2200 m	Underfloor heating	Yes	Solar thermal, 2 x immersion heaters in buffer tank
48	10/07/2014	Residential home	14	1	G/A	Energy Fence: one third buried in ground, two thirds in air above ground	Underfloor heating on ground floor; radiators on first floor.	No	None
51	03/07/2014	Recreational building	38	1	GV	Vertical boreholes: ~10	Radiators	Yes	Gas boiler; immersion heater in DHW cylinder

Table 2 – Summary of sites monitored (phase 1)

** This system is classified in the RHI database as a water-source heat pump. However, the heat source is groundwater, so it could also be considered to be a ground-source heat pump (see MIS 3005 [1]).

Source types:

GH Ground-source, horizontal collector
GV Ground-source, vertical boreholes
WD Water-source, open-loop, direct to evaporator
WX Water-source, open-loop, indirect via heat exchanger
G/A Hybrid ground/air-source

Site ID	Monitoring start date	Building type	Capacity kWTH	No. of heat pumps	Source type	Heat source	Heat emitter	DHW	Auxiliary heat
07	26/03/2015	Refectory and offices	96	1	WD	Water from tarn	Underfloor heating	No	Hot water from LPG boilers, fed to underfloor heating header via thermostatic valve & pump
53	19/03/2015	Offices and warehouse	30	1	WX	River water	Underfloor heating	No	Immersion heater
56	21/03/2015	Retail shop	33	1	GH	Horizontal loops: 1200 m	Underfloor heating	Yes	Immersion heaters: 2 x 6kW in buffer tank; 1 x 3kW in DHW cylinder
57	21/03/2015	Detached 3-storey house used as offices	40	1	GH	Horizontal loops: 6 x 250 m	Radiators	Yes	None
60	25/03/2015	Public hall with a café	40	1	GV	Vertical boreholes: 8 x 100 m	Underfloor heating	Yes	Solar thermal, immersion heater in DHW cylinder
61	19/03/2015	Residential care facility	80	1	GV	Vertical boreholes: 15 x 100 m	Underfloor heating	No	Gas boiler
62	28/03/2015	Large house and outbuilding	268	4	WX	Surface water	Radiators	Yes	Immersion heaters + LPG boiler

Table 3 – Summary of sites monitored (phase 2)

2.4 Data processing methodology

Details of the data processing and analysis methodologies are described in Appendix C of the Interim Report [5]. The data analysis methodology is summarised in the following paragraphs.

Readings of cumulative electrical energy and instantaneous power were recorded from electricity meters at 1-minute intervals. Readings from battery-powered temperature sensors and pulse counters connected to the heat meters were recorded every 2 minutes.

Data was transmitted securely from the monitoring system via the cellular network and the Internet to a data server, where it was recorded in a "raw data" database. The raw data was subsequently processed by custom software to generate a "clean" database of readings from all sensors at 1-minute intervals. Linear interpolation was used to generate the 1-minute temperature and pulse count values from the raw 2-minute data.

The clean data generation process incorporated screening to ensure that each value was within the expected range: out-of-range readings were discarded. Temperature sensor calibration adjustments were also applied at this stage. The clean data was transferred to a Microsoft SQL Server® database for analysis using SQL[13] procedures tailored to suit each heat pump system monitored in this project.

The SPF values were calculated as illustrated by the following example.

[13] Structured Query Language: a programming language used for manipulating data held in a relational database.

- The electrical energy used by the heat pump and the circulating pumps was determined for each 1-minute interval. These values were summed to generate daily, weekly and annual totals.

- The heat output recorded by the heat meter was determined for each 1-minute interval and summed as for the electricity values.

$$SPF_{H2} = \frac{\text{[Heat measured by heat meter]} - \text{[heat added by buffer pump]}}{\text{Electricity used by: [heat pump]} + \text{[brine pump]}}$$

$$SPF_{H4} = \frac{\text{[Heat measured by heat meter } H_{01}\text{]} - \text{[heat loss from buffer tank]} + \text{[heat added by immersion heaters]} + \text{[heat added by heating circulating pumps]}}{\text{Electricity used by: [heat pump]} + \text{[immersion heaters]} + \text{[brine pump]} + \text{[buffer pump]} + \text{[heating circ pumps]}}$$

- The heat added by the buffer pump and heating circulating pumps was estimated as 30% (the assumed pump efficiency[14]) of the electrical energy supplied to the pumps.

- The heat loss from the buffer tank was estimated from published heat loss data for buffer tanks, using the measured flow and return temperatures at each calculation interval to determine the temperature in the tank and an assumed plant room temperature of 15 °C.

Details of the calculations used for each system are presented in the case studies [6].

14 The assumed liquid pump efficiency of 30% is as used in EN 14511-3 [20]. The heat added to the water by the pump is the [hydraulic power required to pump the water (pressure difference x flow rate)] + [the heat generated by the non-pumping churning of the water by the impeller] + [a (small) portion of the heat losses from the electric motor].

3 Results

A summary of performance and examples of the analysis are presented in this section. Analyses for the sites monitored are contained in the site case studies [6].

Results presented in this report are for the monitoring period 1st July 2015 – 30th June 2016.

3.1 Summary of performance data

Table 4 shows a summary of the performance data for the 12-month period from July 2015 to June 2016, for the 19 systems for which the performance data could be determined with an acceptable accuracy[15].

SPF_{H2} represents the performance of the heat pump and the source water or brine pump, whereas SPF_{H4} represents the performance of the complete system including all of the circulating pumps and any auxiliary heaters. Comparison of these two values gives some insight into the efficiency of the overall installation. See Appendix A for further information about the definitions of SPF_{H2} and SPF_{H4}.

3.2 Note about heat metering uncertainty on some systems

As with any measurement, the results presented here are subject to uncertainty of measurement. The uncertainties pertaining to the performance results for each system have been estimated. See Appendix C of the Interim Report [5] for details.

Absolute values for SPF are not presented in Table 4 for systems where the heat meters were installed with strap-on temperature sensors (sites 4, 17, 18, 37, 40, 56) or where a temperature sensor was incorrectly installed using a made-up pipe fitting that did not place the sensor correctly in the flow inside the pipe (site 61). It is known [2] that such temperature sensor arrangements lead to very high uncertainty of measurement which inevitably means that the calculated SPF values also have very high uncertainty.

It is understood that the heat meter installations have subsequently been amended to rectify this problem on at least some of the affected systems, but not within the timeframe of this project.

The SPF values for site 33 where the heat provided to DHW could not be measured or reasonably estimated, and for site 48 where there was insufficient data because of exceptional operational difficulties, have also been omitted from Table 4.

Performance results are presented in this report for 19 systems: sites 01, 02, 05, 07, 10, 13, 14, 27, 28, 29, 30, 34, 35, 39, 51, 53, 57, 60 and 62. Qualitative analysis presented relates to all 28 systems monitored.

3.2.1 Note about SPF_{H4} for SH and DHW operation

The Interim Report [5] contained estimates of SPF_{H4} for space heating and for domestic hot water operation on some systems. Such values are not presented in this report, because a review of the available data led to the conclusion that it is not possible to meaningfully determine the system performance for each mode of operation on any of the monitored systems.

While on some systems it was possible to measure or estimate with reasonable accuracy the amount of heat provided to domestic hot water, it was usually not possible to determine the corresponding electricity use with sufficient accuracy, because there is no available means of accurately apportioning the electricity used by the compressors and ancillary equipment to space heating and to domestic hot water output.

[15] Performance data have been omitted for 8 systems where the heat metering arrangement had unacceptably high uncertainty of measurement. Site 48 has also been omitted because of exceptional operational difficulties leading to very little useful data being available.

In principle, it should be possible to determine the electricity used for domestic hot water operation on those systems that alternately provide heat for space heating or domestic hot water. Only one such system (site 37) has separate heat meters for each mode. However, the heat meters on this system are installed with strap-on temperature sensors which renders the available data unreliable.

For the reasons set out in the foregoing paragraphs, the preliminary values presented in the Interim Report for the breakdown of SPF_{H4} for space heating and for domestic hot water operation should be regarded as having high uncertainty.

Site ID	Source type	Emitter type	DHW mode	E2 kWh	H2 kWh	SPFH2	E4 kWh	H4 kWh	SPFH4	Comments	Notes
01	WD	U	-	12 091	54 261	**4.49**	13 207	54 433	**4.12**		1
02	GH	R	-	30 316	104 253	**3.44**	37 103	105 948	**2.86**		
04	GH	R	A	43 170	n/a		46 393	n/a		Heat metering issues	2
05	GH	R	SI	10 987	37 525	**3.42**	12 202	38 740	**3.17**		
07	WD	U	-	8 273	22 537	**2.72**	16 291	24 317	**1.49**		
10	GH	R	-	8 247	18 467	**2.24**	8 617	15 781	**1.83**		
13	GH	P	-	62 038	139 142	**2.24**	234 370	282 581	**1.21**	Bivalent operation	3
14	WX	U	-	72 973	177 288	**2.43**	83 118	178 248	**2.14**		1
17	GV	U+R	A	14 310	n/a		15 776	n/a		Heat metering issues	2
18	GV	U	A	100 249	n/a		145 579	n/a		Heat metering issues	2
27	GV	U	-	26 640	73 468	**2.76**	29 046	73 638	**2.54**	Partial data	4
28	GV	R	A	109 051	297 810	**2.73**	148 074	330 599	**2.23**	SH only during winter	5
29	WX	R	SI	19 837	57 150	**2.88**	22 872	57 939	**2.53**	Partial data	6
30	GH	U	A	5 174	20 103	**3.89**	6 689	21 485	**3.21**		
33	GH	U	SS	3 491	n/a		4 359	n/a		Heat metering issues	7
34	GV	U	-	41 119	105 341	**2.56**	45 941	106 209	**2.31**		
35	GV	U	SC	10 938	26 040	**2.38**	12 852	20 632	**1.61**		
37	GH	U+R	A	6 022	n/a		9 398	n/a		Heat metering issues	2
39	GH	R	AS	18 818	57 350	**3.05**	19 340	57 169	**2.96**		
40	GH	U	SC	35 482	n/a		40 302	n/a		Heat metering issues	2
48	G/A	U+R	-	1 553	n/a			n/a		Operational problems	8
51	GV	R	SC	45 420	122 084	**2.69**	49 168	122 486	**2.49**		
53	WX	U	-	8 043	20 958	**2.61**	9 451	20 988	**2.22**		
56	GH	U	SI	30 927	n/a		32 914	n/a		Heat metering issues	2
57	GH	R	AS	35 988	103 658	**2.88**	38 135	104 039	**2.73**		
60	GV	U	AS	8 161	25 733	**3.15**	11 578	27 660	**2.39**	Solar heat not included in H4	9
61	GV	U	-	67 842	n/a		79 309	n/a		Heat metering issues	2
62	WX	R	SC	208 016	545 716	**2.62**	242 083	481 710	**1.99**	Operational problems	10
					Max:	**4.49**		Max:	**4.12**		
					Median:	**2.73**		Median:	**2.39**		
					Min:	**2.24**		Min:	**1.21**		

All results are for the period 1/7/2015 to 30/6/2016 except where noted (see notes below).

Source type	
GH	Ground-source, horizontal collector
GV	Ground-source, vertical boreholes
WD	Water-source, direct to evaporator
WX	Water-source, via heat exchanger
G/A	Hybrid ground/air-source

Emitter type	
R	Radiators
U	Underfloor
P	Pipes (greenhouse)

DHW mode	
A	Alternate. The heat pump provides either SH or DHW.
AS	Alternate/simultaneous. One vapour compression module alternately provides SH or DHW. The other provides SH only.
SI	Simultaneous (intermittent). The heat pump intermittently provides DHW at the same time as providing SH.
SC	Simultaneous (continuous). The heat pump output provides SH and DHW via a combined high temperature system.
SS	Simultaneous (desuperheater). The DHW is provided using the superheated vapour from the compressor discharge.

Data	
E2	Total electricity used by the heat pump and the source pump, and excluding buffer pumps (kWh).
H2	Total heat output excluding auxiliary heaters and heat added by buffer pumps (kWh).
SPFH2	Ratio of H2 to E2.
E4	Total electricity used by the heat pump, source pump, heat distribution pumps and auxiliary heaters (kWh).
H4	Total heat output including auxiliary heaters but excluding heat losses from buffer tanks (kWh).
SPFH4	Ratio of H4 to E4.

Notes
1 The heat source is groundwater. The system could also be considered as ground-source.
2 The heat metering arrangement has characteristics known to cause high uncertainty of measurement. SPF results for these systems are not presented.
3 The SPFH4 value is calculated for bivalent operation of the heat pumps & oil-fired boiler.
4 The results presented are for the period 1/7/2015 - 1/3/2016. There was a fault with the heat metering after the latter date.
5 The hotel was closed from 3/11/2015 - 8/3/2016. The heat pumps provided space heating only during that period.
6 The results presented are for the periods 1/7/2015 - 29/2/2016 & 5/4/2016 - 11/4/2016. Data for other dates is unavailable because of data communications difficulties on the site.
7 The DHW on this system is provided using a desuperheater which is not metered.
8 There were exceptional operational difficulties with the heat pump system on this site (not related to the hybrid heat source). Consequently, very little useful data is available. SPF results are not presented for this system.
9 Heat provided to DHW from the solar thermal collector on this site has not been included in the SPFH4 calculation.
10 There were operational problems on this site due to equipment damage caused by exceptional weather. The heat pumps were unable to operate normally and the LPG backup boiler was used for extended periods.

Table 4 – Summary of performance data for the period from 1st July 2015 to 30th June 2016

Figure 2 shows the SPF_{H1}, SPF_{H2} and SPF_{H4} values for the period July 2015 to June 2016. There is considerable variation in the differences between the three values for each system, which illustrates the variation in system efficiency (as distinct from heat pump efficiency).

For example, compare sites 14 and 39. Site 14 extracts heat from groundwater to provide space heating only via underfloor heating pipes in a modern well-insulated building. It has the higher SPF_{H1} of the two systems. However, the power required for open-loop pumping of the groundwater from boreholes was very high. These and the other ancillary pumps used 33% of the total electricity used by the heat pump system - resulting in the relatively low SPF_{H2} and SPF_{H4} values. Site 39 extracts heat from horizontal ground loops and provides space heating via radiators and domestic hot water to refurbished dwelling houses and a farm office. The modest power requirements of the closed-loop brine circulating pump and the high-efficiency heating circulating pumps resulted in the ancillary pumps using only 8.6% of the total electricity. Consequently, even though the SPF_{H1} is lower than for site 14, the SPF_{H2} and SPF_{H4} values are higher.

This illustrates the importance of understanding the design principles of heat pumps and heating systems in general. For heat pumps, it is always important to pay particular attention to the source and output temperatures and to the energy used by ancillary equipment. For any heating system, heat losses can have a significant negative influence on efficiency.

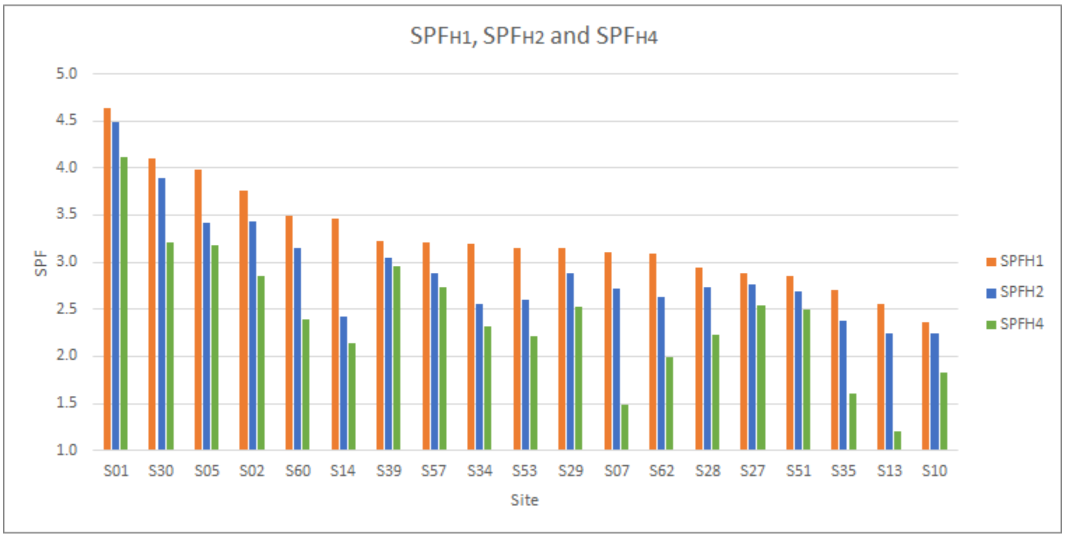

Figure 2 – SPF_{H1}, SPF_{H2} & SPF_{H4} for each system for the period from 1st July 2015 to 30th June 2016

3.3 System efficiency – SPF_{H2} and SPF_{H4}

Figure 3 shows the heat pump performance SPF_{H2} of the systems for which performance results can be presented, for the period July 2015 to June 2016.

The results are presented in the form of a histogram, using an SPF bin size of 0.2. Each system is represented by a coloured rectangle, showing summary details of the system and the SPF value.

15 of the heat pump installations (79%)[16] have SPF_{H2} values of 2.5 or higher. This is the minimum value for heat pumps to qualify as renewable energy technologies under the rules of the EU Renewable Energy Directive [7] [17]. The grey-shaded area on the chart indicates systems with $SPF_{H2} < 2.5$ that are not considered renewable energy technology.

[16] Note that this proportion is different to that reported in the interim report because seven additional sites have been monitored, but results for nine sites have been excluded due to heat metering issues and operational difficulties (see Section 3.1).

[17] The Renewable Energy Directive [7] states: "*Only heat pumps for which SPF > 1.15 * 1/η shall be taken into account, where SPF = the estimated average seasonal performance factor for those heat pumps η = is the ratio between total gross production of electricity and the primary energy consumption for electricity production, calculated as an EU average based on Eurostat data*". This equates to a minimum SPF_{H2} of 2.5.

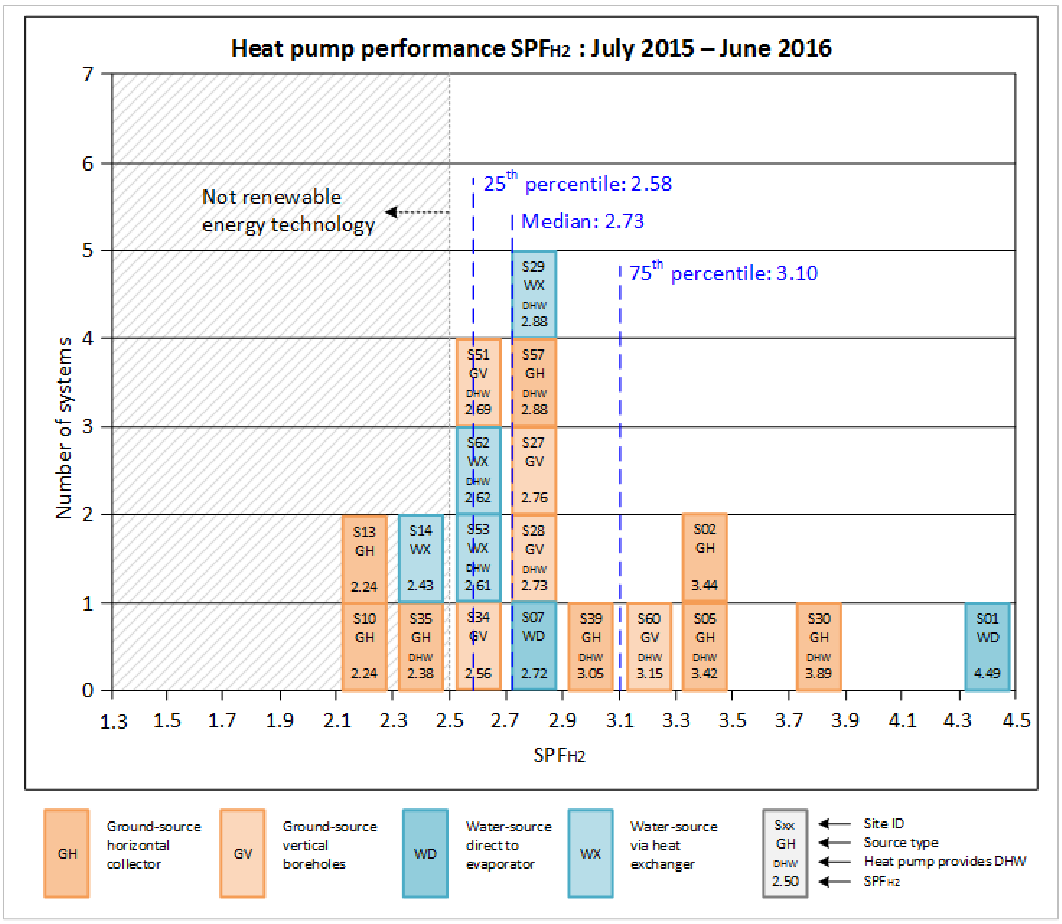

Figure 3 – Histogram showing SPFH2 heat pump performance of the 19 systems for which values are presented, for the period July 2015 to June 2016

The range of measured SPFH2 values is large – from 2.24 for the lowest performing systems to 4.49 for the highest performing one.

The median value of SPFH2 was 2.73.

The mean value of SPFH2 (calculated as the sum of the relevant heat output divided by the sum of the relevant electricity used by all 19 systems) was 2.71.

Summary information about each system is presented in Table 2, Table 3 and Table 5.

All monitored installations will have been required to meet minimum quality standards of demonstrating that the heat pump units achieved a COP of at least 2.9 to become RHI accredited. Note that all of the systems monitored were accredited before May 2014, when the RHI scheme requirements were revised with a view toward improving in-situ performance standards. All heat pump systems accredited since May 2014 have been required to demonstrate a minimum design SPFH2 of 2.5 [8].

Figure 4 shows the system performance SPFH4 of the systems for which performance results can be presented, for the period July 2015 to June 2016.

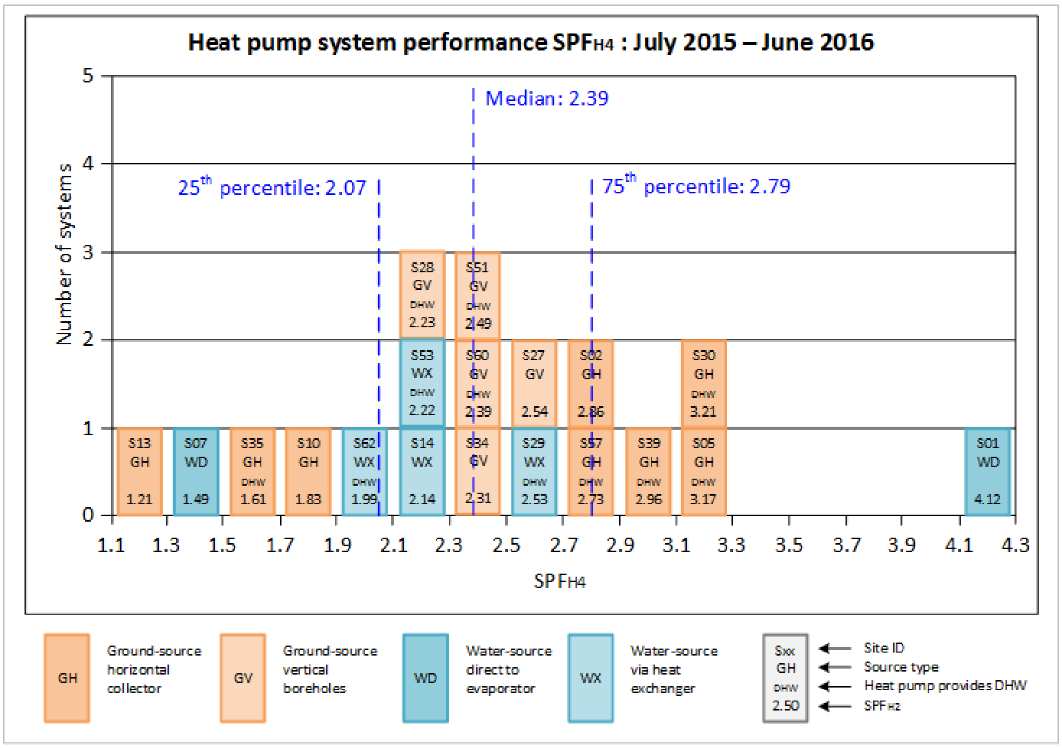

Figure 4 – Histogram showing system performance of the 19 systems for which SPFs are presented for the period July 2015 to June 2016

Again, the spread of SPF values is large – from 1.21 to 4.12.

The median value of SPFH4 was 2.39.

The mean value of SPFH4 (calculated as the sum of the total heat output divided by the sum of the total electricity used by all 19 systems) was 2.04.

The significant difference between the mean and median values is explained by the low SPFH4 of two systems (sites 13 and 62) that together represent 36% of the total heat provided by the 19 systems. As both of these systems have SPFH4 values in the lower quartile of the performance results, the effect on the overall mean value is significant.

The system with the highest SPFH4 was site 01 (direct water-source, SH only via underfloor heating), where no auxiliary heat was used and the electricity used by the source, buffer and heating circulating pumps was below average compared to other systems monitored.

Site 30 (ground-source, underfloor heating & DHW) had the second-highest SPFH4. This was achieved in spite of the heating circulating pump running more than necessary, using 18.7% % of the total electricity used by the heat pump system, and the internal immersion heater being used to provide some of the heat to DHW, using an estimated 5.1% of total electricity.

Site 07 (direct water-source, SH only via underfloor heating) had the lowest SPFH4 of the monovalent systems. This system appeared to have significant system design or control issues which resulted in performance much lower than should have been possible.

Site 35 (ground-source, underfloor heating & DHW) had the second-lowest performance. This system uses a central heat pump plant to provide heat for space heating and domestic hot water using a combined hot water circuit, via underground (insulated) distribution pipes. The low SPFH4 is due largely to the need for a continuously high heat pump output temperature, coupled with high heat losses from the heat distribution system.

Site 13, which had the lowest SPFH4 of all systems, is a bivalent (heat pump + oil-fired boiler) system that provides space heating only to a greenhouse (which has heating characteristics completely different to those of other systems monitored) using heat emitters (simple pipes) that require high temperature (> 70 °C) hot water during cold weather.

See the site case studies [6] for more information about these and other systems.

3.4 Variation of heat demand with outdoor temperature

The amount of heat required to heat a building depends on various factors, particularly the outdoor temperature.

The heating load characteristics are different for each site. Figure 5 shows the behaviour of site 60 (a public hall with underfloor heating) which has the lowest heating cut-off temperature (14 °C) of all sites monitored.

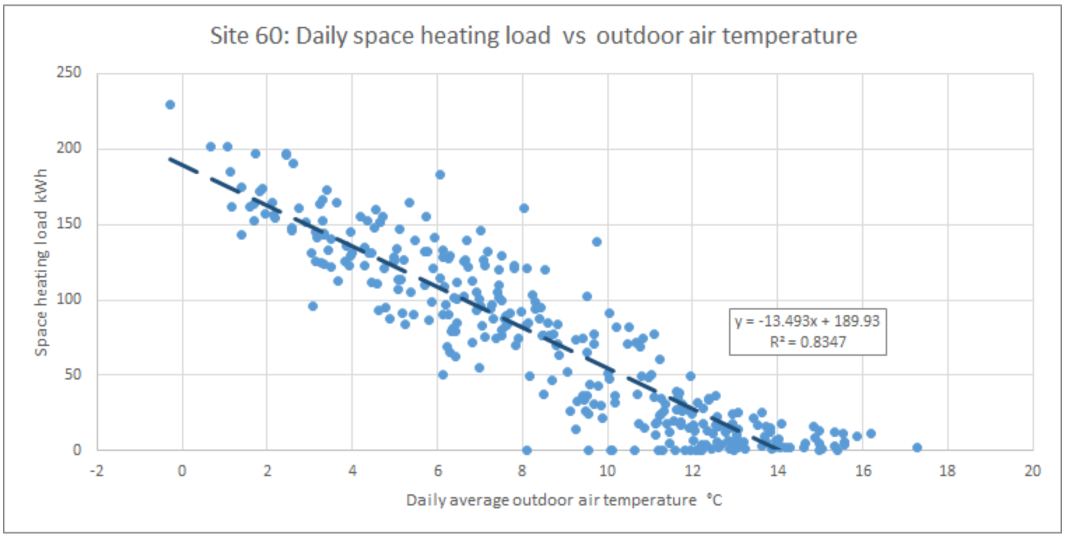

Figure 5 – Site 60: Daily space heating load vs outdoor air temperature

Figure 6 shows the behaviour at site 39 (dwellings and office, with radiators) which is more typical of most sites. The heating cut-off temperature is 17 °C.

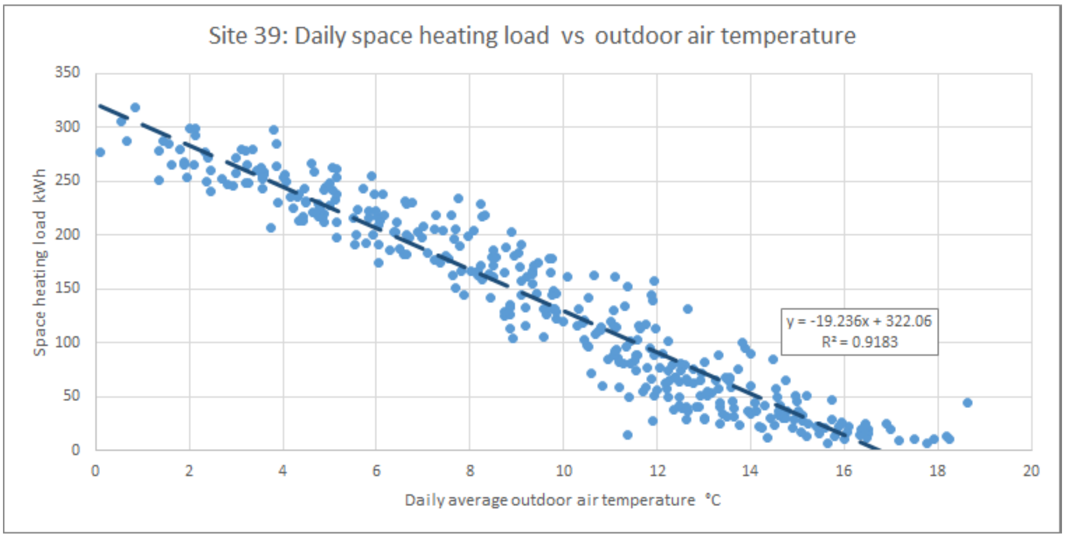

Figure 6 – Site 39: Daily space heating load vs outdoor air temperature

Figure 7 shows the behaviour of site 28 (a 19th century hotel with radiators) which has a higher heating cut-off temperature of 22 °C – presumably because of the less efficient thermal performance of an elderly cut-stone building.

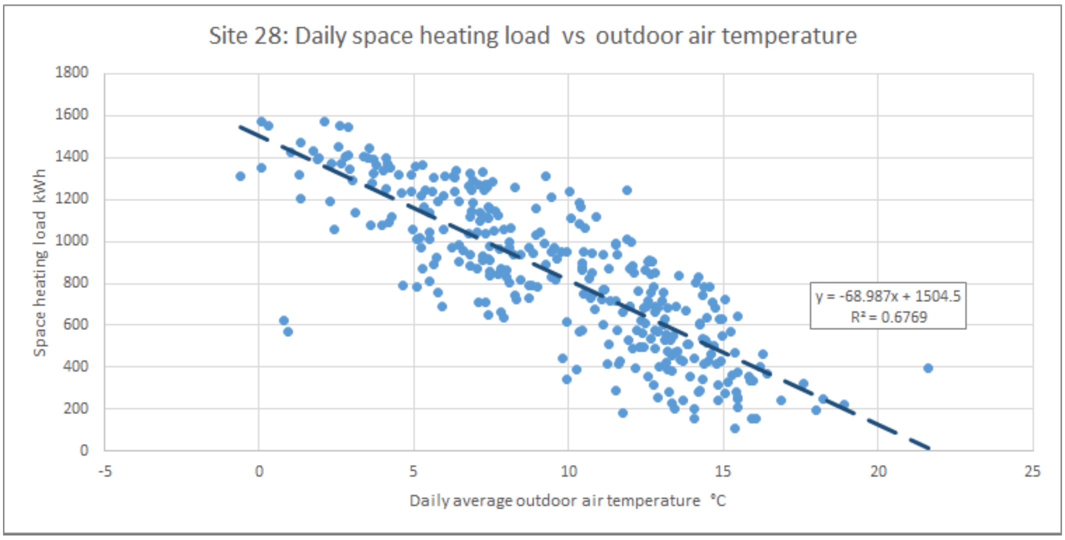

Figure 7 – Site 28: Daily space heating load vs outdoor air temperature

The median heating cut-off temperature for all sites[18] for which the heating load could be measured, determined from linear regression analysis of daily values, was 17.9 °C with a standard deviation of 2.08 °C.

Figure 8 shows a summary of the normalised[19] heating characteristics of all sites where the space heating load could be measured. There is no significant correlation, for the monitored sample, between the slope of the heating characteristic and the SPF_{H4}.

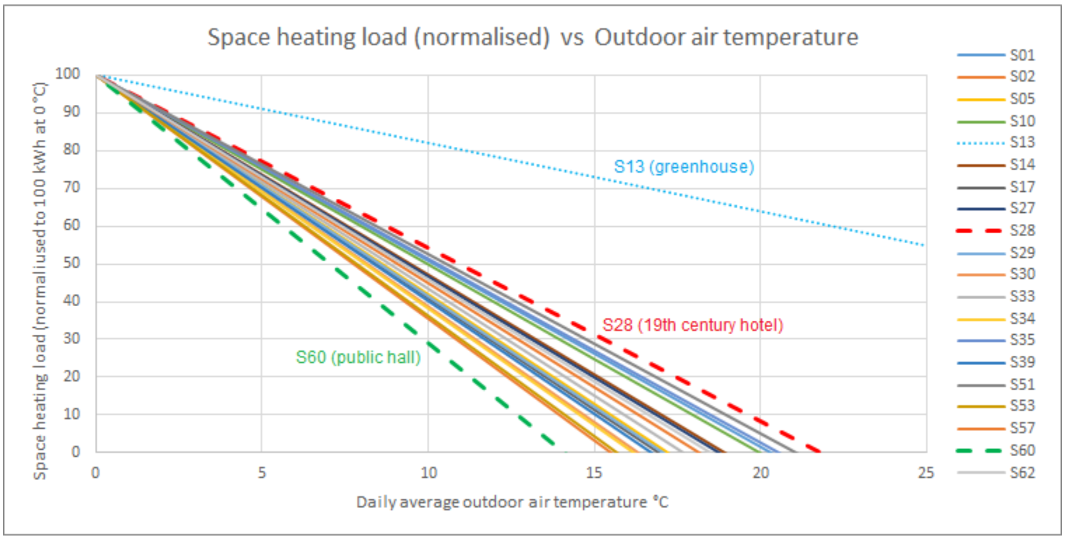

Figure 8 – Normalised space heating load vs outdoor air temperature for all sites for which the space heating load could be measured

3.5 Integration with solar collectors

On some sites, the heat pumps operate in parallel with other heat generators – e.g. solar collectors, gas boilers, etc. Where solar collectors are in use, the techniques used to integrate the heat pump and solar systems vary somewhat from one site to another. The integration of solar collectors and heat pumps has the potential to be problematic in that both of these technologies are sensitive to output temperature. For

[18] All sites except sites 07 & 48 for which useful data is not available and site 13 which is a greenhouse.

[19] To facilitate comparison, the heating characteristic for each system has been scaled so that the daily heat load at 0 °C is 100 kWh.

example, if the solar collector is connected to the lower coil in the DHW cylinder, then it can potentially operate at a low temperature and therefore more efficiently. On the other hand, the heat pump will then be connected to the upper coil in the DHW cylinder, and will be forced to work at a higher temperature. Either way, the heat from the solar collector should reduce the requirement for heat from the heat pump and thereby reduce the total electricity use, even if at a slightly lower SPF.

Solar thermal collectors are installed at sites 17, 40 and 60. The arrangements adopted for integration are described in the following paragraphs.

3.5.1 Site 17

This site is a public hall. The solar thermal collector was installed during a major refurbishment, at the same time as the heat pump, and is intended as the principal source of heat for DHW.

A special type of thermal accumulator is installed for the DHW system. This accumulator is specifically designed for integration of a heat pump and solar thermal collector, and comprises a tank within a tank, as illustrated in Figure 9. The outer tank is heated directly by the heat pump, while the solar collector is connected to a heating coil at the bottom of the outer tank. The inner tank contains the potable water to be heated for supply to the DHW distribution in the building.

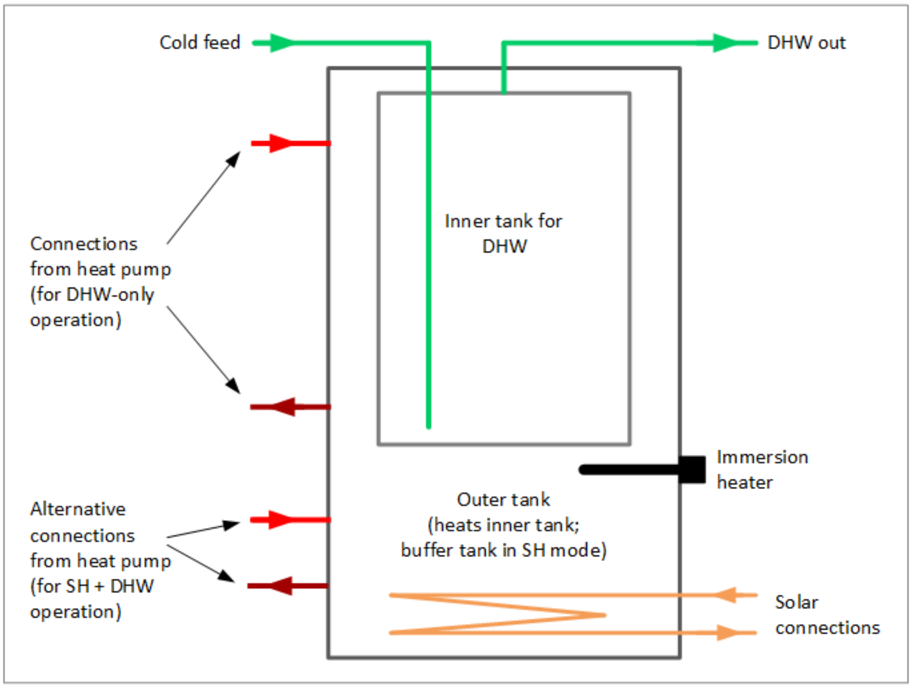

Figure 9 – Thermal accumulator used for domestic hot water at site 17

The design of the thermal accumulator is such that it can be used for both SH and DHW (when a separate SH buffer tank is not installed), or for DHW duty only. Two sets of fittings are provided for the connections to the heat pump: one set at the bottom of the outer tank is for use in combined SH + DHW operation; the other set is positioned higher up the outer tank for DHW-only operation.

On this system, the tank is used for DHW only, although the heat pump is connected to the lower fittings. It is not known whether this arrangement is intentional, but it may influence the operation of the solar collector: the water at the bottom of the outer tank where the solar heating coil is located will be heated by the heat pump, thereby reducing the efficiency of the solar collector because it will need to work at a higher temperature. On the other hand, the heat pump may be able to operate more efficiently by having a lower return temperature from the accumulator.

The heat provided to DHW was not metered on this system, so it is not possible to accurately report the relative contributions of the heat pump and solar collector to DHW. However, it is possible to provide an indication of the heat provided by the solar collector using the measured flow and return temperatures (the flow rates are unknown). Figure 10 shows an indication of the solar contribution. The heat from the heat pump to DHW was estimated as described in Appendix D of the Interim Report [5]. The contribution of the immersion heater was measured and is estimated to have been less than 0.1% of the total DHW heat.

This graph shows an approximately constant amount of heat delivered to DHW throughout the year with the solar contribution being highest during the summer and very low or zero in the winter –corresponding to the variation of solar energy available during the year. The peaks during January and May were presumably due to events in the hall requiring higher than usual demand for hot water.

It appears that this arrangement for integration of a solar collector and a heat pump works reasonably well.

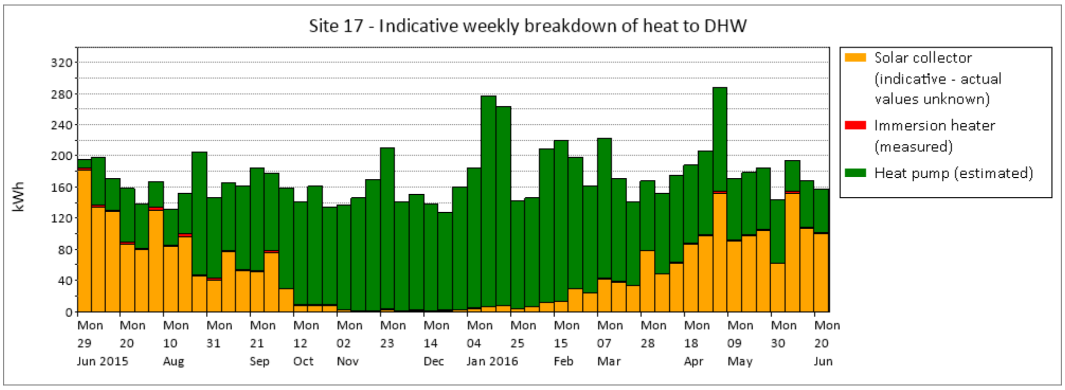

Figure 10 – Site 17: Weekly breakdown of heat provided to DHW (proportion of heat from solar is only indicative)

3.5.2 Site 40

At site 40 (a short-term rental apartment complex) the heat pump and a solar thermal collector both provide heat to a thermal store from which hot water is circulated to the apartments using a combined circuit that provides both space heating and domestic hot water.

Details of the internal arrangement of the thermal store are not available, but site inspection indicates an arrangement as shown in Figure 11.

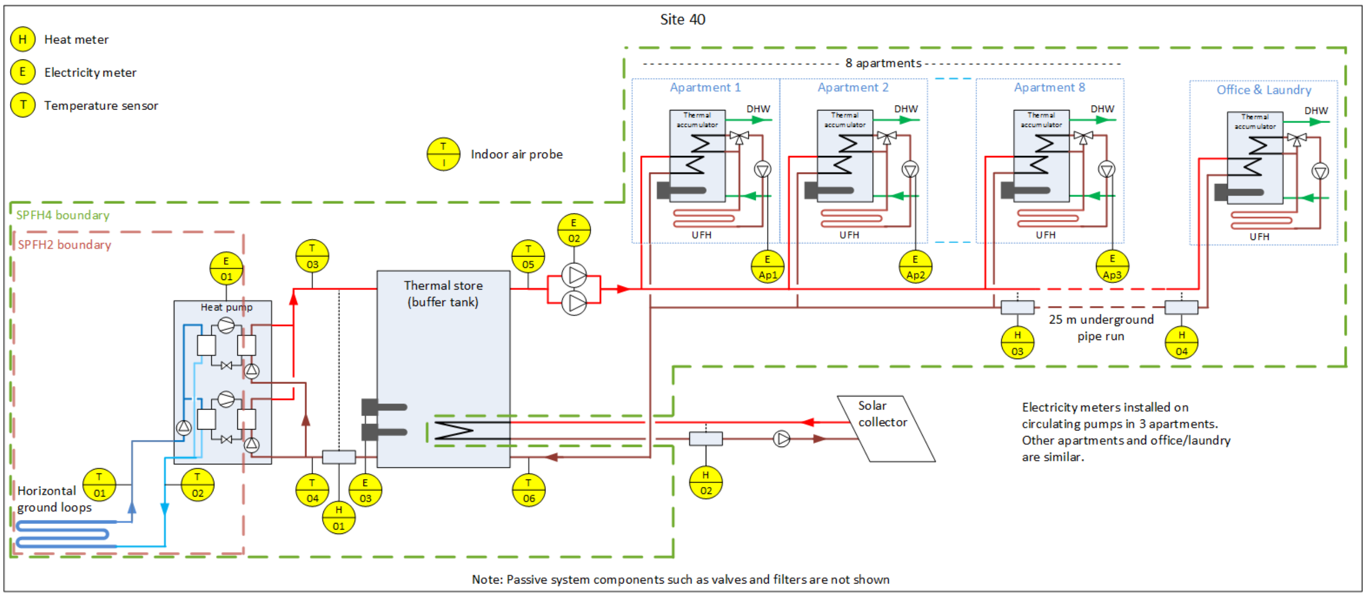

Figure 11 – Site 40 system schematic

Figure 12 shows the breakdown of heat delivered to the thermal store by the heat pump and by the solar collector over the 104-week period from 30th June 2014 to 26th June 2016. The temperature of the return from the heat distribution circuit is shown in the lower graph and should be indicative of the temperature at the bottom of the thermal store.

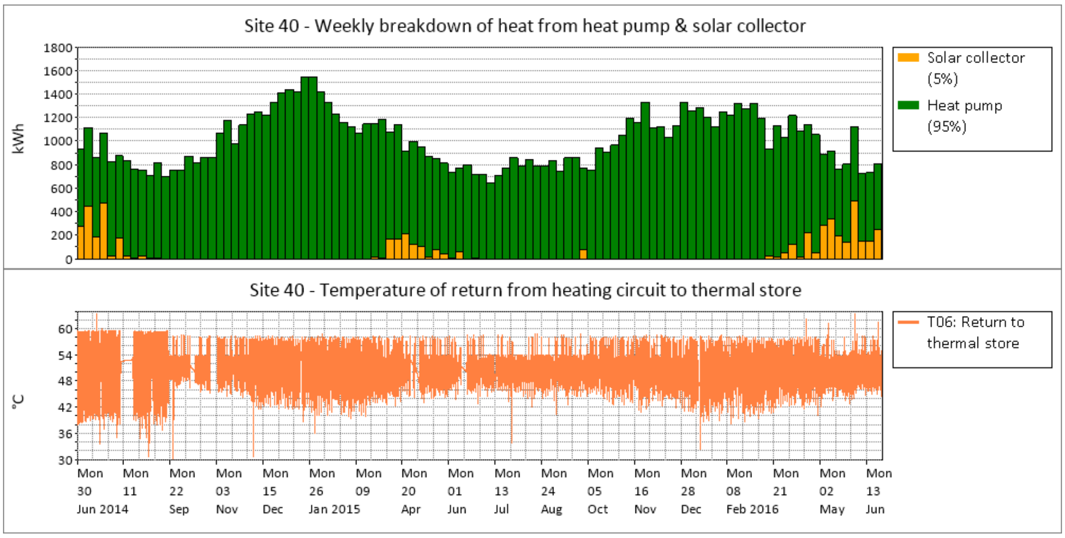

Figure 12 – Site 40: weekly heat delivered by the heat pump and the solar collector and the temperature of the return from the heat distribution circuit, from July 2014 to June 2016

Examination of the hourly heat meter data shows that 82% of the solar heat was delivered while the heat pump was not running, and it was thought that this behaviour might provide an explanation for the relatively low heat obtained from the solar collector.

Figure 13 shows the tapestry of operation of the heat pump and the solar collector from April to August 2015. The heat pump ran for generally shorter times during July and August than it did during April. The return temperatures (see Figure 12) were similar throughout the period. It is therefore surprising that the solar collector did not provide more heat during July and August, when there would have been high levels of insolation on at least some days.

It must be assumed that some unknown variation of the temperature distribution inside the thermal store has a significant effect on the performance of the solar collector, or that there was an unknown fault in the solar system. Whatever the reason, integration of the heat pump and solar collector on site 40 did not work well.

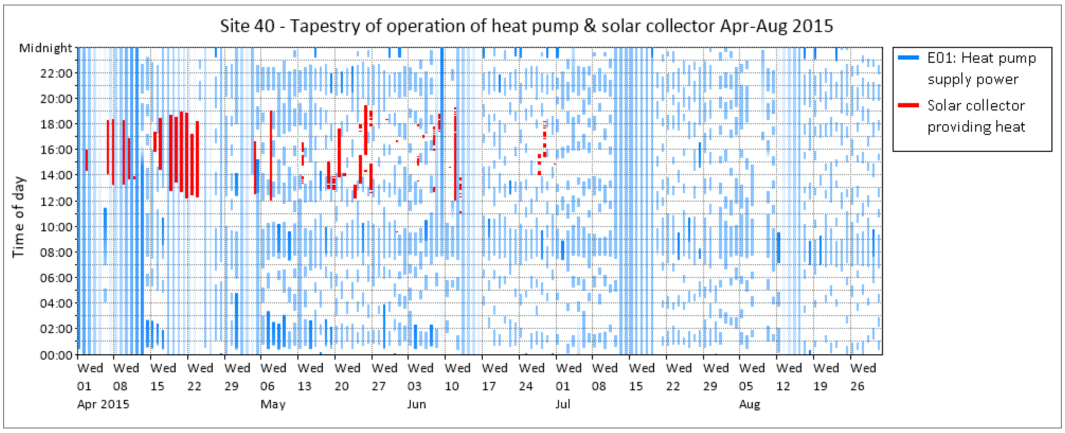

Figure 13 – Site 40: tapestry of operation of the heat pump and solar collector, April to August 2015

3.5.3 Site 60

Site 60 is a public hall. The solar thermal collector was installed at the same time as the heat pump during a major refurbishment and is intended as a supplementary source of heat for DHW.

The DHW cylinder is of a design intended for combined heat pump and solar heating. The solar heating coil is at the bottom of the cylinder, and the larger coil for the heat pump is in the upper half of the cylinder. An immersion heater is installed between the two coils. The system schematic is shown in Figure 14.

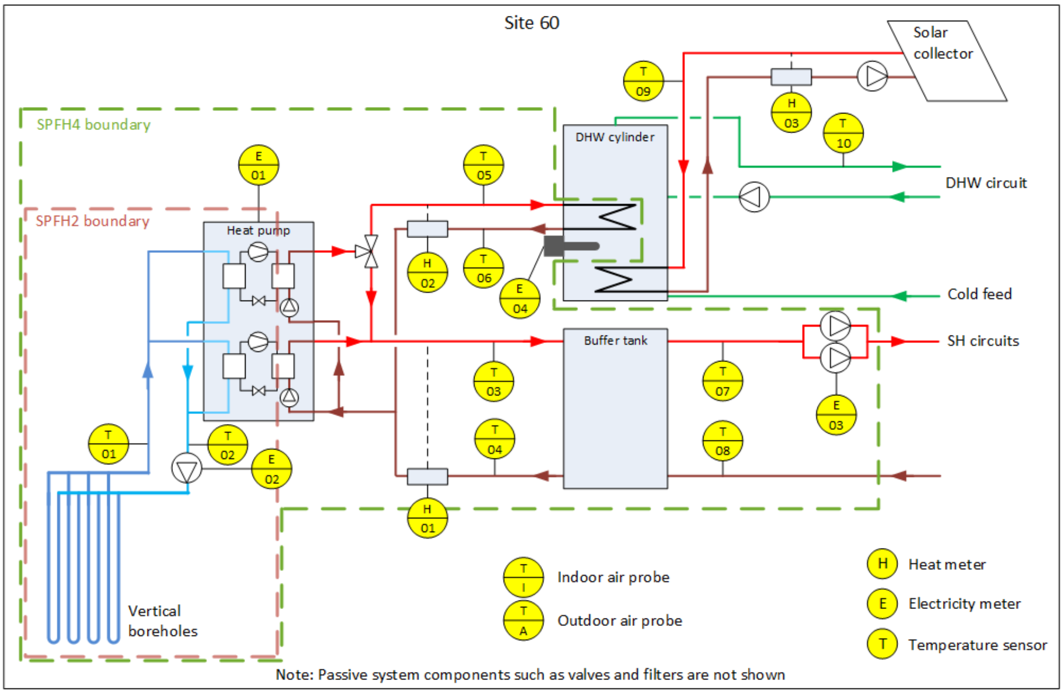

Figure 14 – Site 60 system schematic

Figure 15 shows the breakdown of the heat provided to DHW during the 12-month period from July 2015 to June 2016. The solar collector provided 25% of the heat to DHW and contributed some heat during all but 9 weeks – compared to site 40, where the solar collector provided heat during only 16 weeks. This apparently better performance is presumably due to the lower temperature at which the solar collector was able to operate, as shown by the red line on the lower graph.

The immersion heater was used extensively during the first half of the year to boost the temperature for Legionella control, but stopped being used in January[20]. After that, the temperature of the water in the DHW cylinder was rather lower, as evidenced by the change in the daily maximum DHW draw-off temperature shown by the green line in the lower graph.

[20] A timeswitch fault is suspected, but has not been confirmed.

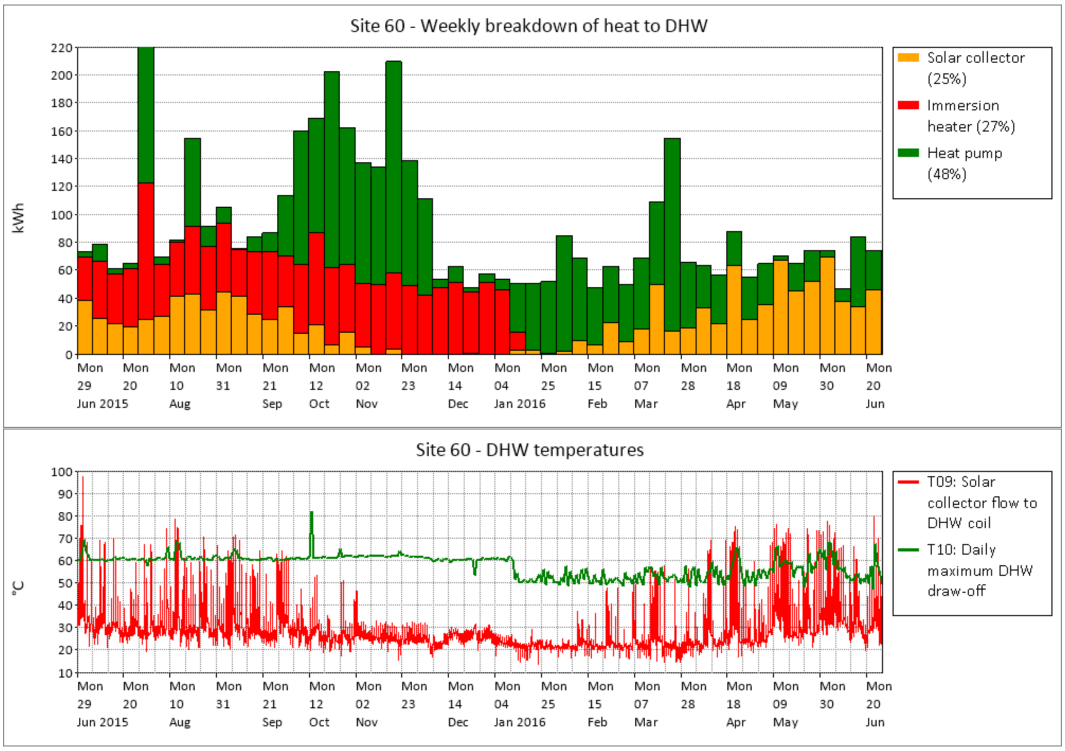

Figure 15 – Site 60: weekly breakdown of heat delivered to DHW[21] and the temperatures of the flow from the solar collector and the DHW draw-off, during the period from July 2015 to June 2016

The observations from sites 17, 40 and 60 show that it is important to pay particular attention to facilitating the lowest possible temperatures for both the solar collector and the heat pump. There appears to be an inevitable trade-off between maximising the heat provided by the solar collector and maximising the heat pump performance – both of which require low temperatures.

At sites 17 and 60 the solar collector was used only for providing heat to DHW. The thermal accumulator arrangements on both systems were broadly similar, with the solar heat being introduced at the bottom of the accumulator where the temperature should be lowest. The solar integration on these systems appears to work reasonably well, although the effect on overall heat pump performance (because of increased heat pump output temperatures while the solar collector is active) is unknown.

At site 40, the solar collector was connected to the thermal accumulator used for combined space heating and domestic hot water provision. With this arrangement, there was continuous flow through the accumulator to and from the heat distribution circuit. This flow probably caused a certain amount of mixing and de-stratification inside the accumulator which is likely to be the main reason for the reduced effectiveness of the solar collector.

As the proprietor of one site has commented, it may well have been better to install solar photovoltaic collectors (instead of solar thermal) and to use the electricity generated to offset the grid supply to the heat pump. This arrangement has been used on a number of other systems monitored. However, the effectiveness of either method was not evaluated within the scope of this project.

[21] The high demand for DHW during the fifth week was due to an event in the hall at the end of July.

4 Factors that Influence Performance

It must be borne in mind, when reading the following observations about factors that influence performance, that the sample size was small, so it cannot be concluded that the behaviour of other heat pump systems would be as found in this study.

4.1 Matrix of factors that influence performance

Table 5 is a matrix of system attributes, operational characteristics and factors that influence performance. Details of the various attributes, characteristics and factors are discussed in the system case studies that have been prepared for each of the systems monitored [6]. A summary of explanatory notes is presented below the table.

4.1.1 SPFs

The values for SPF_{H1}, SPF_{H2} and SPF_{H4} for the period July 2015 to June 2016 are shown at the top of the table for the 19 systems for which performance results can be presented.

The colour shading uses a colour gradient that shows the highest value in green and the lowest value in red.

4.1.2 Mean outdoor air temperature

The mean value of the 15-minute outdoor air temperature measurements is shown for each site. The values for sites 01, 29, 40 and 51 were estimated from measurements at other sites and are shown in italics.

The colour shading uses a colour gradient that shows the highest value in green and the lowest value in red.

4.1.3 System attributes

- Source type. The type of heat source is shown using a 2-letter code, explained in the notes below the table.
- Thermal capacity. The total thermal capacity of the heat pump(s) installed in each system is shown. Systems that incorporate heat pumps with a design output not exceeding 45 kW_{TH} were defined in the MCS standard MIS 3005 [1] as microgeneration systems[22] and therefore required certification to the MCS standard. Such systems are shaded green in the matrix.
- Space heating emitter type. Explained in the notes below the table.
- DHW mode. For systems that provide space heating and domestic hot water. Explained in the notes below the table.
- Auxiliary heat. For systems equipped with auxiliary heaters. Explained in the notes below the table.
- Buffer tank. The code denotes the buffering arrangement. Explained in the notes below the table.
- Bivalent system. The system is designed to work as a bivalent system: i.e. the heat pump operates in parallel with another heat provider.
  O = heat pump + oil-fired boiler; S = heat pump + solar thermal.
- SPF_{H4} boundary excludes solar heat. The SPF_{H4} results for bivalent solar systems in this project do not include the heat output of or electricity used by the solar thermal collectors. See Appendix A for an explanation of why this approach has been adopted.
- Incomplete data set. At site 27 a fault developed in the heat metering on 1st March 2016 and no useful heat meter data was available after then. At site 29 the shared Internet connection to the monitoring system failed on 29th February 2016. Some further data was obtained for 5th – 11th April 2016, but it was not possible to reinstate the connection thereafter.

[22] As defined in MIS 3005 (Issue 3.1 or 3.2) that was current at the time of installation.

- Heat metering issues. The various heat metering issues encountered are described in Section 3.1.
- Open-loop source pumping. Four water-source systems use open-loop pumping of the water from the source to the heat pump. This usually requires more power than closed-loop circulation through a brine loop.

4.1.4 Operational characteristics

The colour shading for each characteristic uses a colour gradient that shows the value corresponding to the highest system performance in green and the value corresponding to the lowest performance in red.

- Mean source temperature at heat pump inlet. The value shown for each system is the mean of the 2-minute temperature measurements at the inlet to the heat pump evaporator while the heat pump was running, during the period 1st July 2015 to 30th June 2016.
- Mean heat pump output temperature. The value shown for each system is the mean of the 2-minute temperature measurements of the heat pump output to space heating while the heat pump was running, during the period 1st July 2015 to 30th June 2016.
- Mean temperature lift. The value for each system is the difference between the mean output and mean source temperatures.
- Source pumping electricity (% of system total). The value for each system is the percentage of the total electricity used by the heat pump system that was used by the source pump(s).
- Sink pumping electricity (% of system total). The value for each system is the percentage of the total electricity used by the heat pump system that was used by the buffer pump(s) and heating distribution pump(s) within the SPF_{H4} performance boundary.
- Auxiliary heater electricity (% of system total). The value for each system is the percentage of the total electricity used by the heat pump system that was used by the electric auxiliary heater(s).
- Auxiliary heat (% of total heat output). The value for each system is the percentage of the total system heat output that was provided by auxiliary heaters. At site 07 the value is noted as “high” because, while the auxiliary heat from the gas boiler could not be measured, it was assessed to be high. See the case study [6] for more information.
- DHW heat from heat pump (% of total). The value for each system is the percentage of the total heat output of the heat pump system that is provided to domestic hot water by the heat pump vapour compression system.
  x = system does not provide DHW.
  n/a = system provides DHW but the data is not available.
- DHW heat from immersion heaters (% of total). The value for each system is the percentage of the total heat output of the heat pump system that is provided to domestic hot water by immersion heaters.
  x = system does not provide DHW.
  n/a = system provides DHW but the data is not available.
- Buffer tank temperature loss (°C). The value shown for each system is the maximum recorded loss of temperature between the inlet to the buffer tank (from the heat pump) and the outlet from the buffer tank to the space heating system.

4.1.5 Factors that influence performance

- Source direct to evaporator. This applies to some water-source systems where the source water is pumped directly to the heat pump evaporator, rather than transferring heat via a heat exchanger to a closed-circuit brine loop. While it is theoretically more efficient not to use a heat exchanger, there are potential problems of dirt and freezing of the source water. There is also the matter of the open-loop pumping usually needed with direct source systems. An open-loop pumping arrangement (e.g. to pump water up from a borehole) will usually require more power than for circulating brine through a loop. This can and does have a significant negative influence on overall system performance.

- Weather compensation. The matrix indicates the systems that utilise weather compensation (reduction of the heat pump output temperature when the outdoor temperature increases), but the degree to which compensation is used varies from one system to another. A ✓ mark indicates a site where weather compensation was observed to operate; A • mark indicates a site where weather compensation is understood to be configured but where no evidence of its operation was observed.
- High source temperature. A system with mean source temperature higher than the median value for all systems is indicated by a ✓ mark.
- Low sink temperature. A system with mean heat pump output temperature lower than the median value for all systems is indicated by a ✓ mark.
- Low electricity use by ancillary equipment. A system where the electricity use by ancillary equipment is lower than the median value for all sites is indicated by a ✓ mark.
- System design. The conceptual design of some systems was found to be poor. For example, a few systems were designed to provide both space heating and domestic hot water using a common output circuit that caused the heat pump to always have to operate with high output temperatures. Provision of small quantities of DHW by the heat pump on some sites may not be justified: point-of-use water heaters could be more efficient. Some systems use a high heat pump output temperature with mixing valves to reduce the temperature supplied to the heat emitters, thereby not benefitting from better heat pump performance at lower output temperatures.
- Control issues. Various types of control issue were discovered: auxiliary heaters being used when apparently not needed; a smart controller intended for use with a boiler causing cycling of a heat pump; circulating pumps running when not needed, or – more seriously – not running when needed; heat pump output temperatures higher than needed (i.e. weather compensation not working); lack of temperature controls in individual rooms sometimes leading to wasted energy; heating and cooling occurred during the same day on a couple of sites; on one site with four heat pumps, the controls started all four heat pumps at the same time with consequent short run times: starting one at a time would have been more efficient.
- Short cycling. Two systems experienced short cycling (runs of less than 6 minutes) – something that is known from previous research [3] to be a cause of reduced performance and excessive equipment wear.
- Poor building energy performance. This mainly applied to old buildings, especially those with listed status, where the building fabric could not be effectively insulated and usually led to higher space heating flow temperatures being required. The sites in this category are those that are not new (< 10 years old) or have not been extensively refurbished with the last 10 years.
- Heat emitter sizing. Six systems utilised radiators or simple pipe heat emitters that had been installed and sized for use with oil-fired boilers, but were not increased in size for use with heat pumps. This sometimes led to higher space heating flow temperatures being required. The sites in this category were selected based on information received from proprietors.
- High output temperature to DHW for Legionella control. This applies to many of the systems that provide domestic hot water: the temperature in the DHW cylinder is periodically raised to 60 °C (every day on some sites, less frequently on some others) – something that requires either a high heat pump output temperature or use of immersion heaters.

- High use of auxiliary heat (> 1% of total heat output of the system[23]). This is usually related to a control issue, where immersion heaters were used when not needed, but may also be related to Legionella control where immersion heaters are used to raise the DHW temperature.
- Problems with source availability. One system appeared to have some problems due to low aquifer level in the boreholes.
- Unexpectedly low source temperature. The brine temperature on some ground-source systems was lower than would have been expected. This was most likely due to incorrect sizing of the ground collector. The selected sites either had horizontal ground collectors where the mean ground-to-brine temperature difference was 4 °C or more, or vertical ground collectors where the daily mean source temperature was below the 25th percentile of values recorded for all sites monitored.
- Unexpectedly high output temperature. The selected sites had daily mean output temperatures to space heating that were sometimes above the 75th percentile of values recorded for all systems. Systems with a combined output for SH + DHW are not included in this category.
- DHW system design. On some systems it appears that much of the heat provided to DHW is wasted in pump-circulated systems with long pipe runs or where the actual demand for hot water is considerably less than the heat delivered to DHW by the heat pump - with much heat simply lost from the DHW tanks and pipes – although in some cases the “lost” heat would have provided space heating, albeit in an uncontrolled and inefficient manner.
- Equipment faults (as distinct from control issues). Various types of fault were encountered: loss of refrigerant from a heat pump; storm damage to pumping equipment; a leaky 3-port valve; disruption of a heat pump controller caused by a loose cable connection. Some of the more serious faults (on six sites) caused loss of output from the heat pumps for days or weeks, with backup heating systems then being required.
- Source water freezing. There was an occurrence of source water freezing in a water-to-brine heat exchanger on one system.
- Heat loss from underground pipes. Three systems have underground insulated pipes for heat distribution between the heat pump plant room and one or more buildings. The heat loss from these pipes was estimated to have a significant negative effect on the system performance – although it is worth noting that this would also have negatively affected the performance of a boiler.
- Operational problems. Two systems suffered operational problems that significantly affected the overall performance. At site 48, the heat pump could not be used for much of the monitoring period, mainly because of loss of refrigerant. It was eventually replaced by an air-source heat pump. At site 62, some of the plant suffered damage during a storm, resulting in poor heat pump performance during part of the monitoring period, with the backup boiler also being used for a lengthy period. See the case studies [6] for more details of the problems.

In summary, there any many factors that influence performance. To realise a very high performing system, all or most things need to be right: the source temperature should be as high as possible; the output (sink) temperature should be as low as possible; the electricity used by ancillary equipment should be as low as possible; the auxiliary heat from immersion heaters or fossil fuel fired boilers should be zero or as low as possible; controls should be selected and configured appropriately for heat pump duty; heat losses should be minimised.

[23] The somewhat arbitrary figure of 1% corresponds to the most pessimistic interpretation of the MIS 3005 requirement that supplementary space heating will be required for the coldest 1% of the hours in a year (assuming that all heat output is provided by the auxiliary heaters during those hours) and represents a reduction in SPF_{H4} of approximately 0.02%.

Matrix of factors that influence performance																													
	Site	01	02	04	05	07	10	13	14	17	18	27	28	29	30	33	34	35	37	39	40	48	51	53	56	57	60	61	62
SPFs	SPFH1 (July 2015 - June 2016)	4.64	3.76		3.98	3.11	2.36	2.55	3.46			2.88	2.95	3.14	4.11		3.20	2.70	3.67	3.22			2.85	3.15		3.21	3.48		3.10
SPFs	SPFH2 (July 2015 - June 2016)	4.49	3.44		3.42	2.72	2.24	2.24	2.43			2.76	2.73	2.88	3.89		2.56	2.38		3.05			2.69	2.61		2.88	3.15		2.62
SPFs	SPFH4 (July 2015 - June 2016)	4.12	2.86		3.17	1.49	1.83	1.21	2.14			2.54	2.23	2.53	3.21		2.31	1.61		2.96			2.49	2.22		2.73	2.39		1.99
	Mean outdoor air temperature	9.4	8.3	8.4	11.0	9.7	10.5	10.9	9.5	11.0	9.5	9.7	9.1	11.2	10.4	10.6	11.9	12.7	9.7	9.6	9.5	10.5	9.7	10.7	10.5	10.5	9.3	10.4	11.1
Attributes	Source type	WD	GH	GH	GH	WD	GH	GH	WX	GV	GV	GV	GV	WX	GH	GH	GV	GV	GH	GH	GH	G/A	GV	WX	GH	GH	GV	GV	WX
Attributes	Thermal capacity (kWTH)	26	93	57	21	96	22	144	60	30	79	54	71	126	14	10	64	20	17	23	31	14	38	30	33	40	40	80	268
Attributes	Space heating emitter type	U	R	R	R	U	R	P	U	U+R	U	U	R	R	U	U	U	U	U+R	R	U	U+R	R	U	U	R	U	U	R
Attributes	DHW mode	-	-	A	SI	-	-	-	-	A	A	-	A	SI	A	SS	-	SC	A	AS	SC	-	SC	-	SI	AS	AS	-	SC
Attributes	Auxiliary heat	-	-	ED	EB+ED	-	-	OB	EB	ED	ED	-	EB+ED	EB+ED	EH	EH	GFB	ED	EH	ED	EB	EH	GB+ED	EF	EB+ED	-	ED	GB	GB
Attributes	Buffer tank	4p	2pF	none	2pF	4p	2pF	4p	4p	4p	4p	4p	2pF	2pF	4p	4pi	2pF	4p	4p	4p	4p	2pR	4p	4p	4p	3p	4p	2pR	4p
Attributes	Bivalent system (O: oil-fired boiler S: solar)							O		S											S						S		
Attributes	SPFH4 boundary excludes solar heat									•											•						•		
Attributes	Incomplete data set											•		•															
Attributes	Heat metering issues			•						•	•	•				•			•		•				•			•	
Attributes	Open loop source pumping	•				•			•															•					
Operational characteristics	Mean source temperature at heat pump inlet	11.1	5.5	6.5	6.7	10.1	3.7	4.3	5.1	4.4	3.7	3.0	3.0	4.2	5.2	8.4	7.1	n/a	6.9	5.7	8.4	n/a	3.7	5.7	8.2	2.9	8.4	5.3	8.1
Operational characteristics	Mean heat pump output temperature (to SH)	31.6	42.2	37.3	36.6	40.8	50.8	49.0	40.8	35.4	50.9	37.1	46.8	37.3	33.2	28.2	32.6	48.3	43.7	44.9	54.5	n/a	58.3	37.4	48.6	47.5	43.2	42.9	56.5
Operational characteristics	Mean temperature lift	20.5	36.7	30.8	29.9	30.7	47.1	44.7	35.7	31.0	47.2	34.1	43.8	33.1	28.0	19.8	25.5	n/a	36.8	39.2	46.1	n/a	54.6	31.7	40.4	44.6	34.8	37.6	48.4
Operational characteristics	Source pumping electricity (% of system total)	4.1%	12%	11%	13%	6.2%	4.8%	11%	26%	3.0%	4.6%	4.0%	5.3%	14%	4.2%	2.2%	18%	10%	2.4%	5.3%	12%	n/a	n/a	16%	7.7%	10%	6.7%	15%	13%
Operational characteristics	Sink pumping electricity (% of system total)	8.4%	18%	6.1%	10%	49%	4.3%	3.7%	3.9%	8.9%	4.5%	7.3%	4.3%	9.7%	17%	20%	11%	13%	7.0%	2.7%	5.4%	n/a	6.8%	10%	5.7%	5.6%	18%	15%	10%
Operational characteristics	Auxiliary heater electricity (% of system total)	0%	0%	0.0%	0%	0%	0%	0%	0%	0.4%	24%	0%	20%	0%	5.9%	0%	0%	2.4%	28%	0%	0%	n/a	0.8%	0%	0%	0%	12%	0%	3.8%
Operational characteristics	Auxiliary heat (% of total heat output)	0%	0%	0%	0%	0%	0%	35%	0%	0.2%	9.4%	0%	9.0%	0%	1.8%	0%	0%	1.5%	11%	0%	0%	0%	0.3%	0%	0%	0%	4.8%	0%	1.9%
Operational characteristics	DHW heat from heat pump (% of total heat)	x	x	30%	5.1%	x	x	x	x	12%	14%	x	8.3%	38%	25.9%	n/a	x	n/a	13%	12%	n/a	x	n/a	x	n/a	2.6%	8.6%	x	n/a
Operational characteristics	DHW heat from immersion heaters (% of total heat)	x	x	0.0%	0%	x	x	x	x	0.1%	14%	x	8.6%	0%	1.9%	0%	x	n/a	12%	0%	n/a	x	n/a	x	n/a	0%	4.8%	x	n/a
Operational characteristics	Buffer tank temperature loss (°C)	5	1	n/a	0	2	5	7	5	2	6	1	0	0	1	n/a	0	6	4	5	1	n/a	n/a	2	0	3	5	0	4
Factors that influence performance	Source direct to evaporator	✓				✓																							
Factors that influence performance	Weather compensation		✓		✓	•			✓	✓	✓	✓	✓	✓	✓	✓	✓		✓	✓		✓		✓		✓	✓	✓	•
Factors that influence performance	High source temperature	✓		✓	✓	✓										✓	✓	✓	✓	✓	✓	✓		✓	✓		✓		✓
Factors that influence performance	Low sink temperature	✓	✓	✓	✓	✓			✓	✓		✓		✓	✓	✓	✓							✓					
Factors that influence performance	Low electricity use by ancillary equipment	✓		✓	✓		✓	✓		✓		✓		✓		✓				✓	✓	✓	✓		✓	✓			
Factors that influence performance	System design																	✗			✗		✗	✗	✗				
Factors that influence performance	Control issues	✗		✗		✗	✗		✗	✗	✗	✗	✗	✗	✗	✗	✗	✗	✗	✗	✗	✗	✗	✗	✗		✗	✗	✗
Factors that influence performance	Short cycling (run times < 6 minutes)						✗																		✗				
Factors that influence performance	Poor building energy performance		✗	✗				✗					✗	✗								✗	✗			✗			✗
Factors that influence performance	Heat emitter sizing		✗	✗	✗			✗						✗									✗						
Factors that influence performance	High DHW temperatures for Legionella control			✗	✗						✗			✗	✗				✗		✗		✗			✗			✗
Factors that influence performance	High use of auxiliary heat					✗		✗			✗		✗		✗			✗	✗				✗				✗		✗
Factors that influence performance	Problems with source availability								✗																				
Factors that influence performance	Unexpectedly low source temperature						✗	✗		✗		✗	✗										✗			✗			
Factors that influence performance	Unexpectedly high output temperature						✗				✗														✗	✗			✗
Factors that influence performance	DHW system design													✗		✓					✗		✗			✗			✗
Factors that influence performance	Equipment faults		✗	✗					✗													✗					✗		✗
Factors that influence performance	Source water freezing																							✗					
Factors that influence performance	Heat loss from underground pipes																	✗			✗								✗
Factors that influence performance	Operational problems																					✗							✗
	Notes	1		2		3			1	2	2	4		5					2		2	6	3		2			2	3

Table 5 – Matrix of system attributes, operational characteristics and factors that influence performance (see notes on the next page)

Notes

1	The heat source is groundwater. The system could also be considered as ground-source.
2	SPF results not presented because of high uncertainty of heat meter readings.
3	The backup boiler provided a large amount of heat because of operational or control issues. This heat is excluded from auxiliary heat figure and from SPF calculations.
4	SPF results calculated for 1/7/2015 - 1/3/2016. There was an apparent fault with heat metering after this date.
5	SPF results calculated for 1/7/2015-29/2/2016 & 5/4/2016-11/4/2016. There was a data communications fault after this date.
6	SPF results not presented because of severe operational problems with the system.

Source types

GH	Ground-source, horizontal collector
GV	Ground-source, vertical boreholes
WD	Water-source, open-loop water circuit direct to evaporator
WX	Water-source, open-loop water circuit to heat exchanger, closed-loop brine circuit
G/A	Hybrid ground/air-source

Emitter types

R	Radiators
U	Underfloor
P	Pipes (greenhouse)

DHW modes

A	Alternate. The heat pump provides either SH or DHW.
AS	Alternate/simultaneous. One vapour compression module alternately provides SH or DHW. The other module provides SH only.
SI	Simultaneous (intermittent). The heat pump intermittently provides DHW at the same time as providing SH.
SC	Simultaneous (continuous). The heat pump output is used to provide SH and DHW via a combined high temperature system.
SS	Simultaneous (desuperheater). The DHW is provided using the superheated vapour from the compressor discharge.

Auxiliary heat

EB	Electric heater in the buffer tank
ED	Electric heater in the DHW cylinder
EF	Electric heater in the flow pipe
EH	Electric heater in the heat pump
GB	Gas-fired boiler
OB	Oil-fired boiler

Buffer tanks

2pF	2-pipe in flow
2pR	2-pipe in return
3p	3-pipe (2-pipe with bypass)
4p	4-pipe
4pi	4-pipe inside the heat pump

4.2 Influence of type of heat emitter on performance

Comparison of the SPF_{H4} values for systems using underfloor heating with those of systems using radiators shows no statistically significant difference in the median values of the two samples. On first analysis, this is rather surprising – as radiators generally need higher temperatures and could be expected to have lower performance than underfloor heating. The foregoing remark about the small sample size should be borne in mind.

Figure 16 shows the effect of the mean temperature of the heat pump output to space heating on the system performance SPF_{H4}, for radiators and underfloor heating. The systems where none of the radiators has been changed from those previously installed for oil-fired heating to larger ones for a heat pump are shown in red. As expected, the general trend is for SPF_{H4} to reduce as the output temperature increases and for radiator systems generally to operate with higher temperatures.

The mean values of SPF_{H4} for the underfloor systems and for the radiator systems are not significantly different, nor is there any significant difference for the systems with radiators that have not been upsized.

It appears visually that radiator systems have higher performance at similar heat pump output temperatures. Figure 17 shows the same groups (underfloor and radiators) plotted against mean temperature lift (= mean output temperature – mean source temperature) which takes into account both the source and output temperatures. Again, it appears that the systems using radiators have higher SPF_{H4} than underfloor systems at similar values of temperature lift.

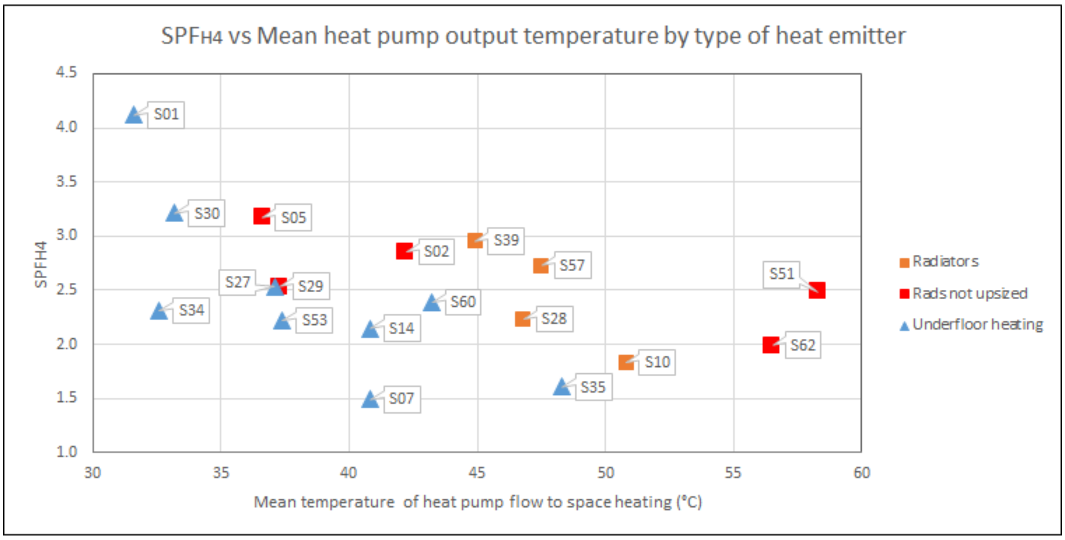

Figure 16 – The effect of mean heat pump output temperature on SPF_{H4} by type of heat emitter

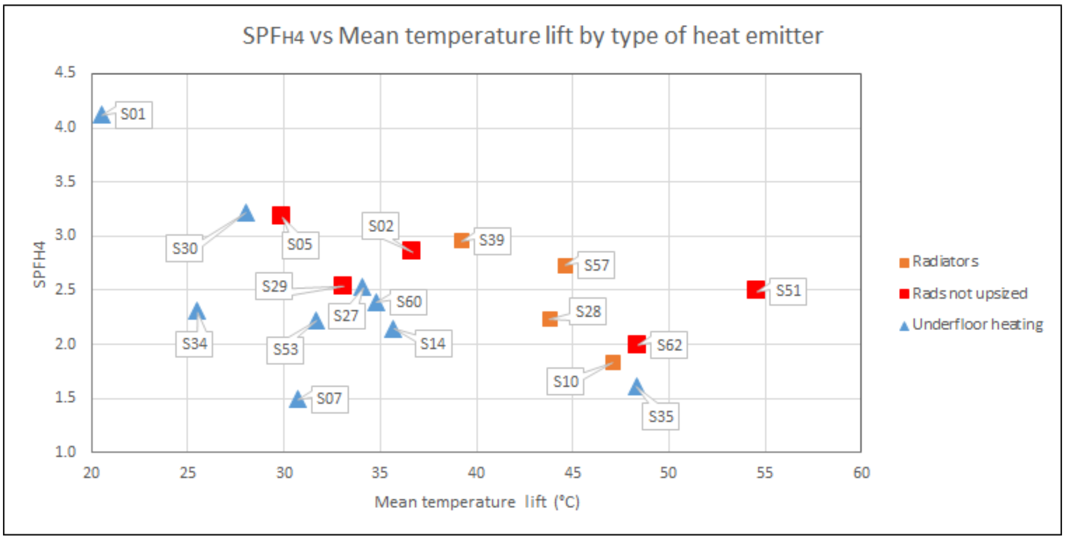

Figure 17 – The effect of mean temperature lift on SPF_{H4} by type of heat emitter

Other factors that might have a significant influence on this characteristic are

- DHW was provided by 4 of the underfloor systems and 7 of the radiator systems
- Electricity use by ancillary equipment was above average (median) on 7 of the underfloor systems and 4 of the radiator system
- Auxiliary heat (> 1% of total heat) was used on 2 of the underfloor systems and 2 of the radiator systems

The only one of these other factors that might account for radiator systems performing better than underfloor systems at a given temperature lift is the electricity used by ancillary equipment. The influence of this factor is examined in section 4.13.

4.3 Influence of ground-source or water-source on performance

Figure 18 shows the system performance SPF_{H4} as a function of temperature lift for ground-source and water-source systems. The sample does not indicate any systematic advantage of either type of source.

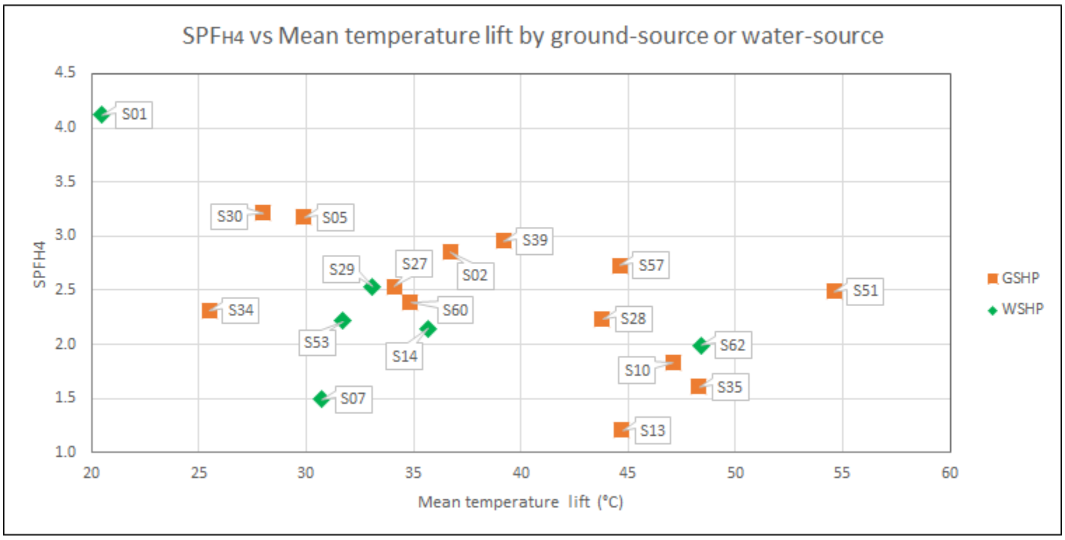

Figure 18 – The effect of mean temperature lift on SPF_{H4} by ground- or water-source

4.4 Influence of ground collector type on performance

Figure 19 shows the system performance SPF_{H4} plotted against mean temperature lift for systems with horizontal and vertical brine-loop ground collectors.

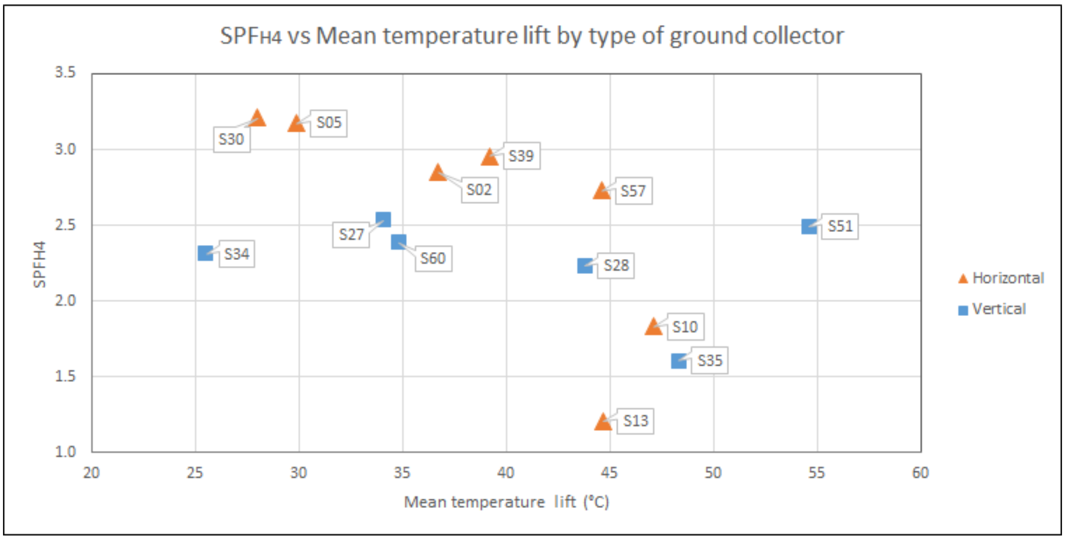

Figure 19 – The effect of mean temperature lift on SPF_{H4} by type of ground collector

There is no obvious systematic difference in performance between the two arrangements, although the selected samples are very small.

4.5 Influence of outdoor air temperature on performance

The mean outdoor air temperature at the sites monitored varied from 8.3 °C to 12.7 °C.

Figure 20 shows the system SPF_{H4} values plotted against mean outdoor air temperature, for sites where the outdoor air temperature was recorded. The data is grouped by type of heat source.

There is no statistically significant correlation between mean outdoor temperature and system performance. (Linear regression analysis has a low coefficient of determination $R^2 = 0.09$.)

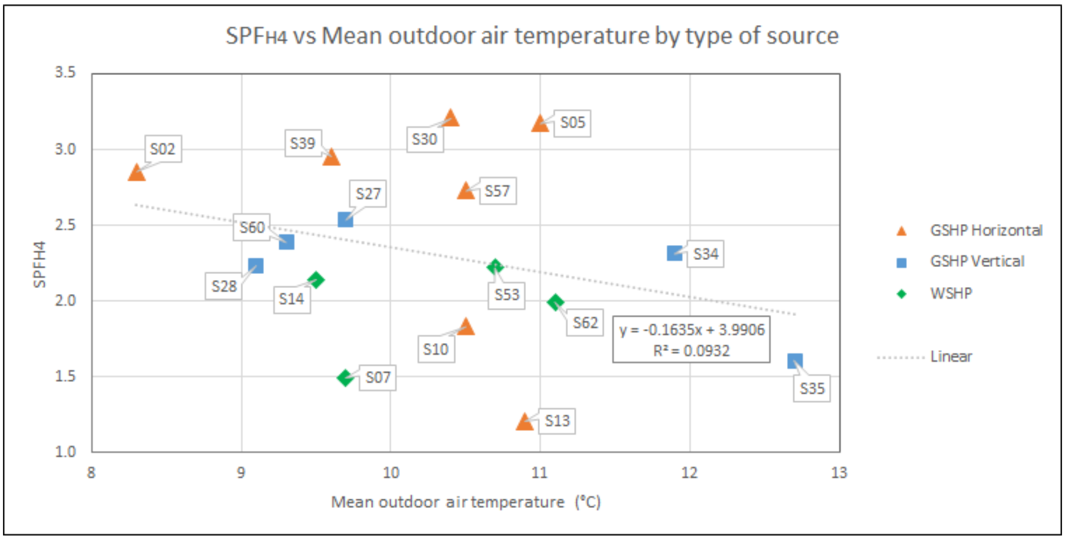

Figure 20 – The influence of mean outdoor air temperature on SPF_{H4} by type of source

4.6 Influence of weather compensation on performance

Weather compensation is a technique used to vary the heat pump output temperature as the outdoor temperature changes. Its purpose is to improve performance at times of reduced space heating demand.

Figure 21 shows an example of weather compensation, at site 57. The hourly mean heat pump output temperature is plotted against the outdoor air temperature. The variation is according to the weather compensation function configured into the heat pump controller. The weather compensation characteristics of other systems are broadly similar, but the slope of the function is different on each system.

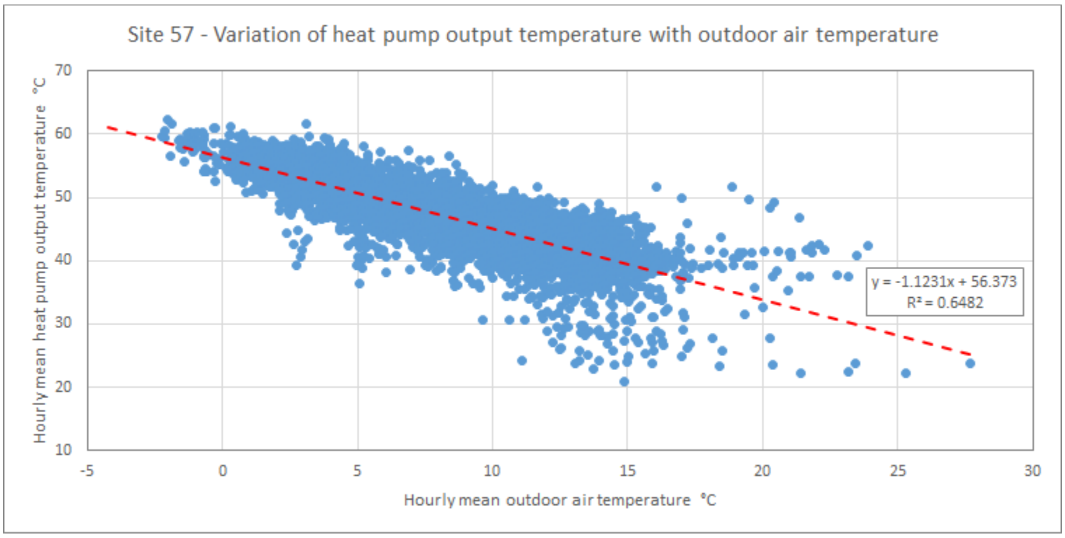

Figure 21 – Variation of heat pump output temperature with outdoor air temperature at site 57

Figure 22 shows the linear regression data for each of the systems monitored.

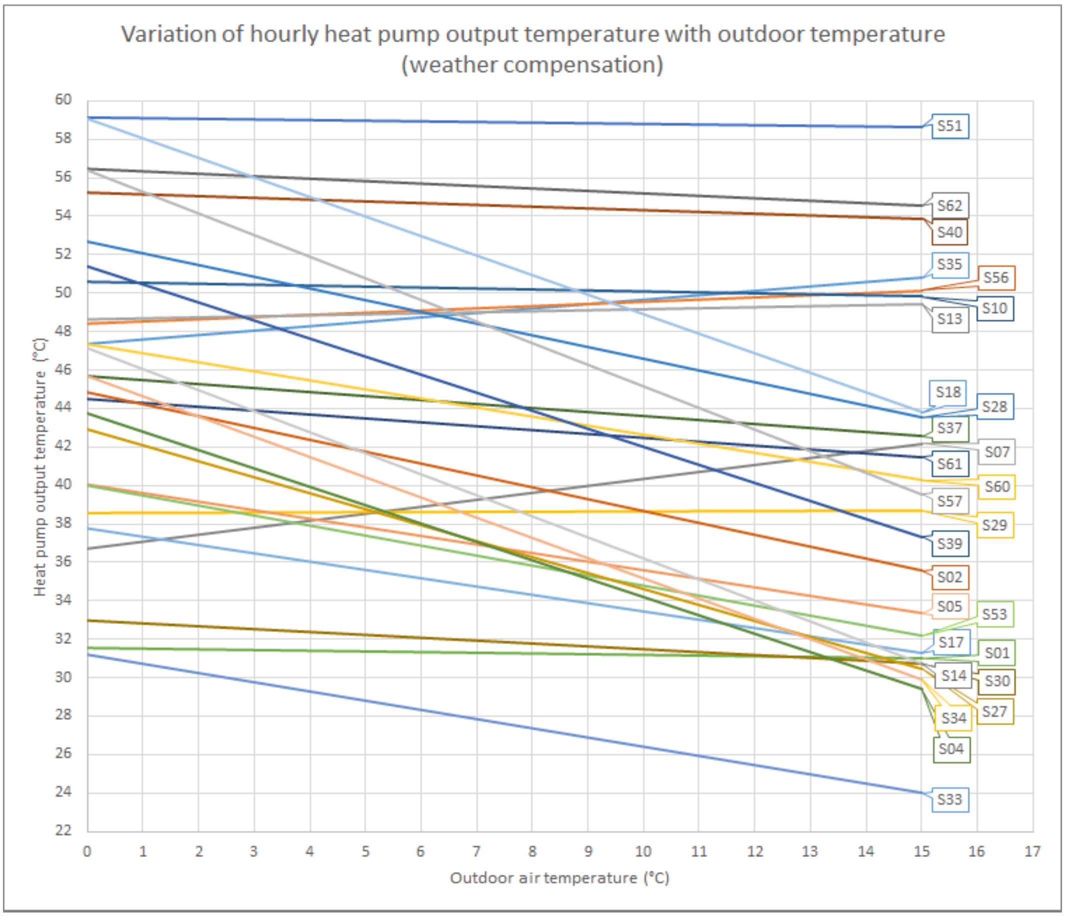

Figure 22 – Weather compensation characteristics of systems monitored (linear regression of hourly data)

The graph in Figure 22 is rather difficult to follow, because the range of temperatures as well as the slope of the compensation function varied from one system to another. Figure 23 uses the same regression data, adjusted to show the change of heat pump output temperature referenced to 0 °C. From this graph, it can be seen that the weather compensation characteristics of the systems range from +0.36 °C / °C (change in heat pump output temperature per degree Celsius rise in outdoor temperature) to -1.10 °C / °C.

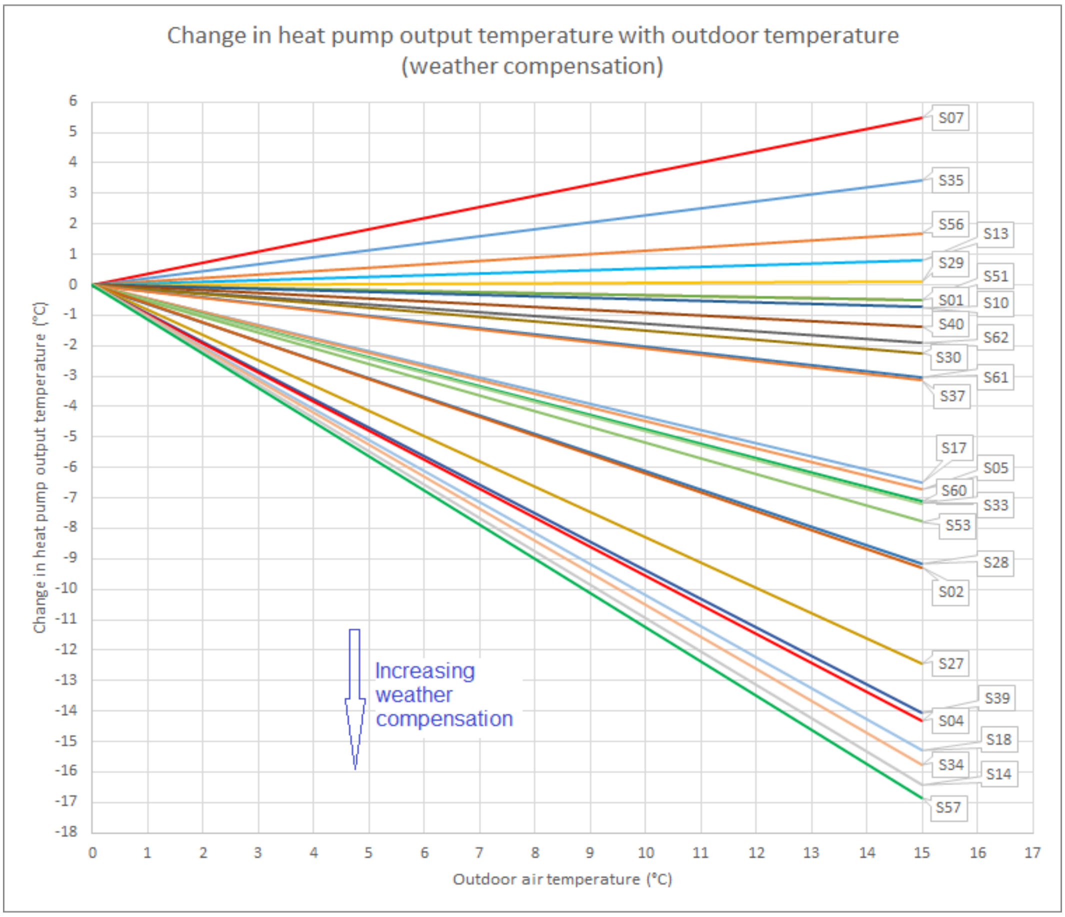

Figure 23 – Weather compensation characteristics of systems monitored (change of heat pump output temperature vs outdoor temperature – normalised at 0 °C)

Weather compensation should have a negative characteristic, with the heat pump output temperature reducing as the outdoor temperature rises. The opposite behaviour of some systems is rather surprising – although these systems did not use weather compensation, or, in the case of site 07, had obvious operational problems.

Figure 24 shows the effect of weather compensation on the heat pump performance SPFH2. The data from the studied sample shows no discernible effect of weather compensation on SPF_{H2}. This does not mean that weather compensation has no effect on performance: just that it is probably masked by the effects of other factors.

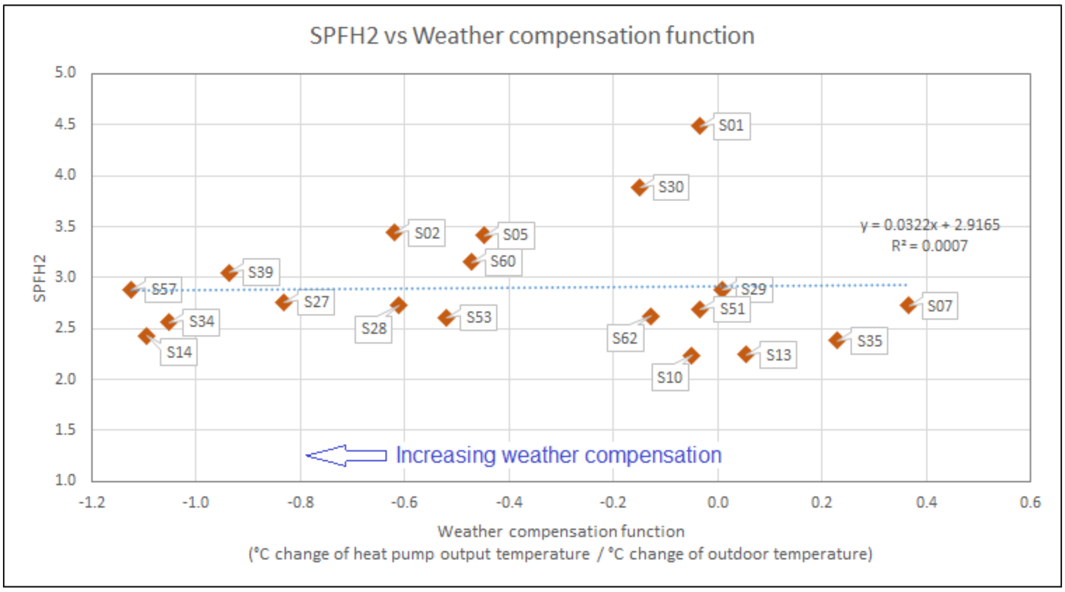

Figure 24 – The effect of weather compensation on SPF_{H2}

4.7 Influence of hours of operation on performance

All of the systems monitored operated 24 hours a day, 7 days a week (24/7) during the winter months (October – March), with many of the systems also operating for much of the summer period – especially those that provided domestic hot water. However, heat pump operation did not always coincide with demand for heat: the hours during which there was demand for heat was less than 24/7 on some sites. The reasons for the heat pumps operating outside the heating demand periods varied. Figure 25 shows a histogram of the SPF_{H4} of systems that operated with a 24/7 demand for heat during winter months.

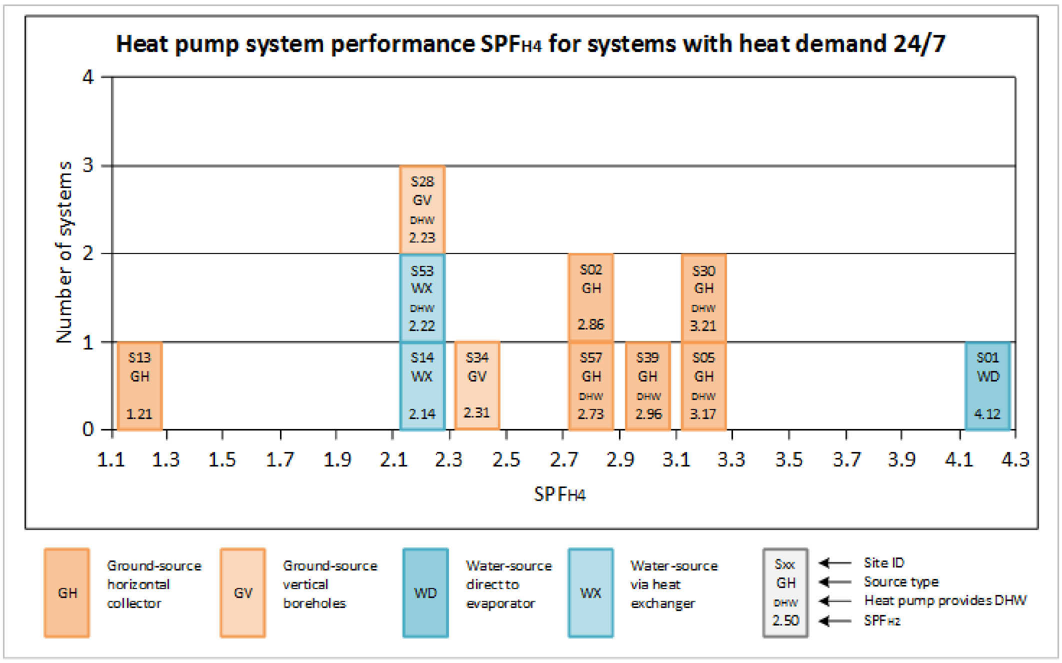

Figure 25 – Histogram of the SPF_{H4} of systems with demand for heat 24 hours/day, 7 days/week during winter months

Figure 26 shows the SPF_{H4} values for systems that operated with heat demand for fewer than 24 hours per day during the winter months. At site 10, the heat pump system continued to operate with no load during times when there was no demand for heat (sometimes called "dry cycling"), causing wastage of energy and probable reduction in overall performance. At other sites, there was reduced demand for heat during the night and at weekends.

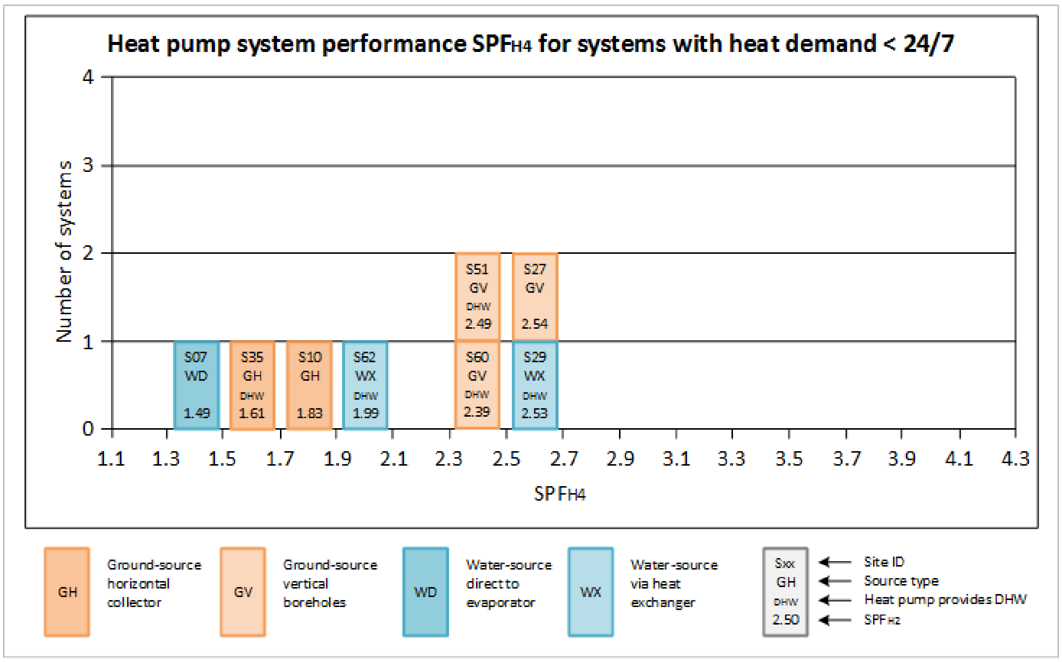

Figure 26 – Histogram of the SPF_{H4} of systems with heat demand for less than 24 hours/day during winter months

Statistical analysis of two groups (24/7 heat demand and <12h/day heat demand) using the T-Test[24] does not show any statistically significant difference between the mean SPF_{H4} of the groups. It can therefore be concluded that the daily hours of heat demand had no significant influence on performance in the monitored sample.

4.8 System performance by heat pump manufacturer

The systems being monitored use heat pumps from 10 different manufacturers, installed by 23 different installers. The system performance by manufacturer has been assessed, and there is no obvious clustering of the systems from any manufacturer.

Factors that have positive or negative influence on performance will be further considered in section 4 below.

4.9 Influence of source & sink temperatures on performance

One of the main factors that influences the performance of a heat pump is the difference between the source (input) and sink (output) temperatures – referred to as the “temperature lift”. In theory, and generally in practice, performance is best when the temperature lift is lowest. Therefore, it is desirable for the source temperature to be as high as possible and for the output temperature to be as low as possible.

For the systems in this study, the source from which heat is being extracted is the ground or the water source; the sink is the space being heated and on some systems the domestic hot water.

Heat extraction from the source to the heat pump is usually via a brine circuit (except on some water-source systems where the water is pumped directly to the evaporator). The brine will always be colder than the source, because a temperature difference between the source and the brine is needed to transfer the heat.

Heat delivery from the heat pump to the heated space is via a hot water circuit. This water will always be warmer than the space because of the need for a temperature difference to transfer the heat. The same applies to the heating of domestic hot water: the heat pump output temperature needs to be higher than that of the water being heated.

An objective of good system design and operation should be to maximise the temperature of the brine at the input to the heat pump, and to minimise the temperature at the heat pump output.

4.9.1 Influence of source temperature on performance

Figure 27 shows the range of source temperatures (at the inlet to the heat pump) for each system (except site 35 where the brine temperature was not measured and site 48 where insufficient useful data was recorded). The squares indicate the mean temperature during the monitoring period July 2015 to June 2016. The vertical bars indicate the range of values (mean ± 2 x standard deviation).

It can be seen that the two systems where the source water is pumped directly to the evaporator have the highest mean source temperatures at the heat pump. There does not appear to be any obvious grouping for other types of system.

[24] The T-Test is a statistical method that assesses whether the mean values of two groups are different from each other.

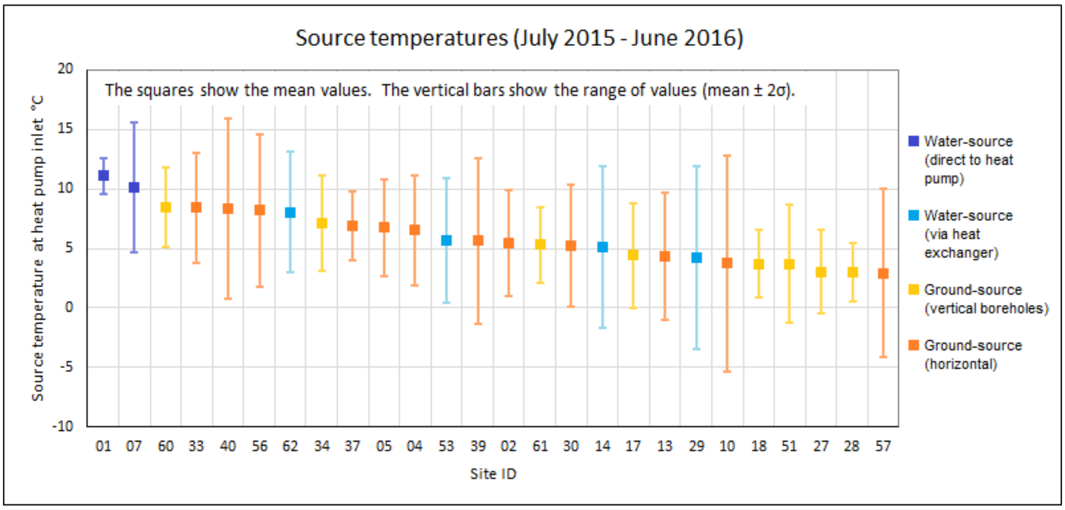

Figure 27 – Range of source temperature at heat pump inlet for each system

Figure 28 shows a scatter plot of the SPFH4 values for each system, plotted against the mean source temperature at the inlet to the heat pump evaporator. The regression line shows the trend for the performance to improve with increasing source temperature as would be expected, although the coefficient of determination (R^2) is not very high. The scatter in the data points is due to there being other unrelated factors that also affect performance (e.g. output temperature, use of auxiliary heat, electricity use by ancillary pumps).

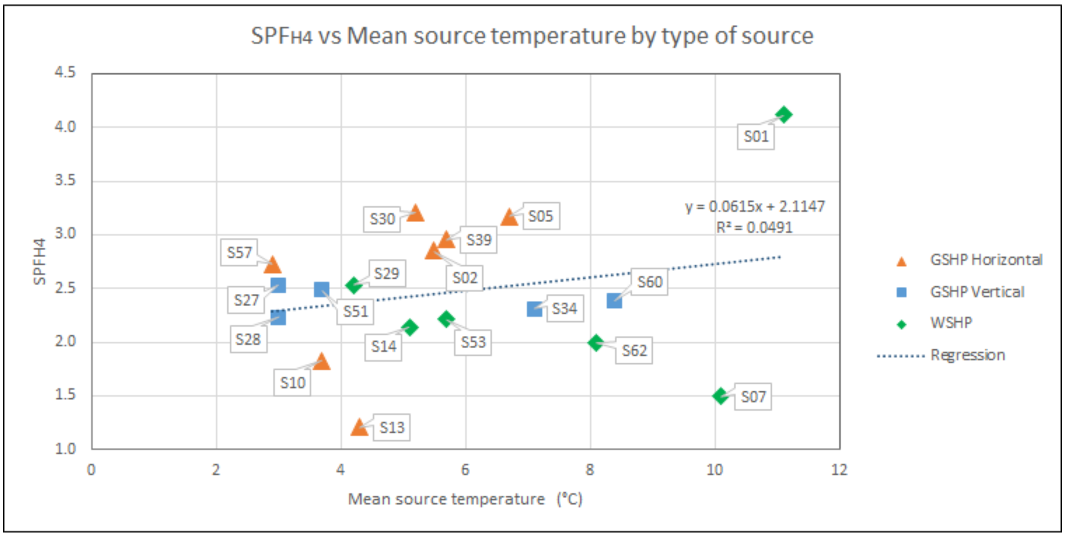

Figure 28 – Influence of source temperature on SPFH4

4.9.2 Ground collector effectiveness

The source temperature at the heat pump is affected by the effectiveness of the source heat exchanger.

For ground-source systems with horizontal collectors the temperature of the ground was measured at a location remote from the collector, 1 metre below the surface. Figure 31 shows the range of temperature difference between ground and brine flow for each system, during winter operation (October 2015 to March 2016) when the heat load would have been greatest.

It can be seen that some collectors apparently work rather better than others: a small temperature difference between the ground and the brine indicates effective heat transfer – although it should be noted that the ground temperature on each site is measured by a single probe that may not have been in the same type of soil or have received exactly the same hours of direct solar radiation as the collectors. The variation in the range of temperature differences from one site to another may be due to the different types of soil and to different collector loading at each sites. However, this was not systematically investigated in this project.

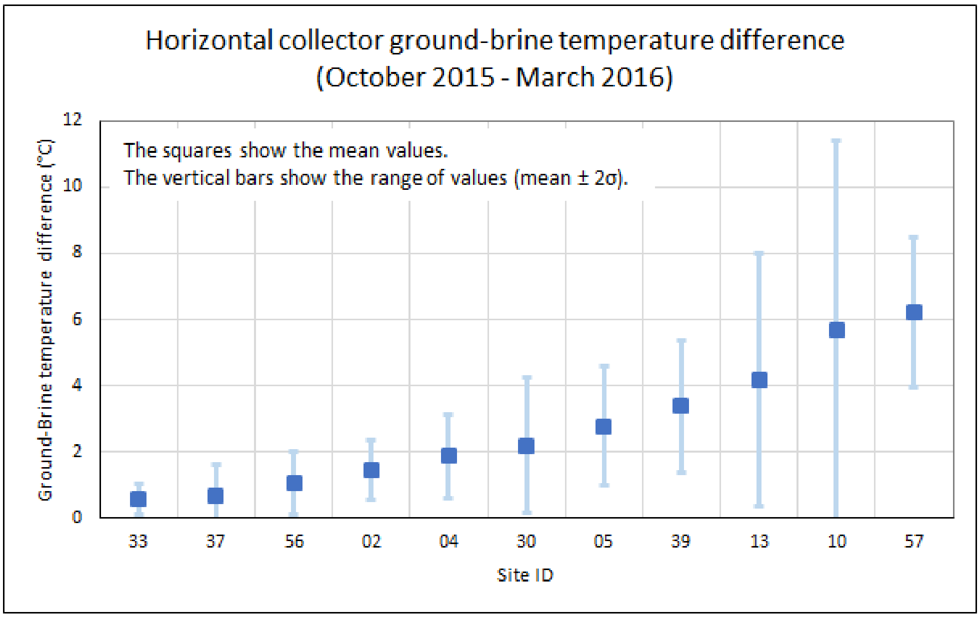

Figure 29 – Horizontal ground collectors: temperature difference between ground and the brine flow

4.9.3 Influence of heat pump output temperature on performance

Figure 30 shows the range of heat pump output temperatures for each system. Note that two markers are shown for some systems: one for SH operation and one for DHW operation. The blue markers indicate outputs to SH; the red markers show outputs to DHW (to the heating coils); the orange markers show the heat pump output temperatures of systems that provide SH and DHW simultaneously in a combined circuit. As would be expected, the markers at the lower (left-hand) end of the chart are for outputs to space heating, while those toward the upper (right-hand) end are for systems that provide DHW. The variation in the range of temperature for each system is due to the various operating characteristics – e.g. the degree of weather compensation used.

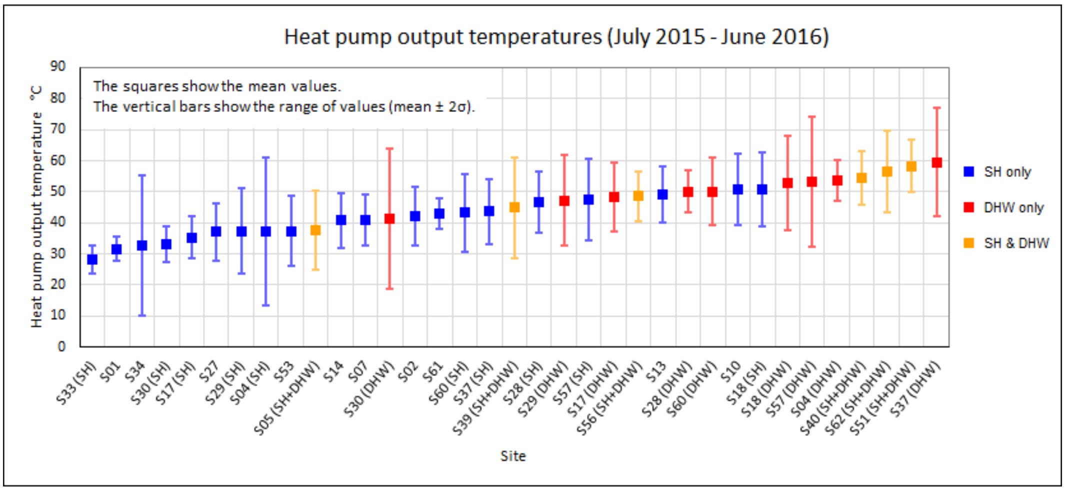

Figure 30 – Heat pump output temperatures for space heating and domestic hot water operation

Figure 31 shows a scatter plot of the SPFH4 values for each system, plotted against the mean temperature of the heat pump output[25]. The regression line shows the expected trend for the performance to increase at lower output temperature, with a reasonable determination (R^2) of 0.28.

25 For systems that have outputs to SH and to DHW, the mean temperature on the chart is a weighted mean temperature, calculated using the number of minutes that the system operated in each mode.

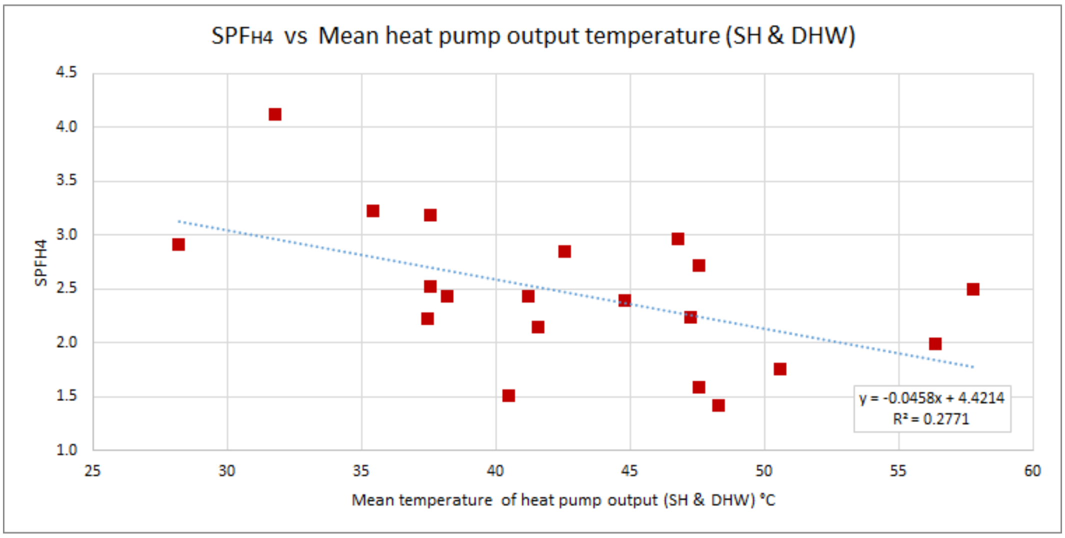

Figure 31 – Effect of mean temperature of heat pump output on SPFH4

4.9.4 Influence of temperature lift on performance

Figure 32 shows the scatter plot of the SPFH2 values for each system, plotted against the temperature lift (between the inlet to the heat pump evaporator and the outlet from the heat pump condenser). The regression line shows the expected trend with reasonably good determination ($R^2 = 0.36$).

This graph may provide a useful basis for estimating the performance of practical heat pumps – e.g. for feasibility analysis or at an early stage of system proposal, before carrying out detailed performance calculations.

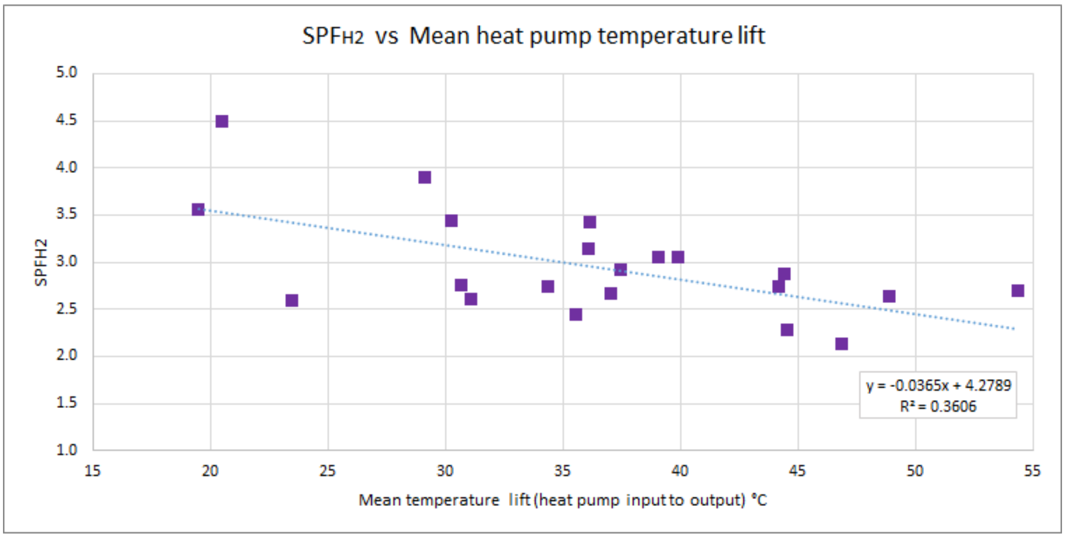

Figure 32 – Effect of mean temperature lift on heat pump performance SPFH2

Figure 33 shows the effect of mean temperature lift on SPFH1 which represents the performance of the heat pump excluding electricity used by the source pump. The linear regression has a better determination ($R^2 = 0.50$) than for the SPFH2 values. This suggests that there is reasonably good consistency of performance between the various types of heat pump used on different systems, notwithstanding the other factors that influence heat pump performance, such as short cycling, voltage and harmonic distortion of the electricity supply, etc. All of the heat pumps monitored used fixed-speed compressors. Models with variable-speed compressors may behave differently.

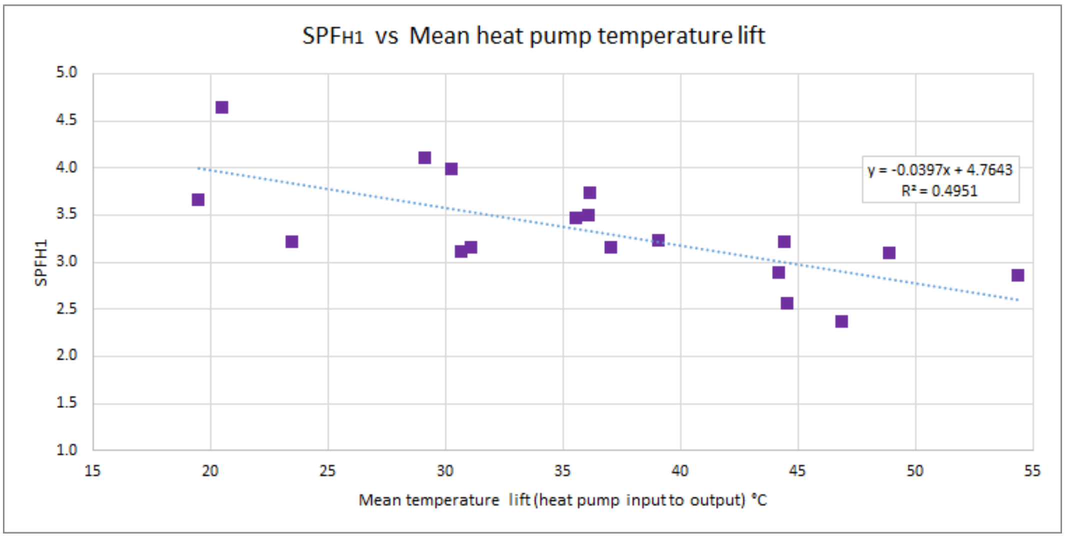

Figure 33 – Effect of mean temperature lift on heat pump performance SPFH1

Figure 34 shows the SPFH1 and the Carnot COPH (the theoretical coefficient of performance for heating) for each heat pump at its mean input and output temperatures. (The small deviation of individual Carnot COPH values from the curve is due to the systems operating at different output temperatures.) This graph shows that the practical performance of the heat pumps is rather less than theoretically possible and that the deviation from theory is greater at low values of temperature lift.

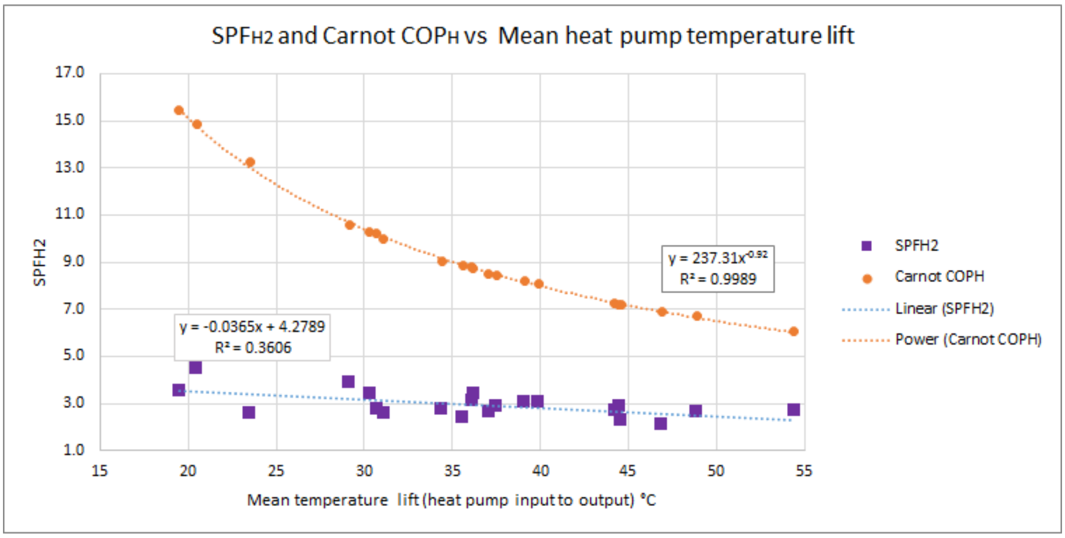

Figure 34 – SPFH1 and Carnot COPH vs temperature lift

The ratio of actual COPH to Carnot COPH can be expressed as the Carnot effectiveness. SPFH1 is essentially the average COPH over a period of time, so can be used in place of COPH. The Carnot effectiveness is thus a measure of how close the SPFH1 performance is to the theoretical maximum.

Figure 35 shows the Carnot effectiveness plotted against mean temperature lift. These values give an indication of the heat pump thermodynamic efficiency (together with the various factors that influence its performance: cycling, electricity supply characteristics, etc.).

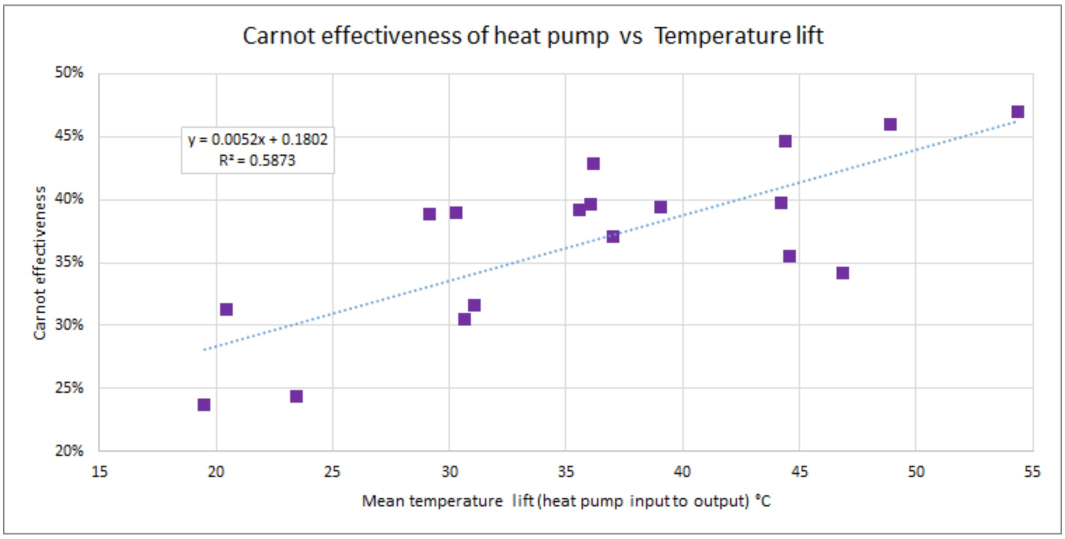

Figure 35 – Carnot effectiveness (based on SPFH1) vs temperature lift

It is notable that the Carnot effectiveness increases as the temperature lift increases, meaning that the units that operate at higher mean temperature lifts are operating more closely to their theoretical maximums. This characteristic can be at least partly explained by two practical considerations.

- Heat exchanger temperature difference:
  A heat pump incorporates two main heat exchangers: the evaporator and the condenser. Heat transfer across any heat exchanger is driven by temperature difference. In the evaporator, the refrigerant (the internal working fluid) being evaporated will be colder than the external source fluid (water or brine). Similarly, in the condenser, the condensing refrigerant will be hotter than the external water being heated. Because of these heat exchanger temperature differences, the internal (i.e. inside the heat pump) temperature lift will always be greater than the external temperature lift.
  As the external temperature lift reduces, the evaporator and condenser temperature differences remain approximately constant[26], but become a greater proportion of the internal temperature lift, causing the internal lift to deviate further from the external lift and causing the efficiency to deviate more from the theoretical efficiency.

- Compressor efficiency:
  The isentropic efficiency[27] of a typical compressor varies with the pressure ratio at which it operates. The low pressure at the compressor inlet (suction) is dependent on the temperature in the evaporator and on the refrigerant properties: the pressure reduces as the temperature reduces. Similarly, the pressure at the compressor outlet (discharge) depends on the temperature in the condenser: a higher temperature requires a higher pressure. Thus, as the temperature lift increases, so the pressure ratio increases. A typical scroll compressor, as used in all of the heat pumps monitored in this study, has an optimum working pressure ratio at which its isentropic efficiency is maximum. Its isentropic efficiency will therefore be lower at other pressure ratios and this affects the overall efficiency of the heat pump as the temperature lift varies. However, it is outside the scope of this project to examine the effects of compressor characteristics on heat pump performance.

In summary, the variation of Carnot effectiveness with temperature lift is mainly a consequence of the practicalities of the way a heat pump operates, rather than anything necessarily to do with the overall system design or operation.

[26] The temperature difference across a heat exchanger is approximately dependent on the amount of heat being transferred, not on the temperature at which the transfer is happening.

[27] Isentropic efficiency is a measure of the power used by a compressor compared to the power theoretically needed.

4.9.5 Influence of continuously high output temperature on performance

Several systems (35, 40, 51, & 62) operate with continuously high output temperature, to provide both space heating and domestic hot water via a combined system. Figure 36 shows the SPF_{H4} plotted against mean temperature lift for all systems. Those with combined output to SH+DHW (except site 40 for which SPF values are not presented) are shown in red. The high temperature lift of these systems is clearly seen. Their performance appears to be lower than for other systems, although the small sample does not yield a statistically significant indication of different performance.

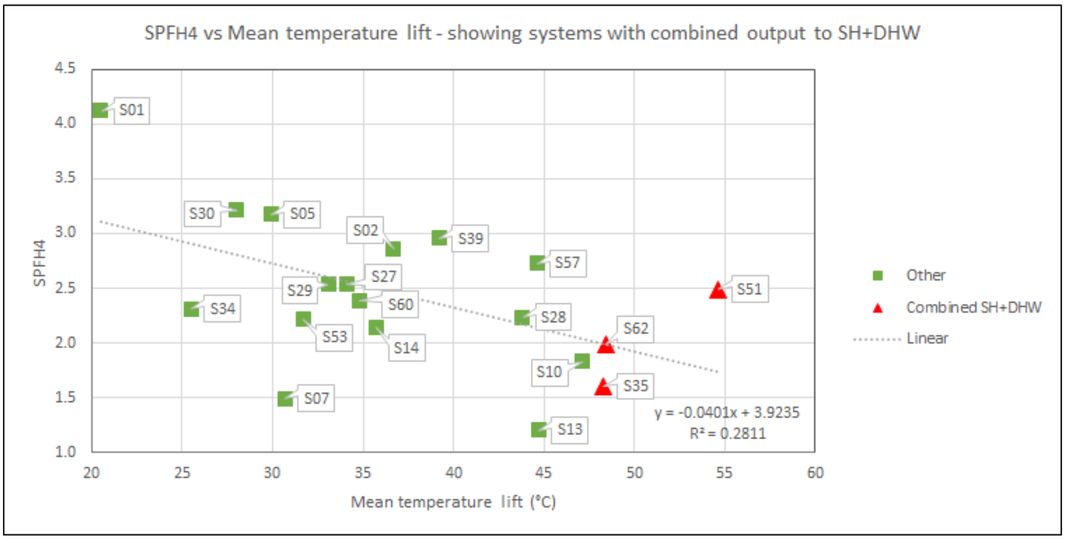

Figure 36 – Effect of continuously high output temperature on performance

4.10 Influence of building age on performance

Some of the systems monitored are used to heat old buildings, where the energy performance would be expected to be well below modern standards. It could be expected that these systems would have lower performance than other systems.

Table 6 shows a summary of the sites with buildings constructed at least 50 years ago that have not been refurbished to modern energy performance standards. These sites all use radiators.

Site	Building type	Constructed	Heat emitters	Comments
02	Large house	18th century	Radiators as installed for oil heating.	Listed. Used as dwelling.
04	Large house	18th century	Radiators as installed for oil heating.	Used as dwelling.
28	Cut-stone castle	19th century	Radiators as installed for oil heating. Some have been replaced with larger sizes since heat pump installed.	Used as hotel.
29	Large house	17th century	Radiators as installed for oil heating.	Used as dwelling and for events.
51	Large house	circa 1960	Radiators as installed for oil heating.	Used as offices.
57	Large house	circa 1960	Radiators as installed for oil heating. Some have been replaced with larger sizes since heat pump installed.	Used for recreational activities.
62	Large house	18th century	Radiators as installed for oil heating.	Used as dwelling and for events.

Table 6 – Sites with old buildings

Figure 37 shows the plot of SPFH4 versus mean temperature lift, with the old buildings shown with orange markers. It can be readily seen that the sites with old buildings do not have lower performance. There is no statistically significant difference between the mean values of SPFH4 for old or new buildings.

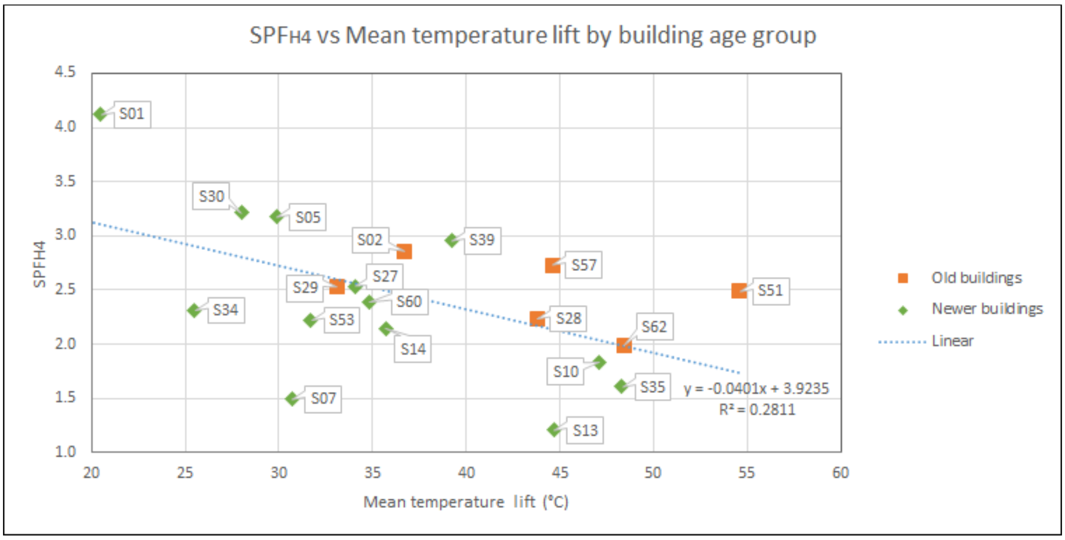

Figure 37 – Influence of building age on system performance

4.11 Cycling

A heat pump will usually cycle on and off a number of times each day, depending on the heat demand. The number and length of the run times will tend to reduce during warmer weather because of the reduced heat demand.

Previous research [4] recommended that systems be designed to achieve a minimum run time of circa 6 minutes under all conditions, to avoid the worst excesses of performance impairment due to short cycling.

The only systems that were found to have short-cycling behaviour likely to impair the performance were at sites 10 and 56.

At site 10, the heat pump was observed to run for very short times throughout the year.

Figure 38 shows the behaviour during a typical winter weekday: the blue line shows the electrical power drawn by the heat pump and the red line shows the thermal power measured by the heat meter. It can be seen that the heat pump ran for a number of short runs during the night, although there was no demand for heat. (The small amount of heat measured by the heat meter during the night was caused by a small flow through the heating circuit.)

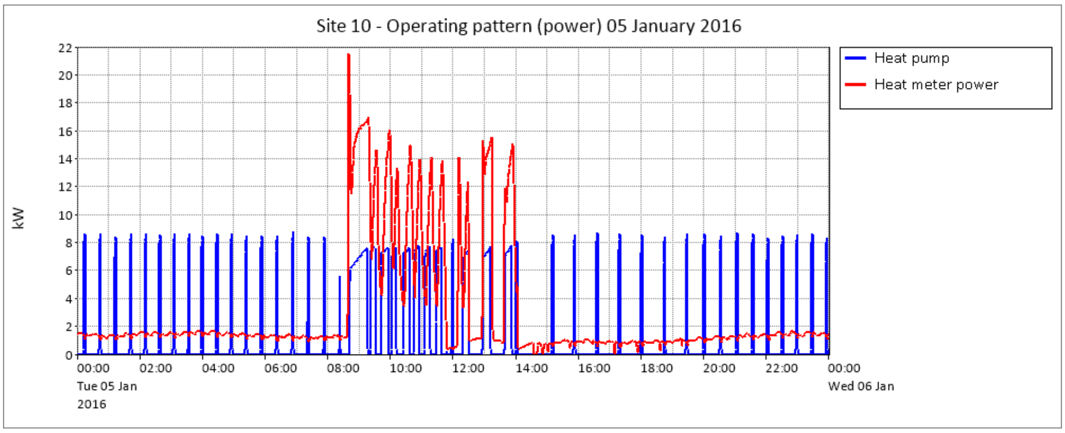

Figure 38 – Site 10: typical operating behaviour on 5th January 2016

Most of the short runs during periods of low load (at night and from May to October) were for less than 4 minutes. The longest run during the year was for 32 minutes, but most runs were much shorter than that. Figure 39 shows the distribution of run times. The very large proportion of runs of less than 4 minutes is clearly seen.

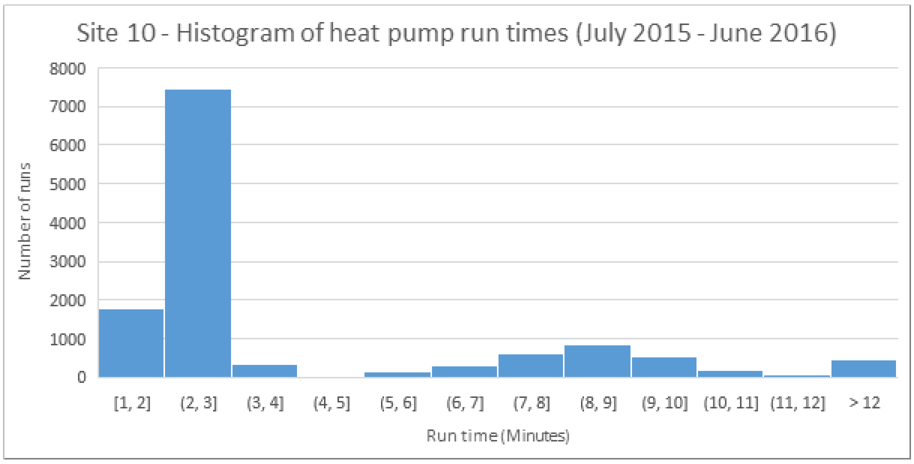

Figure 39 – Site 10: histogram of heat pump run times (July 2015 to June 2016)

At site 56, short cycling of the heat pump was sometimes observed for 2 or 3 hours after the call for heat started for the day.

Figure 40 shows the behaviour during a 6-hour period on 9th January 2016, at the beginning of the daily heating cycle. The heat pump ran for 10 minutes at 04:00 to top up the domestic hot water.

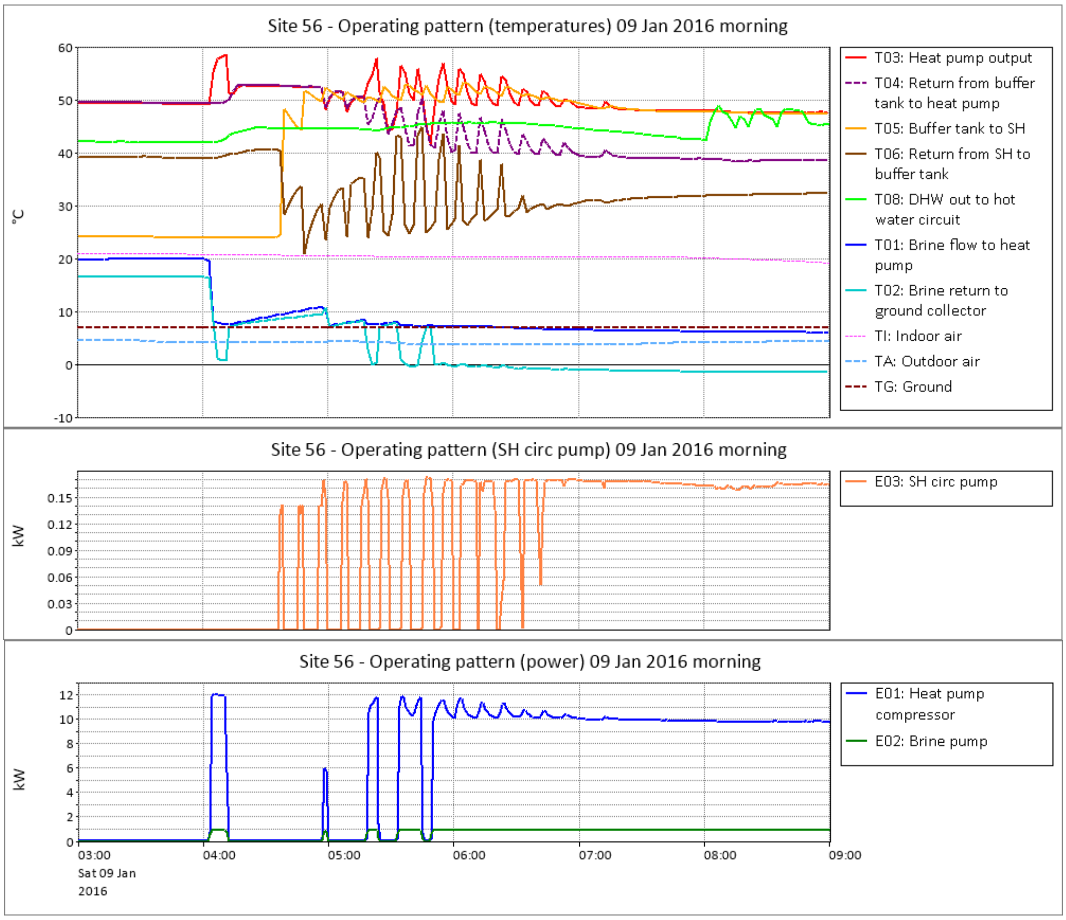

Figure 40 – Site 56: behaviour after startup

The space heating circulating pump started running at 04:36 – initially for a series of short runs of 3 – 9 minutes. After 06:44 it ran continuously until 17:45. The first two short runs apparently circulated heat from the buffer tank. The heat pump started during the third run of the circulating pump and ran for three progressively longer runs of 3, 6 and 12 minutes, before starting its continuous run at 05:50. This behaviour is believed to have been caused by the use of a smart thermostat (programmable controller) that is intended for use with a boiler. It appears that this controller imposes on/off cycles that are not appropriate for a heat pump system.

The extent to which the observed cycling behaviour on these two systems influenced performance is unknown.

4.12 Ancillary equipment

Pumps are needed to pump water from the source or brine through the ground collector or source heat exchanger, and to circulate the hot water from the heat pump via the buffer tank to the heat emitters in the building. These pumps use electricity, which can be a significant part of the total electricity used by the system.

Some heat pump systems incorporate electric immersion heaters to provide auxiliary heat at times when the heat pump is unable to provide the heat required by the load – usually during very cold conditions. Other systems may have immersion heaters in the buffer tank or domestic hot water cylinder.

It is desirable to minimise the electricity used by ancillary equipment.

4.12.1 Breakdown of electricity use by ancillary equipment

Figure 41 shows the breakdown of electricity use for each system. The data is presented as stacked bars, where each colour represents the percentage of the total heat pump system electricity used by each type of ancillary equipment and by the heat pumps. Table 7 shows the same data as well as the totals for all ancillary equipment on each system and the median value for each type of ancillary.

The total electricity used by ancillary equipment varied from 8.0% on site 39 to 55% on site 07. The median value was 23.7%.

The very low figure for site 39 was due to the use of high-efficiency heating circulating pumps and the absence of any auxiliary heat.

The electricity used for open-loop source pumping of water from borewells on site 14 and from a river on site 53 was high. Site 07 also uses open-loop pumping of source water, but benefits from gravity with the lagoon from which water is drawn being several metres above the heat pump, so the electricity used by the source pump was a rather lower percentage of the total.

Site 07 had an unusual issue with high electricity use by the buffer pump, which was run periodically when the heat pump was not running to allow the control system to determine the temperature of the return from the buffer tank – although this system also had other issues that caused the heat pump to be used much less than it could have been. The buffer pump electricity would probably have been a much lower percentage of total electricity had the heat pump been put to greater use.

Site 60 also had fairly high electricity use by the buffer pumps which ran continuously, instead of only when the heat pump was running.

The heating circulating pumps at sites 30, 33 and 61 appeared to be run more than necessary, with resulting above-average electricity use. At sites 02 and 62, the above-average electricity use by the heating circulating pumps appeared to be the result of long pipe runs from the plant rooms to the heat emitters.

At sites 28, 30, 37 and 60, immersion heaters were used to boost the DHW temperature for Legionella control. The high use of immersion heaters at site 18 was unexplained.

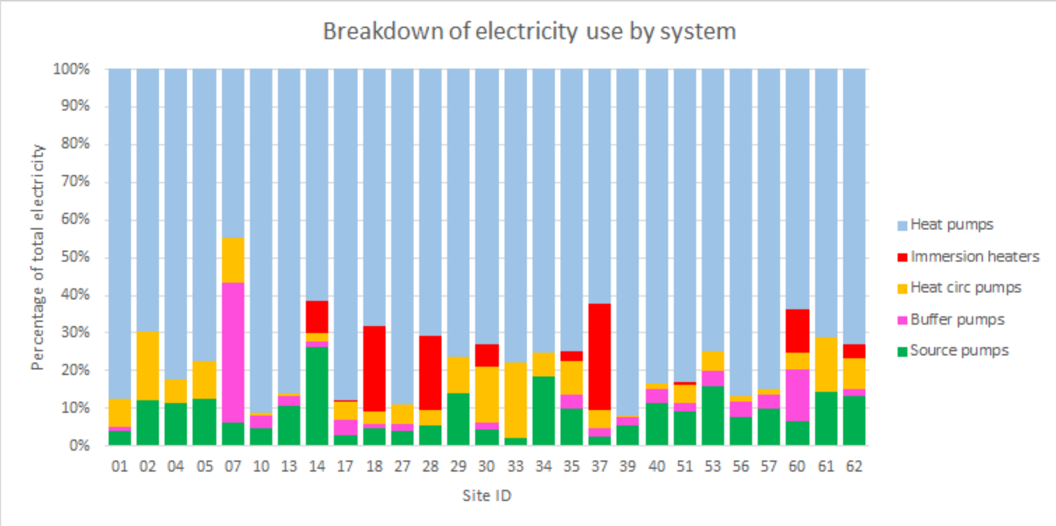

Figure 41 – Breakdown of electricity use by system

Site ID	Source pumps	Buffer pumps	Heat circ pumps	Immersion heaters	Total ancillaries	Heat pumps
01	4.1%	1.2%	7.2%		**12.4%**	87.6%
02	12.2%		18.1%		**30.3%**	69.7%
04	11.5%		6.1%	0.0%	**17.6%**	82.4%
05	12.7%		10.0%	0.0%	**22.7%**	77.3%
07	6.3%	36.9%	12.1%		**55.3%**	44.7%
10	4.9%	3.3%	0.8%		**9.0%**	91.0%
13	10.5%	2.7%	1.0%		**14.2%**	85.8%
14	26.2%	1.7%	2.2%	8.4%	**38.5%**	61.5%
17	3.0%	4.1%	4.8%	0.4%	**12.3%**	87.7%
18	4.6%	1.4%	3.1%	22.9%	**32.0%**	68.0%
27	4.0%	1.8%	5.4%		**11.2%**	88.8%
28	5.3%	0.0%	4.2%	19.8%	**29.3%**	70.7%
29	14.0%	0.0%	9.7%		**23.7%**	76.3%
30	4.2%	1.9%	14.9%	5.9%	**26.9%**	73.1%
33	2.2%		19.9%		**22.1%**	77.9%
34	18.6%		6.3%		**24.9%**	75.1%
35	10.0%	3.7%	8.9%	2.4%	**25.0%**	75.0%
37	2.4%	2.4%	4.6%	28.3%	**37.7%**	62.3%
39	5.3%	2.4%	0.3%		**8.0%**	92.0%
40	11.4%	3.9%	1.4%		**16.7%**	83.3%
51	*8.8% ***	2.1%	4.7%	0.8%	*16.4%*	83.5%
53	15.7%	4.3%	5.3%		**25.3%**	74.7%
56	7.7%	4.2%	1.5%		**13.4%**	86.6%
57	9.8%	3.7%	1.8%		**15.3%**	84.7%
60	6.7%	13.6%	4.4%	11.5%	**36.2%**	63.8%
61	14.5%		14.5%		**29.0%**	71.0%
62	13.1%	1.9%	8.4%	3.8%	**27.2%**	72.8%
Median	**8.8%**	**2.4%**	**5.3%**	**4.9%**	**23.7%**	**76.3%**
		** Site 51: The power used by the internal source pump is unknown. The figure shown is the median of values for all other systems.				

Table 7 – Breakdown of electricity use by system

4.13 Influence of total ancillary electricity use on performance

Figure 42 shows the effect of the electricity used by ancillary equipment on system performance SPFH4. Unsurprisingly, the trend is for SPFH4 to reduce as the ancillary equipment uses a higher proportion of the total electricity. The low coefficient of determination (R^2) for the linear regression can be explained by the influence of other factors on the performance.

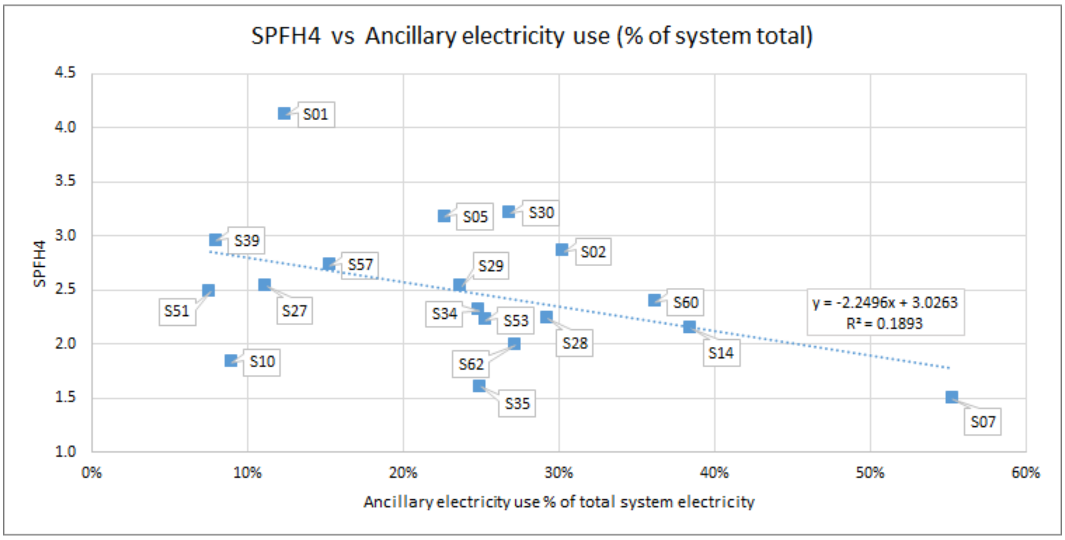

Figure 42 – Effect of total ancillary electricity use on SPFH4

4.14 Influence of source pumping power on performance

Figure 43 shows a scatter plot of the SPFH4 values for each system, plotted against the percentage electricity use of the source pumps. The linear regression line is shown, but it should be noted that the coefficient of determination (R^2) is very low, so the relationship between the source pumping electricity percentage and SPFH4 is rather weak for the sample analysed.

However, common sense tells us that, other factors being equal, higher electricity use by the source pump must detract from the SPFH4, because the pumping energy is a factor in the denominator of the SPF equation.

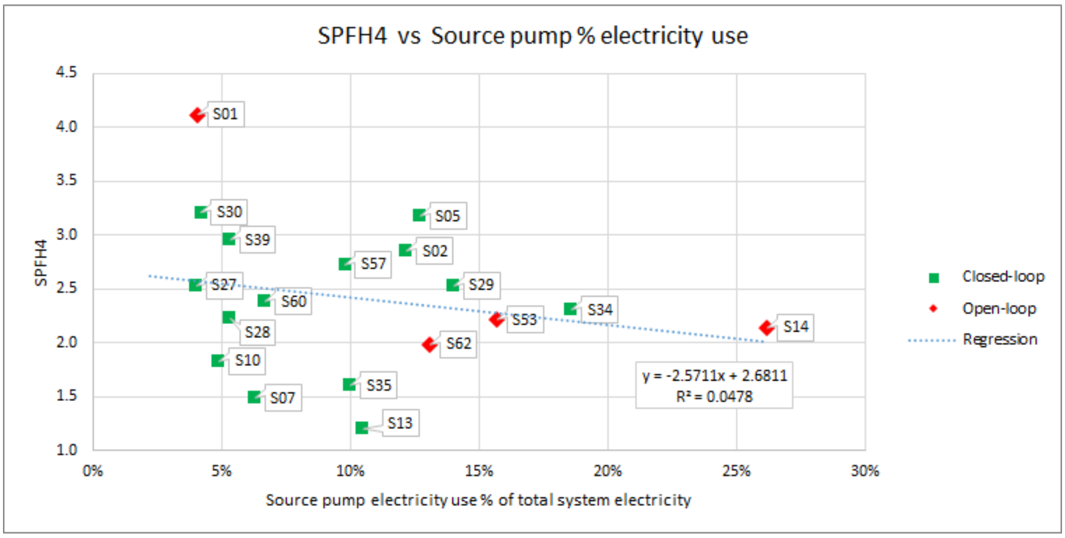

Figure 43 – Effect of source pumping electricity % use on SPFH4

An example of how energy savings could be made by using variable speed pumps was identified at site 04. The system has two heat pumps, each with two compressors. Four different levels of operating capacity are thereby possible. The brine pump used 5325 kWh between 1/7/2015 and 30/6/2016 – 11.4% of the total electricity used by the heat pump system. A system schematic provided by the proprietor showed that an inverter (variable-speed) drive had been considered for the brine pump, but had evidently not been installed. Had it been installed, some useful electricity savings could have been made.

Analysis of the heat pump electrical power data shows that only 1 compressor was used for 60% of the total heat pump run time, 2 compressors for 24% and 3 compressors for 16% of the time. (4 compressors were used simultaneously for only 0.01% of the time.) The full brine flow rate should only be needed when 3 or 4 compressors are running. So, for 84% of the time, a lower brine flow rate would be adequate and reduced pump speed should reduce the overall electricity use of the brine pump[28] by around 60% – yielding an annual energy saving of around 2684 kWh and a potential cost saving of £395 per annum[29].

The situation at some of the other sites monitored (e.g. sites 02, 05, 07, 29, 34, 61) with multiple heat pump compressors and a single brine pump is similar: the brine pump speed could be reduced for much of the time when only one of the compressors is in use. Variable-speed source pumps are used at sites 14, 27 and 62. Other systems have heat pumps with internal brine pumps controlled by the heat pump controllers.

4.15 Influence of sink pumping power on performance

Figure 44 shows a scatter plot of the SPFH4 values for each system, plotted against the percentage electricity use of the sink (buffer and heating circulating) pumps. The linear regression line is shown, but the coefficient of determination (R^2) is very low and the line appears to slope in the wrong direction: the SPF would be expected to decrease with increasing pumping electricity use. The only conclusion that can be drawn is that, for this sample, the sink pumping power does not appear to be a dominant factor in determining performance.

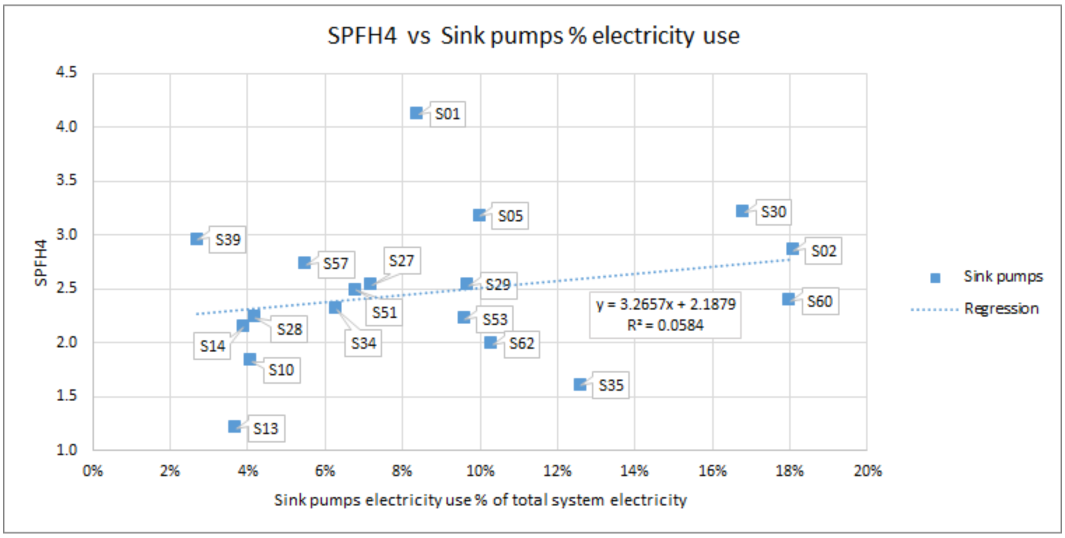

Figure 44 – Effect of sink pumping electricity % use on SPFH4

4.16 Influence of electric auxiliary heat on performance

High use of auxiliary heat would be expected to reduce the system performance because the heat output from an auxiliary heater is never greater than the energy input and such auxiliary heat therefore reduces the overall effect of the heat pump which has an output higher than the paid-for energy input.

Figure 45 shows the SPFH4 and the percentage of total heat output provided by auxiliary heaters for each system.

Note: backup heaters were used during the monitoring period at some sites (07, 14, 51 & 62) – e.g. when a heat pump had failed or needed to be taken out of service, or if there was an obvious issue with the controls, as at site 07. Backup heat has been excluded from SPF calculations, so the auxiliary heat values shown are only for heat provided during normal operation.

[28] See the Carbon Trust report "Estimating savings from VSDs" [21]

[29] Assuming an electricity unit cost of 14.7 p/kWh [16]

Site 13 is a bivalent system where the oil-fired boiler provided 35% of the total heat. For the other systems, the auxiliary heat was used for DHW.

Other than for site 13, the small sample does not permit any conclusions to be drawn about the influence of auxiliary heat on system performance.

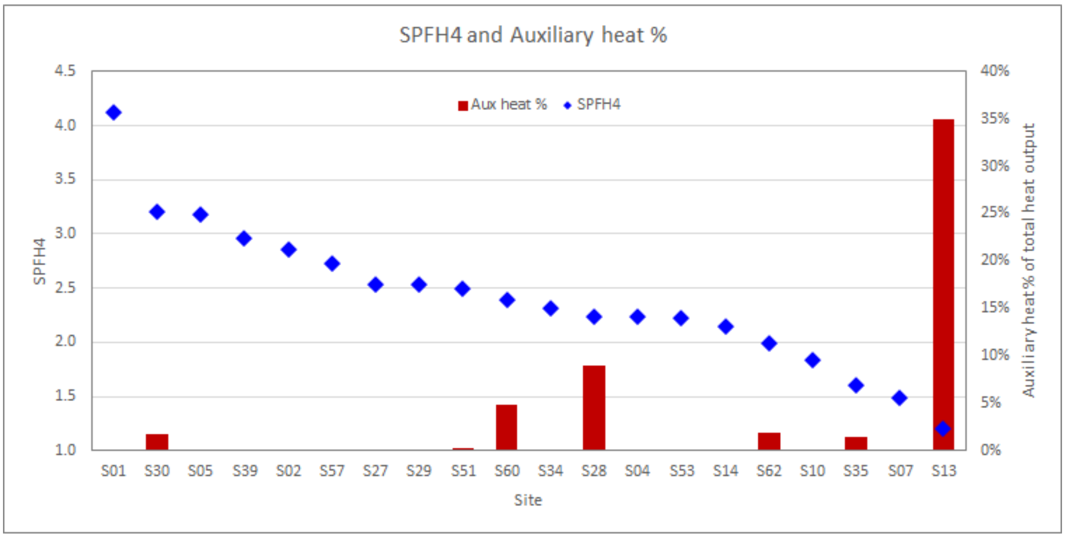

Figure 45 – SPFH4 and auxiliary heat for each system

4.17 Influence of DHW provision on performance

It could be expected that heat pump systems that provide domestic hot water (DHW) as well as space heating (SH) would have lower performance than those that provide SH only, because the temperatures needed for DHW are usually higher than those needed for SH.

Figure 46 shows the SPFH4 plotted against mean temperature lift, grouped by heating duty (SH only or SH & DHW). Visually, it looks as though the systems that provide SH & DHW may have higher performance than systems providing SH only. However, a T-Test analysis shows that there is no statistically significant difference between the mean values of the two groups of data, so this sample provides no evidence that either group has higher performance.

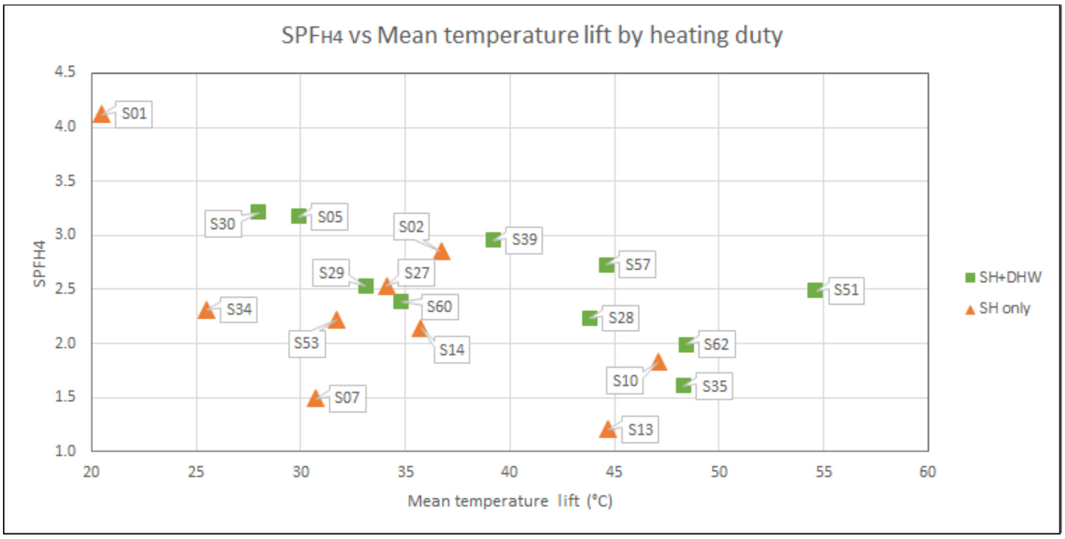

Figure 46 – The effect of mean temperature lift on SPFH4 grouped by heating duty (SH only or SH+DHW)

4.17.1 Note about the measurement or estimation of heat provided to DHW

17 of the systems monitored provide heat to domestic hot water (DHW) as well as space heating. Of these 17 systems, five (sites 18, 37, 39, 57, 60) are equipped with heat meters that directly measure the heat provided to DHW, although two of these (18, 37) use heat metering that has high uncertainty of measurement. Table 8 summarises the methods of providing DHW and the methods used to measure or estimate the proportion of total heat provided to DHW.

Site	DHW provision method		DHW Metered?	Comments	Estimation method
04	A	Alternate			2
05	SI	Simultaneous, intermittent			3
17	A	Alternate			2
18	A	Alternate	Yes	Metering uncertainty	1
28	A	Alternate			2
29	SI	Simultaneous, intermittent			3
30	A	Alternate			2
33	SS	Simultaneous, desuperheater			n/a
35	SC	Simultaneous, continuous			n/a
37	A	Alternate	Yes	Metering uncertainty	1
39	AS	Alternate/simultaneous	Yes		1
40	SC	Simultaneous, continuous			n/a
51	SC	Simultaneous, continuous			n/a
56	SI	Simultaneous, intermittent			3
57	AS	Alternate/simultaneous	Yes		1
60	AS	Alternate/simultaneous	Yes		1
62	SC	Simultaneous, continuous			n/a

Table 8 – Summary of methods of providing and of estimating the heat to DHW

The various methods of providing DHW are:

- Alternate: the heat pump provides either SH or DHW
- Alternate/simultaneous: one vapour compression module of the heat pump alternately provides SH or DHW. The other module provides SH only
- Simultaneous (intermittent): the heat pump intermittently provides both DHW and SH
- Simultaneous (continuous): the heat pump output is used to provide SH and DHW via a combined high temperature system
- Simultaneous (desuperheater): the DHW is provided using the superheated vapour from the compressor discharge.

The methods of metering or estimating the heat provided to DHW were, in outline, as follows:

1. Metered. The heat meter measurements were used. In cases where there was high uncertainty of measurement, it was nevertheless considered that this was the best method of estimating the heat to DHW.
2. Digital filtering. The data processing software determined for each 1-minute interval whether the heat pump was operating in DHW mode, by examining the temperatures of the output to the heating coil in the DHW cylinder. The heat output during each interval was then added to the total for SH or DHW as appropriate.
3. Digital filtering. Similar to method 2, but with reduced accuracy. Used for systems that provide SH and DHW simultaneously but intermittently. The details of the method used varied from one system to

another, but used whatever data from electricity meters and temperature sensors was available to detect DHW operation and to estimate the heat to DHW.

n/a. For other systems, it was not possible to estimate the heat provided to DHW with any useful accuracy.

4.17.2 Breakdown of heat supplied to space heating and to domestic hot water

Figure 47 shows the measured or estimated breakdown of heat delivery for the systems that provide domestic hot water and for which it was possible to determine the breakdown. The data is presented in tabular format in Table 9.

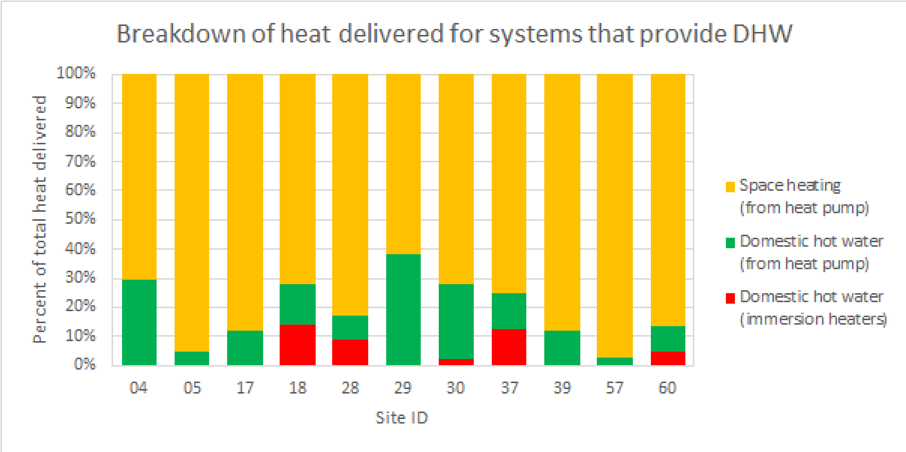

Figure 47 – Breakdown of heat delivered for systems that provide domestic hot water for the period July 2015 to June 2016

Site	Building use	SPFH4	Space heating (from heat pump)	Domestic hot water (from heat pump)	Domestic hot water (immersion heaters)
04	Large house	n/a	70%	30%	0%
05	Public hall	3.17	95%	5%	0%
17	Public hall	n/a	88%	12%	0%
18	Apartments	n/a	72%	14%	14%
28	Hotel	2.23	83%	8%	9%
29	Large house	2.53	62%	38%	0%
30	Public hall	3.21	72%	26%	2%
33	Healthcare facility	n/a	n/a	n/a	n/a
35	Dwellings	n/a	n/a	n/a	n/a
37	Public hall	n/a	75%	13%	12%
39	Dwellings & office	2.96	88%	12%	0%
40	Short-rental apartments	n/a	n/a	n/a	n/a
51	Recreational	2.49	n/a	n/a	n/a
56	Retail shop	n/a	n/a	n/a	n/a
57	Offices	2.73	97%	3%	0%
60	Public hall	2.39	87%	9%	5%
62	Large house	1.99	n/a	n/a	n/a

Table 9 – Breakdown of heat delivered for systems that provide domestic hot water for the period July 2015 to June 2016

The systems with the highest proportions of heat to domestic hot water are 04 and 29. These are both large, old houses with relatively long hot water distribution circuits that, from inspection of recorded data, apparently lose much of the heat provided to the hot water cylinders. The extent or quality of insulation on the hot water circuits in these houses is unknown. At least some of the heat lost from the pipes will provide space heating – albeit in a rather uncontrolled and inefficient manner. Alternative DHW arrangements may be appropriate for these houses. See for example the remarks about point-of-use water heaters in section 4.19.

Sites 30 and 37 are public halls with showering facilities for sports changing and have consequently fairly high demand for domestic hot water. The higher immersion heat figure at site 37 was a consequence of the proprietor becoming concerned about Legionella control and making a number of adjustments to the heat pump controls that resulted in the internal immersion heater being used to generate output at a high temperature (up to 70 °C).

The immersion heaters at sites 28 (a hotel), 30, 37 and 60 (all public halls) were used mainly to boost the temperature in the domestic hot water system for Legionella control.

The extensive use of immersion heaters at site 18 (apartments) was for unknown reasons. The heat pump is evidently capable of meeting the domestic hot water demand as it had previously done so (in January/February 2015), so the significant use of immersion heaters may have been due to an issue with the controls or to inappropriate manual overriding of the automatic controls.

4.17.3 Influence of DHW fraction on system performance

Figure 48 shows a comparison of the SPFH4 and the percentage of total heat output provided as DHW (by the heat pump and by immersion heaters) for each system. The blue markers are the SPFH4 values in descending order and the red bars are the DHW fraction of total heat output.

It appears that the DHW fraction is not of itself a determinant of the system performance SPFH4.

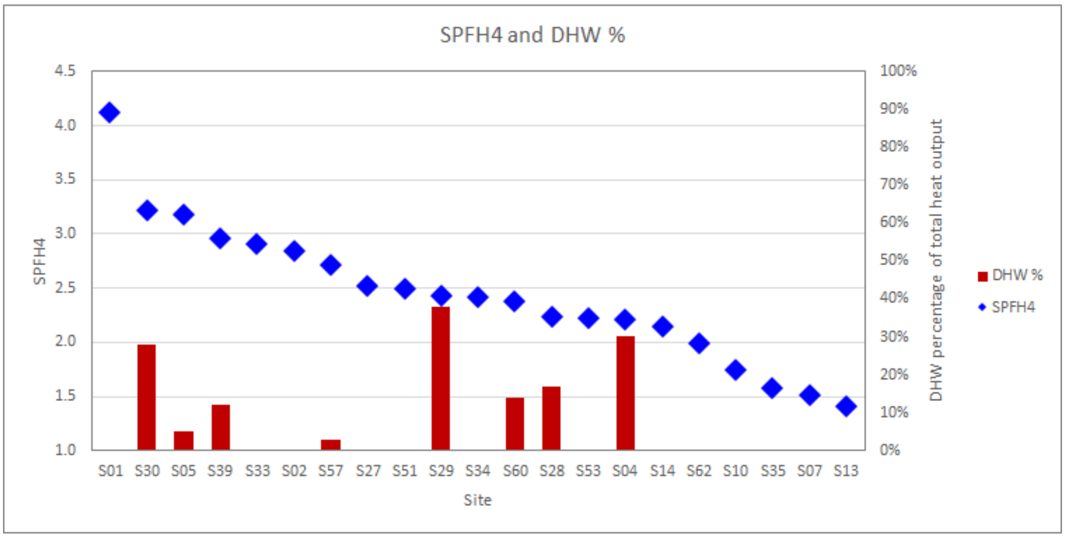

Figure 48 – SPFH4 and DHW % of total heat output

4.18 Legionella control

There is a further complication with the provision of DHW in that growth of Legionella bacteria in the DHW system must be prevented[30]. One method of doing this is to maintain the temperature of the water at the top of the DHW tank at 60 °C and to heat the whole tank to 60 °C for at least an hour a day[31]. If the heat for this is provided by the heat pump, an output temperature of at least 62 °C or more will probably be needed (when the temperature difference across the heat transfer coil in the DHW cylinder is taken into account) – much higher than desirable for high efficiency.

[30] See the HSE code of practice L8 [18]

[31] See HSG274 part 2 [19]

4.18.1 Temperatures of output to DHW

Figure 49 shows the daily maximum temperature of the output to the DHW coil for each relevant system. The top graph shows the systems (6 out of 17) that achieve 62 °C or more on most days (≥ 75% of days). On all but two systems, the high temperature is provided by the heat pump vapour compression system.

At site 37 (a public hall / sports pavilion), an immersion heater in the heat pump is used to achieve the daily high temperature.

At site 30 (a public hall), an immersion heater in the heat pump is used to achieve 62 °C or more at weekly or sometimes longer intervals – as shown in the middle graph.

The system at site 62 (a large house), which achieved 62 °C on only 61% of the days, also used ultraviolet light sterilisation (see section 4.18.3). This system had operational problems for part of the monitoring period. Had the system been operating normally, it would have achieved 62 °C every day.

The bottom graph shows the 9 out of the 17 systems that rarely or never (i.e. < 10% of the days) achieved 62 °C from the heat pump. Systems 17, 28 and 60 used immersion heaters in the DHW cylinders to boost the temperature, although 62 °C was not always achieved.

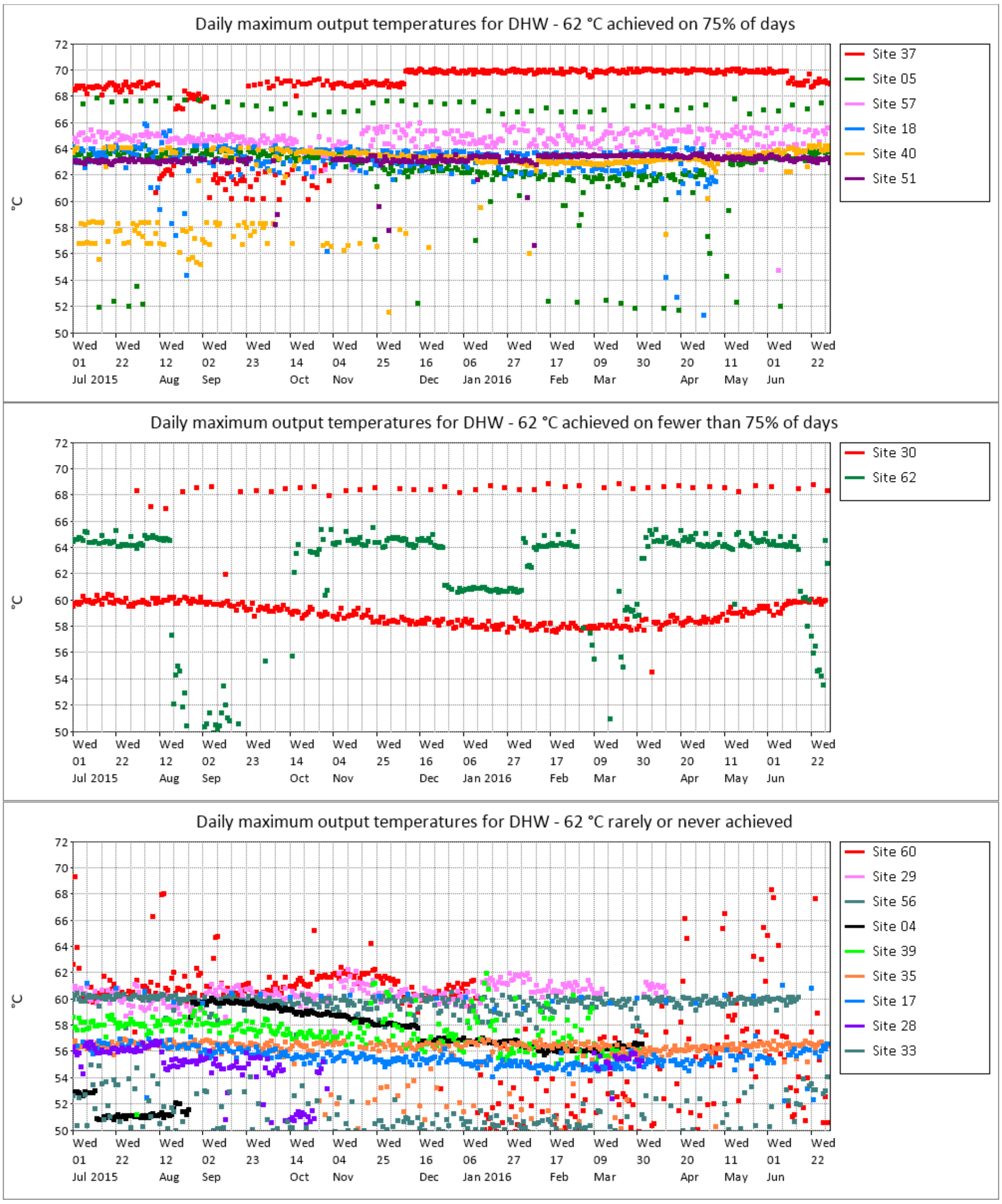

Figure 49 – Daily maximum temperature of output to DHW coil on systems that provide heat to DHW

At site 35 (dwelling houses), the DHW flow temperature was measured as it leaves the central plant. However, there is an immersion heater in the DHW cylinder in each house, under control of the occupants: the operational strategies and the effect on the DHW temperatures achieved in the tanks are unknown.

There are immersion heaters in the DHW cylinders at sites 04, 29 and 39, but these were not used. It appears therefore that the DHW temperature on these systems was never above 60 °C during the monitoring period.

At site 33 (a healthcare clinic), the heat pump has an internal buffer tank that provides both space heating and DHW, with the DHW temperature boosted by a desuperheater coil. The temperatures shown for this site are of the DHW draw-off.

At site 60 (a public hall), where an immersion heater in the DHW cylinder was used to boost the temperature, the measured DHW draw-off temperature was above 60 °C every day until the immersion heater stopped working on 17th January 2016. Thereafter, the DHW draw-off temperature was between 45 and 55 °C.

Table 10 summarises the arrangements used for raising the DHW temperature.

Site	Heat pump provides temperature ≥ 62 °C on ≥ 75% days	Heat pump provides temperature ≥ 62 °C on < 75% days	Immersion heater inside the heat pump used to boost the DHW temperature	Immersion heater in the DHW cylinder used to boost the temperature	Desuperheater	Legionella control possibly unsatisfactory
04				✓		×
05	✓ (76%)					
17				✓		×
18	✓ (91%)					
28				✓		×
29						×
30		✓(12%)	✓			
33					✓	
35						×
37	✓ (92%)		✓			
39						×
40	✓ (78%)					
51	✓ (98%)					
56						×
57	✓ (94%)					
60				✓		×
62		✓(61%)				

Table 10 – Arrangements for raising the DHW temperature for Legionella control

It is possible that sites 04, 17, 28, 29, 35, 39, 56 & 60 are not being operated in line with current best practice guidance with regard to Legionella control in domestic hot water systems.

4.18.2 Influence of DHW temperature boost method on system performance

Figure 50 shows the plot of SPF_{H4} versus mean temperature lift, grouped by the method of DHW temperature boosting. T-Testing of each data sub-group compared to the total sample does not show any significant differences between the mean SPF_{H4} values for each method of boosting the temperature.

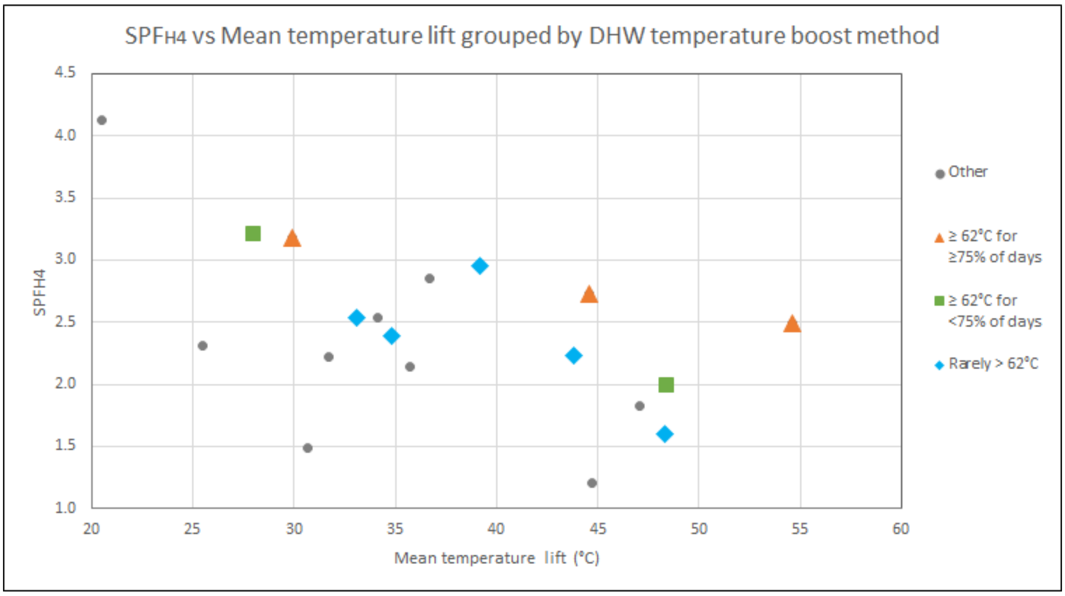

Figure 50 – SPFH4 versus mean temperature lift, grouped by DHW temperature boost method

4.18.3 Ultraviolet light for DHW disinfection

Another method of Legionella control is to use ultraviolet disinfection.

Ultraviolet light of an appropriate wavelength can kill legionellae and is an approved[32] method of treating potable water. However, unlike heat, it has no residual downstream effect, which means that organisms that survive the ultraviolet treatment can subsequently colonise downstream [9].

This method is used at site 62, where ultraviolet disinfection lamps are installed in the DHW draw-off pipes from each DHW cylinder. Note that the system also has high output temperatures (above 62 °C) to the DHW cylinders. It is possible that the high temperatures may not be needed, or that the disinfection schedule could be adjusted to allow the heat pumps to operate with lower output temperatures for at least part of the time.

4.19 Point-of-use water heaters

At some sites there is only a small demand for domestic hot water (see Table 9). For example, at site 05 (a public hall) the estimated heat provided to DHW was 5% (2000 kWh/year), and at site 57 (offices) only 3% (2662 kWh/year) was for DHW.

At these sites, it would be worth considering using point-of-use electric water heaters instead of using the heat pump system. This would allow the heat pump to be switched off during the summer months and should improve the performance of the overall installation by avoiding the need for high heat pump output temperatures.

4.19.1 Site 05

At site 05 (a public hall), there was no demand for space heating from 1/7/2015 – 13/9/2015 and from 11/5/2016 – 30/6/2016. During these periods the heat pump system was used to provide heat for DHW only, although not very efficiently. The measured data was as follows:

Site 05 : DHW-only operation during summer periods	
Number of days	**126**
Electricity used by the heat pump system	**674 kWh**
Heat output to the DHW cylinder	**570 kWh**
Effective SPFH4	**0.85**

Table 11 – Site 05: DHW-only operation

[32] Some ultraviolet disinfection products are approved by the Water Regulations Advisory Scheme (WRAS) www.wras.co.uk .

The effective SPF_{H4} of 0.85 for DHW-only operation indicates that there was no benefit[33] from using the heat pump during the summer to provide DHW. The SPF_{H4} value of less than 1.0 is due to the heat losses from the heat pump and pipework between it and the heat meter being a significant proportion of the total heat generated under low load conditions.

It would have been better to use the immersion heater in the DHW cylinder at times when there is no space heating requirement. This would reduce the annual electricity use by 104 kWh and would yield a small increase in the annual SPF_{H4} of the system from 3.18 to 3.21.

The mean daily heat to DHW during the summer periods was 4.5 kWh. It is possible that a large proportion of this heat is simply lost from the DHW cylinder and the distribution pipework. A reassessment of the actual hot water requirements in the premises would help determine whether point-of-use water heaters might offer a better solution.

However, it is questionable whether the capital expenditure on the high-temperature heat pump, DHW cylinder and other equipment was justified for providing a small amount of DHW. Point-of-use water heaters may have been a more cost-effective option, although it is not suggested that the existing system be changed.

4.19.2 Site 57

The situation at site 57 (offices) is less clear, as there was demand for space heating throughout the year. The total heat provided to DHW was just 2662 kWh during the 12 months from July 2015 to June 2016 (7.3 kWh/day).

Inspection of the recorded data for operating cycles when there was no output to space heating shows that a COP (coefficient of performance[34]) of approximately 2.3 was achieved for DHW-only operation. This is considered to be reasonably good performance.

As for site 05, it is questionable whether the capital expenditure on the high-temperature heat pump, DHW cylinder and other equipment was justified for providing a small amount of DHW. Point-of-use water heaters may have been a more cost-effective option, although it is again not suggested that the existing system be changed.

[33] Ignoring any financial benefit from RHI payments for heat recorded by the heat meter.

[34] COP has been used here to refer to instantaneous performance of the heat pump, rather than SPF which relates to performance over a period of time.

4.20 Buffer tanks

Most of the systems monitored incorporate buffer tanks in the space heating circuits. The usual purpose of a buffer tank is to increase the thermal inertia of the heating circuit to increase the run time of the heat pump and thereby reduce short cycling.

Different buffering arrangements were used:

4.20.1 4-pipe buffer tank

This arrangement decouples the primary (heat pump output) circuit from the secondary (heat emitter) circuit, so that the flow rate in each circuit can be set to suit the requirements of that circuit.

A characteristic of a 4-pipe buffer tank is that there is usually a loss of temperature between the flow into the tank from the primary circuit and the flow out of the tank to the secondary circuit. The size of the temperature loss depends on a number of factors, especially mixing within the tank – which is inevitable when there are different in and out flow rates. However, it is possible that the benefits of having optimal flow rates in each circuit may outweigh the disadvantage of the temperature loss. 18 of the systems monitored use 4-pipe buffer tanks.

4.20.2 2-pipe buffer tank

Eight systems incorporate a 2-pipe buffer tank in series with the heat emitter circuit. Two systems have the buffer tank in the return pipe; the others are in the flow pipe.

The loss of temperature through a 2-pipe buffer tank is usually very small or unmeasurable. However, the flow rate through the heat pump with this arrangement is constrained to be the same as that through the heat emitter circuit. This means that the temperature profile through either or both the heat pump condenser and the heat emitters will not be optimal.

4.20.3 3-pipe buffer tank

One system (site 57) has a 3-pipe buffer tank. This is essentially a 2-pipe tank in the flow to the heat emitter circuit, with a bypass from the bottom of the tank back to the return to the heat pump. See the case study for site 57 [6] for further information.

4.20.4 No buffer tank

One system (site 04) operates without a buffer tank.

The loss of temperature through the buffer tanks in the systems monitored varied from one system to another – up to 7 °C. The maximum values recorded on each system are presented in Table 5.

It is not clear from the data collected in this study how buffer tank temperature loss influences system performance. The topic of buffer tank design is a complex one and beyond the scope of this project. It is therefore not proposed to attempt to draw conclusions about the effects of the type of buffer tank or the measured temperature losses on system performance.

For further information about buffer tanks in heat pump systems, the reader is referred to previous research:

- An investigation of the interaction between hot water cylinders, buffer tanks and heat pumps [10] was carried out by Kiwa for DECC.
- A study of the design and sizing of buffer tanks and recommendations for good practice is presented in a French report on the design and sizing of buffer tanks [11], published as part of an environmental research programme in Grenelle.

5 Estimated CO_2 & Fuel Bill savings

It is important to consider whether heat pumps produce savings in greenhouse gas emissions and in fuel bills.

In a previous report on the preliminary results from the RHPP heat pump monitoring programme [12] it was considered appropriate to make the comparison with alternative technologies using two different assumptions about the heat delivered by heat pumps. The author of that report noted that in the EST heat pump field trial [13] it had been observed that indoor temperatures were, on average, 1 °C higher than in EST's condensing boiler field trial. Consideration of degree-days shows that an increase of 1 °C in indoor temperature requires an increase in heating energy of approximately 10%. The comparison of heat pumps with other technologies was therefore presented with two scenarios:

- heat pump heat delivery the same as for alternative technologies
- heat pump heat delivery 10% higher than for alternative technologies

The same approach is used here, as it is possible that the sites studied also have slightly higher indoor temperatures than they would have had previously or if they were using different heating technologies.

Table 12 shows the minimum SPF_{H4} values that must be achieved by heat pump systems in order to break even with oil-fired boilers (kerosene) or gas-fired boilers (natural gas) in terms of CO_2 emissions and running costs. The Carbon (CO_2 equivalent) intensity values are as published in the UK Government Conversion Factors for greenhouse gas [14]. The efficiency data for gas boilers is based on measurements of in-service operation [15]. The same system efficiency has been assumed for oil-fired boilers[35].

The monthly price for standard grade burning oil [16] was at a 10-year low in February 2016, but has been increasing again since then. Looking at the trend over the past 10 years, the average price of 33.8 p/litre during the monitoring period (July 2015 – June 2016) was exceptionally low and it seems somewhat unrealistic to make a comparison using that figure. A second cost comparison for oil is also shown, using an oil price of 48.8 p/litre which was the mean monthly oil price for the 10 years from December 2006 to November 2016.

July 2015 - June 2016				Scenario 1: Heat delivered by heat pump same as for alternative technology		Scenario 2: Heat delivered by heat pump 10% more than for alternative technology	
	Fuel cost (p/kWh)	Carbon intensity (gCO2/kWh)	System efficiency	Fuel bills reduction breakeven SPF	Carbon reduction breakeven SPF	Fuel bills reduction breakeven SPF	Carbon reduction breakeven SPF
Electricity (standard)	14.72	412					
Oil (price 33.8 p/litre)	5.51	247	81%	2.16	1.35	2.38	1.49
Oil (price 48.8 p/litre)	7.95	247	81%	1.50	1.35	1.65	1.49
Natural gas	4.34	184	81%	2.75	1.81	3.02	2.00

1. Gross calorific value of kerosene: 47 MJ/kg [http://www.kayelaby.npl.co.uk/chemistry/3_11/3_11_4.html]
2. Density of kerosene: 790 kg/m3 [http://www.engineeringtoolbox.com/fuels-densities-specific-volumes-d_166.html]

Table 12 – SPF_{H4} values that must be achieved by heat pumps to break even with alternative technologies

[35] Inspection of typical boiler efficiency data in the Building Energy Performance Assessment Product Characteristics Database [22] indicates that this is a reasonable assumption.

On the basis of scenario 1 and the measured SPF_{H4} results, the 18 monovalent[36] heat pump systems monitored compare to alternative technologies as follows:

- All 18 systems had CO_2 emissions lower than for oil-fired heating
- 16 systems had CO_2 emissions lower than for natural gas heating
- 1 system cost less to run than oil-fired heating (using an oil price of 33.8 p/litre)
- 8 systems cost less to run than oil-fired heating (using a higher oil price of 48.8 p/litre)
- 5 systems cost less to run than natural gas heating.

On the basis of scenario 2 and the measured SPF_{H4} results, the heat pump systems monitored compare to alternative technologies as follows:

- All 18 systems had CO_2 emissions lower than for oil-fired heating
- 14 systems had CO_2 emissions lower than for natural gas heating
- 1 system cost less to run than oil-fired heating (using an oil price of 33.8 p/litre)
- 5 systems cost less to run than oil-fired heating (using a higher oil price of 48.8 p/litre)
- 3 systems cost less to run than natural gas heating.

[36] Site 13, which has a bivalent system, has been excluded from the comparisons.

6 Findings and Conclusions

6.1 Findings

The main findings can be summarised as:

- The systems studied vary widely in application, design and complexity.
- 21 heat pumps have been monitored for 24 months and a further 7 for at least 12 months. With a combined installed capacity of 1601 kW_{TH}, this sample was equivalent to 10.5% of the total non-domestic RHI ground- and water-source heat pump capacity accredited as of 1st June 2015.
- Of the 19 heat pumps for which performance data can be reported, four (21%) demonstrated levels of performance below that required to be considered “renewable” ($SPF_{H2} \geq 2.5$) under the Renewable Energy Directive[37] and when wider system energy use was taken into account, the number operating with an $SPF_{H4} < 2.5$ increased to 11 (58%). Only six of the sample (32%) achieved an $SPF_{H2} > 3.0$ and three (16%) achieved an $SPF_{H4} > 3.0$.
- When the 18 monovalent (electric only) heat pump systems are compared to oil-fired heating systems, all would have lower CO_2 emissions but only eight of the sample would cost less to run[38].
- When the 18 monovalent (electric only) heat pump systems are compared to natural gas-fired heating systems, 16 have lower CO_2 emissions but only five of the sample cost less to run[39].
- The heat metering arrangements on eight installations, whilst likely to have been in line with RHI requirements at the time (the requirements were updated in 2014) were not considered by this study to be of sufficient standard for performance analysis. Some of the heat meter installations are understood by the author to have been improved subsequent to the end of the monitoring period.
- There was no significant difference in performance between heat pumps from different manufacturers.
- Heat pump systems using underfloor heating were not found to have significantly higher system performance than those using radiators – not even those with radiators that had been installed and sized for use with oil-fired boilers, but were not increased in size for use with heat pumps.
- The hours of operation of the heating system (i.e. times of heat demand – ranging from weekday office hours to 24/7 operation) were not shown to have a significant influence on system performance. However, longer-than necessary heat pump operating hours potentially caused energy wastage which is undesirable.
- Eight systems appear not to be operated in line with current best practice guidance with regard to Legionella control in domestic hot water systems.
- The mean temperature lift (heat pump output temperature minus source temperature at the heat pump inlet) during the monitoring period ranged from 21 °C to 55 °C. Systems with low temperature lift tended to have higher performance, and the system with the lowest temperature lift also had the highest SPF_{H2} and SPF_{H4} performance.

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[37] Heat pump installations accredited onto the RHI are all required to meet minimum quality standards. All of the monitored installations were accredited before May 2014, and hence were required to demonstrate that the heat pump units achieved a COP of at least 2.9. Since May 2014, newly accredited RHI installations have also been required to demonstrate a minimum design SPF of 2.5. As 16 of the monitored systems have a capacity below 45kW_{TH}, they will also have been required to achieve MCS certification standards. Further information on scheme eligibility and minimum standard requirements is available from Ofgem: https://www.ofgem.gov.uk/environmental-programmes/non-domestic-renewable-heat-incentive-rhi/eligibility-non-domestic-rhi .

[38] Based on CO_2 emission factors as of June 2016 [14] and the mean oil price for the period December 2006 to November 2016 [16].

[39] Based on CO_2 emission factors as of June 2016 [14] and the mean gas price for the period July 2015 to June 2016 [16].

- Source water pumped directly to the heat pump evaporator can potentially yield high performance because there is no intermediate heat exchanger and the temperature at the evaporator is maximised. However, such arrangements often require open-loop pumping which can use significantly more energy than closed-loop circulation, with consequent negative influence on performance. Care must also be taken in the design and operation of direct source systems to avoid problems of dirt and freezing of the water in the evaporator. Two of the systems monitored used source water pumped directly to the heat pump. One of these was the system with the highest performance.
- Systems that provide heat at a continuously high temperature to a combined space heating and domestic hot water system were found to have performance lower than most other systems[40].
- Some systems incorporated mixing valves that were used to reduce the temperature supplied to the heat emitters. As a consequence, the heat pumps on these systems were required to deliver continuously high temperatures that resulted in reduced system performance.
- The energy used by ancillary equipment (pumps and immersion heaters) varied from 8% to 55% of total electricity used by the heat pump system. The median value was 24%. Systems with lower energy use by ancillary equipment typically had higher performance.
- Various control issues were identified:
  - auxiliary heaters being used when apparently not needed
  - a smart controller intended for use with a boiler causing cycling of a heat pump
  - heat pumps being used during periods when the building was not occupied
  - circulating pumps running when not needed, or – more seriously – not running when needed
  - lack of temperature controls in individual rooms sometimes leading to wasted energy
  - heating and cooling occurring during the same day causing waste of energy
  - on systems with multiple heat pumps, all heat pumps were often started at the same time, with consequent short run times, when starting one at a time would have been more efficient. Essentially, this issue was the control of the system in an on/off manner when the multiple heat pumps would have permitted step-wise capacity modulation.
- Weather compensation (reduction of the heat pump output temperature to space heating when the outdoor temperature rises) varied widely from one system to another. Weather compensation was not shown to influence system performance – although this does not suggest that there is no effect: only that there was no statistically significant evidence from this sample.
- The mean outdoor air temperature at the sites monitored varied from 8.3 °C to 12.7 °C, but was not found to have a significant influence on system performance.
- Provision of small quantities of domestic hot water by a heat pump, as identified on two sites, may not be justified. Point-of-use water heaters could be more efficient for some applications.
- Short cycling of the heat pump (known to be a cause of reduced performance and excessive equipment wear [3]) was found to occur on only two of the systems monitored, and on one of these only during the first two hours following commencement of heating demand each day. The extent to which this short cycling influenced performance is unknown.
- Systems used to heat buildings older than 50 years (and with poor energy performance compared to modern or refurbished buildings) were not found to have lower performance.
- System proprietors all stated at the start of the monitoring project that they were satisfied or very satisfied with their heat pump installation. One proprietor of a system that uses a common high-temperature output to provide both space heating and domestic hot water has subsequently commented

[40] The apparently lower performance of the small sample (three) of these systems was not proven with statistical significance.

that he would consider a different system design if he had the opportunity of starting again. A separate research programme [4] has surveyed a sample of non-domestic RHI applicants.

The observed sample performance should not be taken as representative of the Non-Domestic RHI ground- and water-source heat pump population (or the wider heat pump population) due to the sampling method and site selection process employed. The findings present a range of seasonal performance factors found on a sample of Non-Domestic RHI ground- and water-source heat pumps and outlines issues which may be affecting their performance.

6.2 Conclusions

The project shows that it is possible to design, install and operate a heat pump system to provide a high seasonal performance factor, but that this high level of performance is not being realised on some installations.

From the available data obtained from the small sample of systems, it is difficult to draw general conclusions about the performance of the wider population of non-domestic ground-source and water-source heat pump installations. This is a consequence mainly of the wide variation in the application and design of the systems monitored. A larger sample would have been useful to yield better statistical significance of the analyses performed, but at the time the project was started, the available sample represented 21% of the total non-domestic RHI heat pump population[41].

Numerous factors influence heat pump performance. There is not one overriding factor that needs to be addressed, but more careful design, installation, commissioning and operation are all required to ensure a high-performance system.

It is always important to pay particular attention to

- maximising the source temperature at the heat pump evaporator inlet
- minimising the temperature at the heat pump condenser outlet
- minimising the energy used by ancillary equipment
- avoiding exceptional heat losses (for example from underground heat distribution pipes)
- the use and correct configuration of controls that are appropriate for heat pumps.

Each application has its own particular characteristics and each individual system therefore needs to be designed and optimised to suit its application.

6.3 General Observations

6.3.1 System design

- Good conceptual system design, based on sound thermodynamic principles, is essential for the achievement of good performance. It is important to ensure that all of the key factors that influence performance are addressed: maximum possible source temperature, minimum possible sink temperature, minimum possible energy use by ancillary equipment.
- Using a heat pump to provide domestic hot water may not always be the most efficient means of doing so. The specific requirements of the application should be carefully assessed and alternative means of providing domestic hot water should be considered.
- Control systems need to be designed to suit the special requirements of heat pump systems. They also need to be implemented, configured, maintained and used correctly. The facility for users to override automatic controls (e.g. immersion heater controls) should be removed as far as practicable.

[41] As of December 2013, there were 100 accredited Non-Domestic RHI ground-source and water-source heat pump installations with a total thermal capacity of 4900 kWTH. [23]

6.3.2 Legionella control

Legionella control in domestic hot water is problematical for heat pump systems.

It was observed that on some installations the control of Legionella in the DHW system may not have been properly addressed.

17 of the systems monitored provide domestic hot water. Ultraviolet light sterilisation was used on one site. On the other 16 sites, the technique used to control Legionella was to raise the temperature in the domestic hot water cylinder.

It was found that at least eight of these systems were probably not being operated in accordance with the requirements for Legionella control, as the temperature in the DHW cylinders would rarely, if ever, have reached 60 °C. If there is found to be a requirement to raise the temperature of the heat pump output to DHW on these systems, it can be expected that the overall performance of the systems will be reduced.

6.3.3 Heat metering

Accurate metering of electricity and heat is essential for performance monitoring. It was decided at the outset of this project that, to avoid intervention on completed and accredited installations, the heat meters already installed for RHI would be used for monitoring. This presented challenges on some sites.

- The heat meters were not always installed in the most appropriate position for measurement of heat pump performance and overall system performance. For example, on site 10, the heat pump plant room is outside the building being heated. The heat meter used for measuring the heat delivered to space heating was installed inside the building, some distance from the heat pump – so as not to include heat losses from the heat distribution pipes between the plant room and the heated space in the heat measurement used for RHI. On this system it was therefore not possible to directly measure the heat output of the heat pump and the SPF_{H2} could only be determined by estimating the heat losses from the heat distribution pipes and from the buffer tank.
- It was not feasible to verify the accuracy of the heat metering. While the flow and return temperatures can be verified to a reasonable accuracy, using the sensors installed for monitoring purposes, and the temperature sensors used by the heat meters could have their calibration checked on site, there is no readily available means of verifying the heat meter flow rate measurements without opening and probably extending the pipe to install a second flowmeter – something that might disturb the functioning of the heat meter and which would in most cases be an unacceptable intervention on a working, commissioned installation. The possibility of using clamp-on ultrasonic flow meters for verification was explored, but it was concluded that the uncertainty of measurement of such clamp-on meters is too great for this to be a useful technique. It is understood that the conventional method of checking the calibration of a heat meter is to remove it from the installation and replace it with a calibrated unit of the same type. In hindsight, this might have been a very useful thing to do.
- Satisfactory automatic reading of the heat meters was not always possible.
    - The best solution was to use the M-Bus interface if available, and to have the heat meter mains-powered to permit taking readings every minute.
    - In cases where the meter has an M-Bus interface and is battery-powered, readings can only be taken much less frequently (e.g. hourly) to avoid discharging the battery.
    - Where the heat meter has no M-Bus interface, a pulse logger can be used to record the pulse output. However, sometimes the heat meter will have been configured with a high pulse weight (e.g. one pulse per 100 kWh).
    - With either of the latter two situations, the granularity of the data recorded from heat meters may not be sufficient for analysis of system behaviour. However, it is often possible to use temperature and electrical data to estimate the heat output sufficiently accurately for behavioural analysis. This technique was used for analysis of a number of systems.

- 10 systems were equipped with mains-powered heat meters with M-Bus interfaces, allowing measurement of the heat pump output to be taken every minute.
- The other 18 systems used heat meters with pulse interfaces to measure the heat pump output. Of these, six had a relatively high pulse weight that provided fewer than 12 pulses per hour at the maximum heat output of the system, with one providing a pulse only every 62 minutes at full load. On these systems, the temperature and electrical data was used for system behavioural analysis.

- The performance measurements of systems with heat meters that were found to be inadequately installed (e.g. incorrectly mounted temperature sensors) have not been presented in this report, as it is known from research on heat meter accuracy undertaken by BRE [2] that the measurement uncertainty for such meters can be very high. This information about errors with certain metering configurations only became available after monitoring had been under way on the phase 1 sites for around 12 months. Had the extent of the heat metering issues been known at the outset of this project it seems highly probable that a rather different approach to heat metering would have been taken. However, hindsight is a powerful tool that can never be used, but lessons can certainly be learned for future similar studies.
- One heat meter (site 10) was found to be configured for use with glycol, although none was present in the heating circuit. This caused the heat measurements from the meter to be low by about 5%. The readings from this meter were corrected to allow for this incorrect configuration, as described in the case study [6].

6.4 Recommendations

- Avoid using controllers designed for use with boilers as they may be unsuited to the control of heat pump systems.
- Consider using point-of-use water heaters instead of the heat pump for providing domestic hot water, particularly where the demand is low.
- Avoid using mixing valves (e.g. to reduce the temperature of the flow to underfloor pipes) where their use would be likely to cause unnecessarily high heat pump output temperatures.
- Consider installing a monitoring & diagnostic system to provide continued monitoring of system performance and behaviour after commissioning and to provide early warning of any problems that may arise. Such a system would quickly identify serious faults such as circulating pumps not running or immersion heaters being used unnecessarily.

6.5 Suggestions for further work

- It may be useful to monitor additional non-domestic, ground-source and water-source heat pump installations, to provide data from a larger sample that could now[42] be selected to be more representative of the wider heat pump population. This could be expected to improve understanding of the factors that influence performance by improving the statistical significance of the results and may identify other factors not explored by this study.
- At the time of writing, the monitoring equipment used for this project is still in place. The data communication links are no longer active, but could easily be reactivated. It is understood that, at the request of Ofgem, heat metering installations on affected sites are being modified to bring the measurement uncertainty within acceptable limits. It would be relatively straightforward to continue monitoring these systems and thereby collect some very useful, reliable performance data that would fill some of the gaps in the set of results presented in this report.

[42] Now that there is a larger number of RHI installations.

- Further work on the behaviour and influence of buffer tanks would be useful. It has been found that there were drops in temperature of up to 7 °C through the buffer tanks (especially 4-pipe buffer tanks), but the effect of these temperature drops on performance is not known. A thorough review of previous research and possibly new experimental and simulation work would lead to better understanding of the effects of buffering in heat pump systems and provide guidelines or design tools that would be useful for systems designers.
- Additional research on the requirements and methods used for Legionella control in heat pump systems would be very useful. This should include a thorough assessment of the use of ultraviolet light instead of temperature as a means of sterilisation. The research could be coupled with further analysis of the benefits or otherwise of using heat pumps to provide DHW in non-domestic installations, especially those where DHW demand may be a small proportion of total heat demand. The objective would be to provide improved design tools and guidelines for system designers.
- Further system modelling, to build on work already done by others, to improve understanding of the behaviour of complete systems of different types and sub-systems (ground collectors, pumps, weather compensation, heat emitters, control strategies, variable-speed drives). Much work has undoubtedly already been done in this field, so a thorough review of previous research and available modelling tools should be carried out before any new work is started.

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Appendix A System Boundaries for Performance Calculation

The seasonal performance of heat pumps has been defined by the "SEPEMO-Build" project (SEasonal PErformance factor and MOnitoring for heat pumps in the building sector – http://sepemo.ehpa.org/.)

The SEPEMO Final Report [17] contains definitions for a number of seasonal performance factors (SPFs). The following definitions of are for hydronic heat pump heating systems:

SEPEMO system boundaries

SPF_{H1}:

This system contains only the heat pump unit. SPF_{H1} evaluate the performance of the refrigeration cycle. The system boundaries are similar to COP defined in EN 14511, except that the standard takes, in addition, a small part of the pump consumption to overcome head losses, and most part of fan consumption.

SPF_{H2}:

This system contains of the heat pump unit and the equipment to make the source energy available for the heat pump. SPF_{H2} evaluate the performance of the HP operation, and this level of system boundary responds to $SCOP_{NET}$ in prEN 14825 and the RES-Directive requirements[1].

Note: COP in EN 14511 and $SCOP_{NET}$ in prEN 14825 are more or less between SPF_{H1} and SPF_{H2} (see table 1 at the end of the document)

SPF_{H3}:

This system contains of the heat pump unit, the equipment to make the source energy available and the back-up heater. SPF_{H3} represents the heat pump system and thereby it can be used for comparison to conventional heating systems (e.g. oil, gas,…). This system boundary is similar to the SPF in VDI 4650-1, EN 15316-4-2 and the $SCOP_{ON}$ in prEN 14825. For monovalent heat pump systems SPF_{H3} and SPF_{H2} are identical.

SPF_{H4}:

This system contains of the heat pump unit, the equipment to make the source energy available, the back-up heater and all auxiliary drives including the auxiliary of the heat sink system. SPF_{H4} represents the heat pump heating system including all auxiliary drives which are installed in the heating system.

Figure A 1 (also from the SEPEMO Final Report [17]) illustrates the system boundaries:

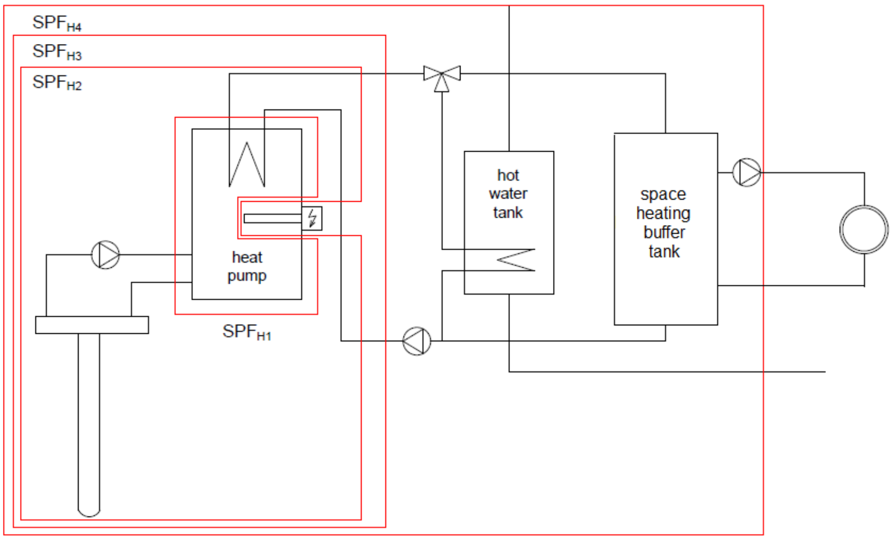

Figure A 1 – SEPEMO system boundaries

Heat loss from DHW cylinders

It can be seen in the SEPEMO diagram (Figure A 1) that the space heating buffer tank and the hot water tank (DHW cylinder) are within the SPFH4 boundary. This implies that heat losses from these tanks should be subtracted from the system heat output.

There is, however, some confusion about whether the DHW cylinder should be inside or outside the SPFH4 boundary. After consultation with industry specialists, it was agreed to exclude the heat loss from space heating buffer tanks but not from DHW cylinders. The SPFH4 boundary adopted for this project is therefore as shown in Figure A 2.

Note that optional immersion heaters in the buffer tank and DHW cylinder have been shown. These are within the SPFH3 boundary and therefore also within the SPFH4 boundary.

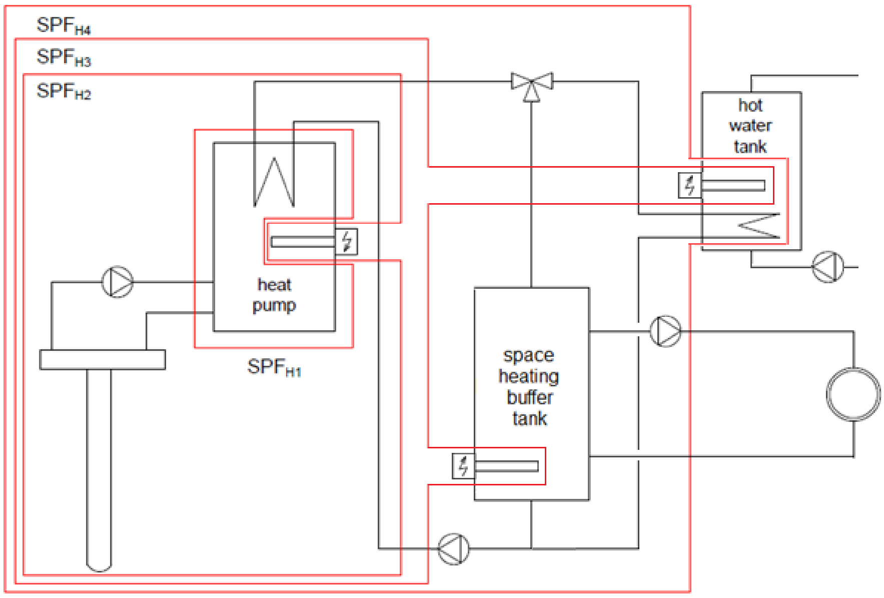

Figure A 2 – Modified SPFH4 boundary used in this project

The effect of not deducting the heat losses from the DHW tanks is very small. For the systems monitored, it is estimated that the reported SPFH4, for systems providing DHW, is between 0.24% and 1.24% (median 0.57%) higher than it would have been had the DHW tank heat losses been deducted from the total heat output.

Bivalent systems

The heat pumps monitored in this project include several that are designed for bivalent operation.

- Heat pump + oil-fired boiler (site 13)
- Heat pump + solar thermal collector (sites 17, 40 & 60)

A number of other systems (sites 02, 14, 28, 34, 51, 61 & 62) incorporate oil-fired, gas-fired or electric boilers intended for backup duty only. These have not been treated as bivalent systems.

Bivalent with oil-fired boiler

The system at site 13 (a greenhouse) is designed to use the oil-fired boiler together with the heat pumps to meet the demand for high output temperatures during cold weather.

This system has been analysed as a bivalent system, whereby the fuel supply to and heat output of the boiler are included in the calculation of SPF_{H4}:

$$SPF_{H4} = \frac{\text{[Heat output from heat pump system] + [heat output from boiler]}}{\text{[Electricity used by heat pump system] + [Energy content of fuel used by the boiler]}}$$

Bivalent with solar thermal collector

The systems at sites 17, 40 & 60 incorporate solar thermal collectors that provide heat to domestic hot water.

The SEPEMO project [17] recommends that for this type of system the heat from the solar collector be included in the numerator of the SPF_{H4} calculation and the electricity used by the solar circulating pump in the denominator. Inclusion of these values in the SPF_{H4} calculation is likely to yield very high SPF_{H4} values that are not very meaningful for comparison with systems without solar collectors – because the ratio of heat to electricity in a solar thermal system can be very high and does not have the same meaning as the SPF of a heat pump.

The SPF_{H4} results for bivalent solar systems in this project do not include the heat output of or electricity used by the solar thermal collectors.

Appendix B Monitoring Requirements and Equipment Used

Monitoring requirements

The monitoring requirements are described in Chapter 3 of the Interim Report [5].

It should be noted that, since the Interim Report was written, the definition of the seasonal performance factor SPF_{H4} has been revised to place the DHW tank outside the SPF_{H4} boundary.

See Appendix A for definitions of the performance factors presented in this report and for the estimated effect of the change in the definition of SPF_{H4}.

Monitoring equipment used

The monitoring equipment used for the project is described in Appendix B of the Interim Report [5].

Appendix C Estimation of Uncertainty of Measurement of SPF

SPF is the ratio of the thermal energy output from the system to the electrical energy input:

SPF = Thermal energy / Electrical energy

= (Heat meter) / Σ (Electricity meters)

= $E_H / \Sigma (E_{E1} \ldots E_{En})$

Where

E_H is the thermal energy

E_E is the electrical energy

The standard uncertainty of measurement of SPF is the combination of the standard uncertainties of measurement of the electricity meters and of the heat meter(s).

Note: "Standard uncertainty" corresponds to a margin whose size can be thought of as 'plus or minus one standard deviation'. "Expanded uncertainty" is taken to be the standard uncertainty multiplied by a coverage factor k (=2) to give 95% confidence limits.

The combined standard uncertainty of electricity metering u_E can be determined by summation in quadrature of the standard uncertainties of the individual electricity meters:

$$u_E = \sqrt{u_1^2 + u_2^2 + \ldots u_n^2}$$

Where

$u_1, u_2 \ldots u_n$ are the standard uncertainties of measurement of the individual meters.

The heat metering standard uncertainty u_H can be determined in a similar manner, although in most cases there is only one heat meter.

The standard uncertainty u_{SPF} of measurement of the SPF can then be determined from:

$$\frac{u_{SPF}}{SPF} = \sqrt{\left(\frac{u_E}{E_E}\right)^2 + \left(\frac{u_H}{E_H}\right)^2}$$

The same types of electricity meter are used on all sites. The heat metering arrangements vary from one site to another.

Electricity metering uncertainty

The electricity monitors used in the project are specified as Class 1 or better. They typically operate at less than 20% of their full load capability. It will therefore be assumed that the maximum permissible error (MPE) of the electricity meters is ±1.5% of the reading.

There are usually 3 or 4 electricity meters used for monitoring an installation. The uncertainty of electricity measurement can be calculated as in the following example for site 01.

The standard uncertainties of each meter are summed in quadrature to determine the overall standard uncertainty of ±0.041 kW, corresponding to a relative standard uncertainty of ±0.7%.

Uncertainty assessment **Site 01**

Coverage factor:	k	Coverage
	2.0	95%

Electricity meters IEC class 1

Meter	Type	Full scale kW	Average kW	% FSD	FSD error	MPE %	Assumed max error	Standard uncertainty	Relative standard uncertainty %	Expanded uncertainty %
E01	ZEM-61-120	82.8	5.50	6.6%	1.0%	1.5%	0.083	0.041		
E02	ZEM-30-10i	2.3	0.21	9.1%	1.0%	1.5%	0.003	0.002		
E03	ZEM-30-10i	2.3	0.15	6.5%	1.0%	1.5%	0.002	0.001		
Total			**5.86**					**0.041**	**0.7%**	**1.4%**

Table 13 – Example calculation of electricity metering uncertainty

As most sites have quite similar electricity metering arrangements, it will be assumed that a relative standard uncertainty of electricity metering of ±0.7% can be used for all systems.

Heat metering uncertainty

Some of the heat meters installed on the systems monitored used strapped-on temperature sensors. This arrangement can lead to very large measurement error, as determined by previous research on heat meter accuracy testing [2]. While performance figures for these systems were reported with caveats in the Interim Report, it has subsequently been decided not to include their performance results in this report – because the uncertainties are so large.

All of the heat meters used on systems with results presented here use temperature sensors mounted inside the pipes, and either ultrasonic or vortex flow meters. The previous research on heat metering accuracy [2] reported the overall (expanded) uncertainty of heat metering for these types of meter to be between -5.9% and +2.8%, with a 95% confidence interval. These values have been used to determine the uncertainties of measurement used in this report.

The uncertainty of measurement for SPF is calculated as follows:

$$\frac{U_{SPF}}{SPF} = \sqrt{\left(\frac{U_E}{E_E}\right)^2 + \left(\frac{U_H}{E_H}\right)^2}$$

Relative standard uncertainty of electricity metering = $\frac{u_E}{E_E}$ ±0.7%

Relative standard uncertainty of heat metering = - $\frac{u_H}{E_H}$ 3.0% | +1.4%

The expanded relative uncertainty of measurement of SPF for systems using heat metering with ultrasonic or vortex flow metering and temperature sensors in the pipes is: -6.0% | +2.8%
(95% confidence interval).

Appendix D Summary of Heat Pump Installations

Table 14 contains summary information about the heat pump installations that were monitored during the period covered by this report.

Site ID	Monitoring start date	Type	Building type	Capacity kWTH	No. of heat pumps	Heat source			Heat emitter	DHW	DHW method	DHW cylinders	Auxiliary heat	Weather compensation	Mean outdoor temperature (7/15-6/16)	Mean source temp at heat pump inlet	Mean heat pump output temp to SH or SH+DHW	Mean daily max output temp to DHW
						Source	Open / closed loop	Direct to evap / indirect										
01	10/07/2014	WSHP (1)	Offices	26	1	Ground water from borehole	Open loop	Direct	Underfloor heating	No			None	No	9.4	11.1	31.8	N/A
02	27/06/2014	GSHP	Large house	93	1	Horizontal ground loops: 12 x 200 m	Closed loop	Indirect	Radiators	No			Oil-fired boiler	Yes	8.3	6.0	42.6	N/A
04	23/06/2014	GSHP	Large house	57	2	Horizontal ground loops	Closed loop	Indirect	Radiators	Yes	Alternate DHW/SH (using one of two heat pumps)	2 x 300 litre	4 x 3 kW immersion heaters: controlled manually	No	8.4	8.0	44.4	56.4
05	09/06/2014	GSHP	Public hall	21.4	1	Horizontal ground loops: 6 x 200 m	Closed loop	Indirect	Radiators	Yes	Alternate DHW/SH	1 x 300 litre	Immersion heater in buffer tank (only used in emergency); immersion heater in DHW cylinder	Yes	11.0	7.3	37.6	62.3
07	26/03/2015	WSHP	Refectory & offices	96	1	Water from tarn	Open loop	Direct	Underfloor heating	No			LPG-fired boiler	Yes	9.7	10.1	40.5	N/A
10	09/06/2014	GSHP	Offices	22	1	Horizontal ground loops: 8 x 100 m	Closed loop	Indirect	Radiators	No			None	No	10.5	3.9	50.6	N/A
13	27/05/2014	GSHP	Agricultural	144	3	Horizontal ground loops: 4000 m	Closed loop	Indirect	Pipes at high and low level	No			Oil-fired boiler	No	10.9	4.4	48.3	N/A

						Heat source											
Site ID	Monitoring start date	Type	Building type	Capacity kWTH	No. of heat pumps	Source	Open / closed loop	Direct to evap / indirect	Heat emitter	DHW	DHW method	DHW cylinders	Auxiliary heat	Weather compensation	Mean outdoor temperature (7/15-6/16)	Mean source temp at heat pump inlet	Mean heat pump output temp to SH or SH+DHW	Mean daily max output temp to DHW
14	09/07/2014	WSHP (1)	Healthcare clinic	60	2	Ground water from 2 x vertical boreholes	Open loop	Direct	Underfloor heating	No			21.6 kW electric boiler; immersion heater in buffer tank (backup only)	Yes	9.5	5.1	41.6	N/A
17	08/07/2014	GSHP	Public hall	30	1	Vertical boreholes: 1 x 65 m, 6 x 75 m	Closed loop	Indirect	Underfloor heating (part of building); radiators	Yes (top-up of solar heat)	Simultaneous DHW & SH (using one of two compressors in the heat pump)	1 x 450 litre	3 kW immersion heater in DHW cylinder	Yes	11.0	4.4	37.2	55.0
18	12/06/2014	GSHP	Apartment block	79.2	2	Vertical boreholes: 12 x 100 m	Closed loop	Indirect	Underfloor heating	Yes	Alternate DHW/SH (using one of two heat pumps)	3 x 1000 litre	3 x 9 kW immersion heaters in DHW cylinders	No	9.5	3.7	53.7	63.1
27	26/06/2014	GSHP	Accommodation building	54	1	Vertical boreholes: 10 x 150 m	Closed loop	Indirect	Underfloor heating	No			None	Yes	9.7	3.1	37.6	N/A
28	11/07/2014	GSHP	Hospitality	70.8	2	Vertical boreholes: 12 x 125 m	Closed loop	Indirect	Radiators	Yes	Alternate DHW/SH (using one of two heat pumps)		4 x 6 kW imm htrs in DHW cylinders; 7.5 kW imm htr in buffer tank; oil-fired boiler for back-up.	Yes	9.1	3.0	47.3	49.7
29	06/06/2014	WSHP	Large house	126	1	Coils in river	Closed loop	Indirect	Radiators	Yes	Simultaneous DHW & SH (via a common output)		9 kW immersion heater in buffer tank + 9 kW immersion heater in DHW cylinder	Yes	11.2*	4.2	41.3	60.3
30	09/07/2014	GSHP	Public hall	14	1	Horizontal ground loops	Closed loop	Indirect	Underfloor heating	Yes	Alternate DHW/SH	1 x 100 litre	9 kW immersion heater in heat pump	Yes	10.4	6.3	35.5	59.7
33	09/07/2014	GSHP	Healthcare clinic	10.3	1	Horizontal ground loops: 500 m	Closed loop	Indirect	Underfloor heating	Yes	Simultaneous DHW & SH (DHW provided by a desuperheater)	None	4 kW immersion heater in heat pump	Yes	10.6	8.7	28.2	49.0
34	14/07/2014	GSHP	Healthcare clinic	64	1	Vertical boreholes	Closed loop	Indirect	Underfloor heating	No			Gas boilers.	Yes	11.9	9.1	38.2	N/A

						Heat source									Mean outdoor temperature (7/15-6/16)	Mean source temp at heat pump inlet	Mean heat pump output temp to SH or SH+DHW	
Site ID	Monitoring start date	Type	Building type	Capacity kWTH	No. of heat pumps	Source	Open / closed loop	Direct to evap / indirect	Heat emitter	DHW	DHW method	DHW cylinders	Auxiliary heat	Weather compensation				Mean daily max output temp to DHW
35	15/07/2014	GSHP	Dwelling houses	19.8	2	Vertical boreholes: 5 x 90 - 140 m	Closed loop	Indirect	Underfloor heating (ground floor); radiators (first floor).	Yes	Simultaneous DHW & SH (via a common output)	3	3kW immersion heat in DHW cylinder in each house	No	12.7	N/a	47.6	55.9
37	29/05/2014	GSHP	Public hall	17	1	Horizontal ground loop: 880 m	Closed loop	Indirect	Underfloor (ground floor); radiators (first floor)	Yes	Alternate DHW/SH	1 x 500 litre	7 kW immersion heater in heat pump	Yes	9.7	11.1	49.5	68.3
39	25/06/2014	GSHP	Dwelling houses and offices	22.9	1	Horizontal ground loops: 3 x 400 m.	Closed loop	Indirect	Radiators	Yes	Alternate DHW/SH	1 x 500 litre	9 kW immersion heater in DHW cylinder	Yes	9.6	5.7	46.8	57.5
40	11/07/2014	GSHP	Rental apartments	31	1	Horizontal ground loops: 2.2 km	Closed loop	Indirect	Underfloor heating	Yes	Simultaneous DHW & SH (via a common output)	8	2 x immersion heaters in buffer tank	No	9.5*	8.4	54.7	62.1
48	10/07/2014	GSHP / ASHP	Care home	14	1	Energy Fence: 1/3 buried in ground, 2/3 in air	Closed loop	Indirect	Underfloor heating on ground floor; radiators on first floor	No			No	Yes	10.5	N/A	N/A	N/A
51	03/07/2014	GSHP	Recreational building	38.3	1	Vertical boreholes: 10?	Closed loop	Indirect	Radiators	Yes	Simultaneous DHW & SH (via a common output)	1 x 500 litre	Gas boiler; immersion heater in DHW cylinder	No	9.7*	3.8	57.8	63.1
53	19/03/2015	WSHP	Offices & warehouse	30	1	River water	Open loop	Indirect	Underfloor heating + fan-coil units in warehouse	No			Immersion heater in flow pipe	Yes	10.7	6.3	37.5	N/A
56	21/03/2015	GSHP	Retail shop	33	1	Horizontal ground loops: 1200 m	Closed loop	Indirect	Underfloor heating	Yes	Simultaneous DHW & SH (via a common output)	1 x 300 litre	2 x 6kW imm htrs in buffer tank; 1 x 3kW imm htr in DHW cyl	No	10.5	8.7	48.4	59.6
57	21/03/2015	GSHP	Detached house used as offices	40	1	Horizontal ground loops: 6 x 250 m	Closed loop	Indirect	Radiators	Yes	Alternate DHW/SH	1 x 300 litre	None	Yes	10.5	3.1	47.6	62.9

						Heat source												
Site ID	Monitoring start date	Type	Building type	Capacity kWTH	No. of heat pumps	Source	Open / closed loop	Direct to evap / indirect	Heat emitter	DHW	DHW method	DHW cylinders	Auxiliary heat	Weather compensation	Mean outdoor temperature (7/15-6/16)	Mean source temp at heat pump inlet	Mean heat pump output temp to SH or SH+DHW	Mean daily max output temp to DHW
60	25/03/2015	GSHP	Public hall with a cafe	40	1	Vertical boreholes: 8 x 100 m	Closed loop	Indirect	Underfloor heating	Yes	Alternate DHW/SH	1 x 400 litre	9 kW immersion heater in DHW cylinder	Yes	9.3	8.7	44.8	57.8
61	19/03/2015	GSHP	Residential care facility	80	1	Vertical boreholes: 15 x 100 m	Closed loop	Indirect	Underfloor heating	No			Gas-fired boiler (backup only)	Yes	10.4	5.4	43.1	N/A
62	28/03/2015	WSHP	Large house & outbuilding	268	4	Open water	Closed loop	Indirect	Radiators	Yes	Simultaneous DHW & SH (via a common output)	1 x 500 litre 1 x 300 litre	Imm htrs in DHW cylinders LPG-fired boiler (backup duty)	Yes	11.1	7.6	56.4	59.6
															* Estimate			

Table 14 – Summary of heat pump installations

<u>Notes</u>

(1) This system is classified in the RHI database as a water-source heat pump. However, the heat source is groundwater from a borehole, so it could also be considered as a ground-source heat pump. MIS 3005 [1] provides the following description:

"Heat pumps may utilise different heat sources:
** Ground Source, where heat energy is extracted from the ground (e.g. from boreholes, horizontal trenches or aquifers)*
** Water Source, in which heat energy is extracted from water (e.g. lakes, ponds or rivers)*
** Air Source, where heat energy is directly extracted from ambient air. This includes solar assisted heat pumps."*