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Analysis of Heat Pump data from the Renewable Heat Premium Payment Scheme (RHPP) to the Department of Business, Energy and Industrial Strategy: Compliance with MCS Installation Standards.

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Nomenclature

PERFORMANCE EFFICIENCY NOMENCLATURE

COP Heat pump (HP) coefficient of performance

SPF_{Hn} HP seasonal performance factor for heating at SEPEMO boundary Hn

MONITORED VARIABLES

Eb	Electricity for whole system boost only
Edhw	Electricity for domestic hot water (typically an immersion heater)
Ehp	Electricity for the heat pump unit (may include a booster heater and circulation pump)
Esp	Electricity for boost to space heating only
Fhp	Flow rate of water from heat pump (may be space heating only)
Fhw	Flow rate of water to DHW cylinder (if separately monitored)
Hhp	Heat from heat pump (may be space heating only)
Hhw	Heat to DHW cylinder (if separately monitored)
Tco	Temperature of water leaving the condenser
Tin	For ASHP: Temperature of refrigerant leaving the evaporator For GSHP: Temperature of ground loop water into the heat pump
Tsf	Flow temperature of water to space heating
Twf	Flow temperature of water to cylinder

(Note that external temperature, Tex, was not measured directly. Data from a publicly available database were used in the analysis.)

RHPP ENERGY AND POWER UNITS

Energy	J	Joule	SI unit of energy
Energy	kWh	3.6 MJ	Customary unit of energy for residential energy use
Energy	MWh, GWh	3.6 GJ, 3.6 TJ	
Power	W	Watt, J/s	SI unit of power and heat flow
Power	Wh/2 minutes	30 W	Base unit of energy for monitored data in RHPP trial, limit of resolution of power – note that power and heat have been recorded at 2 minute intervals
Power	kWh/year	3.6 MJ/year	Customary unit for rate of residential energy use. 1kWh/yr ≈ 0.11416 W
Power	kW	1000 W	Typical unit for measurement of heating system ratings

KEY ACRONYMS AND ABBREVIATIONS

BEIS	Department of Business, Energy and Industrial Strategy
DECC	Department of Energy and Climate Change
EST	Energy Saving Trust
Preliminary Assessment	Preliminary assessment of the RHPP data performed by DECC (Wickins, 2014)
RAPID-HPC	Research and Analysis on Performance and Installation Data – Heat Pump Consortium
RHPP	Renewable Heat Premium Payment Scheme
MCS	Microgeneration Certification Scheme - a nationally recognised quality assurance scheme. MCS certifies microgeneration technologies used to produce electricity and heat from renewable sources.

MIS	Microgeneration Installation Standards. MIS 3005 sets out requirements for MCS contractors undertaking the supply, design, installation, set to work, commissioning and handover of microgeneration heat pump systems.
SEPEMO	SEasonal PErformance factor and MOnitoring
DHW	Domestic Hot Water
DHDG	Domestic Heating Design Guide
HEG	Heat Emitter Guide
Likely SPF	HEG values of SPF based on heat pump type and space heating flow temperature

Context

The RHPP policy provided subsidies for private householders, Registered Social Landlords and communities to install renewable heat measures in residential properties. Eligible measures included air and ground-source heat pumps, biomass boilers and solar thermal panels.

Around 14,000 heat pumps were installed via this scheme. BEIS funded a detailed monitoring campaign, which covered 700 heat pumps (around 5% of the total). The aim of this monitoring campaign was to provide data to enable an assessment of the efficiencies of the heat pumps and to gain greater insight into their performance. The RHPP scheme was administered by the Energy Savings Trust (EST) who engaged the Buildings Research Establishment (BRE) to run the meter installation and data collection phases of the monitoring program. They collected data from 31 October 2013 to 31 March 2015.

RHPP heat pumps were installed between 2009 and 2014. Since the start of the RHPP Scheme, the installation requirements set by MCS standards and processes have been updated.

BEIS contracted RAPID-HPC to analyse this data. The data provided to RAPID-HPC included physical monitoring data, and metadata describing the features of the heat pump installations and the dwellings in which they were installed.

The work of RAPID-HPC consisted of cleaning the data, selection of sites and data for analysis, analysis, and the development of conclusions and interpretations. The monitoring data and contextual information provided to RAPID-HPC are imperfect and the analyses presented in this report should be considered with this in mind. Discussion of the data limitations is provided in the reports and is essential to the conclusions and interpretations presented. This report does not assess the degree to which the heat pumps assessed are representative of the general sample of domestic heat pumps in the UK. Therefore these results should not be assumed to be representative of any sample of heat pumps other than that described.

Acknowledgements

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Executive Summary

The Microgeneration Certification Scheme Installation Standard (MCS MIS) 3005 provides the 'requirements for contractors undertaking the supply, design, installation, set to work commissioning and handover' of microgeneration heat pump systems for compliance with the certification scheme.

The aim of this report was to use monitored data from sites enrolled in the Department of Energy & Climate Change's (DECC) Renewable Heat Premium Payment (RHPP) scheme to assess how well RHPP Trial installations reflect the design requirements of MCS MIS 3005. In July 2016, the Department of Energy & Climate Change was merged with the Department for Business, Innovation and Skills to create the Department for Business, Energy & Industrial Strategy (BEIS). The appellation BEIS is applied where appropriate to reflect that change.

The MCS MIS 3005 standard was changed several times during the period over which RHPP heat pumps were designed and installed. Some of these changes were significant. In principle, systems should have been designed to whichever version of the standard was mandatory at the time of quotation, not installation. The metadata supplied with the RHPP data does not include the MIS 3005 version used for the design and quotation dates are only provided for privately owned properties. It is therefore not possible to determine with certainty which version of the standards was applied by the designer in each case.

For these reasons, the assessment in this report cannot provide precise estimates of how many systems complied with the standards. However, we examine eight elements of the MIS 3005 standard, namely:

- Calculation of heat loss
- Heat pump sizing
- Radiator sizing
- Calculation of measured annual energy use and comparison with the installers' estimates and EPC calculations
- Sterilisation of domestic hot water
- Specification of flow temperature at design conditions.
- Weather compensation
- Actual measured SPF_{H2} for space heating compared to the SPF_{H2} predicted from the MCS Heat Emitter Guide (MCS 021).

Data taken between 1/11/2011 to 31/10/2015 was used in the analysis. A subset of the data for just one year (the concurrent dataset) between 1/11/2013 and 31/10/2014 was also used when appropriate[1].

Heat loss calculations

The estimate of heat loss affects both the annual energy estimate and the sizing of the heat pump.

The RHPP data do not allow an assessment of the reliability of the installers' estimate of heat loss, since design criteria such as areas and volumes, thermal transmittance values (U-values), allowances for ventilation, etc., were not provided in the metadata. However, the report demonstrates that calculations of heat loss are influenced by subjective assessment of U-values and ventilation rates, in particular for retrofit situations. For example, calculations for an end-terraced house showed a 24% increase in estimated heat loss when ventilation assumptions were changed from ventilation class C to ventilation class A, as defined in BS 12831. It should be noted that this issue is not specific to heat pumps.

Heat loss calculations may be carried out by hand, spreadsheet or by proprietary software. In order to support installers MCS introduced an approved heat loss calculator on 03 November 2015.

Heat pump sizing

Heat pump sizing for design conditions was found to be either poorly understood or expressed. At the time of the trials, installers predominantly assessed 'net capacity' as manufacturers' nominal capacity and not at the site specific design conditions. RAPID-HPC used the measured heat output from the RHPP sample to extrapolate to design conditions and compared this estimated power with the nominal capacity as quoted. Manufacturers' data was used to provide estimates of the difference between nominal capacity and capacity at design conditions for a range of heat pump types. Comparing two extrapolated figures is necessarily subject to error. However, a majority of heat pumps appear to be adequately sized when compared to peak measured load.

As mentioned, heat pump sizing is affected by the calculated heat loss, which is sensitive to assumptions in ventilation and U-values. The same issues apply for boiler sizing, but the cost implications of over-sizing boilers are lower.

Comparison of installers' estimate of annual heat demand with measured values

The installers' estimates of annual energy use are influenced primarily by five factors; weather, estimated heat loss and assumed SPF of the heat pump at the chosen flow temperature, proportion of space heating relative to water heating and the assumed SPF of water heating.

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[1] Further information on the process of selecting data for analysis is contained in a companion report, Investigating Variations in Performance of Heat Pumps Installed via the Renewable Heat Premium Payment (RHPP) Scheme.

A comparison of installers' estimates of annual heat demand with measured values indicates a relatively poor correlation. Calculating annual energy use by integrating the technical complexities of heat pump whole-system performance and occupant operational preference is a complex socio-technical challenge.

The comparison of kWh/year from Energy Performance Certificates (EPC) with trial data also showed a discrepancy, but the most significant source of disagreement may be prevailing, wide variations in commercial EPC ratings of dwellings (DECC, 2014b).

The poor agreement between measured and estimated energy use may be due to mild winters during the trial, or may suggest that calculation procedures were too complex.

Radiator Sizing

Radiator sizing analysis indicates that 'star rating oversize factors' as described in the Heat Emitter Guide (HEG) may be inadequately understood or ignored due to practical and aesthetic considerations of size and location.

Sterilisation

There is no clear understanding of the number of systems installed after compliance with MIS 3005 v3.1a was made compulsory from 01 March 2012. Metadata quotation dates are given only for private housing and installation/commissioning dates from MCS certificates are only broadly indicative of the period of design and installation since, for a domestic heating installation, the installer's quotation may precede or be followed by a full technical specification compliant with MIS 3005.

Compliance with protection from legionella exemplifies this uncertainty. Those systems quoted for after version 3.1a became mandatory should have included appropriate measures to ensure protection against legionella. For RHPP heat pumps capable of producing water at an appropriate temperature to achieve cylinder temperatures of 60C or above, sterilisation is unnecessary although immersion heaters may be present for back up purposes only.

Where cylinder storage temperatures are below 60°C, the installation should have incorporated regular sterilisation of the domestic hot water. Whilst cylinder hot water temperatures were not monitored, examination of the data from 220 metered immersion heaters in the sample indicates that between one quarter and one third of these exhibit immersion operation consistent with regular sterilisation; predominantly either weekly or daily, although other patterns also emerged.

Design Flow Temperatures

The analysis of maximum flow temperatures at minimum outdoor design temperature indicates a mean of between 40 and 45°C for both radiator and underfloor heating corresponding to 4 star operation from the Heat Emitter Guide (HEG). This would be expected to result in good performance.

However, a wide range of temperatures is observed; 17% of systems examined had design flow temperatures of 50-60ºC, indicating 2 or 1 star operation in the HEG, and 34% had design flow temperatures of <40ºC, indicating 5 or 6 star operation[2]. Note that regression calculations down to zero degrees centigrade have been necessary due to the low number of days at design outdoor temperature.

Weather Compensation

The same analysis indicates weather compensation was used for 64% of installations. Weather compensation is recommended by MCS MIS 3005 version 4.0; however, under some circumstances, for example, intermittent heating, weather compensation may not be the most effective strategy.

Comparison between measured space heating SPF and Heat Emitter Guide "Likely space heating SPF"

Using the design flow temperatures calculated for each site, the Heat Emitter Guide (HEG) 'likely space heating SPF' has been compared to the actual, measured space heating SPF. Correlations are poor, with the observed SPF's being significantly lower than the HEG values. This is more pronounced for GSHP than ASHP.

In conclusion, MCS heat pump installation standards were updated significantly and on several occasions during the RHPP period. Any changes inevitably take time to embed. The analysis presented here refers to the monitored RHPP sample; it should not be assumed to apply to heat pumps installed after the RHPP.

[2] The flow temperature for UFH is a function of the floor and envelope resistances.
Tf = {(Tin –Tout) * envelope area/ [floor area * (Rfloor / Renv)]} + Tin
Where Tf = flow, Tin and Tout = inside and outside temperature, Rfloor & Renv = thermal resistances of floor and envelope construction.
Where, e.g. UFH is embedded in timber floors with carpets, higher flow temperature may be necessary to ensure adequate heat emission.

1 Report on compliance with MCS installation standards

Introduction

The Microgeneration Certification Scheme (MCS) is an industry-led, nationally recognised, BS EN ISO/IEC 17065:2012-compliant quality assurance scheme for microgeneration, launched in 2008. It publishes product and installation standards, which are periodically updated to reflect evolving technical understanding. The installation standards specifically relating to heat pumps are MCS MIS 3005, MCS 021 plus a number of guidance documents relating to estimating heat loss, heat emitter design and designing ground loops and hydraulics.

An historical review of MCS installation standards MIS 3005 is necessary to contextualize the development of the installation standard over the period covered by RHPP and its evolving role in promoting quality design and installation. The RHPP ran from 1 August 2011 until 31 March 2014[3]. The design, installation and commissioning of the RHPP heat pump installations that took place over this period therefore involves more than one version of MIS 3005. MIS 3005 v2.0 was introduced in August 2010 and updated to v3.0 in September 2011. Version 3.1 was initially introduced on 1st February 2012 but was updated within the same month to 3.1a with the instruction that "all new quotes must be compliant with MIS 3005 v3.1 from 1st March 2012". MIS 3005 v3.2, published on 22nd July 2013, states: "Installers …. may commence working in accordance with this update from 22/07/2013", Table **1–1**.

Publication	Publication Date	Date at which standard became mandatory
MIS 3005 v2.0	26/08/2010	No date specified[4]
Heat Emitter Guide	09/08/2011[a]	No date specified
MIS 3005 v3.0	05/09/2011	No date specified
MIS 3005 v3.1	01/02/2012	No date specified
MIS 3005 v3.1a	20/02/2012	01/03/2012[b]
MIS 3005 v3.2	22/07/2013	22/10/2013

a) Publication reference:

b) Or on expiry of existing quotes. Period of quote validity to be determined by installation company. (MIS 3005 v3.1a. Important Information. (15/03/2013)

Table 1–1 MCS heat pump installation standards in place during the RHPP programme

[3] https://www.gov.uk/guidance/renewable-heat-premium-payment-scheme

[4] Where no date is specified, we assume that the standard became mandatory when published.

The Provisional Report (Wickins, 2014) states of installations referred to in the present report: "the final sets of meters started reporting valid data in October 2013", thus establishing v3.2 as the latest probable version for designing RHPP installations that were included in the RHPP trial. The design and installation of systems covered by the present report would have involved versions v2.0; v3.0; v3.1a; and possibly v3.2 of the MIS.

For each installation, there will be some ambiguity over the actual dates when installers changed to the latest MIS version, since they were allowed to work with the previous version where they had already provided the quotation/tender. It is expected that the version introduction date varied across installations, depending mainly on the dates of and time elapsed between the quotation and installation start date.

Figure **1-1** shows the quotation, installation and stated "commissioning" dates from the metadata and MCS certificates. The green vertical lines indicate the date on which each version of MIS 3005 was published. Note that the metadata only provides quotation dates for heat pumps installed in private properties. From this analysis, 47% of the ASHPs and 51% of GSHPs installed in private properties were quoted for before 01/03/2012 and so may not have complied with all aspects of MCS 3005 issue 3.1a. Commissioning dates are thought to be related to the date of registering the installation with MCS.

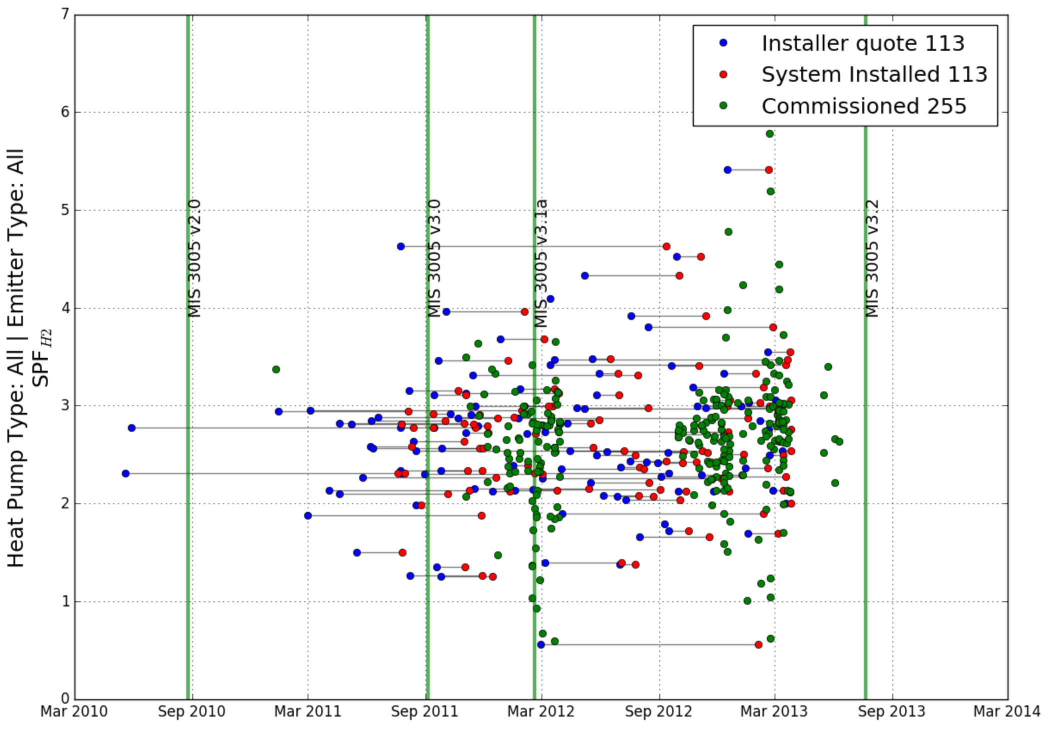

Figure 1-1 RHPP quote, installation and commission dates and dates of MCS standards

Substantive MIS 3005 changes

This report identifies the development of the MIS 3005 design criteria including the change from earlier versions that allowed monoenergetic bi-valent systems (to provide non-heat-pump backup during cold weather) to later versions that require the heat pump to supply 100% of the design-day space heating load without recourse to the backup heater; the removal of reference to SAP 2005 default SPFs and adjusted efficiencies; the introduction of BS EN 12831:2003 Heating systems in buildings. Method for calculation of the design heat load for room by room heat loss calculations; and the publication of the Heat Emitter Guide (HEG).

MIS v3.0 (09/2011) introduced for the first time a standard method for calculating heat loss and radiatior sizes and for estimating the system performance[5], that is, BS 12831 along with the HEG. In 2011, DECC supported a series of installer training sessions, the DECC Heat Pump Training road shows. These included online webinars and spreadsheets produced and presented by David Matthews, then Chief Executive of the Ground Source Heat Pump Association. Their purpose was to initiate the introduction of MIS 3005 v3.0 including heat loss calculations based on the CIBSE Domestic heating design guide (DHDG) and the national annex from BS EN 12831. Whilst BS 12831 provides guidance to the designer, the calculation of U-values and ventilation rates for heat loss calculations is still dependent on a qualitative, site-based assessment by the designer. The assessment of e.g. ground floor or window U-values and air change rates can lead to significant variation in calculated room heat loss and the whole-house heat pump power needed to satisfy the design day heat load.

The design assessment of SPF is necessary for the calculation of system annual energy use; it is a requirement of MIS 3005 that annual energy use is calculated and given to customers as part of the handover. The online webinars show how SPF could be derived from manufacturers' technical literature for both space heating and hot water, thus bypassing the use of SAP. However, a review of manufacturers' literature from the period shows that relatively few published sufficient COP data to identify performance at design outside temperature for space heating and design hot water flow temperatures. In addition, since all published COP values were from BS EN 14511 testing, the assessment of likely SPF would be based on COP measurements at outside design temperature and system flow temperature, in effect, a laboratory test condition of fixed source and sink temperatures and heat sink load (albeit with possible defrost cycle for ASHPs) and thus not reflecting the annual variations in these factors experienced in real installations. The installer also needs to assess a value for domestic hot water (DHW) SPF by interpreting COP space heating test results at the most appropriate DHW flow temperature, such as 55°C.

[5] MCS 3005 issue 2.0 stated that the installer should provide an estimate of heat loss but gave no details as to what temperature this should be for and contained no information on radiator sizing.

The calculation of SPF was rationalised with MIS 3005 v3.1/v3.1a, from which all references to SAP efficiencies were removed and which required space heating SPF to be assessed using the Heat Emitter Guide (HEG) based on flow temperatures[6]. The HEG is a look-up table which allows installers to select a space heating SPF based on the design heating system flow temperature, calculated through radiator oversize factors[7]. The look-up table values for the HEG "likely space heating SPF" are calculated based on weather conditions for a single location (Leeds, UK) and assuming that "the SPF values for ASHP are 0.7 less than for GSHP, which is consistent with SAP", plus an allowance of "100W for the electrical consumption of heating circulation pumps". The HEG definition of SPF is thus a combination of SPF_{H2} with circulating pump, or SPF_{H4} where there is no additional boost. The HEG SPF is for space heating, "heating circuit flow temperature" with the assumption that flow temperature is weather compensated. For the designer, likely SPF at a particular flow temperature is a function of radiator "oversize factor", the ratio of manufacturer's radiator output at 50K water-air ΔT to the room heat loss. This is expressed as a "star rating" where the estimates of SPFs for GSHPs and ASHPs are provided at the relevant flow temperature, Figure 1-2.

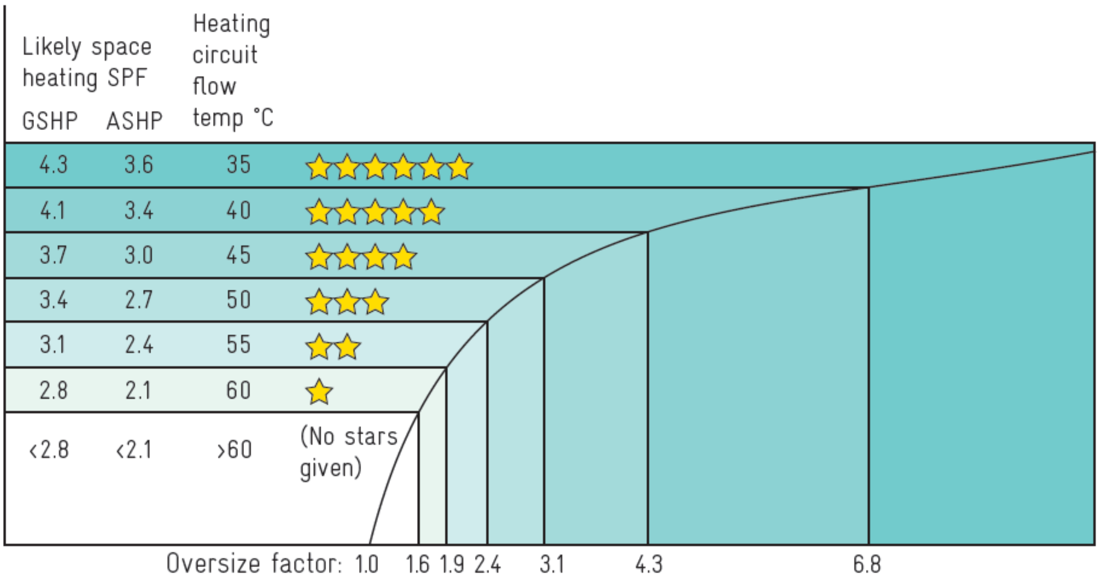

Figure 1-2 SPF, flow temperature and star rating (HEG, 2011)

The HEG SPF also provided the designer with a guide to SPF for DHW, also a requirement for the annual energy calculation; indeed the online MCS webinar annual energy calculations in Spreadsheet 5 are based on HEG values of likely space heating SPF for both space heating and domestic hot water[8]. Note that DHW heating is qualitatively different from space heating since the temperature difference between the primary flow and hot water in the cylinder reduces, approximately exponentially, from the start to the

[6] Section 4.3 of MCS 3005 issue 3.1

[7] Note that version 2.1 of the heat emitter guide (01/05/2015) no longer provides space heating SPF's, since these have been replaced by estimates from the MCS SCOP calculator, which was developed after the adoption of the Energy Related Products Directive.

[8] MSC Spreadsheet 5: http://www.microgenerationcertification.org/component/content/article/2-uncategorised/123-archived-installers-standards
http://www.screencast.com/users/Gemserv/folders/MCS%20Heat%20Pump%20Webinars/media/7416084e-63cf-4f50-88fd-1a005ff350ec

end of each heat-up cycle. The instantaneous performance of the heat pump will be determined by the temperature of the water around the coil, stratification and the location of the indirect heat exchanger coil within the cylinder. Actual SPF for each DHW cycle will be an integral from the start of the cycle, with the heat exchanger coil likely surrounded by water at a temperature between room and cold water feed temperature, to the end of the cycle, when the heat exchanger coil will be surrounded by water close to the maximum primary water temperature or cylinder thermostat set-point[9].

Instantaneous COP for DHW production is covered by BS EN 16147:2011 although, in practice, few manufacturers of GSHPs and ASHPs provide DHW data. Given that COP for DHW is not available to the designer, the HEG provides a solution for addressing DHW SPF.

Installer qualifications and handover

MIS 3005 has been the engine for driving enhanced installer training and quality management with the expectation that such an approach would raise the overall performance of heat pump installations. From as early as MIS 3005 v1.2 (2008), MCS has demanded a range of specific competences of designers and installers. Alongside a list of formal qualifications, MCS have been careful not to disadvantage those working on heat pumps without formal qualifications but who can show accredited prior learning for registration under the 'Experienced Worker Route', through:

- Manufacturer's product training – this is product specific and requires independent verification
- Experience gained through a mentoring process – this requires independent verification
- Demonstrable track record of successful installation – this requires independent verification

The original competence requirements have since been updated to establish 'nominated technical' and 'designer' contractual roles along with online qualifications and experience mapping. For MCS company registration, evidence of these competencies is evaluated by MCS Certification Bodies.

Number of heat pumps, manufacturers and models

Of the total 699 sites in the RHPP sample supplied to RAPID-HPC (referred to as "Sample A"), 99 sites were excluded at the outset of the project due to technical issues with the metering equipment and a further 104 sites were omitted due to missing data streams and other issues affecting the calculation of SPFs. The RHPP sample available for consideration in this analysis therefore comprised 496 sites. A further series of checks and filters were then applied to generate two sub-samples:

- Sample B2 (Broad data) with 417 sites (318 ASHP and 99 GSHPs) with sufficiently complete and stable (based on circulation rates) monitoring data of the HP system over a year to enable

[9] …unless the cylinder thermostat is set to a temperature below the maximum supply temperature from the heat pump. This would represent a failure of commissioning, but it is not impossible.

calculation of SPFs. The specific annual period selected for SPF analysis differs from site to site.

- Sample C (Concurrent data) with 299 sites (223 ASHP and 76 GSHPs) is a subset of sites in Sample B, but where data from the same annual monitoring period of 1/11/2013 to 31/10/2014 was selected. Sample C was used for adjustment of data to calculate the SPFs under UKSET conditions.

This report on compliance with MCS installation standards will use Sample B2 where actual recorded data provides the clearest explanation of installation performance, and use Sample C where weather corrected data provides the best solution to the questions posed. RHPP metadata will also be used where appropriate.

The RHPP MCS certificates show that sample A contains 23 separate manufacturers and up to 115 different heat pump models.

System boundaries and monitored data

Appendix A presents information on system boundaries and monitored data for the RHPP sample.

2 Heat pump sizing; have installers complied with MCS standards?

Introduction

Analysis of heat pump sizing for the RHPP must be referenced against the design requirements of MIS 3005 from version 2.0 to version 3.0 and 3.1a. Before MIS 3005 v 3.0 there was no specific guidance for designers regarding heat loss calculations and emitter sizing. Any method that the designer thought appropriate would have satisfied the requirements of MIS 3005 v2.0, whether rule of thumb, a manual calculator or manufacturers' software. Version 3.0 introduced the specific requirement that sizing should comply with the CIBSE Domestic Heating Design Guide (DHDG) and the BS EN 12831 National Annex.

DHDG and the National annex of BS 12831 are based on fabric and ventilation heat losses as expressed by $\left(\boldsymbol{\Sigma UA} + \frac{\boldsymbol{NV}}{\mathbf{3}}\right)\boldsymbol{\Delta T}$. Thus a heating system design requires the assessment of U-values (U) for fabric elements and their areas (A), the assessment of appropriate ventilation rates (N - air changes per hour) for each room and its volume (V) and the difference between the room temperature and the design outdoor air temperature for the geographic region (ΔT)[10]. The designer therefore has to assess three factors in order to meet MCS criteria: U-values for structural elements, ventilation rates for room-type (living, kitchen, bedroom, etc) and room temperatures along with the appropriate design outside air temperature from three sources – the DHDG, BS 12831 and MIS 3005. The designer has to measure the building for areas and volumes and to complete room-by-room heat loss calculations. Allowances for thermal bridging (BRE, 2006), and technical underperformance of thermal elements are not specifically addressed in either the National annex of BS 12831, DHDG or MIS 3005 v3.0[11].

Following the introduction of MIS 3005 v 3.0, DECC subsidised a series of roadshows and webinars for installers, exploring heat loss calculations and heat pump sizing. When assessing whether RHPP installations comply with MCS standards, RAPID-HPC consider it appropriate to reference system design calculations to this online resource, which can be found on the MCS website[12].

Heat loss calculations

Assessing fabric element U-values from the DHDG is not always straightforward. Wall construction in the UK is generally either solid brick, solid stone or cavity wall. Cavity walls may be brick and brick, or

[10] The problem with ventilation heat loss is that, over periods of the order of 24 hours or less, it actually appears mainly in those rooms in the infiltration zone of the dwelling. Those rooms in the exfiltration zone experience low or no ventilation heat loss. But the respective in/exfiltration zones move around depending on weather conditions. In a 2 storey dwelling, infiltration is mostly downstairs, but on windy days it may be the whole of the windward façade. Thus, the heat output required in each room varies with the weather. This aspect of design is not taken into account in the guides cited. Such calculation techniques have to be seen primarily as pragmatic rather than theoretically consistent representations of heat loss.

[11] Party wall heat loss is included, based on the assumption that the other side of the wall is at 10°C on average.

[12] http://www.microgenerationcertification.org/component/content/article/2-uncategorised/123-archived-installers-standards

brick and block where block conductivity varies significantly depending on the block materials. The eligibility criteria for the RHPP included cavity wall insulation (where appropriate) and at least 250mm of loft insulation where possible. Commonly used cavity-fill materials include mineral wool, polystyrene beads and foam, which all have varying thermal properties. The designer will generally not have the technical specification for the insulation and, allied to the variable quality of installation, must make an assessment of the likely U-value. The DHDG provides a range of U-values for brick/brick and brick/block walls with 13 mm plaster and 50 mm mineral fibre-filled cavity from 0.56 to 0.39 W/m²K resulting in a potential difference of 35%[13]. Even if the most appropriate value is selected from the DHDG, the actual performance of the wall is likely to be substantially different. Wingfield et al. (2010) present measurements of the in situ performance of insulated cavity walls and conclude that the ratio of actual to design U-values is typically between 1.5 and 2. Hulme and Doran (2014) concluded that the ratio of actual to RDSAP estimated U-values was between 0.86 and 0.99 for uninsulated cavities (based on 50 homes) and that the ratio for insulated cavity walls was 1.29-1.34 (based on 109 homes). U-values for solid walls are known to be overestimated in many cases (Li et al. 2014).

Conversely, the DHDG provides a double glazing default of 2.8 W/m²K whereas most double glazed windows installed since 2002 will outperform this value. For example, 10 year old 12 mm gap double glazing that complied with Approved Document L1 2000 (DCLG, 2000:23) may have a U value ranging from 2.8 to 1.8 W/m²K (Table A1). It is therefore possible that estimates of window heat losses may vary between designers by 40%. Additional losses, such as at window reveals and lintels, may be significant, and are unlikely to be taken into account.

Ventilation rates offer yet another example of the potential for variation in heat loss assessment. The DHDG provides design ventilation rates ranging from 3.0 air changes per hour (ac/h) for a bathroom, 2.0 ac/h for a kitchen, 1.5 ac/h for a living room and 1.0 ac/h for a bedroom. These air change rates are all significantly higher than the whole house ventilation requirement of 0.5 ac/h. MIS 3005 v3.1a suggests the use of lower ventilation rates where appropriate: "However, this option should only be taken with caution as field trials indicate these ventilation rates tend to provide good in-use estimates of power and energy consumption"[14]. BS 12831 introduces three ventilation categories (A, B and C) based on year of build – pre-2000, 2000 to 2006 and post 2006 where Category B and C provide 1.0 and 0.5 ac/h for living rooms. The RHPP required homes to have cavity insulation where practical and statistics from the BRE Housing tool suggest that they are likely to have had reasonable draught stripping with double glazed windows that reduce air infiltration[15], indicating that lower estimates of ventilation rates may have been chosen by the installer. Further complexity is introduced for rooms with open fires, a condition identified in many of the RHPP off-gas grid houses. A 40m² room with chimney with a throat restrictor, typical of a wood burner, requires 3 ac/h.

[13] Percentage difference = $((V_1 - V_2) / ((V_1 + V_2)/2)) * 100$

[14] Room heat loss is based on the worst-case assumption that the room is on the windward side of the building (see earlier footnote). This cannot be true for all rooms at the same time, so summing individual room heat losses, will lead to an overestimate of whole house heat loss. This does not invalidate the procedure, whose purpose is purely pragmatic.

[15] *BRE housing tool* indicates that, in 2011-12, 88% of RSL and local authority houses had full double glazing. In the owner occupier sector, the figure was 77%. http://housingdata.bre.co.uk/Home/Crosstab

The summation of room losses provides the power required for space heating in continuous mode. For intermittent heating, an additional allowance of 15% (DHDG) or 20% (BS 12831) is added to the room heat losses to provide an estimate of the heat pump heat output rating required for the dwelling. BS 12831 also suggests that for intermittent heating, outdoor design temperature is reduced in accordance with Table NA.1c External Design Temperatures and resulting in a percentage difference in heat loss of an additional 1.5 to 2%. The DHDG suggests a further 10% allowance for distribution losses and hot water allowance from 2.0 to 3.0 kW. We may comment that uninsulated pipes within the insulated envelope would provide useful space heating during the space heating season whilst primary pipework to the hot water cylinder (which operates in both winter and summer) has been required to be insulated under Part L of the Building Regulations for new dwellings since 2002. Observations in the RHPP case studies indicate that whilst pipe insulation around the heat pump and cylinder is generally provided, the quality of that insulation is variable. For these reasons it is difficult to say whether that DHDG recommendations for allowances for distribution losses will be met. Heat pump DHW switching is generally by diverter valve, since DHW is required at the highest heat pump output temperature. The heat pump will switch between space heating and DHW rather than supply both at the same time. With diverter supplied DHW, it is most unlikely that installers would add the hot water power load when sizing for maximum heat pump output. The situation is best described with an example: for a house with an 8 kW design day heat loss, the additional allowances would result in the selection of a circa 13 kW heat pump. For heat pumps, unlike for gas boilers, the additional cost of the larger unit output would also be significant, at approximately 40% of the base case price[16].

The final heat loss assessment and heat pump sizing therefore reflects the designer's technical ability to navigate through the DHDG, BS 12831 and MIS 3005 as well their commercial judgement when tendering. Apart from the heat pump specifics of MIS 3005, many of these design decisions apply to boilers.

Radiator sizing

For the installation designer, maximising heat pump Carnot efficiency is a primarily a function of sink temperature, that is the temperature of the water flowing to the emitters, whether underfloor heating, radiators or hot water cylinder[17]. Thus to maximise the SPF of an installation, a designer should select the lowest flow temperature consistent with meeting the heat demand of the dwelling. This selection procedure will comprise several factors including typical default flow temperatures. Providing the designer understands this relationship, we would expect the selection of radiators to be a compromise between radiator size (high SPF requires larger radiators) and annual efficiency. Lower temperatures require larger radiators or, where wall space is limited, progressively higher outputs are achieved for the

[16] http://www.airconwarehouse.com/acatalog/Mitsubishi_Ecodan_Air_Source_Heat_Pumps.html
[17] Carnot efficiency = $T_{sink} / (T_{sink} – T_{source})$. Actual SPFs are typically around half the Carnot efficiency.

same height and length through the addition of double or triple panels and convectors, Figure **2**-**1**.

50 Δt (75/65/20°C)

Height	Length mm	Sections	P1 UIN	P1 Heat output Watt	P1 Heat output Btu/hr	K1 UIN	K1 Heat output Watt	K1 Heat output Btu/hr	P+ UIN	P+ Heat output Watt	P+ Heat output Btu/hr	K2 UIN	K2 Heat output Watt	K2 Heat output Btu/hr	K3 UIN	K3 Heat output Watt	K3 Heat output Btu/hr
300	500	15	-	-	-	143650	255	870	143656	373	1273	143662	491	1675	-	-	-
	1000	30	-	-	-	143651	509	1737	143657	745	2542	143663	982	3351	3033310	1349	4603
	1500	45	-	-	-	143652	764	2607	143658	1118	3815	143664	1473	5026	-	-	-
	2000	60	-	-	-	143653	1018	3473	143659	1490	5084	143665	1964	6701	3033320	2698	9206
	2500	75	-	-	-	143654	1273	4343	143660	1863	6357	143666	2455	8376	-	-	-
	3000	90	-	-	-	143655	1527	5210	143661	2235	7626	143667	2946	10052	-	-	-

Figure 2-1 Example radiator catalogue

The design of installations before the publication of the HEG required a knowledge of radiator correction factors since catalogue outputs are based on a 50K temperature difference (50K delta T) between mean radiator temperature and room air temperature. For a 45° to 35°C flow and return (40°C mean) and 20°C room air temperature (a 20K difference), the radiator output is only about 30% of catalogue emission and a larger surface area is required. The designer must adjust the catalogue output by the appropriate correction factor (Table **2–1**) in order to select the right sized radiator.

delta T (K) between radiator & room	Radiator Output Correction Factor
20	0.30
25	0.41
30	0.51
35	0.63
40	0.75
45	0.81
50	1.00
55	1.13
60	1.27
65	1.41
70	1.55

Table 2–1 Typical panel radiator output correction factors

The HEG has simplified this process since the designer may start by selecting a target SPF, known as 'likely space heating SPF' based on a maximum space heating flow temperature. From the flow temperature, the star rating provides the oversize factor for the radiators. Continuing the previous example for a 45°C flow temperature and applying the HEG: a 4 star system would require a radiator oversize factor of between 3.1 and 4.3, a mean of 3.7, Figure **1**-**2**. Under these circumstances a radiator with a catalogue output at a 50 K mean temperature difference of 3700 Watt (oversize factor of 3.7) will meet a room design heat loss of 1000 Watts.

Each increase in star rating results in a higher likely space heating SPF and therefore greater energy efficiency in annual operation. However, when replacing, for example an oil boiler-fed high temperature radiator system with low temperature heat pump-fed radiators, achieving a 4 star output is not just a technical decision of radiator replacement. A typical 600 x 1100 single panel-single convector with an actual output at 50K of 1078 W cannot be replaced by the same size radiator to achieve a heat pump 4 star design. A 4 star design requires an output at 50K of 3700 Watts and the same size double panel-double convector or triple panel-triple convector have outputs at 50K of only 1905 W and 2628 W respectively – they are not large enough. The triple panel would achieve a star rating of only 2.6 (against a requirement for 3.7) and cost approximately three times as much as the original double panel-single convector. Achieving a 4 star design would require either more than one radiator or replacing the 600 x 1100 single panel-single convector with a 600 x 1800 triple panel-triple convector with a catalogue output of 4300 W[18]. It is worth noting that triple panel-triple convectors of 600 x 1100 and 600 x 1800 weigh over 50 kg and 80 kg respectively[19] with implications for occupational health and safety and increased labour costs.

It is apparent that, even for the technically competent designer, radiator selection will be based on multiple factors that include space availability, client demand, room aesthetics and capital and labour cost. The likely result is that radiator star ratings will vary from room to room as illustrated in Section 2.4.

Heat pump sizing, compliance with MCS

Compliance may be illustrated with an example from the case studies carried out by RAPID-HPC. An end terrace house in South Wales with an ASHP and radiators provides a graphic example of the multiple factors that installers need to negotiate to produce a heat loss calculation and radiator design. Heat loss calculations based on DHDG are compared to BS 12831 continuous; BS 12831 intermittent; and BS 12831 intermittent with wood burners in two rooms, resulting in four different heat pump sizing calculations, Table 2–2, Table 2–3, Table 2–4 and Table 2–5. Radiator sizes were measured on site and catalogue outputs produced for 50K delta T to provide room by room star ratings.

[18] https://www.plumbnation.co.uk/site/stelrad-compact-triple-panel-triple-convector-radiators/

[19] http://www.screwfix.com/p/kudox-premium-triple-panel-convector-radiator-white-600-x-1800mm/8945f
An 80 kg radiator may require a four person team to lift and fit to the radiator brackets (HSE, 11/2012, Manual handling at work. A brief guide) .

Ventilation (Air-changes per hour)	Room	Heat output at a water-air ΔT of 50^0 C[20] (Watts)	Heat loss (Watts)	Star rating from the HEG
1.5	Living room total	2868	1087	3
2.0	Kitchen	873	531	2
1.5	Dining room	1598	776	2
1.5	Hall & Landing	1991	767	3
2.0	Bathroom	1055	358	3
1.0	Study	964	401	2
1.0	Small bedroom	852	473	2
1.0	Main bedroom	1106	386	3
	HP output		4778	

Table 2–2 DHDG estimates of heat loss for continuous heating at -1.6°C, ventilation Class A

Ventilation (Air-changes per hour)ac/h	Room	Heat output at a water-air ΔT of 50^0 C (Watts)	Heat loss (Watts)	Star rating from the HEG
0.5	Living room total	2868	802	4
1.5	Kitchen	873	477	2
0.5	Dining room	1598	592	3
0.5	Hall & Landing	1991	616	3
1.5	Bathroom	1055	326	3
0.5	Study	964	340	3
0.5	Small bedroom	852	393	2
0.5	Main bedroom	1106	291	4
	HP output		3837	

Table 2–3 BS 12831 estimates of heat loss for continuous heating at -1.6°C, ventilation Class C

Ventilation (Air-changes per hour)ac/h	Room	Heat output at a water-air ΔT of 50^0 C (Watts)	Heat loss (Watts)	Star rating from the HEG
0.5	Living room total	2868	912	3
1.5	Kitchen	873	633	1
0.5	Dining room	1598	758	2
0.5	Hall & Landing	1991	796	3
1.5	Bathroom	1055	273	4
0.5	Study	964	457	2
0.5	Small bedroom	852	507	2
0.5	Main bedroom	1106	376	3
	HP output		4712	

Table 2–4 BS 12831 estimates of heat loss for intermittent heating (plus 20%) at -3.2°C, ventilation Class C

[20] Manufacturers typically quote the output of radiators at a mean radiator-to-room temperature difference of 50K. This is often referred to as the water-air ΔT. With a temperature difference across the radiator of 10K, and a room temperature of 20°C, this is equivalent to a flow temperature to the radiator of 75°C and a return temperature of 65°C.

Ventilation (Air-changes per hour)ac/h	Room	Heat output at a water-air ΔT of 50^0 C (Watts)	Heat loss (Watts)	Star rating from the HEG
3.0	Living room total	2868	1704	2
1.5	Kitchen	873	633	1
3.0	Dining room	1598	1347	1
0.5	Hall & Landing	1991	796	3
1.5	Bathroom	1055	273	4
0.5	Study	964	457	2
0.5	Small bedroom	852	507	2
0.5	Main bedroom	1106	376	3
	HP output		6093	

Table 2–5 BS 12831 estimates of heat loss for intermittent heating (plus 20%) at -3.2°C, ventilation Class C with throat restrictor chimneys in living and dining rooms (highlighted in red)

The heat loss calculations in Table **2–2** to Table **2–5** range from 3.8 to 6.1 kW (to 1 decimal place), an increase in output of about one third.

Different radiator sizes that could result under these different design criteria would lead the living room star rating to vary between 4 and 2. For BS 12831 intermittent heating, Table **2–5**, the Case study occupants' operating pattern, achieving the Study room set-point temperature at design outdoor conditions requires a 2 star flow temperature of 55°C resulting in a likely space heating SPF of 2.4 for an ASHP. We may assume that, for comfort conditions, living and dining room radiators are supported by the wood burning stoves, whilst prolonged periods in the kitchen are associated with heat gains from cooking. The examples above illustrate that assessing compliance with MCS design criteria is by no means a simple procedure.

Peak load and Plant size ratio

MIS 3005 expects designers to assess maximum heat pump power requirements or peak load. Peak load is described by reference to the CIBSE DHDG for space heating at design temperatures and may include allowances for intermitancy, pipe losses and DHW. The installer will likely choose a heat pump from a preferred manufacturer that meets this peak load and, since most manufacturers provide a limited number of models with differering outputs in step changes, it is likely that the heat pump chosen will be of a higher capacity, resulting in a higher plant size ratio. This section of the report explores peak load in terms of installer information on maximum power output.

Peak load is defined as the heat pump space heating load at design outdoor temperature – the design heat loss. Plant size ratio compares the heat pump maximum output (the calculation of which may include various factors such as a 15 or 20% allowance for intermittency) to the peak load. Plant size ratio is therefore equal to installed heat emission of the heat pump/design heat loss. Data for maximum heat pump power is found from the RHPP metadata and MCS certificates and is typically expressed by manufacturers at BS EN 14511 test temperatures of 0°C brine/45°C water for GSHPs and at 7°C air/35°C water for ASHPs. However, manufacturers' literature showing output data is not consistent. For maximum power at design conditions, these power outputs from BS EN 14511 testing need to be adjusted for the chosen flow temperature. GSHPs, according to MIS 3005, must be designed for a minimum ground loop flow into the evaporator of 0°C; however, for ASHPs, the heat pump maximum output also needs to be adjusted for COP at outdoor design temperature. It is assumed that for most installers this would not have proved possible since at the time of installation of the RHPP heat pumps, many manufacturers did not supply sufficient test data to allow the installer to interpolate power outputs between maximum (-0.2°C, Plymouth) and minimum (-3.9°C, Glasgow) outdoor design temperatures (MIS 3005, Table 2, after CIBSE Guide A Table 2.4). It is therefore assumed that outputs quoted by installers from the metadata are predominantly those published by manufacturers at standard testing conditions. In addition, as already noted (Section 2.2) additional allowances for intermittency, pipe losses and hot water production may have been added to the space heating load. Finally, the designer must choose a heat pump with an output above that calculated. Thus we would expect the chosen heat pump output to always exceed the maximum operating load.

Assessing peak heat pump power output at the MIS 3005 outdoor design temperature, "hourly dry-bulb temperatures equal to or exceeded for 99% of the hours in a year", is complicated by the lack of heat pump operating data at these low temperatures. It is also complicated by the time lags inherent in the opaque fabric of dwellings, which tends to smooth out the effect of variations in external temperature with periods of less than about 24 hours.

Historically the information from different manufacturers has varied between a single value of kW output at BS EN 14511-2:2007 'Standard rating conditions' to comprehensive tables and graphs across a range of source and sink temperatures. In addition to this literature, manufacturers may also produce non-published information which is made available to heat pump purchasers/registered installers for extrapolation and interpolation of performance data. Such additional data has not been considered although it may have some bearing on the conclusions to this section.

Standard rating conditions for GSHPs are 0°C 'brine', the ground loop flow temperature into the evaporator with 45°C space heating flow temperature from the condenser, and for ASHPs 7°C outdoor air and 35°C space heating flow. The effects on power output of varying both the source and sink temperature are shown in Table **2–6** for a nominal 6 kW output GSHP.

Source Temperature	Sink Temperature		
	35°C	45°C	55°C
0°C	5.9 kW	5.48 kW	5.17 kW
5°C	6.81 kW	6.49 kW	6.08 kW

Table 2–6 GSHP Manufacturer's capacity data sheet

Some manufacturers provided output data in graphical form at, for example, flow temperatures of 35°C and 50°C with graphs of kW output, compressor power and coefficient of performance (COP). An example from a manufacturer's installation instructions illustrates the interrelationship between these variables, Table **2–7**. Whilst in this instance there is little difference in power output at these flow temperatures, 4.95 and 4.75 kW (a 4% percentage difference), the output is maintained by increased power consumption at the compressor with a significant impact on COP and by extension, on seasonal performance factor (SPF).

Flow temperature	Heat Output	Compressor power	COP
35°C	4.95 kW	1.25 kW	3.9
50°C	4.75 kW	1.75 kW	2.7

Table 2–7 GSHP capacity derived from manufacturer's graphical data

Whilst some ASHP manufacturers provided just a single value of performance at, typically, 7°C outdoor air temperature and 35°C flow temperature, others provided tables of outputs for a range of source and sink temperatures. These values may be given as 'peak values' or as 'integrated values', that is, excluding or including the effects of defrosting at low outdoor temperatures. Table **2–8** and Table **2–9** illustrate this difference for a 6 kW nominal output ASHP.

Source Temperature	Sink Temperature			
	35°C	40°C	45°C	50°C
-7°C	4.8 kW	4.5 kW	4.3 kW	4.2 kW
-2°C	5.6 kW	5.3 kW	5.0 kW	4.9 kW
2°C	6.4 kW	6.0 kW	5.7 kW	5.3 kW
7°C	7.5 kW	7.0 kW	6.7 kW	6.5kW

Table 2–8 ASHP peak capacity values at different sink and source temperatures for a nominal 6 kW heat pump

Source	Sink			
	35°C	40°C	45°C	50°C
-7°C	4.2 kW	4.0 kW	3.8 kW	3.7 kW
-2°C	4.9 kW	4.6 kW	4.4 kW	4.4 kW
2°C	5.5 kW	5.2 kW	4.9 kW	4.8 kW
7°C	5 kW	7.0 kW	6.7 kW	6.5 kW

Table 2–9 ASHP 'integrated capacity values' at different sink and source temperatures for a nominal for 6 kW heat pump

Comparing the tables, at -2°C (indicative of mean outdoor design temperature for UK locations) the effects of defrost on output are evident e.g. at 45°C (a typical design temperature for radiators) where a peak output of 5.0 kW is reduced to 4.4 kW, a 12% difference and where 4.4 kW represents a 26% difference from the 6 kW nominal output. This relationship between net capacity, source and sink temperature on kW output is illustrated graphically for an ASHP in Figure **2-2**.

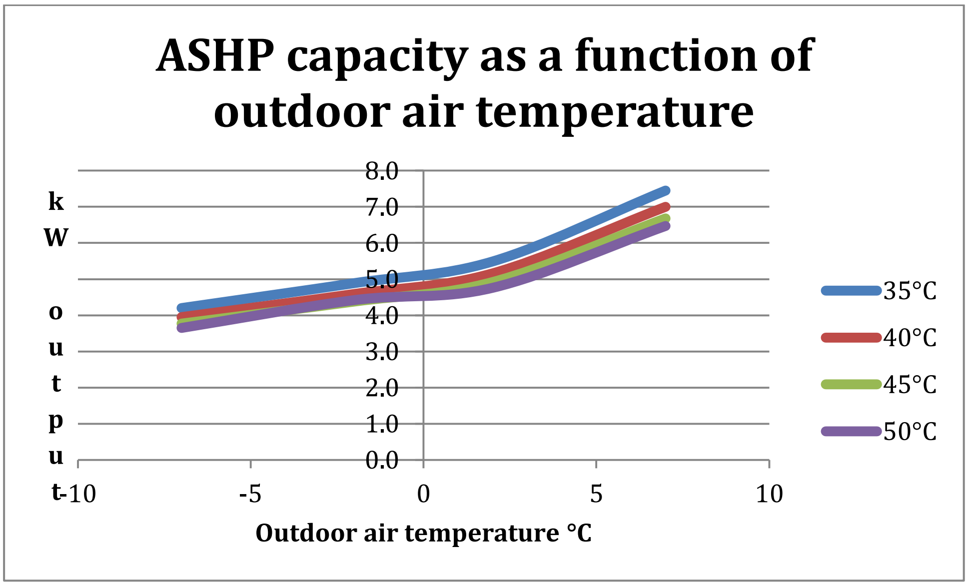

Figure 2-2 Manufacturer's ASHP capacity as a function of outdoor air temperature for 6 kW nominal output

MIS 3005 v4.0 (16/12/2013) introduced the MCS Compliance Certificate. This certificate requires the installer to quote the estimated power at design conditions. However, the monitored RHPP heat pumps were installed in 2011-2013, prior to existence of this compliance certificate.

The RHPP metadata contained a column headed "installer net capacity". RAPID-HPC mapped this "installer net capacity" against MCS-supplied heat pump product codes for 332 heat pumps, selected to cover a wide range of installation companies. For the 240 ASHP examined, 86% of the installer net capacities corresponded to the nominal capacities provided by the manufacturers at standard conditions. A similar exercise was carried out for 92 GSHP, of which 89 net capacity values reflected the nominal capacities. The mapping of installer net capacity to manufacturers' nominal capacity indicates that metadata 'installer net capacity' is overwhelmingly listed as manufacturers' nominal capacity, or for the installer, 'what it says on the box'.

Net capacity

In order to assess the maximum power output at close to design conditions from the metered data, an algorithm was developed based on the following procedure, the results are shown in Figure 2-3 and Figure 2-4:

- For each site find the coldest external temperature reading[21].
- For every day in the calendar month with the coldest external temperature reading, calculate the average external temperature and get the 99th percentile space heating power value from the meter readings for each site.

[21] External weather conditions were not monitored as part of the RHPP monitoring programme. Weather data has therefore been obtained from the United States National Centres for Environmental Prediction (NCEP) Climate Forecast System Reanalysis (CFSR, Saha et al 2011). This data has been shown to be as accurate as any historic weather data set while providing a higher geographical resolution with no missing data (Sharp et al 2015, Fuka et al 2014).

- These values are plotted in the top right hand panel. There are therefore approximately 30 points per site.
- Weather compensation curves are essentially straight line plots providing space heating supply or return temperature against the difference between room and outdoor temperature (for derivation of curves see Appendix C). We therefore fit a straight line and extrapolate down to zero degrees (0°C) external temperature to provide the expected peak heating power output at a daily average temperature of zero degrees to model low outdoor temperature performance. The straight line fit is sensitive to uncertainties in the space heat readings and the external temperature estimates. If some of the data has been corrupted it is possible for the gradient to be miscalculated as zero or even positive, meaning that, un-realistically, heat output increases as external temperature rises. Those sites where a negative gradient is not returned are given a peak space heating power output equivalent to the maximum daily value of the coldest month. This extrapolated peak heating power output at a daily average temperature of 0°C is shown on the bottom righthand chart.
- We consider that the peak heating power at an average daily temperature of 0ºC will occur when the hourly external temperature is comparable to, or lower than, the design temperatures for maximum flow rate in MCS (-0.2ºC for Plymouth, -3.9ºC for Glasgow). Therefore, extrapolation of peak heating output to a daily average temperature of 0ºC will provide a rough estimate of the peak heating power at the MCS design temperature.
- Installer net capacity is shown in the top left hand side.
- A scatter plot is created of peak heating power versus Installer-estimated net capacity, shown on the bottom left hand side. The line is to guide the eye and shows what a one-to-one mapping would be. For Figure 2-4 the red dots identify those systems with unmetered resistance boost producing more than 10% of annual heat output.

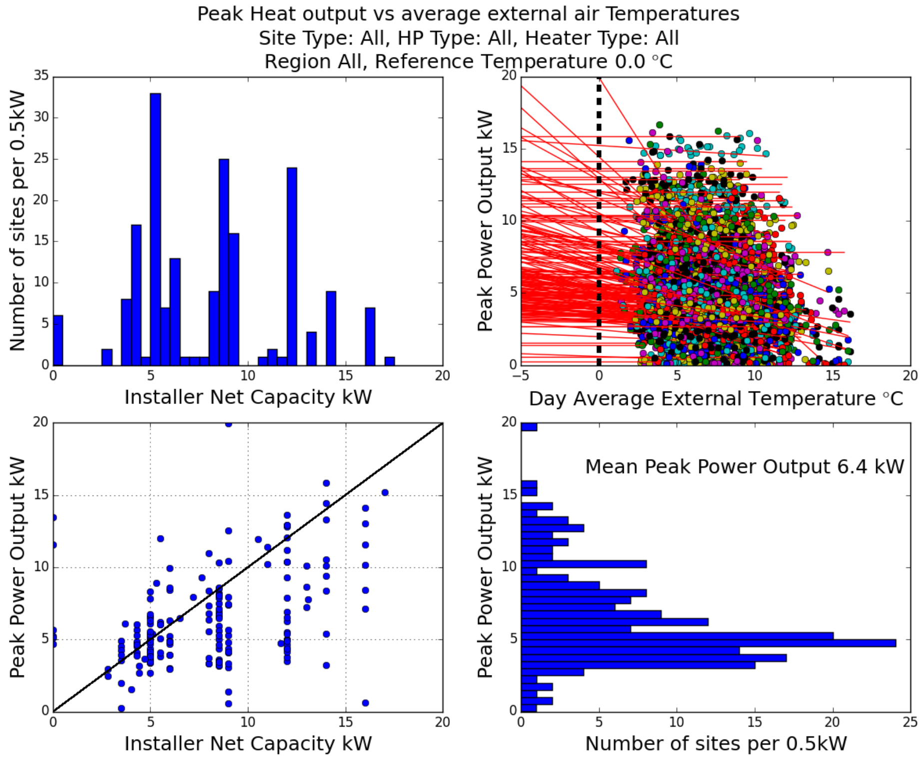

Figure 2-3 Assessment of peak heat power versus installer net capacity (kW).
The red lines in the upper right panel are the straight line extrapolations of the peak power expected at 0C external temperatures. A histogram is then made in the bottom right hand plot of the expected peak power at 0C external temperatures. These peak power values are then compared to the installer net capacity values in the bottom left hand scatter plot. A histogram is made of the installer net capacity values in the upper left hand plot.

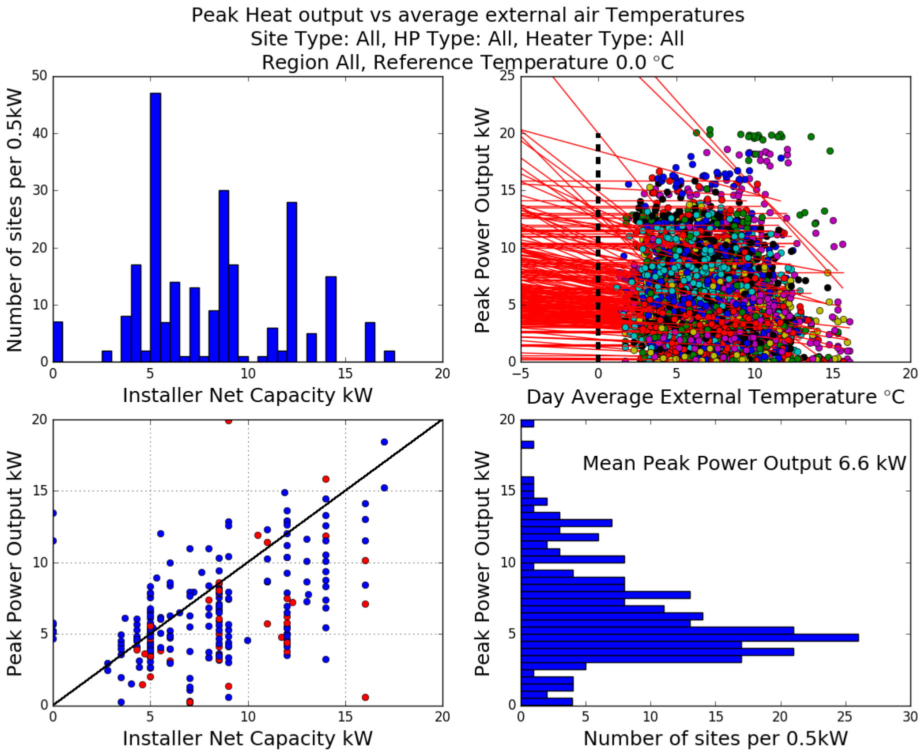

Figure 2-4 Assessment of peak heat power versus installer net capacity (kW) showing HPs with >10% boost
The red lines in the upper right panel are the straight line extrapolations of the peak power expected at 0C external temperatures. A histogram is then made in the bottom right hand plot of the expected peak power at 0C external temperatures. These peak power values are then compared to the installer net capacity values in the bottom left hand scatter plot. Red dots identify heat pumps with >10% boost annual boost. A histogram is made of the installer net capacity values in the upper left hand plot.

The bottom left hand side scatter plots in Figure **2-3** and Figure **2-4** show the peak capacity at a mean daily external temperature of 0°C as a function of installer net capacity or "installer net power" as written on the MCS certificate. As noted previously, at the time of the RHPP installation not all manufacturers provided the required tables or charts to allow installers to estimate the power at design conditions. The procedure to do so was described in the Matthews webinars[22]. In practice, it is not possible to assess how many of the installers actually estimated the power at design conditions; the new MCS compliance certificate includes an explicit requirement to do this. The requirement for meeting heat output at design conditions was introduced in MIS 3005 v3.0 (05/09/2011: Section 4.2.1.c). Downloadable MCS spreadsheet certificates explicitly demanding the heat output at design conditions were introduced in MIS 3005 v4.0 (16/12/2013: Section 6.1).

In general, and particularly for fixed speed compressor ASHPs, the capacity at design conditions of -2°C (the mean outdoor design temperature for MIS 3005 UK locations) is demonstrably lower than manufacturer nominal output or badged capacity. For five fixed speed compressor ASHP models the capacity at design conditions drops by between -8% and -32% depending on the model and flow temperature chosen.

For the two variable speed ASHP models examined, the capacity at design conditions (as calculated from manufacturers' data) is reduced by as little as -2% to -4% for one manufacturer but by -20% to -31% for another.

For GSHPs MIS 3005 design criteria is based on a minimum ground loop temperature 0°C and since manufacturers provide COP data at this temperature no interpolation based on source is required. Output however will vary depending on flow temperature. BS EN 14581 standard rating conditions provide 0/45 for 'brine' source (water with antifreeze) GSHPs. For the 10 GSHP models examined, the listed capacity at 0/45 was generally different from the manufacturer's nominal capacity and varied between plus 5% and minus 23%. Similarly, variations in output are found for 0/35, varying from plus 10% to minus 12% . For a more complete description of output at design temperature see Appendix B.

The impact on output at design temperature, explored in Figure 2–3 and Figure 2–4 and shown in the bottom left hand side chart, demonstrates that some heat pumps in the sample are clearly undersized according to MIS criteria. In order to determine whether the remainder of the heat pumps in the sample are under or oversized, the corrections from nominal capacity to capacity at design conditions would have to be applied to the x-value of each point on the chart. As described in Appendix B, these corrections depend on the make and model of heat pump; for the models examined, we found a range from -2% to -32% for ASHP and from -23% to +10% for GSHP. This calculation has not been carried out. As a result, we are not able to make a more precise assessment of how many of the heat pumps in the sample were

[22] MCS Heat Pump Webinars, MIS 3005 v3 Spreadsheet – 5: http://www.screencast.com/t/s13i9Nrmp7

under or over-sized relative to the MCS criteria. However, there are some points for which the inferred design capacity is much lower than the nominal "net" capacity, which would indicate over-sizing.

Load Factor

As a short digression from the analysis of MCS proper, the analysis of RHPP data for load factor offers the opportunity to consider the impact on annual operation at low loads and hence any connection between annual heat output and SPF. The Heat emitter guide provides likely space heating SPF for a range of flow temperatures whereas analysis of load factor may indicate that annual heat output may also affect SPF. As defined by DECC (2010) – 'the load factor on an unchanged configuration basis for a calendar year', is calculated from:

$$\boldsymbol{Load\ Factor} = \frac{\frac{\text{"}\boldsymbol{Sum\ of\ heat\ output\ in\ Wh\ per\ year}\text{"}}{\boldsymbol{1000\ Wh\ per\ kWh}}}{\text{"installer Net capacity kW"} \times \boldsymbol{8760\ hours\ per\ year}} \times \boldsymbol{100}$$

$$\boldsymbol{LF} = \frac{\boldsymbol{kWh\ actual}}{\boldsymbol{kWh\ potential}} \times \boldsymbol{100\%}$$

The load factor for heat pumps, therefore, is based on heat energy generated and is defined as the ratio of produced energy (kWh) to the maximum possible energy output (kWh) over a year. Data for maximum energy is derived from the nominal heat pump power as found from the RHPP metadata and MCS certificates.

Load factors are shown for all heat pumps in Figure **2**-**5**; for ASHPs in Figure **2**-**6**; and for GSHPs in Figure **2**-**7**.

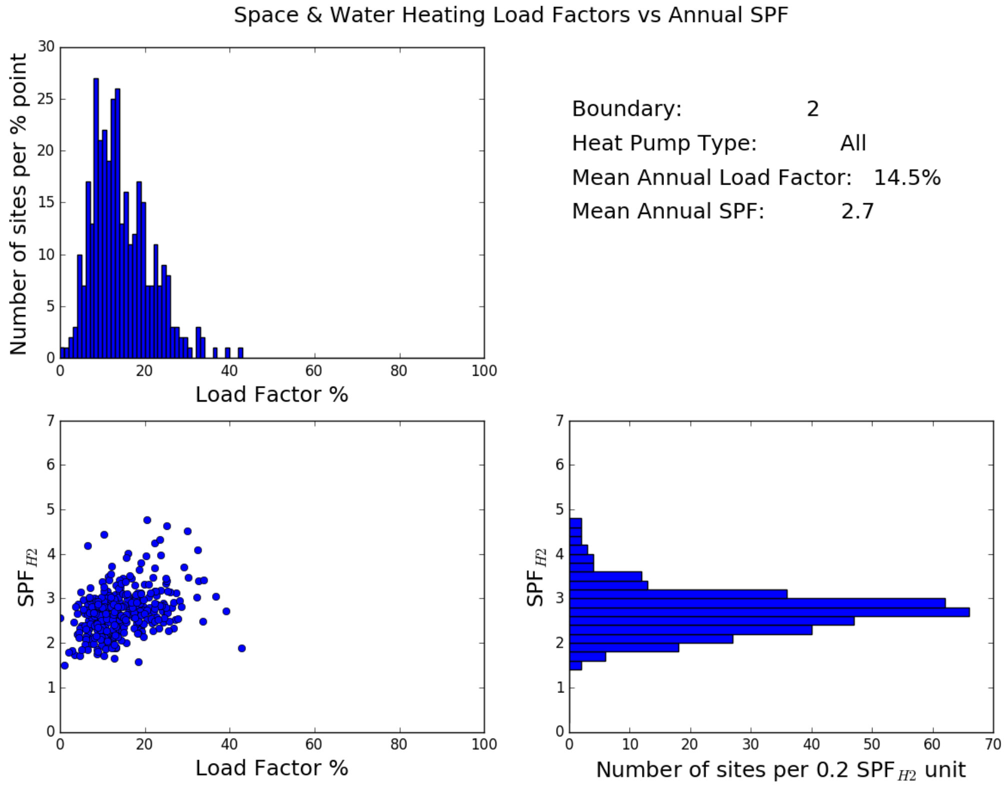

Figure 2-5 Load factor for all heat pumps

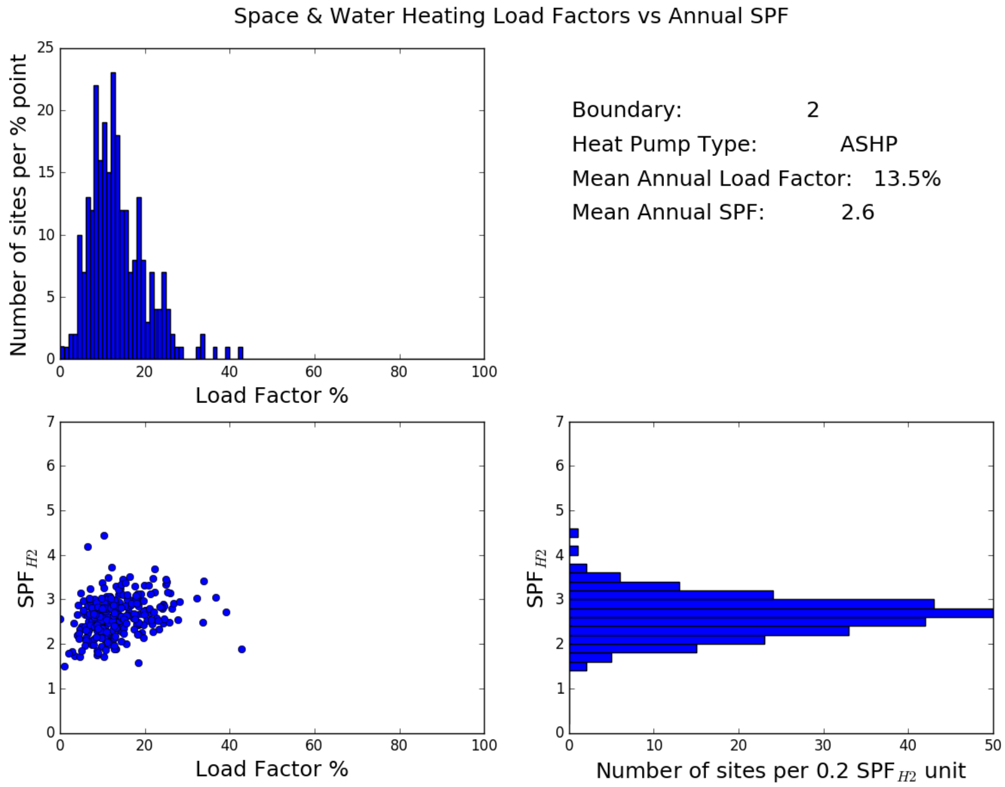

Figure 2-6 Load factors ASHPs

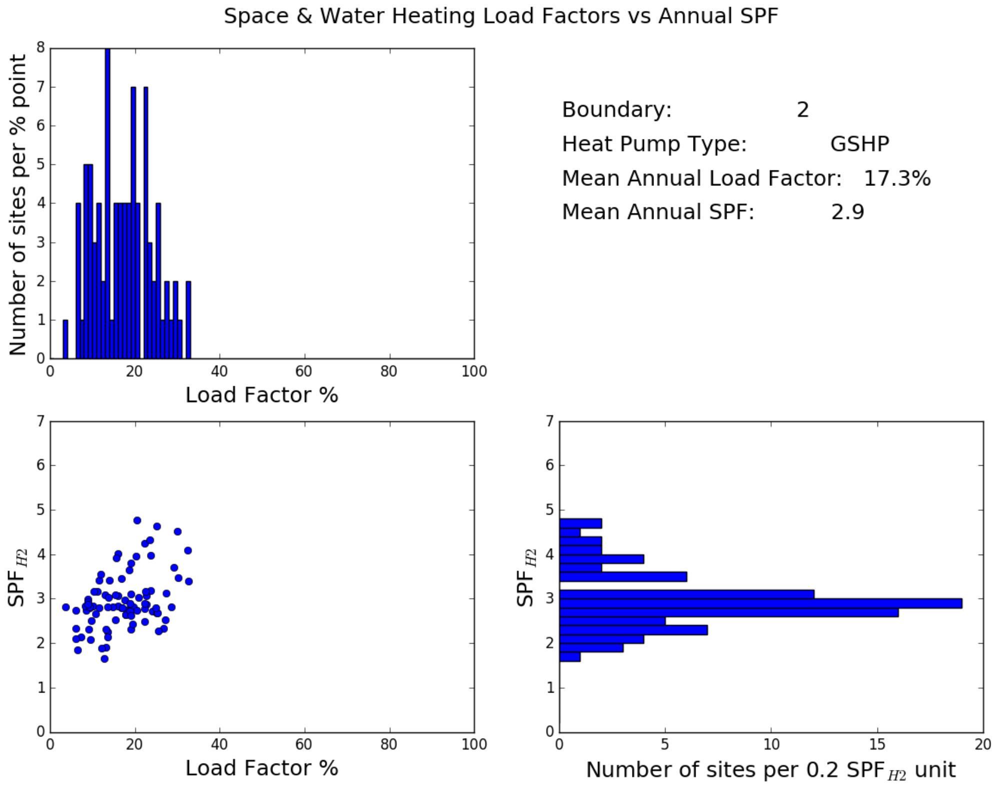

Figure 2-7 Load factors GSHPs

The load factor for all air and ground source heat pumps is 14.5%, for ASHPs 13.5%, and 17.3% for GSHPs. When viewed annually there appears to be some relationship with SPF at low load factors although this is less clear for GSHPs, with the suggestion of a relatively well defined lower boundary to the scatterplot. The relationship between load factor and SPF is further explored in Appendix D.

Ground loop design

MIS 3005 v2.0 states: "The design of Closed-Loop Heat Exchangers shall be in compliance with the Microgeneration Heat Pump manufacturer's specification and shall be clearly documented so that such compliance may be demonstrated." MIS 3005 v3.0 introduced a comprehensive ground loop design process in Section 4.2 which states that: "For all installations, the installer shall complete and provide the customer with Table 3." Unfortunately, no design information for the ground loop was made available to RAPID-HPC for analysis other than that supplied in the metadata file for the 173 GSHPs. The metadata specification requires installers/commissioners to provide information on the type of ground loop, ground pump power, speed setting, type of ground loop and ground thermal conductivity. Ground loop metadata provided by installers is shown in Table **2–10**.

Ground loop Metadata	Number out of 173	Comment
Borehole Horizontal Slinky Total	7 33 12 52	
Pump power (Watts)	54	Range 45 – 415 W
Speed setting	88	3 @ setting 2 85 @ setting 3
Thermal conductivity (W/mK)	38 9[a] 1[a]	Range 0.6 – 3.5 Range 7 – 35 678

a Note that MIS 3005 (Table C.1) gives a recommended maximum ground thermal conductivity of 5.5 W/mK. Note that a decimal point in the right place would bring the 9 values of thermal conductivity within the range 7 to 35 back within the 5.5 W/mK maximum.

Table 2–10 Ground loop metadata provided by installers

Site visits to 21 case studies indicated that not all installers in the trial provided the design details to clients. Few occupants could locate comprehensive design data for heat loss calculations, the radiator schedule or the underfloor heating coils. Of the 10 GSHP case studies, none of the occupants could locate MIS 3005 Table 3, "Details of Ground Heat Exchanger design to be provided to the customer".

MIS 3005 v3.0 (05/09/2011) of Clause 4.2.18, introduced the requirement that "the overall system pumping power at the lowest operating temperature is less than 2.5% of the heat pump heating capacity". This was changed to 3% in MIS 3005 v3.2 (22/07/2013), although it would appear from MCS certificates that no GSHP systems in the monitored sample were installed after August 2013. We would therefore expect the significant numbers of RHPP GSHPs to be designed to the 2.5% standard. It should be noted that ground loop circulation pumps were not separately metered to show the actual pump power or whether these pumps complied with EU Ecodesign of water pumps criteria[23]. The following analysis is based on RHPP metadata supplied by DECC.

Metadata that refers to ground loop pump power is provided in two columns entitled:

- "Rated power of ground loop circulation pump (Watts)"
- "Ground loop pump power (W)".

[23] Commission Regulation (EU) No 547/2012 of 25 June 2012 implementing Directive 2009/125/EC of the European Parliament and of the Council with regard to ecodesign requirements for water pumps.

Metadata includes 174 GSHPs with "Installer net capacity" provided for 165 units. As discussed in Section 2.6, RAPID-HPC has determined that this generally refers to the manufacturers' BS EN 14511 power output rather than the rated power at design conditions. Rated power (expressed in Watts) is provided for 98 sites against 96 with Installer net capacity. For these 96 sites, 58 (60%) exceed the 2.5% design criterion.

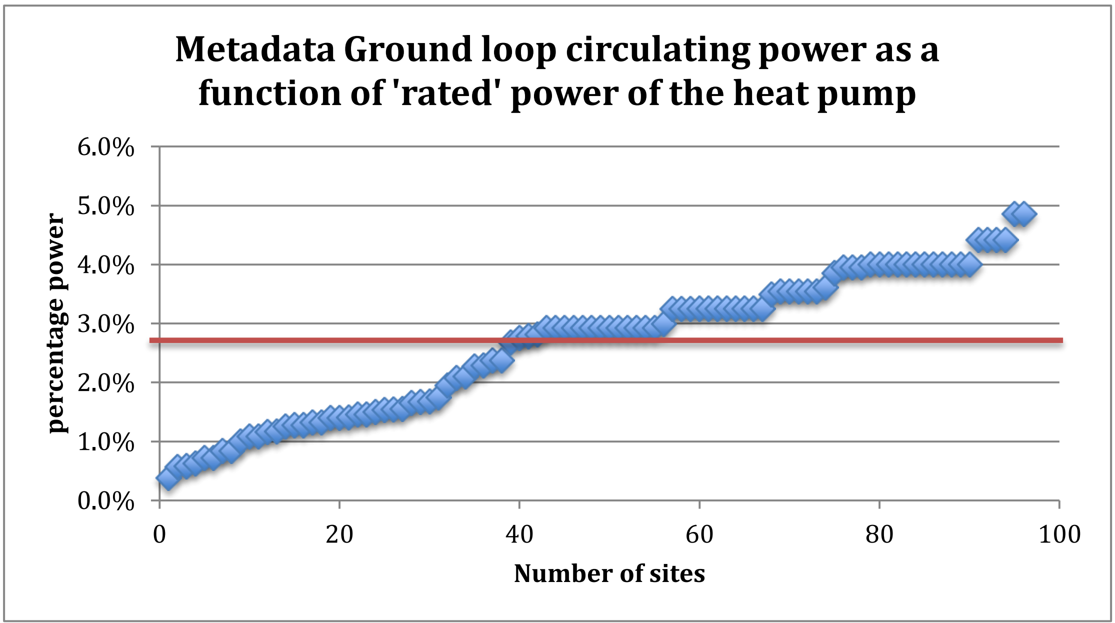

Figure 2-8 "Rated" ground loop circulating pump power as a percentage of heat pump power

A similar picture emerges with metadata for "Ground loop pump power". Counting only those values expressed in Watts, "Ground loop pump power" is provided for 99 sites. However, 45 of these sites are for ASHPs. The remaining 54 GSHP sites also provide Installer net capacity. Of these installations, 39 sites (72%) exceed the 2.5% design criterion.

Since ground loop circulation pumps were not separately metered, the pump power data for Table 2–8 is likely to be the installers' interpretation of manufacturers' nominal pump power. A more realistic approximation of circulation pump power can only be found through the analysis of circulating pump manufacturers' data at the intersection of flow rate and pressure head. Given this proviso, we conclude from the metadata, based on the restricted set of values provided by installers, that 60 – 70% of ground loop circulation pumps exceeded the 2.5% power design criterion.

Since many manufacturers supply GSHPs with a built-in ground loop pump, the designer needs to specify the ground loop index circuit in conjunction with the evaporator pressure drop, that is, the residual pressure available. As illustrated in Figure **2**-**9**, if for example the brine flow rate is 1.5 m^3/h, then the evaporator pressure drop alone is approximately 12,000 Pa (1.2 kPa) or about 1.2 metres head for a typically 5, 6 or 7 metre head brine pump.

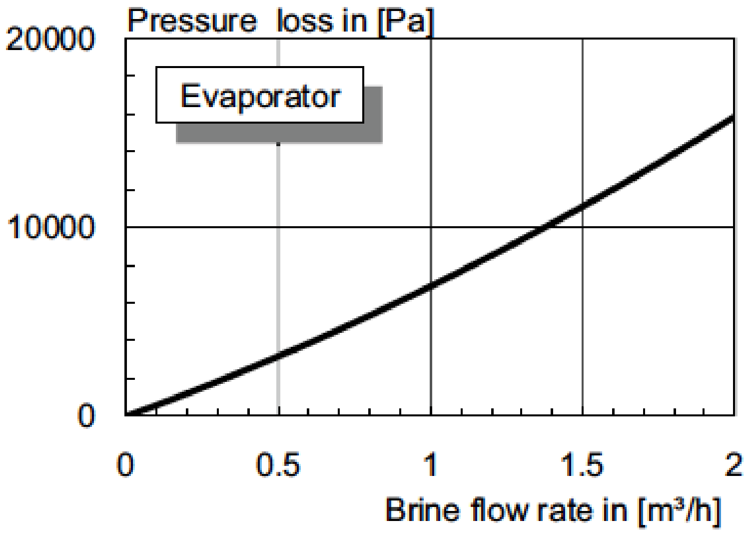

Figure 2-9 Example evaporator pressure drop at design brine flow rate in the ground loop (for GSHPs)

MIS 3005 Clause 4.2.15(j) also requires that: "the Reynolds number of the thermal transfer fluid in the ground heat exchanger active elements should be ≥ 2500 at all times", and therefore achieve turbulent flow. Turbulence breaks up the stagnant 'boundary layer' associated with laminar flow and improves heat transfer from the pipe wall into the circulating fluid, but at the cost of increased frictional resistance and therefore pumping power. The Reynold's number is given by $Re = \rho Vd/\mu$, (where ρ is density, V velocity, d pipe diameter and μ viscosity). Both viscosity and density are dependent on the "brine" water/antifreeze mix. Such calculations are ideally suited to ground loop design software. This combination of maximum power for a mass flow rate, balanced against friction head at minimum Reynolds number requires in practice an iterative ground loop design sizing process. Again with the proviso that installers are likely to have provided the badged nominal pump power and not operating power as commissioned, the graphical analysis in Figure 2-8 suggests that pump power as a function of ground loop design was poorly understood by some installers at the time of the RHPP installations.

It probable that there is a split between those 'larger' GSHP organisations who have the technical capacity and may also use ground loop software that will design for heat transfer at any Reynolds number flow regime, and thus meet design requirements, and those organisations who do not install sufficient numbers to make such an investment.

Conclusions

We have estimated whether heat pumps are correctly sized by extrapolating heat power to cold weather conditions. From analysis of metering error presented in the 'Performance Variations' report, there is a systematic over-estimate of heat by 4 - 7%, due to calibration for water, rather than a water/glycol mix. Furthermore, up to 16 of the larger sites may be affected by the 18 kW limit of the heat meters. Missing heat data is unlikely to have influenced the extrapolation to cold weather conditions. It is possible that a small number of sites were equipped with strap-on sensors, which would affect the estimated peak power for these sites, although most of these sites were excluded from the cropped B2 dataset.

The analysis of heat loss calculations shows that, even when using BS 12831, compliance is not an 'exact science' especially when retrofitting in existing housing with unknown U-values and ventilation rates. Radiator sizing has been shown to depend as much on the exigencies of property and client as on formal compliance with a single 'star rating' output. The RHPP metadata provides predominantly badged heat pump capacity as opposed to capacity at design conditions. Any analysis of installer ability to heat pump size is compromised by the lack of detailed dwelling heat loss characteristics such as floor area or year of build. Similarly, the lack of ground loop design calculations and limited metadata throws into question any quantitative conclusions regarding compliance. The step-change in design criteria iniated in MIS 3005 v3.0 demanded of installer companies the necessary calculation methodologies to ensure quality assurance of all MCS installations against a background of a largely unregulated design approach. For those companies specialising in heat pump installations, where design procedures were well understood and may already have been supported by specialist software, such changes were likely welcomed as recognition that heat performance is sensitive to design, installation and operation. For those companies installing the occasional heat pump the additional demands for MIS 3005 compliance could certainly have required investment in new knowledge and working practices. That the changes occurred during the trials complicates any assessment of their impact on RHPP performance since their effects are likely to require a period of adjustment. Thus a poor understanding of formal design criteria in 2012 does not imply a poor understanding in 2016.

3 How do actual space and water heating demand compare to the estimated figures on the MCS certificate and on the EPC?

Introduction

The installer is expected to assess annual energy performance (kWh/year) for contractual purposes. The calculation for space heating is based on degree day region, building form and fabric, occupant usage patterns and system efficiency. The calculation for energy required for domestic hot water is based on the likely daily hot water use, the maximum temperature achievable by heat pump only (and thus the likely DHW SPF) and any additional energy required for regular pasteurisation by a resistance heater to meet health and safety demands for legionella protection. The assessment of annual energy for MCS compliance may have been unfamiliar to many installers since it is not a demand for either gas or oil heating systems. Hence MIS 3005, as early as v1.2 (25/02/08), has provided an annual energy assessment method:

The means of estimating the annual energy performance is as follows:

a) Assess the annual heat load for the building (space heating or hot water) using any suitable performance calculation method. Such calculation method shall be clearly described and justified.

b) Multiply the result from a) by the proportion of the relevant heat load provided by the Microgeneration Heat Pump system as determined in accordance with Clause 4.2.1.

c) Divide the result from b) by the default efficiency (expressed as a Coefficient of Performance or CoP) for Heat Pumps contained in SAP 2005 Table 4a (note: CoPs corrected for Heat Pumps with auxiliary heaters should not be used).

d) Divide the result from c) by the appropriate efficiency adjustment contained in Table 4c of SAP 2005.

e) Calculate the energy supplied by the auxiliary heater by multiplying the result from a) by the proportion of the relevant heat load not supplied by the Heat Pump.

f) Add the result from d) to the result from e) to give the total energy required for the relevant heat load.

g) The results from f) for space heating and hot water are added together to give an overall energy requirement for the building for these heat loads.

We note that by dividing the annual space heating and hot water load by the “efficiency” the resulting annual energy performance is the electrical “energy required”, from which the installer may directly estimate annual running costs.

The Matthews webinars on the MCS website simplify this process with a spreadsheet containing embedded equations linked to room heat loss sheets. The designer is required to do little more than enter the flow temperatures for space heating and DHW along with the number of occupants (to calculate the daily hot water demand), the "Final HP secondary HW temperature" and the sterilisation/pasteurisation cycle (set for daily).

The calculation of annual heat demand therefore requires the designer to use the spreadsheet where heat loss calculations (Watts) are summed for each room and converted to annual kWh. The final kWh total is clearly dependent on the assessment of U-values and ventilation rates (see Section 2.0) along with degree day region and likely SPF. We should note that the online spreadsheet is not based on room-by-room calculations but on a ground and first floor whole house method somewhat analogous to that in SAP. Adapting the spreadsheet for a room by room approach requires the user to sum each room sheet into the final annual kWh cell; a process that requires some knowledge of spreadsheet manipulation.

Before the publication of MIS 3005 v3.0, likely SPF could have been calculated from either SAP 2005 Table 4a or from manufacturers' data where available. Few manufacturers published sufficient BS EN 14511 test results across a range of outdoor temperatures and flow temperatures to assess SPF for space and DHW heating although such data may have been made available to installers. The introduction of the HEG provided the 'likely space heating SPF' and a source for hot water SPF based on the hot water flow temperature to the cylinder.

For calculation of DHW energy, Matthews states that it should be assumed that the actual cylinder temperature will be 5K lower than the maximum primary flow temperature and that the difference between the cylinder temperature and the minimum for legionella protection of 60°C is made up through electric resistance heating. Matthews provides an equation that generates both the space heating and hot water annual energy. The sum of these numbers is then entered by the installer in the MCS certificate. MCS certificates require the installer to provide "Estimated Annual Generation (kWh)"[24]. We take this to mean heat pump annual heat output although we note that there are certainly cases where installers appear to have entered MIS3005 "annual energy performance". RAPID-HPC have been unable to find MCS/Gemserve documentation from the time of the trials that explains to installers what must be provided under Estimated Annual Generation (kWh).

RHPP annual energy assessment

Analysis of heat pump installers' estimates from MCS Certificates of annual energy delivered versus actual annual heat energy output by dwelling type (RSL or private) is shown for ALL heat pumps, Figure 3-1; for GSHPs,

[24] MIS 3005 v4.0 (16/12/2013) introduced a URL link for the MCS Compliance Certificate . .

Figure **3-2**; and ASHPs, Figure 3-3. An ideal mapping of estimated to actual is shown by the black diagonal lines in the figures. Sites where obvious metering errors occurred have been removed. For the GSHP in the RHPP trial, the actual heat delivered during 2013-14 is lower than the installer's estimates, particularly for houses with expected heat demands of 20,000 kWh or more. For ASHPs, again, the actual heat delivered during the trial is lower than would be expected from the installers' estimates. Scatter increases as expected heat demand increases.

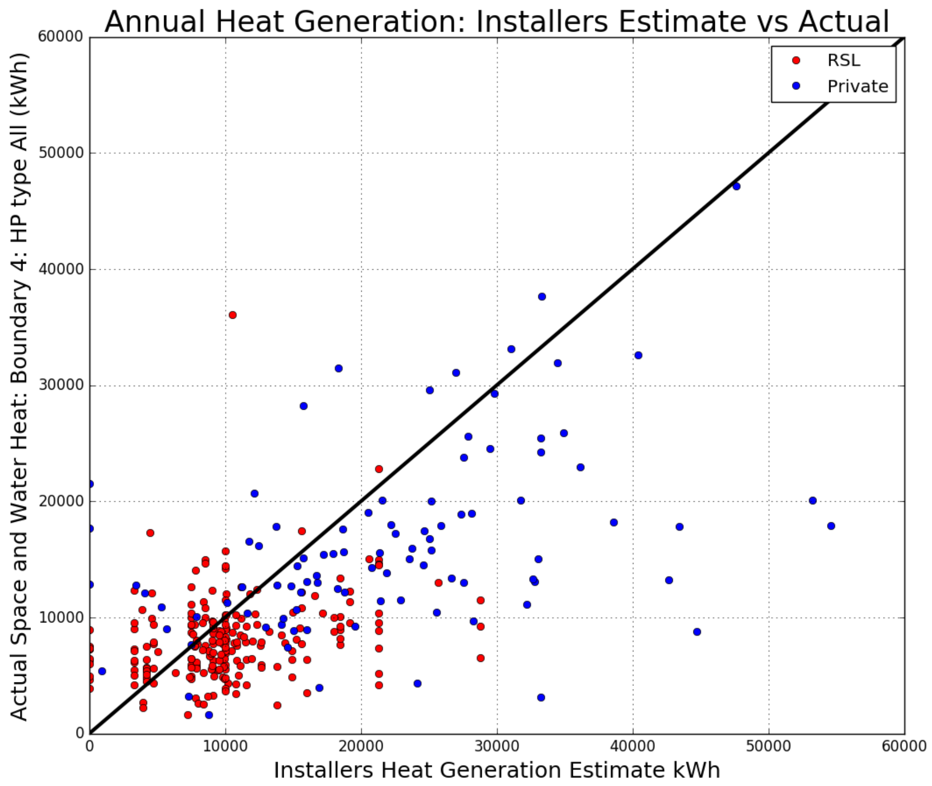

Figure 3-1 Installers' estimate of annual heat demand versus actual heat generated for all heat pumps

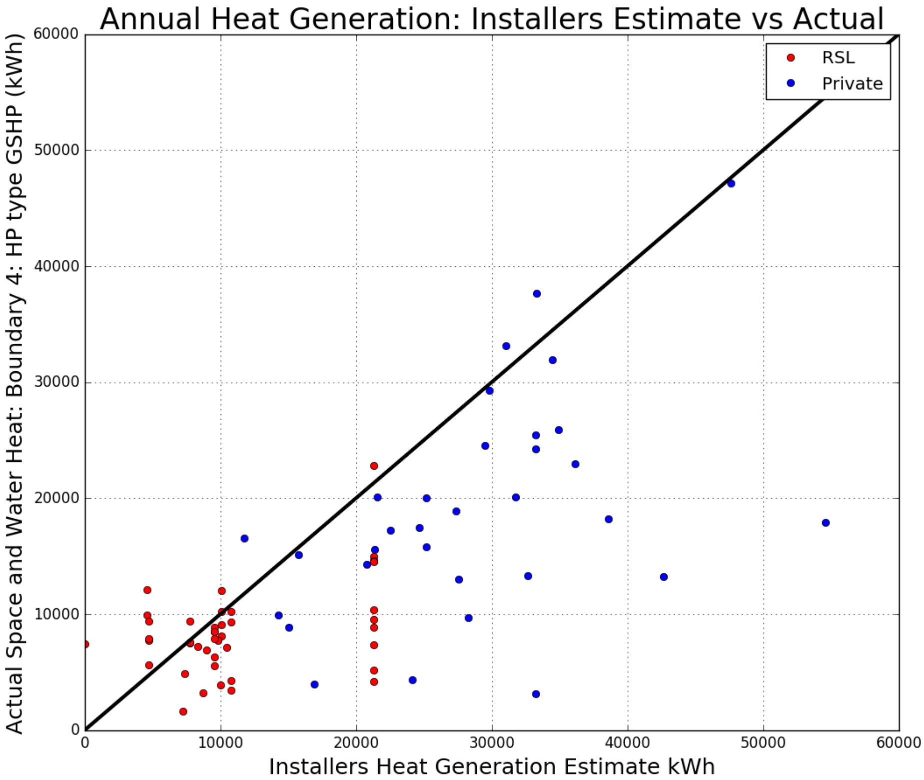

Figure 3-2 Installers' estimate of annual heat demand versus actual heat demand - GSHPs

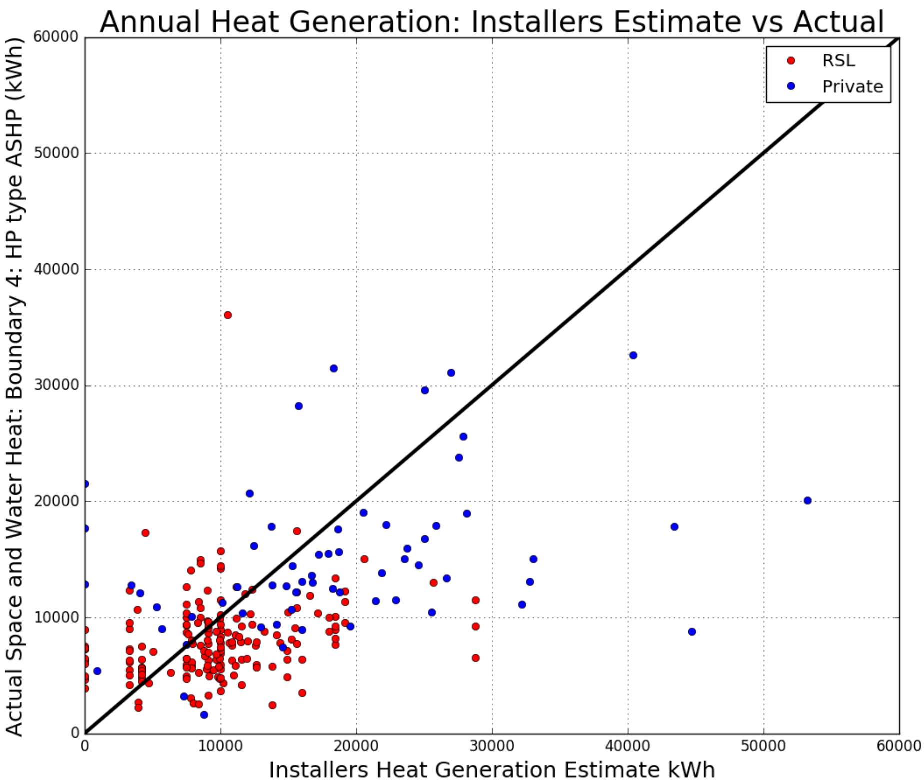

Figure 3-3 Installers' estimate of annual heat demand versus actual annual heat demand ASHPs

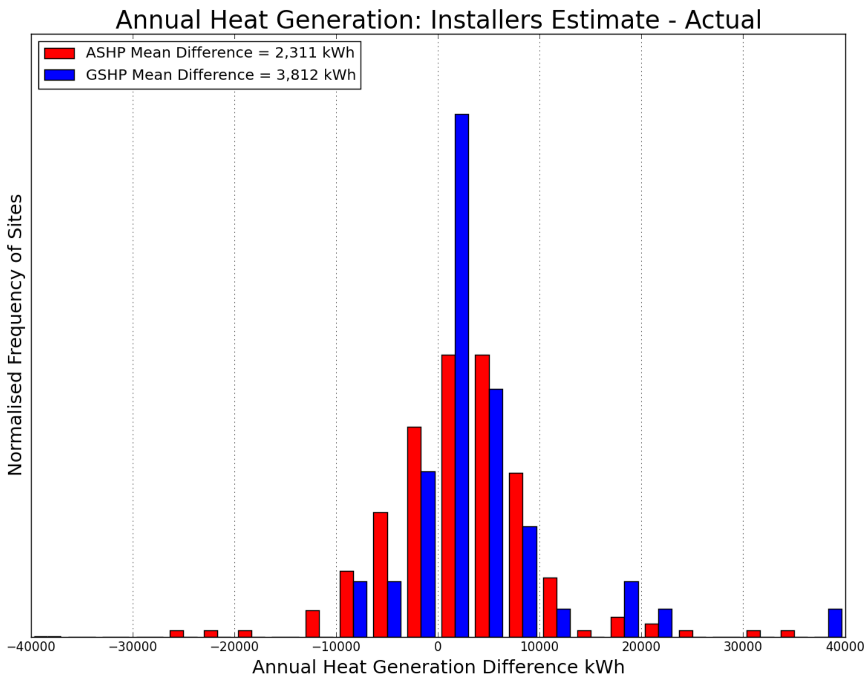

Figure 3-4 Difference between MCS predicted and actual heat energy for ASHPs and GSHPs

EPC annual energy assessment

The RHPP trial required an energy performance certificate (EPC) to be provided as part of the quality control process. EPCs provide an assessment of annual energy demand based on either SAP for new build or the reduced version RDSAP for existing properties. BEIS have provided RAPID-HPC with EPCs for the RHPP sites in order to compare EPC estimated heat demand with actual heat demand and to compare EPC demand with MCS demand. A series of scatter plots illustrating these relationships are provided below. Figure **3-5** illustrates EPC predicted space heating for all heat pumps with measured heat output and shows quite reasonable mapping up to 10,000 kWh/year, but a tendency to over-estimate demand above this level. It is possible to speculate that that this results from technical inaccuracies such as from solid wall U-values (Li, et al, 2014) or from social issues associated with heating poorly insulated or larger houses. It may be that larger houses provide occupants with more options for occupancy and heating patterns. We also note in passing that some EPC certificates record very low energy demand.

The EPC assessment of domestic hot water heat demand, Figure **3-6** is clustered around 2,500 to 3,000 kWh/year where measured demand ranges from almost none through to 15,000 kWh/year. Actual hot water use is highly dependent on number of occupants and presence of children in the household. Adding complexity is the role of showering versus bathing and, as observed in the RHPP case studies, the use of unmetered electric showers, which may have the effect of reducing the amount of heat supplied by the heat pump to provide for domestic hot water.

No significant difference is seen between EPC assessment of space heating demand for ASHPs and GSHPs, Figure **3-7** and Figure **3-8**, although the sample sizes are quite different.

What is interesting is the range of differences between MCS and EPC predictions for space heating and DHW for all heat pumps, Figure **3-9**. Some of these estimates vary by 5,000 kWh/year although less than 1,000 kWh/year is far more common and suggests reasonable mapping even though MCS estimates are based on different indoor/outdoor delta T and MIS 3005 supported SPF assumptions. Finally, Figure **3-10** and Figure **3-11** for ASHPs and GSHPs demonstrate that the observed range of MCS-EPC differences apply to both types of heat pump.

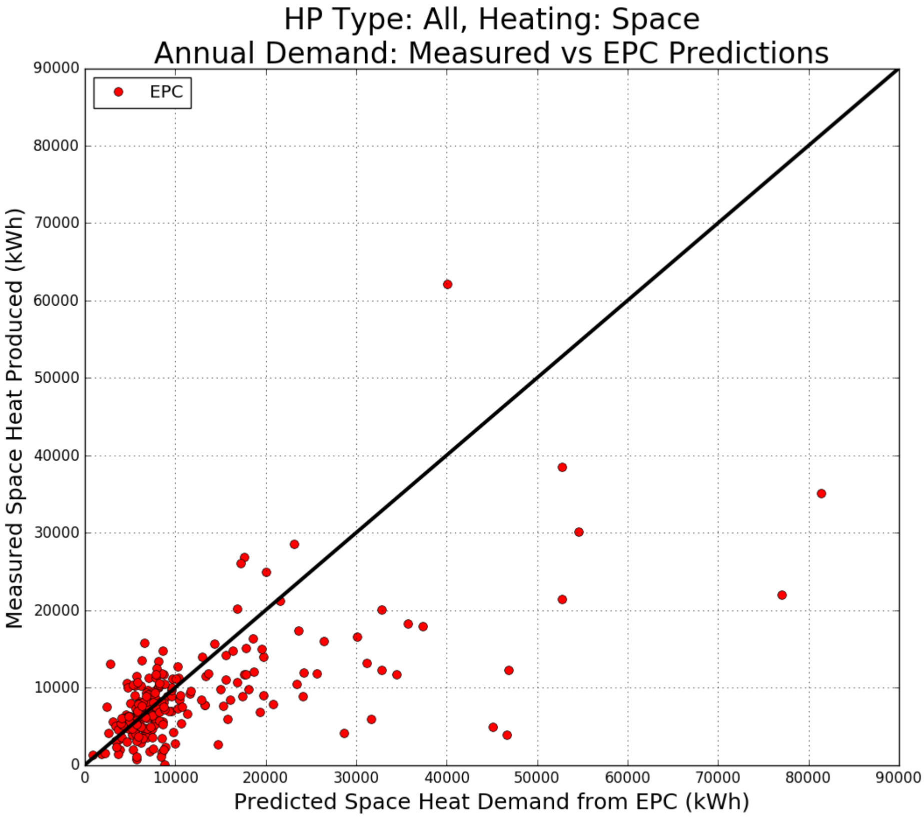

Figure 3-5 EPC predicted annual space heating demand versus measured for all heat pumps

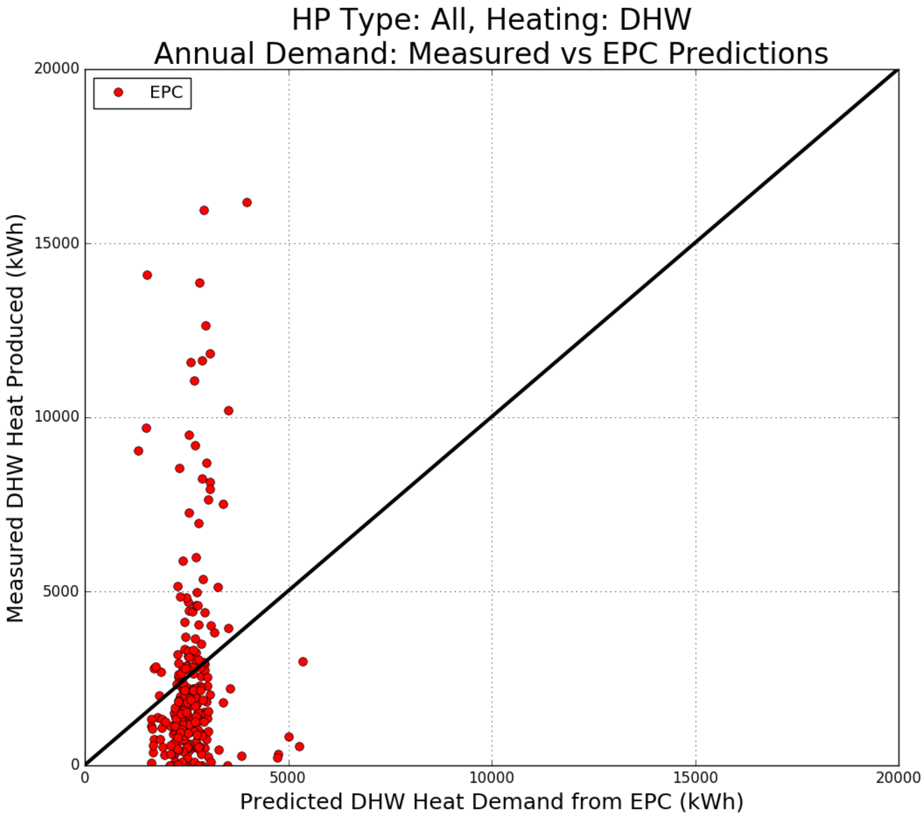

Figure 3-6 EPC predicted annual DHW heat demand versus measured for all heat pumps

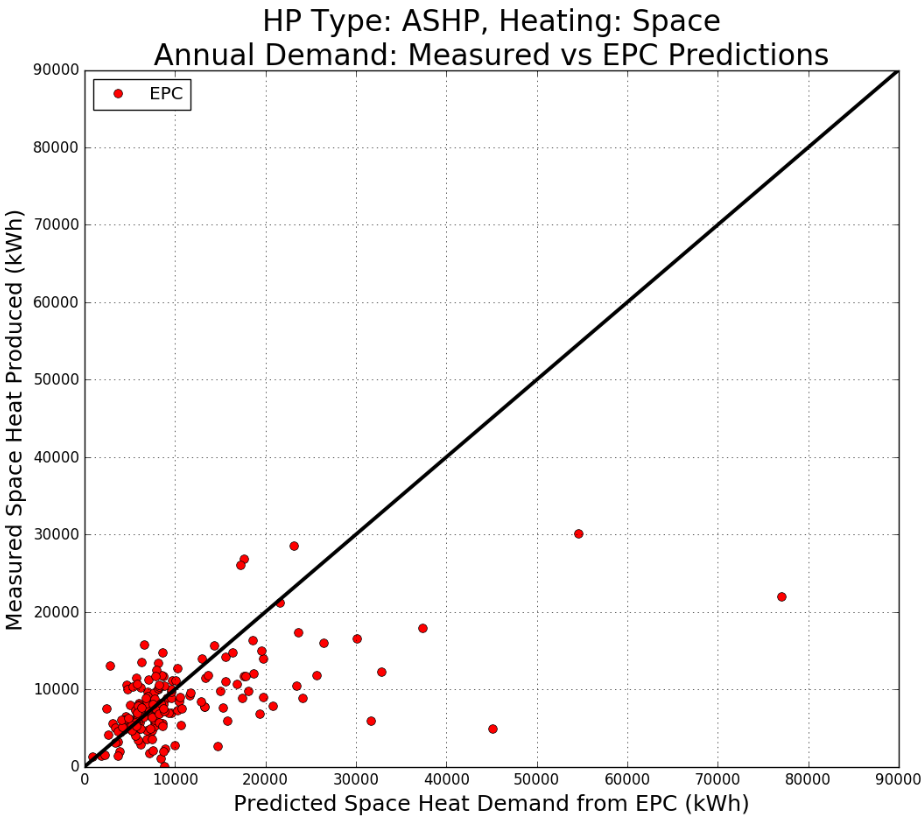

Figure 3-7 EPC predicted ASHP annual space heating heat demand versus measured

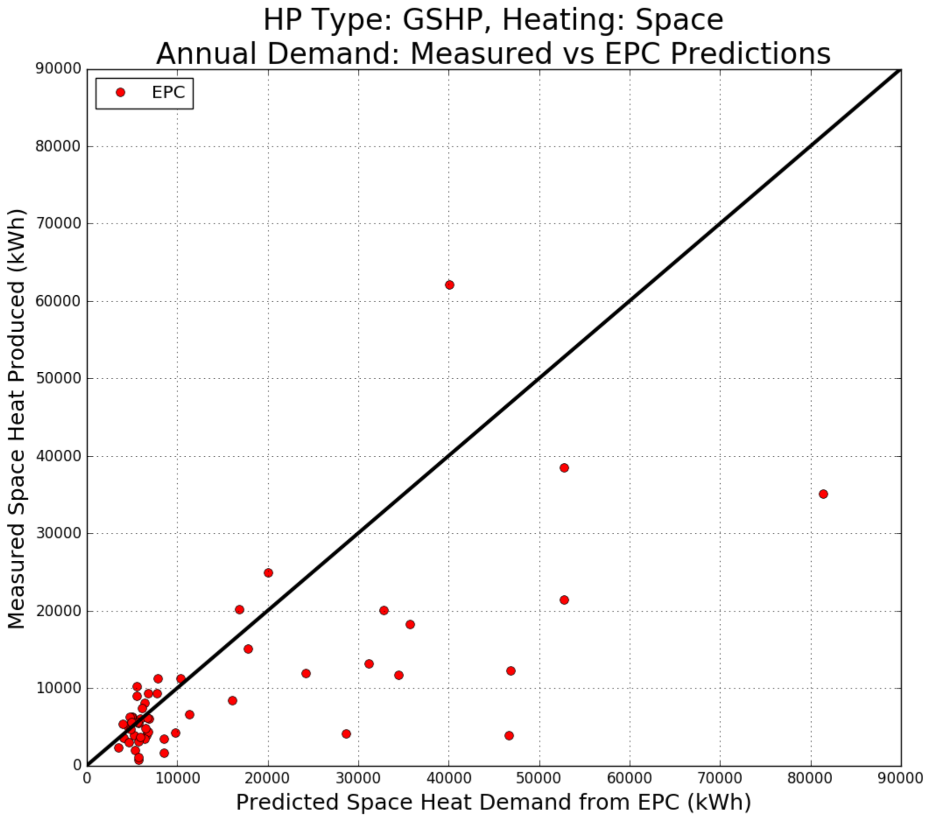

Figure 3-8 EPC predicted GSHP annual space heating heat demand versus measured

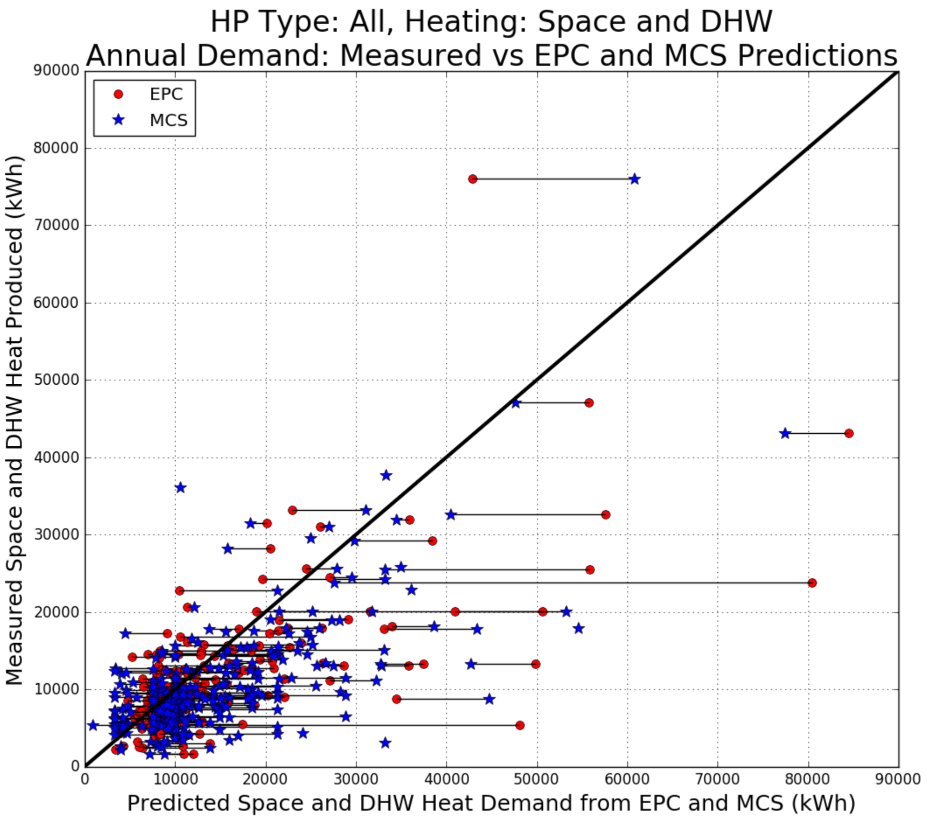

Figure 3-9 EPC and MCS predicted total heat demand versus actual (same sites joined).

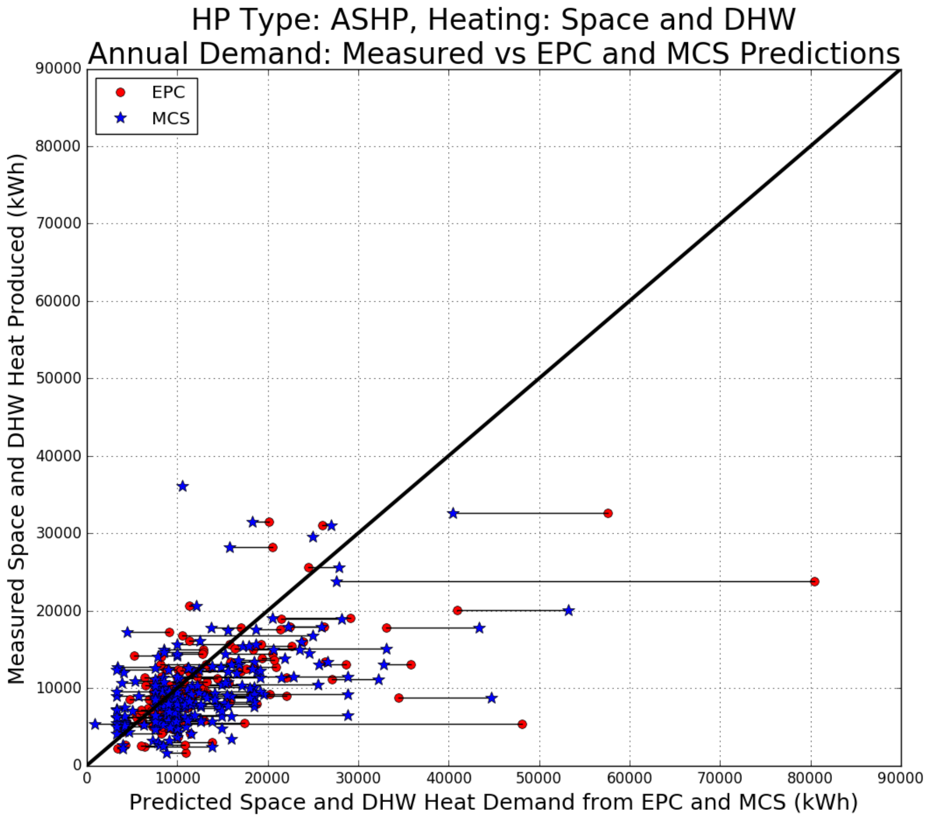

Figure 3-10 EPC and MCS predicted ASHP total heat demand versus actual (same sites joined)

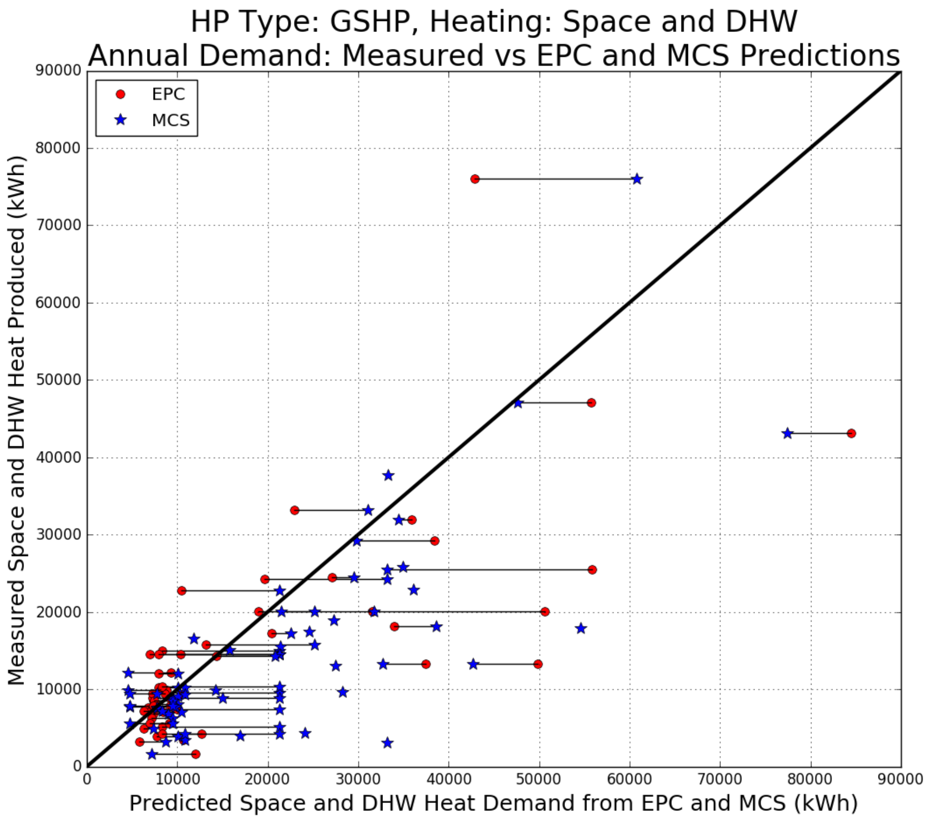

Figure 3-11 EPC and MCS predicted GSHP total heat demand versus actual (same sites joined)

When reviewing Figures 3-5 to 3-11 it is worth considering the literature regarding EPCs and in particular the variation in assessment between individual assessors (DECC, 2014b: iii):

> "There was a lack of consistency in the data, results and advice generated by different assessors for the same property. There was significant variation in the EPC and OA [Occupancy Assessment] results produced by the different assessments conducted at individual properties. The range of EPC ratings spanned at least two EPC bands for almost two thirds of the dwellings analysed. The analysis found many differences in the values recorded for key input variables at the same property. Input variation was observed with EPCs, particularly for total floor area and the energy efficiency rating of building fabric and technologies. The variation in the inputs to the EPC process contributed to the EPC rating varying by, on average, 11 points in each dwelling."

Conclusion

Measured annual heat generated vs EPC and installers' estimates: individual sites may have been affected by small amounts of missing heat data. There is a systematic over-estimate of heat due to calibration of sensors for water rather than glycol. This does not influence the conclusion that, for the period of data examined, the EPC and installers' estimates of annual heat demand are higher than the measured heat produced.

The comparison of actual energy use with its calculated value, whether through MIS 3005 or EPC procedures, is complicated by a range of factors that include the actual annual weather conditions, real as opposed to 'likely' heating system efficiency, occupation pattern and comfort requirements. The graphical analysis of both MIS and EPC estimates indicates a range of variables that are currently poorly expressed in the calculations. Even for those installer companies that fully understand the role of likely SPF and annual degree days, it is unlikely that the calculated results will provide anything other than a broad estimate of annual energy use.

4 Do systems sterilise the hot water tank for legionella control? If so, how often?

Sterilisation

The maximum outlet temperature of a heat pump is fixed by the manufacturer to ensure safe operation and a minimum vapour compression cycle efficiency at high temperatures and pressures. For RHPP heat pumps this temperature ranges between 50 and 65°C and is dependent on the specific refrigerant mix. For heat pumps with maximum condensing temperatures at or below 60°C, the bulk temperature (the mean or overall temperature) of the hot water cylinder will be below 60°C for a variety of reasons including cylinder stratification. Under these circumstances the stored hot water may fail to meet HSE guidance on the protection from Legionnaires disease or legionella. A requirement for sterilisation of DHW cylinders (or pasteurization as it is sometimes called) was introduced in MIS 3005 v 3.0 Section 4.2.5: "domestic hot water systems shall incorporate a means to prevent bacterial growth (including Legionella bacteria). NOTE: Further guidance can be found within the Health and Safety Executive Approved Code of Practice L8 document (HSE ACoP L8)".

ACoP L8 is supported by HSG274 Part 2 which cites temperature as the most common method of Legionella control: "It is recommended that hot water should be stored at 60°C and distributed so that it reaches a temperature of 50°C within one minute at outlets." However, this guidance is aimed primarily at commercial hot water systems with either secondary returns (a pumped circulation pipe loop returning hot water to the cylinder for continuous reheating), or trace heating (an electrical resistance tape attached to the pipe where reheat is achieved by current flow in response to change in electrical resistance due to any drop in temperature). It therefore should be noted that all HSG & ACoP's refer to non-domestic situations and very little specific advice is available for domestic applications.

Whilst secondary returns were observed in the trial case studies, most hot water systems comply with water regulations, and therefore indirectly with HSG274, by limiting the length of hot water draw off pipework or 'deadlegs'. Since under these conditions it is expected that outlet temperatures will reach cylinder temperature within one minute, Legionella control is primarily by regular high temperature sterilisation.

Sterilisation or pasteurisation is achieved by raising the cylinder water to over 60°C. HSG274 (which refers to commercial installations) states that: "Arrangements should therefore be made to heat the whole water content of the calorifier, including that at the base, to a temperature of 60°C for one hour each day." However, a review of heat pump manufacturers' literature identifies varying guidance on the sterilisation process with temperatures ranging from 65°C to 73°C for between 30 and 90 minutes either weekly or fortnightly.

It should also be noted that the MCS online training spreadsheets, produced for DECC training purposes, provide energy calculations based on daily sterilisation, although none of the reviewed manufacturers' literature suggests such a procedure.

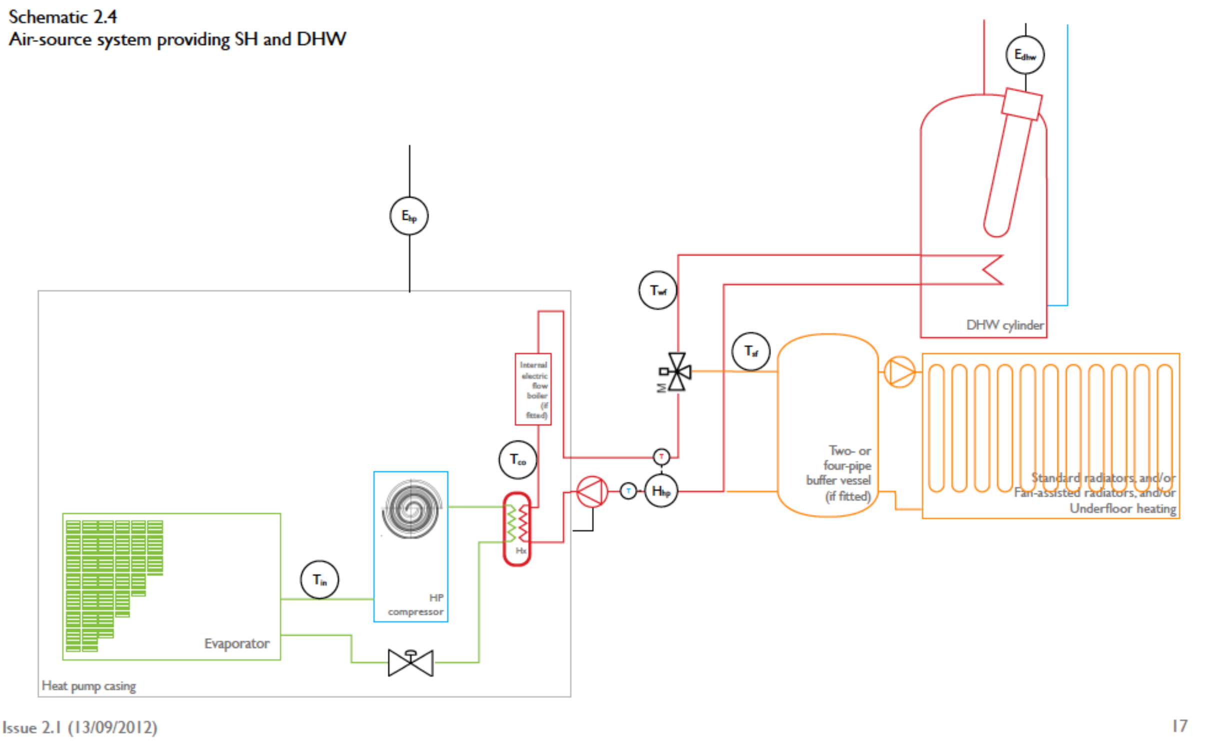

Figure 4-1 Separate DHW cylinder with metered domestic hot water immersion

The MCS online training is focused on heat pumps that will not meet the 60°C minimum cylinder temperature. Maximum flow temperature is provided by a number of manufacturers where, for example, those units using R410a range from 50 to 60°C and those with R407c and R134a range between 60 and 65°C. Therefore it is entirely possible that a separate sterilisation procedure is not required where installer companies can guarantee the appropriate cylinder temperature. What follows from this observation is that the existence of an immersion heater does not necessarily imply the need for sterilisation.

For systems requiring sterilisation and utilising a separate cylinder with immersion heater, Figure **4-1**, the installer must design and commission a control system with a specific sterilisation function. Setting the immersion heater thermostat for 60-65°C will cause the immersion to operate continuously where the immersion thermostat is located in water below this temperature. In order to provide a more effective sterilisation function, immersion operation must therefore be timed, and ideally, coordinated with the operation of the heat pump so that only the final 5-15 degrees of temperature lift are achieved through the use of resistance heat. However, the immersion also acts as a backup for hot water when fast heat up is required at times of high hot water usage. The immersion may also act as a backup system providing the occupant with hot water when the heat pump, for whatever reason, is not working at all. To provide this functionality, the installer must select a programmer/timer with override switching and set the time period for regular sterilisation at the appropriate temperature. It is worth noting that whilst such timers are available, they are not required for either gas or oil fired central heating where cylinder temperatures above 60°C may be achieved simply through setting the boiler thermostat; the immersion operates (normally at 60-65°C) only when required by the occupants. Heat pump installations may therefore exhibit additional complexity with respect to DHW controls and their commissioning.

Recognising the potential for an automated legionella function, some manufacturers provide a controller with these functions, either with factory default sterilisation, generally either weekly or fortnightly operation, or as an option to be set during commissioning.

Analysis methodology

Cylinder temperatures were not monitored during the RHPP trials so analysis of sterilisation cannot be by direct cylinder temperature measurement and must be inferred from immersion operating pattern. Additionally, an installation with an immersion heater does not necessarily imply the need for sterilisation. For example, for heat pumps capable of up to 65°C flow temperature the immersion provides a back up rather than sterilisation role. The analysis of sterilisation for this report is based on the following understanding. Sterilisation of the hot water cylinder contents by heating the water to high temperatures can be achieved in a number of ways:

- The heat pump could provide all the heat.
- The heat pump could start heating until it reaches its maximum temperature limit and either an integrated booster through-flow heater or an immersion heater could be used to bring the water temperature up to the required level.
- The heat pump may not be used at all and the through-flow booster heater or the immersion heater could be used.

In this section we have focused on the use of the separately metered immersion heater as this has a very clear signal in the data. A regular and prolonged signal from the electrical meter to the immersion heater may be assumed to represent sterilisation.

The immersion heater electricity use is recorded as Edhw every 2 minutes in Watthours (Wh). During a sterilisation cycle, the control system will maintain the cylinder at the required temperature for an extended period of time (up to an hour)[25]. For a lot of systems these intense heating episodes appear at regular intervals, depending on how the systems were commissioned and operated. To detect a signal of a high intensity heating episode, the data was scanned for any periods where the Edhw electricity use in a 1 hour period exceeded a threshold of 1.0 kW averaged over the hour[26,27]. This threshold level was chosen

[25] For well insulated cylinders, with temperature decay rates of 1K per hour or less, it is unlikely that there would be any cycling, once pasteurisation temperature had been reached.

[26] Edhw was summed as 30 x 2min Wh samples in each hour. If that was more than 1000 Wh or 1kWh it was taken as a high intensity episode. How this was distributed within the hour was not evaluated so, at one extreme, there could have been a 30 kW immersion working for the first 2 minutes and then nothing else, or at the other, a 1kW immersion working continuously for an hour. In this context the term "power" could be misleading. It is the average power over the hour.

as it is above the normal hourly use of the immersion heater in most systems where the cylinder is heated to a lower temperature and stops once the set temperature has been reached. A histogram is then made of the time interval in days between successive high intensity heating episodes. The peak frequency is obvious in most cases where a regular schedule has been used. When sterilisation is not occurring via the immersion heater or not at all the histogram shows a large peak at zero time interval.

Analysis results

The immersion heater frequency for each site is plotted showing a range of schedules in the sample, Figure **4-2**.

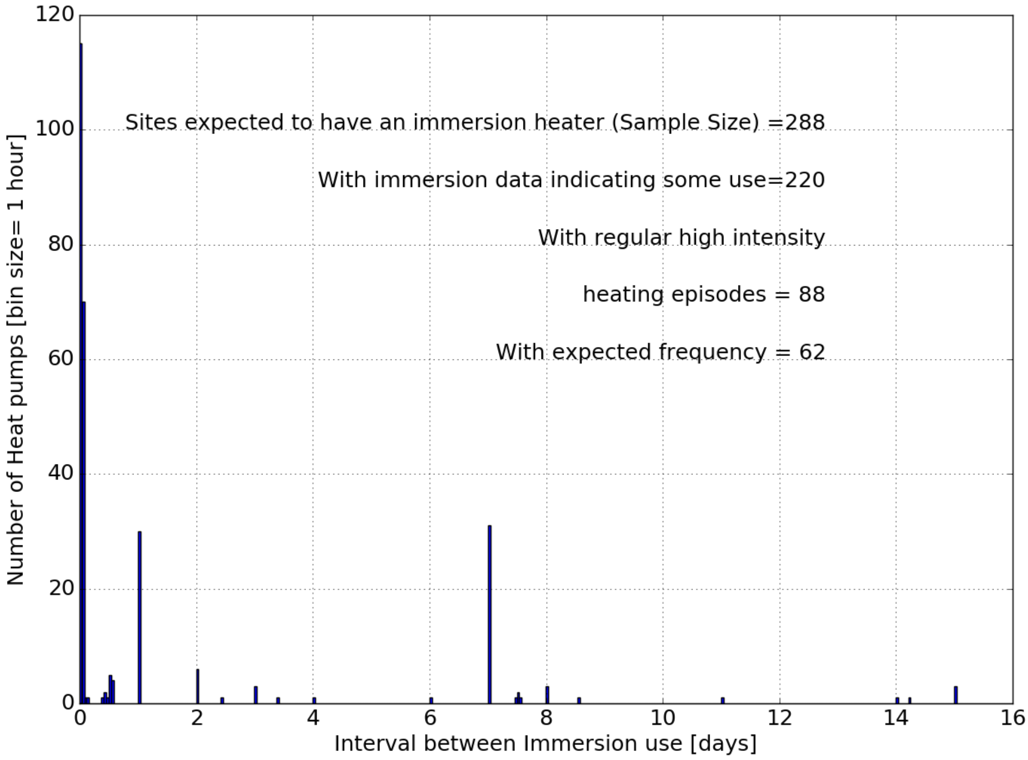

Figure 4-2 Assessment of apparent immersion frequency

Of the 417 heat pumps in Sample B2, monitoring schematics indicate that 288 should have monitored DHW immersion. However, the data showed that only 220 of these sites actually used the immersion (ie the sum of their immersion electricity consumption > 0). To these 220 sites, an algorithm was applied to determine whether their immersion use followed a regular pattern, which may indicate that it was related to sterilisation, and if there was a regular pattern, what its frequency was. 60% of sites with immersion showed no pattern (therefore showing sporadic immersion use instead). These are plotted at x = 0 on Figure 4.2. 40% of sites with immersion showed a regular pattern although sterilisation at these intervals appears unlikely with intervals such as 3, 8 and 15 days. A sterilisation regime would be expected to occur

[27] This is sufficient to raise the temperature of 120 litres of water by approximately 7K, enough to achieve a temperature of over 60C for a cylinder in which a temperature of 55C had already been achieved by the use of the heat pump.

either daily, weekly or fortnightly. The analysis shows that 28% of sites showed this 'expected frequency' of use.

Conclusion

Sterilisation analysis is based on electricity data. Electricity monitoring is generally of high quality; we do not consider that metering errors will have had any effect on the conclusion that between ⅓ and ¼ of the sites examined show regular patterns of immersion.

A protection regime against the build up of legionella bacteria is only required where cylinder temperatures are below 60°C. Importantly, since DHW temperature was not monitored, compliance with HSE guidance cannot be confirmed and must necessarily be inferred by such methods as time-series data analysis. RAPID-HPC recognize the limitations of such an approach and suggest that the assessment of compliance be taken as provisional. Where we expect legionella sterilisation frequency to be daily, weekly or fortnightly, we conclude that between a quarter and one third of RHPP installations with a metered immersion heater exhibit a pattern of operation broadly consistent with MIS 3005 legionella criteria.

5 What are the actual flow temperatures at the MCS 3005 design temperature conditions? What Heat Emitter Guide[28] star rating would apply, based on the actual flow temperatures?

Introduction

The installation 'star rating' and 'likely space heating SPF' would have been assessed from the planned design flow temperature to radiators and underfloor heating but will depend on the individual heat pump's control type and settings. Heat pumps are controlled by a thermistor attached to the flow or to the return. For systems controlled on return temperature, a low return temperature implies a high heat flow from the radiators due to significant temperature difference between radiators and room air temperature, whereas a high return indicates that the room is at or near design temperature. Variable compressor heat pumps are able to adjust compression, and thus indirectly control flow temperature from the condenser heat exchanger, to provide space heating water at the design flow temperature. However for fixed speed units, where the water is either below or above set point, control is limited to on/off. Final room control is commonly by thermostatic radiator valve or room thermostat. The rate of heat transfer from the condenser to the heating circuit is dependent on condenser temperature, the return temperature and mass flow rate. For return temperature control, this results in uncontrolled flow temperatures which may rise to the maximum heat pump output temperature with implications for Carnot efficiency. Such systems often allow the installer to set a maximum flow temperature to ensure a minimum SPF and compliance with HEG design criteria. Thus it is common for heat pumps to provide the option for two maximum set points, one for domestic hot water - controlled in conjunction with the cylinder thermostat signal, and a lower setting for space heating controlled through the room thermostat signal.

Some manufacturers use a 'degree minute' control function that integrates the temperature difference between return temperature and its set point. At a pre-specified negative degree-minute summation an electrical heater may be operated to provide a boost to the water temperature to speed up the heating process. Many such systems also have a default maximum output temperature (such as 60°C) reflecting the heat pump's maximum temperature output which the installer should reset to reflect their design star rating and their choice of HEG likely SPF.

In addition, many manufacturers offer weather compensation control where space heating water temperature is adjusted in response to outdoor temperature resulting in a 'heating curve' that adjusts the flow temperature. As outdoor temperatures rise, the building requires less heat to achieve room temperatures and emitter water temperature is reduced resulting, in theory, in a higher Carnot efficiency.

[28] MCS 021: http://www.microgenerationcertification.org/mcs-standards/installer-standards

Maximum flow temperature must be set by the installer during commissioning to reflect the star rating flow temperature at minimum outdoor design temperature. Only if this has been achieved will maximum flow temperatures monitored during the trials reflect the design star rating of the installation. We note that the installer must have absolute confidence in their system design to reduce the flow temperature since higher flow temperatures will compensate for any potentially undersized emitters or client demand for higher temperatures. It is also possible that re-setting complex heat pump control menus may be either off-putting or require the support of a specialist commissioning technician.

Maximum temperature assessment

An algorithm has been developed to identify the maximum flow temperature in cold weather. Since design outdoor temperature is relatively infrequent, the coldest month has been selected and maximum flow temperature regressed to a daily average outdoor temperature of 0°C as a proxy for outdoor design temperature. In order to assess the maximum flow temperature at close to design conditions, an algorithm was developed based the following procedure, the results are shown in Figure **5**-**1**:

- For each site find the coldest external temperature reading based on NCEP weather data based on a 10km x 10km grid.
- For every day in the calendar month with the coldest external temperature reading, calculate the average external temperature and get the 99th percentile flow temperature value for each site. These values are plotted in the lefthand panel. There are therefore approximately 30 points per site.
- Fit a straight line and extrapolate down to zero degrees (0°C) daily average external temperature to provide the expected peak flow temperature
- We consider that the peak flow temperature at an average daily temperature of 0°C will occur when the hourly external temperature is comparable to, or lower than, the design temperatures for maximum flow rate in MCS (-0.2°C for Plymouth, -3.9°C for Glasgow). Therefore, extrapolation of peak flow temperature to a daily average temperature of 0°C will provide a rough estimate of the peak flow temperature at the MCS design temperature.
- The straight line fit is sensitive to uncertainties in the flow temperature readings and the daily average external temperature estimates. If some of the data has been corrupted it is possible for the gradient to be miscalculated as zero, highly negative or even positive. The latter case would mean that heat output increases as external temperature rises which would be un-realistic. Those sites where a negative gradient is not returned or with a gradient less than -1°C/°C are given a peak space heating temperature output equivalent to the maximum daily value of the coldest month of those sites.
- The frequency of maximum flow temperature and its corresponding star rating are shown on the righthand side.

The results indicate a wide range of installation maximum flow temperatures for all installations, Figure **5-1**, with a mean peak flow temperature of 43.6°C for 189 sites. Of these 189 installations, only 26 have a star rating (14%) provided in the metadata. Where metadata does provide star ratings, Figure **5-2** shows these installer star ratings mapped against maximum flow temperature calculated from the data for a daily average temperature of zero degrees. 'Star rating unknown' has been used to designate installations where predicted star rating is missing from the metadata. Whilst only 14% of the sites provide star rating, the results indicate a low confidence in installer star rating assessment[29]. Separate derived peak flow temperatures for radiator and underfloor systems are shown in Figure **5-3** and Figure **5-4**.

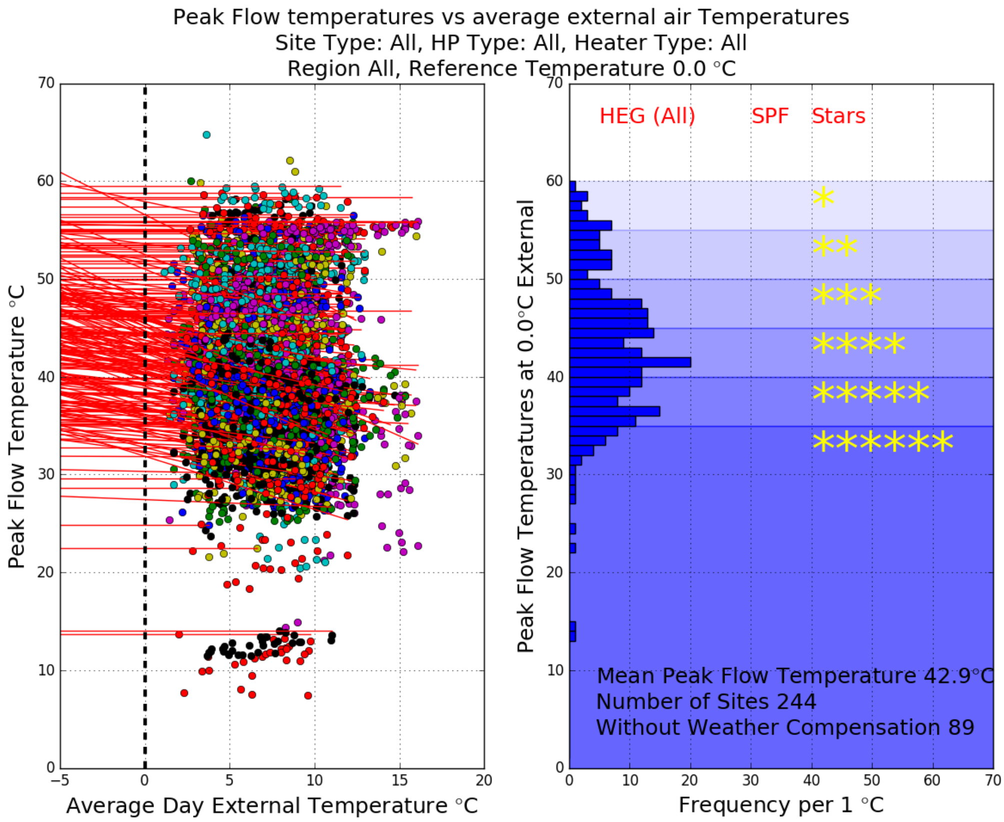

Figure 5-1 Derived peak flow temperature for all heat pumps and all emitters

[29] BS 12831 heat loss calculations do not consider internal heat gains. Emitter sizing and therefore output is based on maximum flow temperature necessary to ensure design conditions. For correctly commissioned weather compensation control, internal gains may mean that the estimated peak temperatures are lower than the design flow temperatures. This will impact on our assessment of design star rating from projected cold weather flow temperatures. This is one of many pragmatic simplifications that surrounds the assessment of the design of heating systems.

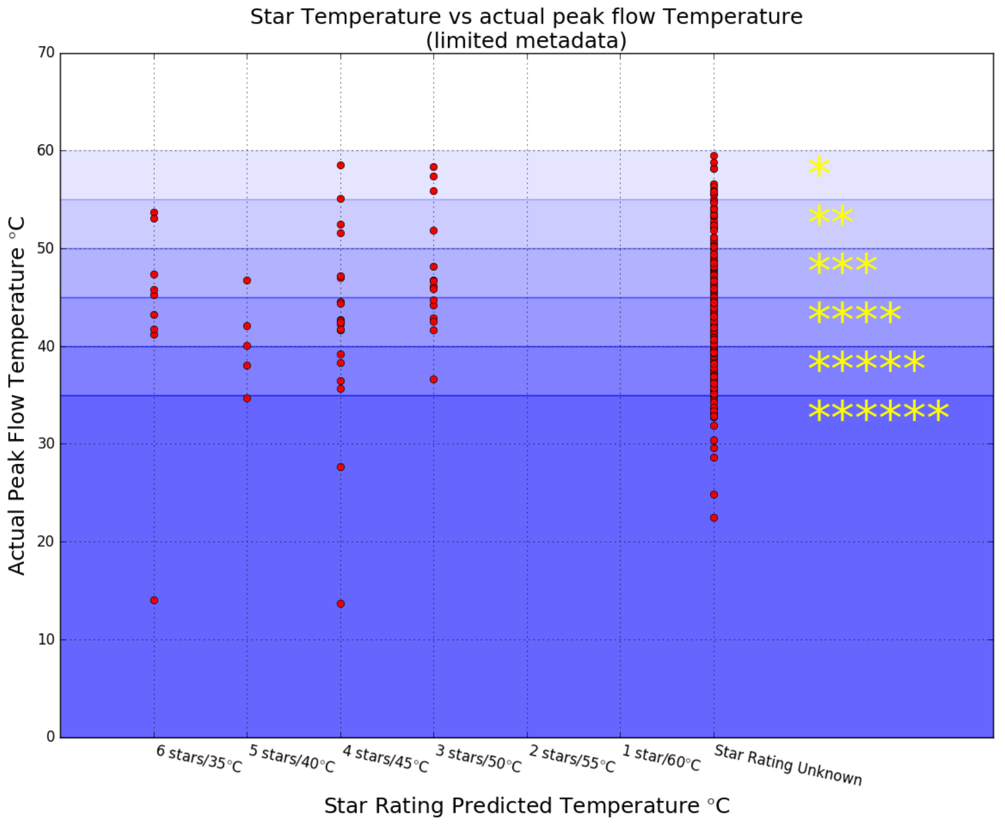

Figure 5-2 Star temperature versus derived peak flow temperature

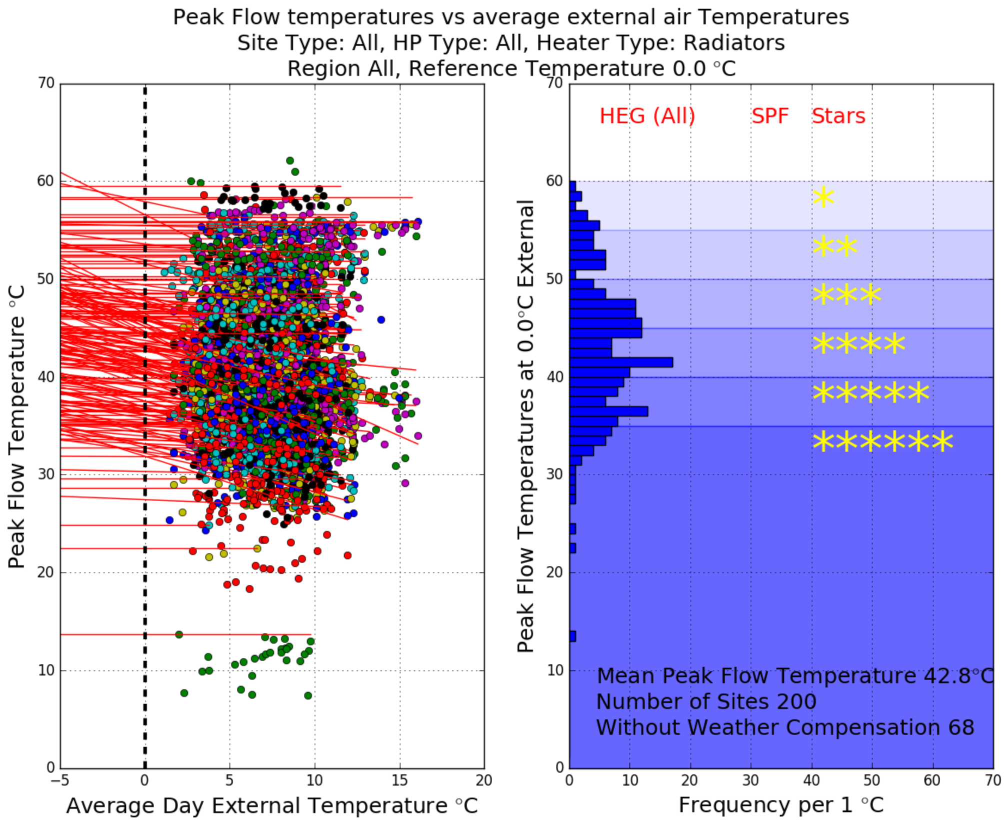

Figure 5-3 Derived peak flow temperatures for all heat pumps with radiators

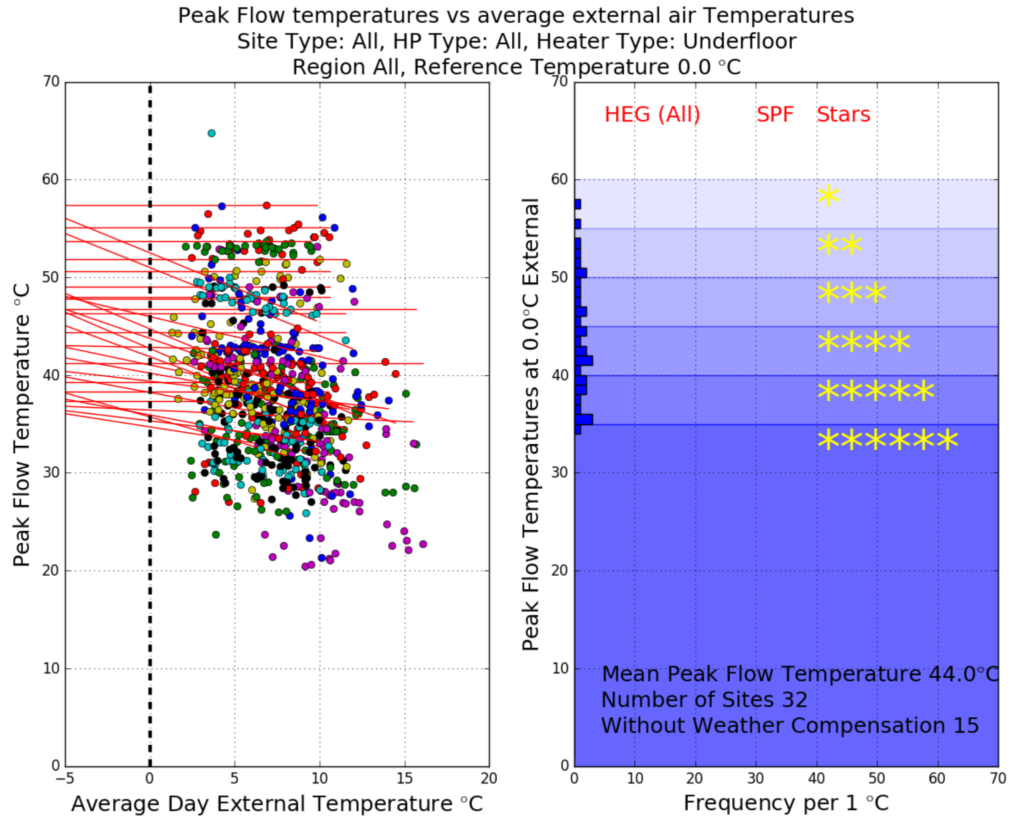

Figure 5-4 Derived peak flow temperatures for all heat pumps with underfloor heating

Weather compensation

This section of the report has identified space heating flow temperature from heat meter temperature sensor readings and concurrent outdoor air temperature weather data files. Regression calculations down the zero degrees centigrade have been necessary due to the low number of days at design outdoor temperature and hence the results are calculated rather than measured. The sample analysis indicates that for all heat pumps and all emitters, a sample size of 244 installations, weather compensation is indicated for 64%. An analysis by radiators (200 installations) and underfloor heating (32 installations) indicates the presence of weather compensation in 66% and 53% respectively. We have inferred the percentage of sites using weather-compensation from the chart of flow temperature versus outdoor temperature. Sites for which there is no slope are assumed not to have weather-compensation. We consider this estimate of the proportion of sites using weather compensation not to be significantly affected by metering errors.

Conclusions

We have estimated the specification of flow temperature at design conditions by extrapolation from 99th percentiles of measured flow temperatures. From the analysis of metering error presented in the 'Performance Variations' report, we would expect temperature errors primarily when heat meters are close to other sources of heat. This is likely to happen in a small number of cases and clearly, is most likely to occur at low flow temperatures. The chart uses 99th percentile of flow temperatures. We would therefore not expect these estimates to be unduely affected by metering error but may be influenced where monitored temperatures are taken from strap-on sensors.

A calculated mean peak temperature for radiators of around 43°C indicates a general tendency to maximise space heating SPF with 4 star installations. However, the analysis also indicates high temperature flow regimes in radiator and, in particular, in underfloor heating systems. With regard to the latter, underfloor heating in carpeted timber floors may require higher temperatures to ensure adequate heat output (see footnote 2). It is apparent that heating temperatures could be better controlled in order to benefit from Carnot performance. Finally, that there is little correlation between flow temperature and installer assigned 'star rating' is perhaps understandable given that, in 2012 when use of the HEG became obligatory, the terminology was entirely new to the heating industry.

6 How do the measured SPFs for space heating compare with SPFs from the Heat Emitter Guide?

Introduction

The Heat Emitter Guide (HEG) provides the SPF for space heating flow temperatures at 5K increments from 35 to 60°C. It is important to understand the derivation of these HEG SPF values. The fifteen footnotes (a to o) in the original HEG are as follows:

Heat pump likely Seasonal Performance Factor (SPF) is calculated for space heating only in accordance with the following notes and assumptions:

(a) Leeds is used for weather data. (b) Provision of domestic hot water is not included. (c) Room temperature is based on European Winter standard 21˚C operative temperature per BS EN ISO 7730. (d) The heat pump is sized to meet 100% of the space heating load and is the only heat source used in the dwelling. (e) GSHP SPF is the SCOP calculated in accordance with prEN 14825. (f) GSHP 0/35 COP = 3.5 (MCS minimum thresholds). (g) Heating flow temperature in heat emitter guide is at peak design conditions (i.e. at the lowest external design temperature). (h) The temperature difference across the heat emitters is fixed at 1/7th of the emitter circuit flow temperature. (i) Weather compensation is used. (j) 100W has been added for the electrical consumption of heating circulation pumps. (k) The heat emitter control system meets current building regulation requirements. (l) No allowance has been made for losses from: cycling, buffer vessels, or associated water pumps. (m) The GSHP ground array is designed with a minimum heat pump entry water temperature of 0˚C. (n) A ground circulation pump is included. (o) The SPF values for ASHP are 0.7 less than for GSHP, which is consistent with SAP.

The aim of this section of the report is to assess how well RHPP SPFs for space heating compare with HEG SPFs: to do so we first need to explore these HEG footnotes. Leeds weather data (a), according to CIBSE TM 41, is based on the East Peninnes degree-day region (CIBSE, 2006) with a 20 year average of 2169 degree-days (Vesma[30]). Following the MIS 3005 99% winter design temperature criteria, Leeds is listed in the CIBSE Guide A, Table 2.5 at (-1.9)°C (2016 : 2-7). However, we note from (f) that HEG SPFs are based on GSHP performance at 0/35°C and that the SPF is actually a laboratory test of SCOP (e) from BS EN 14825. The variation in HEG SPF is dependent on space heating flow temperature and is presumably also from the same standard where provision is made to test at 35, 45, 55 and 65°C. We note that weather compensation (i) is used and so we assume that flow temperature is the maximum in the compensation curve. The SPF quoted is based on the addition of 100 Watts for a circulating pump (j), thus HEG SPF refers to Sepemo SPF_{H2} plus this nominal allowance. We also note that no allowances have been made for losses (l) such as from cycling (the results are for steady state operation) or pipe and

[30] http://vesma.com/ddd/20year07.htm

vessel losses. Thus for GSHPs we would expect RHPP SPFs to be somewhat less than those provided in the HEG. HEG SPF values for ASHPs (o) are derived from GSHP SPF results by the simple subtraction of 0.7. We may note from the RHPP Detailed Report (RAPID-HPC, 2016) that the difference between ground and air source heat pumps at SPF_{H2} in the three samples is an average of 0.33.

Derived maximum flow temperatures

The HEG assumes that SPF is a function of flow temperature. For comparison with RHPP SPF_{H2} it is necessary to identify maximum space heating flow temperatures and the HEG SPF associated with each system for that temperature. Graphs are provided for all GSHPS and ASHPs where maximum flow temperature is plotted as described in Section 5.1. For each site, SPF_{H2} is then compared with the HEG SPF.

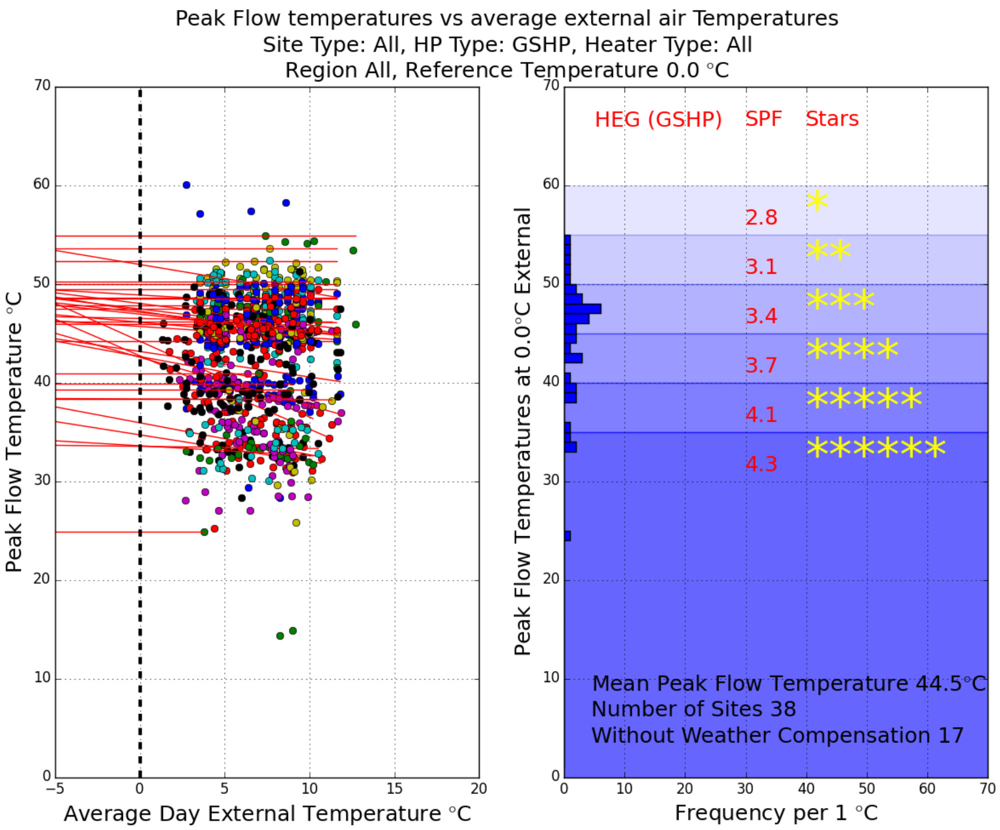

Figure 6-1 Derived peak flow temperatures for GSHPs

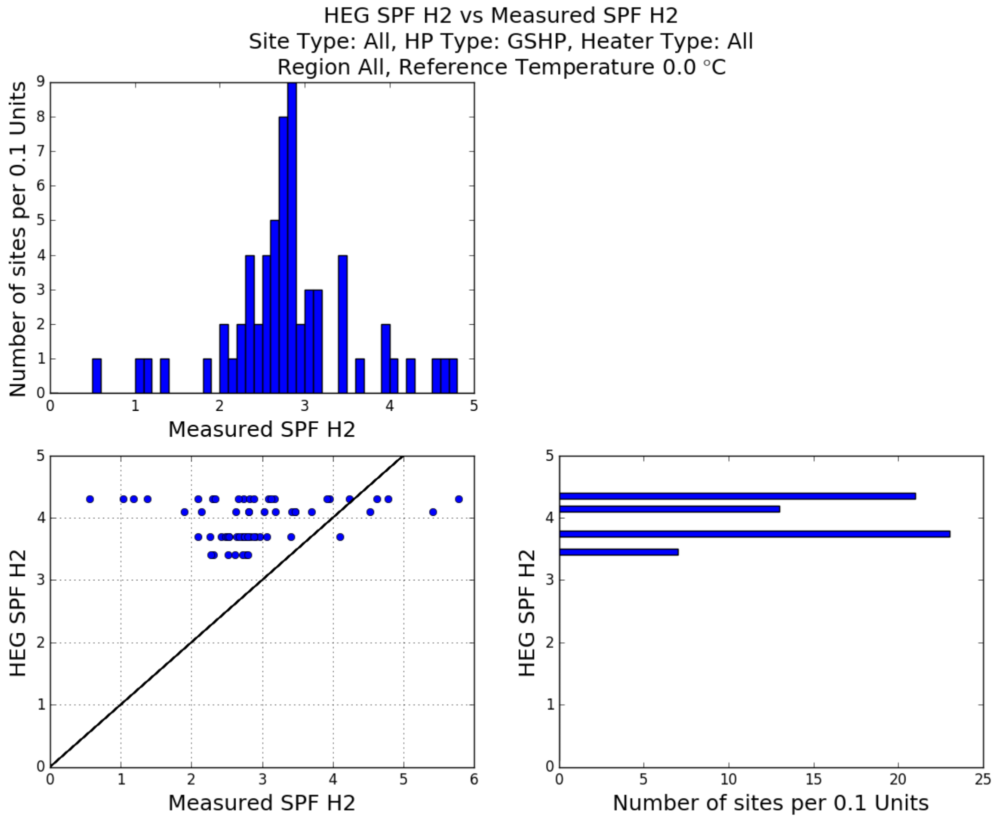

Figure 6-2 GSHP SPF_{H2} versus HEG SPF

It is apparent from Figure **6**-**2** that for GSHPS HEG 'likely space heating SPF' does not match actual SPFs in the RHPP trials and that HEG SPF is an over-estimation.

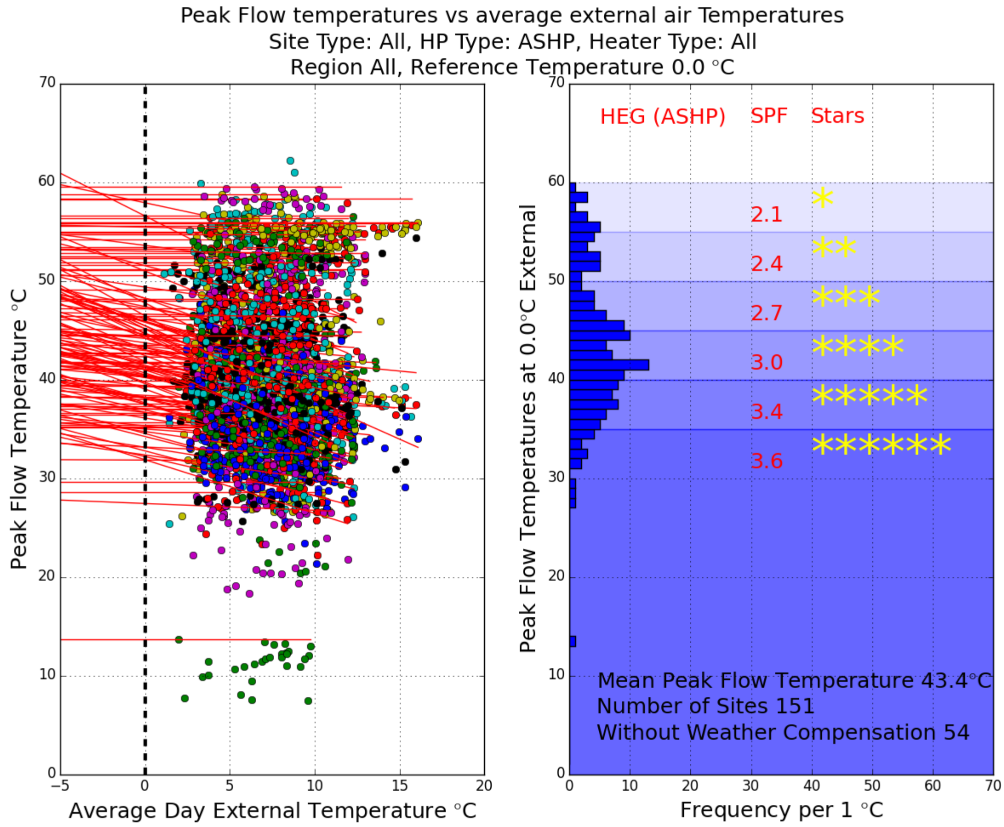

Figure 6-3 Derived peak flow temperatures for ASHPs

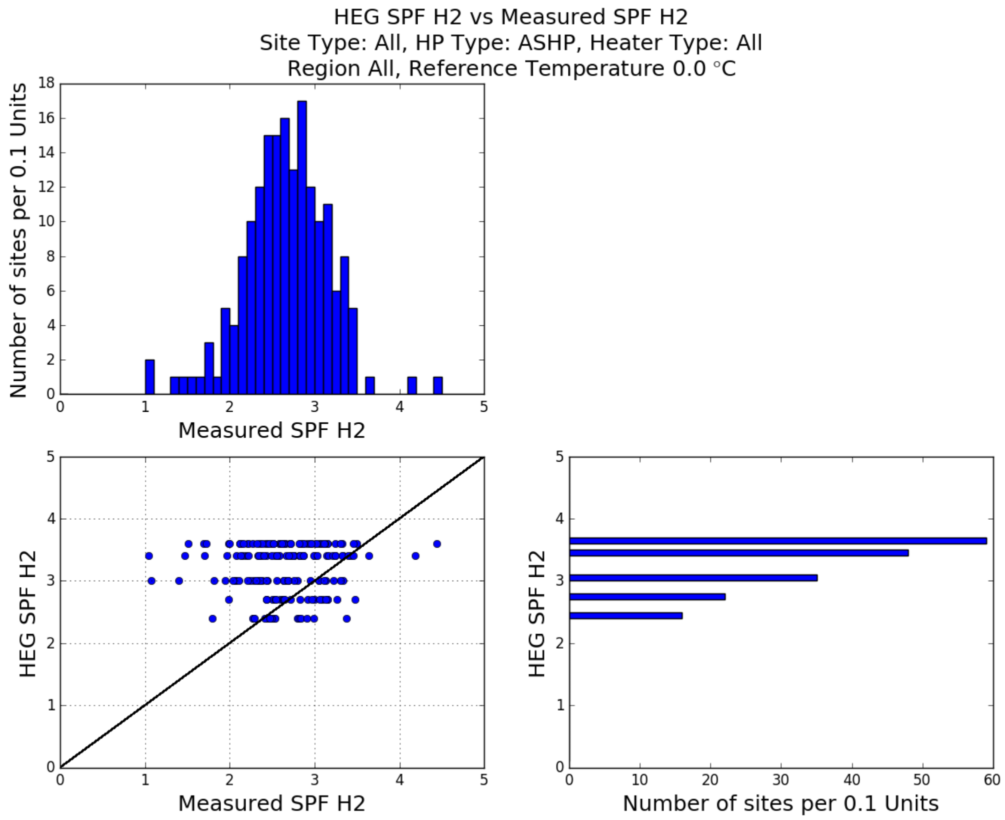

Figure 6-4 ASHP RHPP SPF_{H2} versus HEG SPF

HEG SPFs for ASHPs, are, in comparison to HEG SPFs for GSHPs, more closely matched to measured RHPP SPFs although clearly HEG SPF is predominantly higher than trial SPF. Statistical comparison between measured and HEG SPF is provided in Table **6–1**.

SPF values	ASHPs	GSHPs
median measured SPFs	2.65	2.78
median HEG SPFs	3.40	4.10
IQR measured	0.64	0.64
IQR HEG	0.60	0.60

Table 6–1 Statistical analysis of measured and HEG SPF

Modelling of SPF as a function of load factor

Appendix D shows a comparison of modelled and measured SPF_{H2} as a function of load factor. The modelling indicates that the wide variation observed in the results could, potentially, be the result of parasitic electrical loads within the H2 boundary such as controls, inlet fans (ASHP) or ground loop circulating pumps (GSHP).

Conclusion

As described in the 'Performance Variations' report, a range of metering errors can affect individual estimates of SPF_{H2} with some causing an increase, others a decrease. There is a systematic over-estimate of heat flow of 4 - 7% caused by calibration of heat meters for water instead of glycol, which one would expect to find in all hydronic split ASHPs and externally located GSHPs. However, this margin of error would not be expected to greatly affect the overall result found here, namely that calculated space heating efficiency does not correlate with the expected SPF_{H2} from the MCS Heat Emitter Guide for the heat pumps examined.

The design assessment of 'likely space heating SPF' is required to calculate annual energy costs for both air and ground source heat pumps. However, for GSHPs, the initial assessment of SPF is integral to ground loop sizing and therefore its accurate prediction is critical for ensuring that the system can extract adequate heat from the ground. The comparison of HEG 'likely space heating SPF' with RHPP field trial SPF provides some justification for a change in SPF assessment methodology. Compliance with ErP requirements has resulted in the removal of 'likely space heating SPF' from the latest version of the HEG and its replacement with individual heat pump laboratory test results for seasonal coefficient of performance (SCOP) at a series of space heating flow temperatures and available from the MCS Product search database[31]. Mapping individual manufacturers' heat pump SCOP with trial SPF has not been carried out for this report.

[31] MCS Product search database: http://www.microgenerationcertification.org/consumers/product-search

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7 Appendix A

System boundaries and monitored data parameters

As noted in previous outputs from this project, HP system boundaries are fundamental to the evaluation of annual HP performance using monitored data. The system of boundaries used is that defined by the SEPEMO project (Riviere et al., 2011). The H5 System Boundary, as shown by the outer dotted boundary is not defined by SEPEMO. Instead, it emerged as an extension of the SEPEMO boundary approach (Gleeson & Lowe, 2013 : 641). Performance based on tapped hot water rather than heat flow into the cylinder, was originally defined as "System efficiency" in the Phase I report of the EST HP Field Trials (Dunbabin & Wickins, 2012). All boundaries are shown in Figure **7**-**1**.

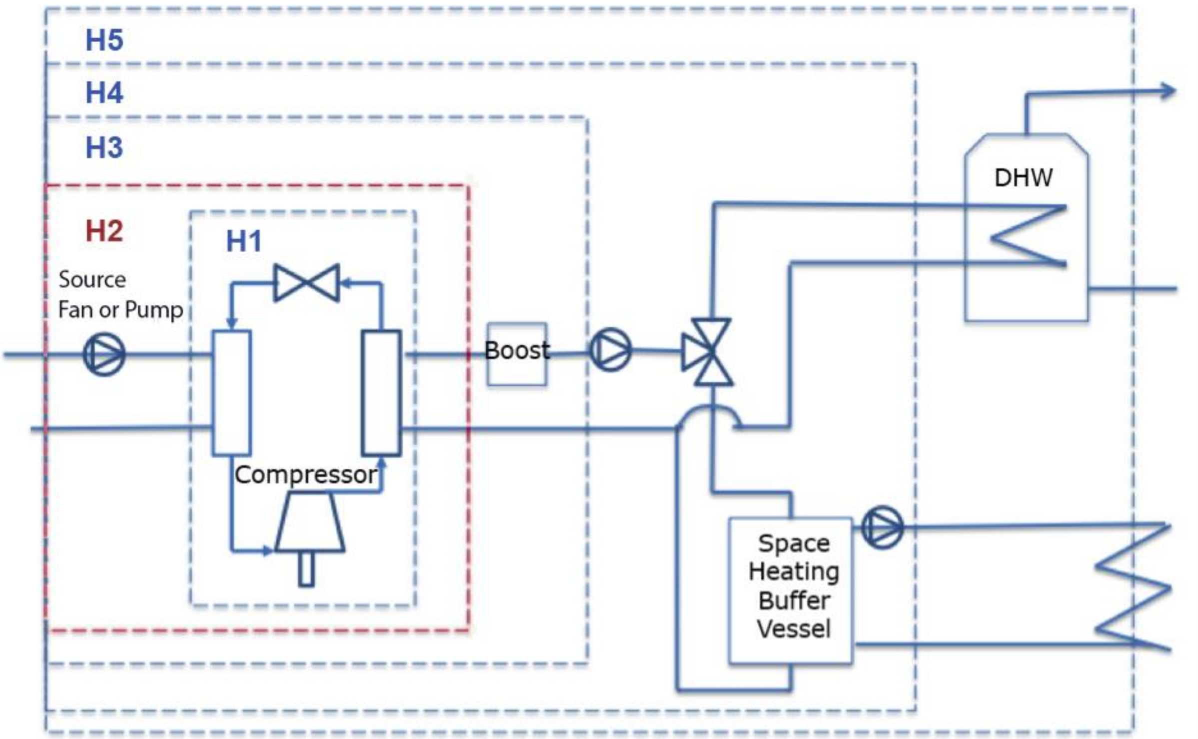

Figure 7-1 SEPEMO system boundaries (derived from Riviere et al., 2011) with the addition of H5 boundary that accounts for heat losses from the hot water cylinder.

Table **7–1** shows the complete set of parameters in the monitored data used to calculate the SPFs, though it should be noted that different sites had different combinations of parameters according to the schematic (or monitoring layout) that was applicable to that installation and plumbing arrangement. Figures 7-2 and 7-3 provide examples of two simple schematic diagrams (for an air source and ground source HP) that illustrate the location of monitoring points corresponding to the monitored parameters in any given heat pump system. Full details of the monitoring programme, including the overall monitoring philosophy and considerations of sensor resolution, can be found in the Preliminary Assessment report (Wickins, 2014); a summary of this report will be published at the end of the project. Further details of the heat metering have been provided in the form of private communications by Martin.

Parameter	Description
Eb	Electricity meter for whole system boost only
Edhw	Electricity meter for domestic hot water (typically an immersion heater)
Ehp	Electricity meter for the HP unit (may include a booster heater and circulation pump)
Esp	Electricity meter for boost to space heating only
Fhp	Flow rate of water from HP (may be space heating only)
Fhw	Flow rate of water to DHW cylinder
Hhp	Heat meter from HP (may be space heating only)
Hhw	Heat meter to DHW cylinder
Tco	Temperature of refrigerant leaving the condenser
Tin	For ASHP: Temperature of refrigerant leaving the evaporator For GSHP: Temperature of ground loop water into the HP
Tsf	Flow temperature of water to space heating
Twf	Flow temperature of water to cylinder

Table 7–1 The complete set of parameters included in the monitored data

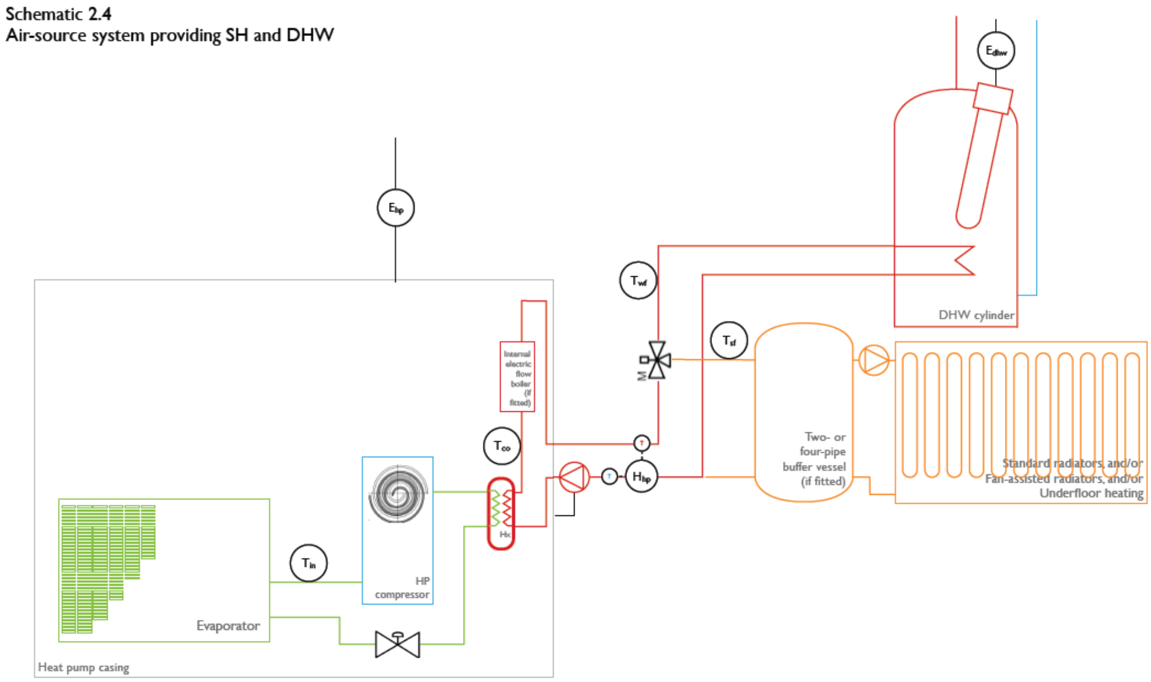

Figure 7-2 An example simplified schematic of the metering arrangement for a monobloc ASHP that provides heat to space heating and a domestic hot water cylinder with an immersion element.

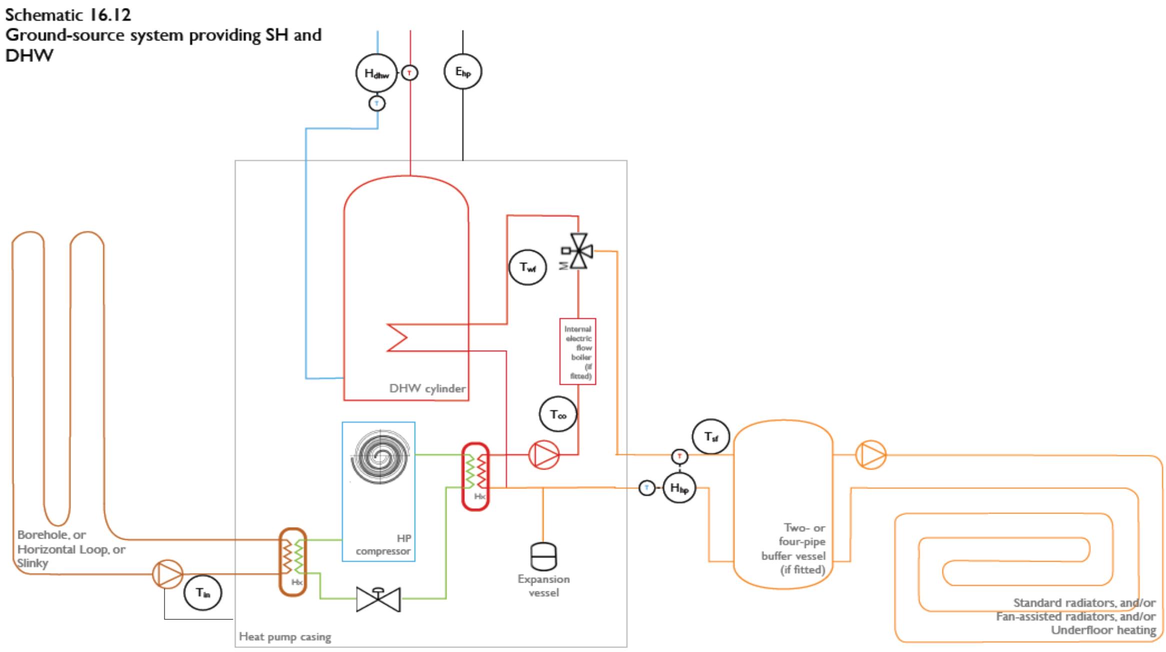

Figure 7-3 An example of a GSHP with an integrated domestic hot water cylinder.

8 Appendix B

Weather compensation curves

Weather compensation curves are derived from the proportionality of building heat loss, radiator output and heat source output (Moss 2003):

$$[\boldsymbol{\Sigma}(\boldsymbol{UA}) + \boldsymbol{Cv}](\boldsymbol{ts} - \boldsymbol{te}) \propto \boldsymbol{KA}(\boldsymbol{tm} - \boldsymbol{ts})^{\boldsymbol{n}} \propto \dot{\boldsymbol{m}}\boldsymbol{C}(\boldsymbol{t}\dot{\boldsymbol{f}} - \boldsymbol{tr})$$

Where UA + Cv represent fabric and ventilation constants; KA and the index n represent radiator constants; $\dot{m}C$ represent mass flow rate and specific heat capacity. Since all are constants they can be removed to provide ratios of temperatures where ts equals internal (space); te equals external; tm equals mean radiator; tf and tr equal flow and return temperatures.

Heat loss must be replaced by heat gain from the radiators, however, heat loss is proportional to the temperature difference between inside and outside. To provide the correct radiator output it is possible to adjust the mean radiator temperature to match heat demand. Three examples of weather compensation curves are derived from this relationship for design return temperatures of 50, 40 and 30°C, Table 9-1 and weather compensation curves plotted, Figure 9-1.

ts	te	tr	dt = (ts-te)	ts	te	tr	dt = (ts-te)	ts	te	tr	dt = (ts-te)
20	-10	50.0	30	20	-10	40.0	30	20	-10	30.0	30
20	-9	49.4	29	20	-9	39.6	29	20	-9	29.8	29
20	-8	48.7	28	20	-8	39.2	28	20	-8	29.6	28
20	-7	48.1	27	20	-7	38.7	27	20	-7	29.4	27
20	-6	47.4	26	20	-6	38.3	26	20	-6	29.2	26
20	-5	46.7	25	20	-5	37.9	25	20	-5	29.0	25
20	-4	46.0	24	20	-4	37.4	24	20	-4	28.8	24
20	-3	45.3	23	20	-3	37.0	23	20	-3	28.6	23
20	-2	44.6	22	20	-2	36.5	22	20	-2	28.4	22
20	-1	43.9	21	20	-1	36.0	21	20	-1	28.1	21
20	0	43.2	20	20	0	35.6	20	20	0	27.9	20
20	1	42.4	19	20	1	35.1	19	20	1	27.7	19
20	2	41.7	18	20	2	34.6	18	20	2	27.4	18
20	3	40.9	17	20	3	34.0	17	20	3	27.2	17
20	4	40.1	16	20	4	33.5	16	20	4	26.9	16
20	5	39.3	15	20	5	33.0	15	20	5	26.7	15
20	6	38.5	14	20	6	32.4	14	20	6	26.4	14
20	7	37.6	13	20	7	31.9	13	20	7	26.1	13
20	8	36.7	12	20	8	31.3	12	20	8	25.9	12
20	9	35.8	11	20	9	30.7	11	20	9	25.6	11
20	10	34.9	10	20	10	30.1	10	20	10	25.2	10
20	11	33.9	9	20	11	29.4	9	20	11	24.9	9
20	12	32.9	8	20	12	28.7	8	20	12	24.6	8
20	13	31.8	7	20	13	28.0	7	20	13	24.2	7
20	14	30.7	6	20	14	27.3	6	20	14	23.8	6
20	15	29.5	5	20	15	26.5	5	20	15	23.4	5

Table 8–1 Weather compensation derived return temperatures

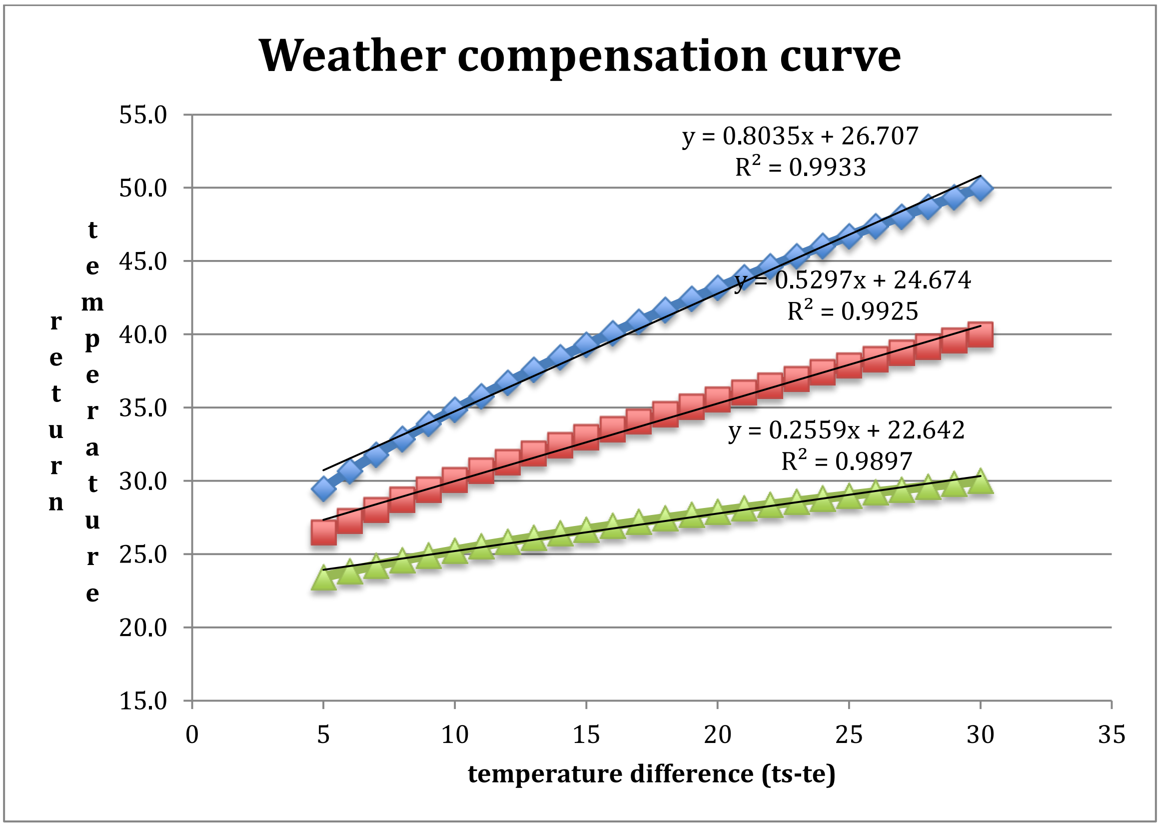

Figure 8-1 Weather compensation curves

9 Appendix C

Heat pump net capacity

Heat pumps must provide sufficient heat at design outdoor temperature to achieve internal design temperatures. The MCS installer must calculate the building heat losses according to MIS 3005 and select a heat pump capable of meeting this heat output. Unlike conventional boilers, whose heat output varies only weakly with external temperature and flow and return temperatures, heat pump heat output is sufficiently dependent on source and sink temperatures that installers need to ensure that 'net capacity' at design conditions is capable of meeting the design heat loss.

Heat pump output is governed by Carnot efficiency where coefficient of performance (COP) is represented by heat out / work in. Work in (compression) is defined as the difference between heat out at the condenser and heat in at the evaporator and is dependent on the thermodynamic temperature in Kelvin of the source (T_{Low}) and sink (T_{High}):

$$\boldsymbol{COP = \frac{desired}{required} = \frac{Heat\ out}{Work\ in} = \frac{T_H}{T_H - T_L}}$$

A Carnot analysis shows that high efficiency COP, and by extension SPF, is dependent on maintaining low sink temperatures (high star rating). Where source or sink temperatures deviate from those published by manufacturers then COP and heat output, 'net capacity', is affected.

Manufacturers supply heat pumps with nominal outputs that match a range of dwelling heat losses typically from 5 to 20 kW. Each unit is tested to EN 14511 standards, Table 9-1, where output is expressed at set source and sink temperatures known as 'standard rating conditions'. For ASHPs these are typically 7°C dry bulb, 6°C wet bulb (80% saturation) and 35°C flow temperature (with a 5 K delta T), expressed as A7/W35. Some manufacturers, at the time of the RHPP trial, are found to have provided outputs at 'application rating conditions', such as A2/W35 or A2/W45.

Table 9 — Air-to-water heat pumps - Heating mode

		Outdoor heat exchanger		Indoor heat exchanger	
		Inlet dry bulb temperature °C	Inlet wet bulb temperature °C	Inlet temperature °C	Outlet temperature °C
Standard rating conditions	Outdoor air	7	6	40	45
	Exhaust air	20	12	40	45
	Outdoor air (for floor heating or similar application)	7	6	30	35
Application rating conditions	Outdoor air (for floor heating or similar application)	2	1	[a]	35
	Outdoor air (for floor heating or similar application)	- 7	- 8	[a]	35
	Outdoor air (for floor heating or similar application)	- 15	-	[a]	35
	Outdoor air	2	1	[a]	45
	Outdoor air	- 7	- 8	[a]	45
	Outdoor air	- 15	-	[a]	45
	Outdoor air	7	6	[a]	55
	Outdoor air	-7	-8	[a]	55

[a] The test is performed at the flow rate obtained during the test at the corresponding standard rating conditions.

Table 9–1 ASHP EN 14511 test regime

Some manufacturers met the minimum requirement to provide COP and heat output at the single condition of A7/W35 whereas others provided outputs at a range of source and sink temperatures. Some manufacturers provided 'integrated' and 'peak' values, that is, with and without defrost cycles respectively. Two manufacturers of variable speed compressor heat pumps supplied a wide range of tabulated outputs. Both provided outputs at a range different source and sink temperatures whereas one added performance at different compressor speeds described as 'steps'. Yet another ASHP manufacturer provided graphical data with source temperature on the x axis, heating capacity on the y axis and performance curves representing low, mid and high compressor speeds at flow temperatures of 35°C and 50°C.

For GSHPs test results are provided for 0°C 'brine' entering temperature (3K delta T) and typically either 35°C or 45°C flow (B0/W35 or B0/W45), whilst some manufacturers included output at 55°C (a typical hot water production temperature), Table 9-2.

Table 7 — Water-to-water heat pumps - Heating mode

		Outdoor heat exchanger		Indoor heat exchanger	
		Inlet temperature °C	Outlet temperature °C	Inlet temperature °C	Outlet temperature °C
Standard rating conditions	Water	10	7 [a]	40	45
	Brine	0	-3 [a]	40	45
	Water (for floor heating or similar application)	10	7 [a]	30	35
	Brine (for floor heating or similar application)	0	-3 [a]	30	35
Application rating conditions	Water	15	[b]	[b]	45
	Brine	5	[b]	[b]	45
	Brine (for floor heating or similar application)	5	[b]	[b]	35
	Brine	- 5	[b]	[b]	45
	Brine	0	[b]	[b]	55
	Water	10	[b]	[b]	55

[a] For units designed for heating and cooling mode, the flow rate obtained during the test at standard rating conditions in cooling mode (see Table 8) is used.
[b] The test is performed at the flow rate obtained during the test at the corresponding standard rating conditions.

Table 9–2 GSHP EN 14511 test regime

ASHP net capacity corrections

ASHPs are supplied with either fixed or variable speed compressors. Performance data from ASHP manufacturers (3 fixed speed, 2 variable speed) has been an analysed for an outdoor temperature of -2°C, the mean MIS 3005 outdoor design temperature for the UK.

ASHP fixed speed

Three manufacturers and five fixed speed compressor models indicate that percentage difference from manufacturer nominal capacity varies from manufacturer to manufacturer and model to model, Table 9-3. Note that manufacturer A only provides performance data for 45°C flow. The results indicate that change in output for A-2/W35 range from -8% through to -19% whilst changes in output at A-2/W45 range from -13% to -32%.

Manufacturer's Fixed Speed ASHP	Nominal kW	kW at (-2)/35	% difference from nominal	kW at (-2)/45	% difference from nominal
A	5	-	-	3.9	-25%
B	5	4.6	-8%	4.4	-13%
B	9	7.45	-19%	7.28	-21%
C	12	10.8	-11%	10	-18%
C	16	13.3	-18%	11.6	-32%

Table 9–3 ASHP Fixed speed compressor corrections for -2°C

ASHP variable speed

Two manufacturers and four variable speed compressor models are shown in Table 9-4. For manufacturer D, at both A-2/W35 and A-2/W45, output differs from nominal by as little as -2% to -4%. In contrast, manufacturer E outputs at A-2/W35 are -20% and -21%, and at A-2/W45, -31% and -29%.

Manufacturer's Inverter ASHP	Nominal kW	kW at (-2)/35	% difference from nominal	kW at (-2)/45	% difference from nominal
D	5	4.8	-4%	4.8	-4%
D	8.5	8.3	-2%	8.3	-2%
E	6	4.89	-20%	4.38	-31%
E	8	6.48	-21%	5.96	-29%

Table 9–4 ASHP variable speed compressor corrections for -2°C

Ground Source Heat Pump fixed speed

At the time of the RHPP trials, most GSHPs were equipped with fixed speed compressors. Four manufacturers and 11 different models are compared for B0/W35 and B0/W45, Table 8-5. The results indicate that change in output for B0/W35 range from -12% through to plus 10%. Changes in output at B0/W45 range from -23% to plus 5%.

Manufacturer's Fixed Speed GSHP	Nominal kW	kW at 0/35	% difference from nominal	kW at 0/45	% difference from nominal
C	5	4.9	-2%	4.6	-8%
C	14	14.8	6%	14.7	5%
F	6	5.33	-12%	5.38	-11%
F	12	11	-9%	10.6	-12%
F	16	16.4	2%	15.6	-3%
G	6	6.5	8%	6.2	3%
G	12	12.6	5%	12	0%
G	16	17.6	10%	16.9	5%
H	5	4.65	-7%	3.98	-23%
H	8	8.15	2%	7.75	-3%
H	12	11.6	-3%	10.99	-9%

Table 9–5 GSHP Fixed speed compressor corrections for -2°C

Conclusions

The impact of design conditions on badged capacity is shown to depend on both source and sink temperatures. Fixed speed ASHPs most clearly show the negative impact of design temperature on badged capacity. For variable speed ASHP the impact ranges from as little as -2% up to -30% depending on the manufacturer. For GSHP fixed speed compressors the general pattern is a loss in capacity but one that is again dependent on the make and model.

10 Appendix D

Potential impact of low Load Factor

The load factor for all air and ground source heat pumps is 14.5%, for ASHPs 13.5%, and 17.3% for GSHPs. When viewed annually there appears to be some relationship with SPF at low load factors although this is less clear for GSHPs, with the suggestion of a relatively well defined lower boundary to the scatterplot first described in the early days of condensing boiler development in Pickup and Miles (1977) and more recently by Orr, et al (2009). This observation prompted the authors to further exploration through the construction a theoretical model of monthly heat pump performance that included:

- physically realistic relationships between monthly space heating operating temperatures and monthly external temperature (on the assumption of perfect external weather compensation and with an unlimited turn-down ratio on the heat pump);
- a simple modified Carnot relationship between condenser-evaporator ΔT and monthly mean COP;
- a highly simplified model of monthly space and water heating demand;
- a continuous fixed parasitic electrical load that can be varied by the model user.

This simplified model assumes away the problem of diurnal dynamics. It was used to explore the range of possible shapes (sign of slope and approximate functional form) for the relationship between monthly COP and monthly load factor. The results of this exploration are shown in Figure 10-1.

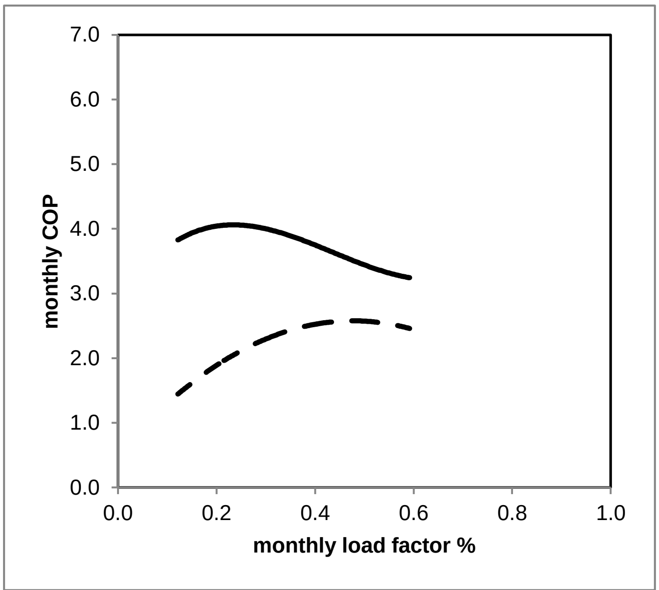

Figure 10-1 Envelope of monthly COPs versus monthly load factors. The upper bound includes continuous parasitic electrical loads. The lower bound includes a 200 W continuous parasitic electrical load.

Our proposition is that the upper and lower bounds theorised in Figure 10-1 are suggestive of the envelope of points in the scatterplot for all heat pumps in the bottom left hand corner of Figure 10-2.

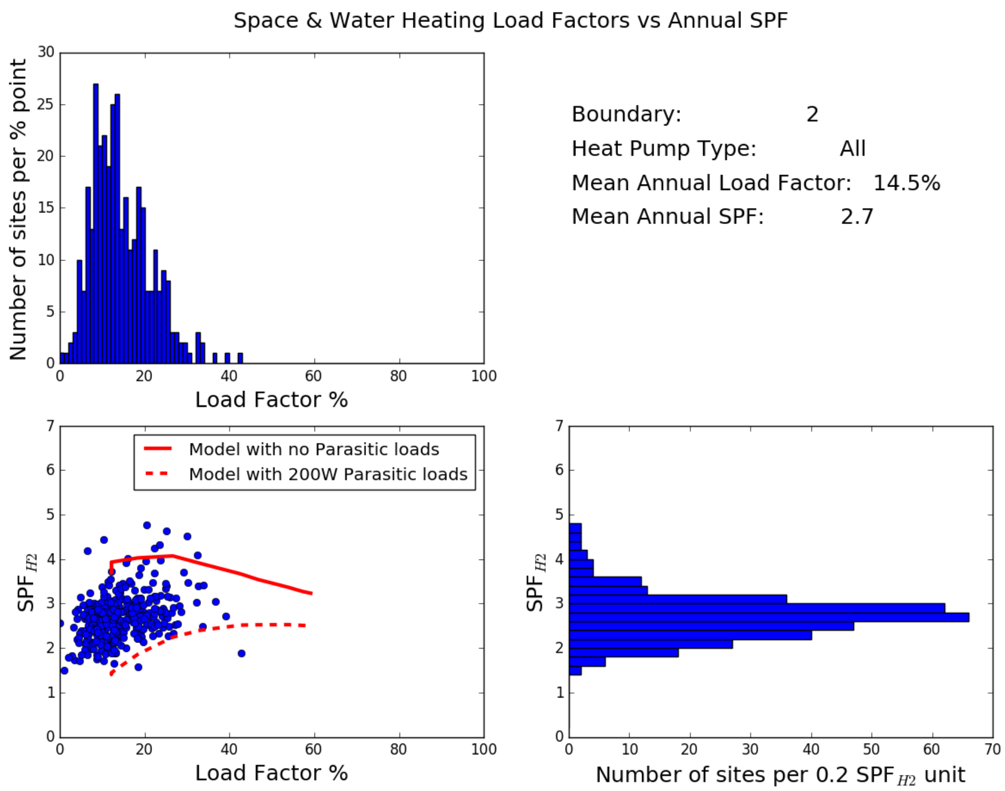

Figure 10-2 Measured SPF as a function of Load Factor for all heat pumps

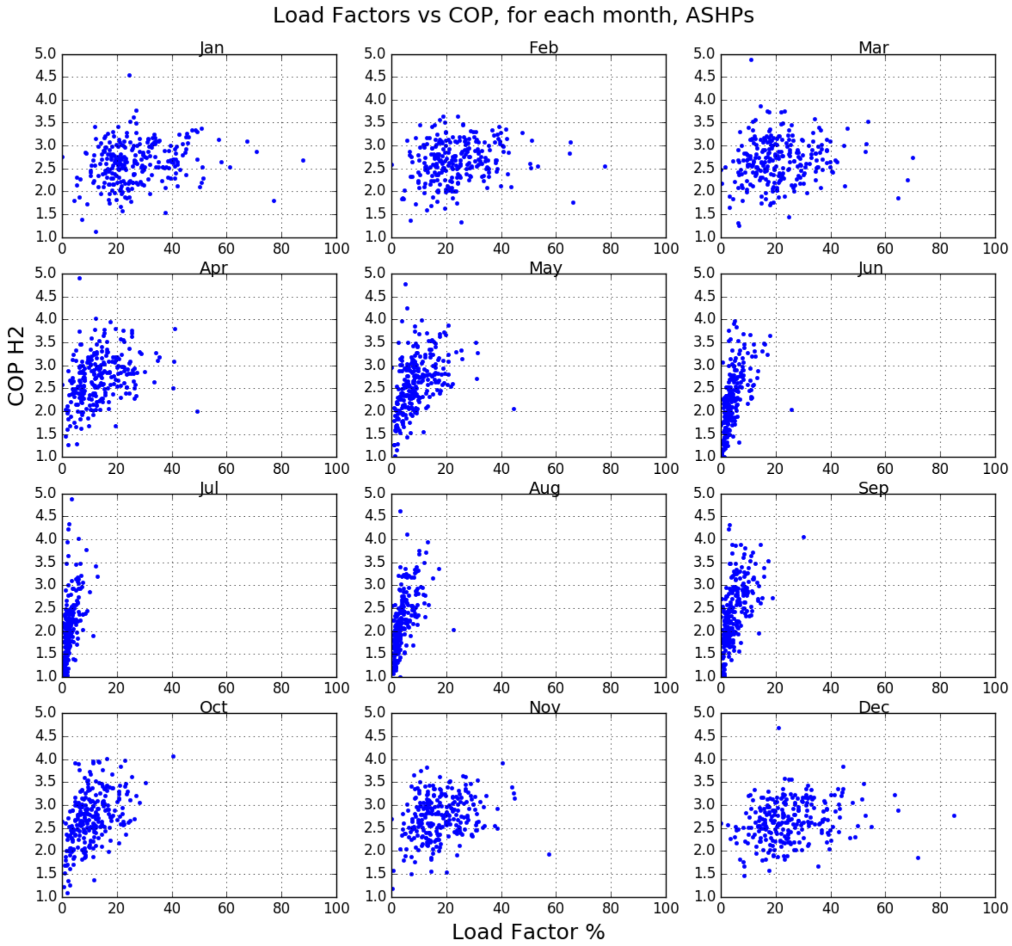

Figure 10-3 Monthly COP and load factors for ASHPs

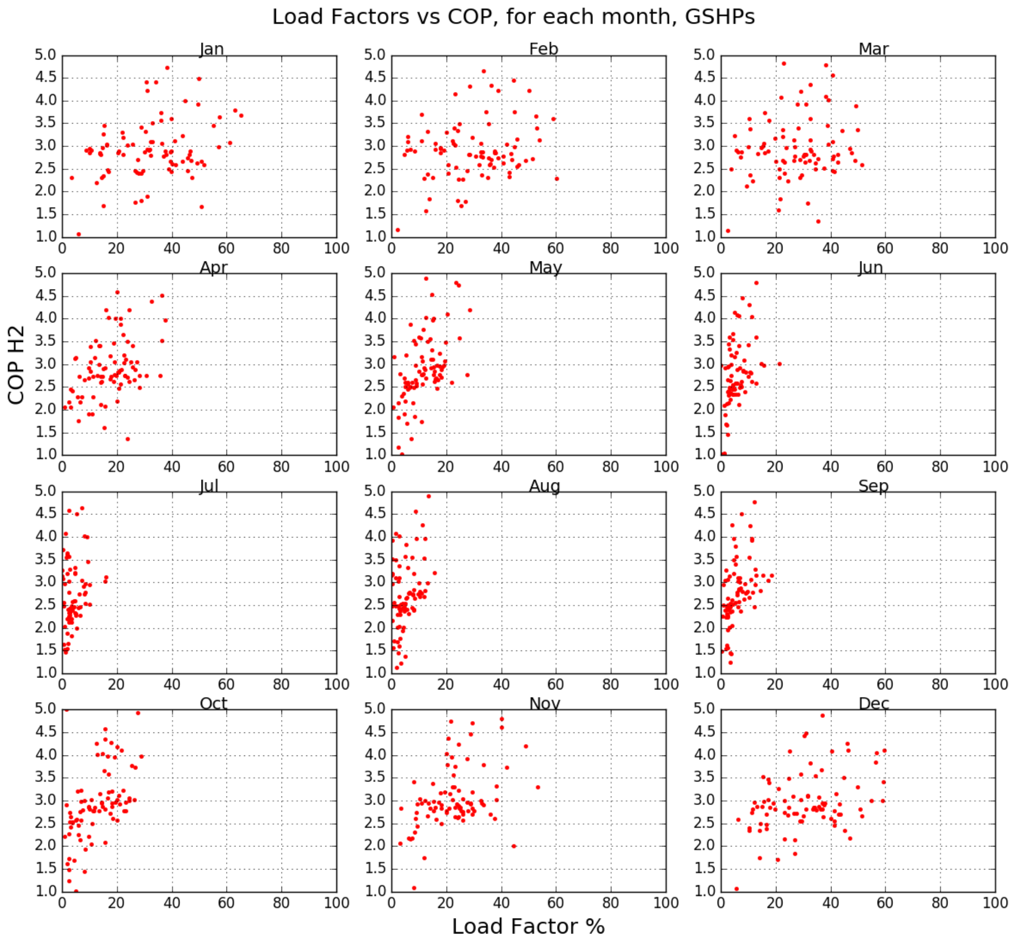

Figure 10-4 Monthly COP and load factors for GSHPs

Monthly load factors for ASHPs and GSHPs are provided in Figure **10-3** and Figure **10-4**. It is not clear whether the summer distribution for July/August identifies any consistent reduction in monthly COP at low loads since a spread of COPs is observed in all months. However, the model does suggest that at low output, parasitic loads from circulating pumps and fans may be dominant and therefore detrimental to annual seasonal efficiency. Somewhat counterintuitively, a householder who wishes to save money by operating the heat pump for limited time periods may reduce the COP in the process because of fixed parasitic loads. The potential effects of low load factor on COP indicates a need for further research.

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