Superhomes 2.0

Best Practice Guide for ASHP Retrofit

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The purpose of the Superhomes 2.0 (SH2.0) project was to research optimisation of real-world residential energy retrofit installations that included Air Source Heat Pumps (ASHPs).

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Executive Summary

The guidelines presented in this report refer specifically to the 20 homes under investigation in the Superhomes 2.0 project that may be extended to homes of similar construction and heating system layout.

The key steps in the process of retro fitting a heat pump to a domestic dwelling are identified. Analysis of data from comprehensive monitoring systems is used to present examples of how the completion of these steps in real life has affected the performance of the systems.

Section 2 considers the steps required to determine the correct sizing of an ASHP taking into account building load, the effect of ice build-up on the evaporator and the use of variable speed compressors. As a component within a low-temperature heating system, the ASHPs in this study were connected to both radiator and underfloor heating emission systems. Section 3 deals with the design considerations for these emission systems with examples from the data showing the effects of installation and commissioning set-up on system performance.

Retrofitted heating systems generally retain some or all of the original heating distribution system. Section 4 investigates the effect of existing zoning and pipework on performance while Section 5 investigates how Domestic Hot Water is produced with examples from the data and key recommendations. Section 6 discussed the main control strategies that were encountered in the Superhomes such as room temperature control, weather compensation curves and compressor control and is followed by the report Discussion and Conclusions.

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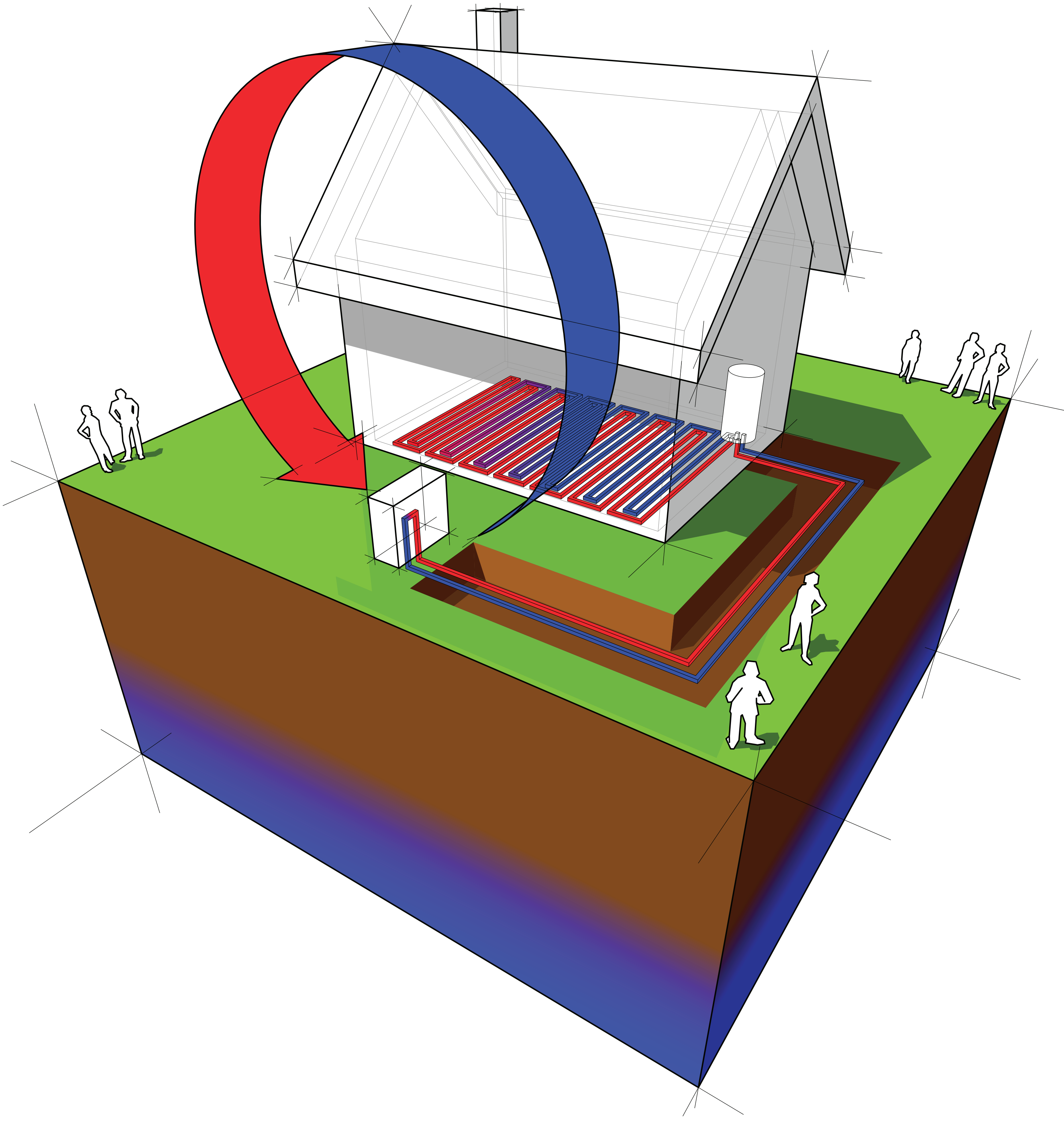
Nomenclature

Symbol	Description	Unit
Tair	Outdoor air temperature	°C
Tdo	Design outside temperature	°C
Tflow	Flow temperature	°C
ΔTfr	Flow-return temperature difference	°C
Tft	Target flow temperature	°C
EnPro	Heat Produced	kWhth
Tout	Outside air temperature	°C
Tret	Return temperature	°C
Troom	Room temperature	°C
Tsb	Room temperature setback	°C
ΔTSZ	Set - zone temperature difference	°C
Tsf	Set flow temperature	°C
Tsr	Set room temperature	°C
ΔT	Temperature difference	°C

Abbreviations

A/W	Outside air temperature/flow temperature
ASHP	Air source heat pumps
ASHP Opto	Optimisation module
CC	Compressor cycling
COP	Coefficient of Performance
CSD	Compressor switching delay
DHW	Domestic hot water
EnCon	Energy consumed
FFT	Fixed flow temperature
HCC	Heating compensation curve settings
HDD	Heating degree days
HLI	Heat loss indicator
HP	Heat pump
HSFR	Heating system water flow rate
HSWV	Heating system water volume
HSWV*min*	Minimum required volume
HSWV*tot*	Total system volume
KPIs	Key performance indicators
LSV	Lock shield valve
RT	Room temperature
SH	Space heating
SH20	Superhome 2.0
SHT	Space heating time
TOU	Time of Use
TRV	Thermostatic radiator valve
TS	Time schedule
UFH	Underfloor heating
Z1	Zone 1
Z2	Zone 2
ZTT	Zone target temperature



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1. Introduction

The purpose of the Superhomes 2.0 (SH2.0) project was to research optimisation of real-world residential energy retrofit installations that included Air Source Heat Pumps (ASHPs).

To this end, the SH2.0 team was furnished with data, access to the installations for the purposes of carrying out tests over a two-year period. Opportunities such as this to assess the energy performance of systems in real world buildings are limited and this is evidenced by the lack of performance data of ASHPs when reviewing the literature. Critically, the 20 homes that were involved in the study had extensive monitoring systems in place that facilitated data collection from the buildings and the ASHP. In many cases, this data was available at a minute-by-minute scale thus allowing for detailed analysis of performance factors.

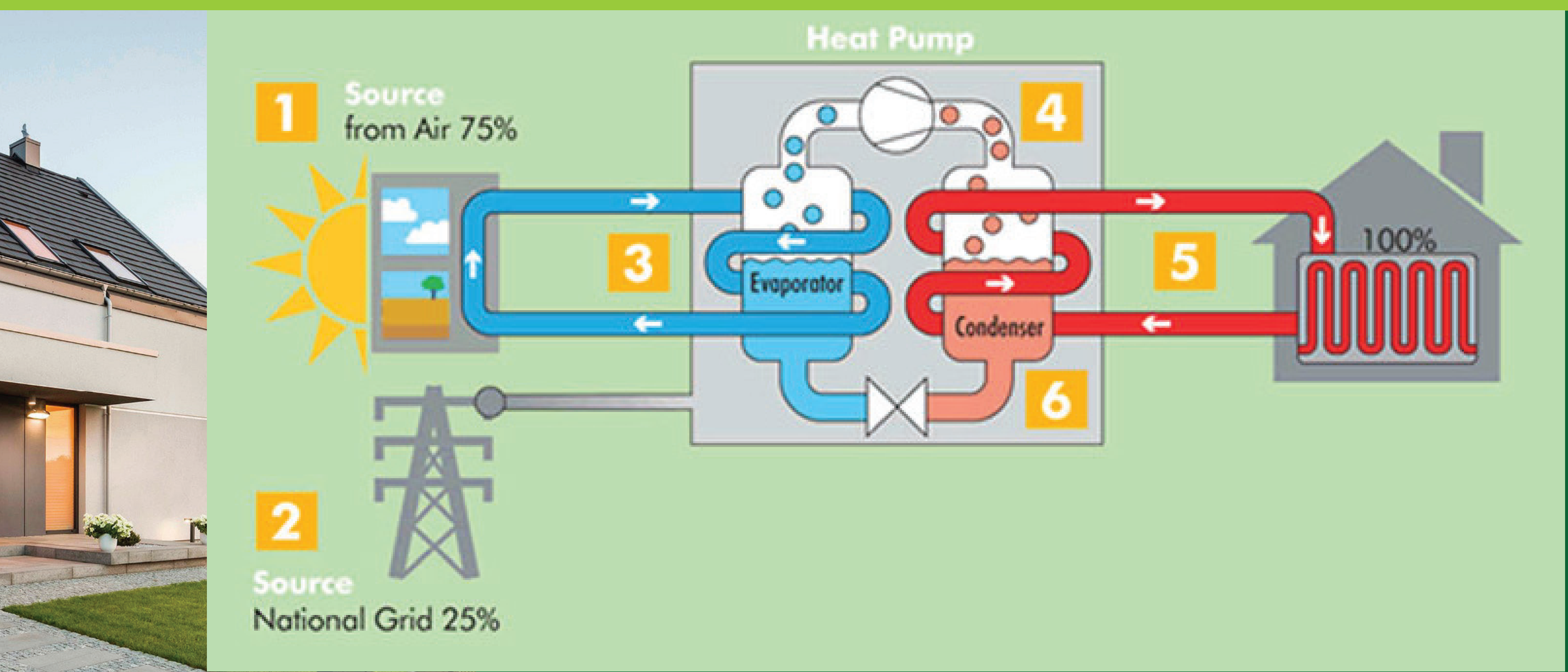
Preliminary research in the SH2.0 project explored different opportunities for optimising ASHP performance. Prominent among these was the negative affect of factors such as excessive compressor modulation and compressor cycling (CC). Heat pumps are selected based on heat output and COP, measured in accordance with EN14511 and published by the manufacturer. These performance metrics are established when the heat pump is operating in a steady state condition. This report will investigate the effect of interruptions to smooth operation on output and COP. During the course of SH2.0, it was very evident that the adoption of the ASHP as a "new technology" was quite daunting for many homeowners. For some, there was a general lack of understanding of how the control systems should or could be interacted with in order to optimise performance. In many cases, space heating operated with no time or temperature setback control, the sole control resting with the zone thermostats.

On the other hand, other homes actively employed time schedules, in some instances quite effectively, especially where underfloor heating (UFH) was present.

The correct handling of these key factors will ensure realistic optimal COP and running cost, long life for the compressor and associated electrical components, and satisfied homeowners. The aim of this report is to provide guidance to designers to help achieve these goals.

There were however examples where a time schedule was working well, but if there was a sudden change in outside temperature, the system was timed to switch off before achieving the zone thermostat target temperature. This could become problematic for the homeowner if they were not accustomed to regularly interacting with the schedule. Optimal performance can be achieved when interruptions are minimised. This allows the ASHP to operate as closely as possible to the outputs and efficiencies in manufacturer's performance charts. In order to achieve this, a clear design philosophy is required which sets realistic performance and COP targets for the system and which ensures that these targets are continuously referred to in the design, installation and commissioning stages of the ASHP installation. Budget and other project considerations will inform decisions of which type of heat emission system is to be used, but once these decisions are made, it is critical that the following key factors are properly managed:

- Correct flow temperature range for the heat emission system
- Appropriate expectations for SPF
- Avoidance of excessive compressor modulation
- Avoidance of compressor cycling
- Appropriate time and set-back temperature control



2. Heat Pump Selection

2.1. Design outside air temperature & ASHP sizing

When sizing an ASHP and associated heat distribution and emission system, the first thing to consider is which design outside air temperature (*Tdo*) is to be used. This decision should be taken by the designer in conjunction with the client, taking into consideration issues such as building thermal mass, control strategy, expected system response time, client's tolerance of internal temperature falling short of target, capital and running costs. CISBSE Guide A provides a method that uses either a 24-hour or a 48-hour mean temperature and then carrying out a risk analysis of the likelihood of the proposed *Tdo* being exceeded. The 24-hour mean temperature would be used for buildings with low thermal mass while the 48-hour temperature is more suitable for buildings with higher thermal mass. Generally, for the homes in the SH2.0 study, *Tdo* was in the range of -2 to -3°C while the construction type in all cases were medium-high thermal mass.

EN12831 sets out a method for calculating the design heat load for a building. A room-by-room heat loss calculation should be carried out which will also be used in the design of the heat emission system. This process calculates the building fabric and ventilation heat losses. In addition, the application of a re-heat factor (RHF) is recommended by EN12831. This is an intermittency factor which is based on the thermal mass properties of the building, the level of setback applied and the desired re-heat time. Details can be found in Appendix A. For example, when a house is unoccupied, it is good practice to maintain the room temperature at 1-2°C below comfort level. For a living area, comfort temperature may be 21°C and setback temperature 19°C. The RHF allows for a small additional capacity within the heat pump size to ensure the comfort temperature is achieved reasonably quickly, especially at lower *Tout*.

All of the systems in the SH2.0 project are controlled to prioritise the production of DHW and so the heat pump is never tasked with simultaneously heating DHW and the Space Heating system. In all cases, the output required for space heating exceeds that required for DHW and so it is not necessary to add a DHW factor to the design capacity of the heat pump as is suggested in EN12831.

2.2. Frost Build-up & effects of defrosting

At low *Tout* and high relative humidity (RH), ice starts to build up on the surfaces of the evaporator during AHSP operation. Eventually this ice causes a reduction in heat transfer from the outside air to the refrigerant. The ASHP controller will detect this occurrence and initiate a de-frost cycle whereby the refrigeration circuit is reversed so that hot gas enters the evaporator thus causing the ice to melt. The process typically lasts 3 – 4 minutes and involves energy consumption by the compressor as well as a temporary interruption of the ASHP seeking to maintain the target flow temperature.

The main considerations for the operation of the ASHP system resulting from the defrost process are:

1. **The energy required to complete the process with no resultant heat to the house**
2. **The effect of reducing output on COP as the ice builds up**
3. **Defrost interruptions prevent the unit from achieving target flow temperatures. In some cases, this can lead to insufficient room temperature.**

2.2.1. Energy Consumed during defrost

The pattern of the defrost cycle was closely observed for an 8.5kW heat pump, during the SH2.0 study, in an attempt to quantify the energy consumed during a typical defrost event. The data on a minute-by-minute basis was analysed and relevant flags that identified heating mode and defrost mode utilised to filter and assess the data. Within the control panel, a signal was initiated when the controller initiated the defrost cycle. It was noted that for approx. one minute, the heat pump continued to operate in heating mode that made it difficult to use the data collected from the controller. As a result the defrost flag was not a reliable means of attributing energy consumed to the defrost cycle. Had this been done, energy consumed and produced while in heating mode would have been incorrectly attributed to the defrost cycle.

MINUTE	CONSUMED ELECTRICAL ENERGY (Wh)	DELIVERED HEAT ENERGY (Wh)	CoP	DEFROST
1	51	144	2.82	0
2	53	123	2.83	0
3	54	125	2.31	0
4	54	120	2.22	0
5	54	112	2.07	2
6	16	48	3.00	2
7	8	0	0.00	2
8	9	0	0.00	2
9	5	0	0.00	2
10	1	0	0.00	0
11	7	0	0.00	0
12	35	88	2.51	0
13	42	83	1.98	0
14	44	92	2.09	0
15	44	111	2.52	0
16	51	121	2.37	0
17	52	127	2.44	0
18	52	128	2.46	0

Table 2.1: Defrost event - example of raw data

As the defrost cycle continued , approx. 1 minute after the cycle started, the fan switched off and the compressor continued to run at a low speed. Some seconds later, the reversing valve was heard to operate, thereby diverting hot gas to the evaporator. This process continued for approx. 2 minutes after which, the defrost signal disappeared and the compressor switched off, signifying the end of the defrost period.

Table 2.1 presents the raw data for a defrost event for SH006 from the 8th November 2018 when *Tout* was 3°C. The controller flags the defrost event in minute 5 but it is not until minute 7 that it can be said that the unit stops delivering heat to the house. Between minutes 7 and 11, a total of 30Wh of electrical energy was consumed without any related heat output. From minute 12 onwards, heat is again delivered to the house with minute-by-minute COPs that are in keeping with the start-up phase of a heating cycle. Figure 2.1 presents the overall number of defrost events for each of the 19 Mitsubishi ASHPs in the study. It should be noted that SH018 was located to the South of Cork city and so experiences a slightly milder climate that the others which are either in Mid/North Co. Tipperary or Co. Dublin. Also, SH018 only uses the heat pump to cater for a base space heat load up to 16 or 17°C with wood burning stoves doing the rest of the work to maintain required comfort levels. Taking the figure of 30Wh/defrost cycle as being representative of the energy consumed during a defrost cycle with no associated heat delivered, then the range for these 18 homes is 31.7 – 55.3kWh electrical input or an average 1% of overall energy consumed within the heating season.
In conclusion the data shows that energy consumption for defrost cycles should be within the 1% of total energy consumed for a well-designed and commissioned system. There will be variations due to external temperatures clearly but this should not be excessive on a yearly average basis.

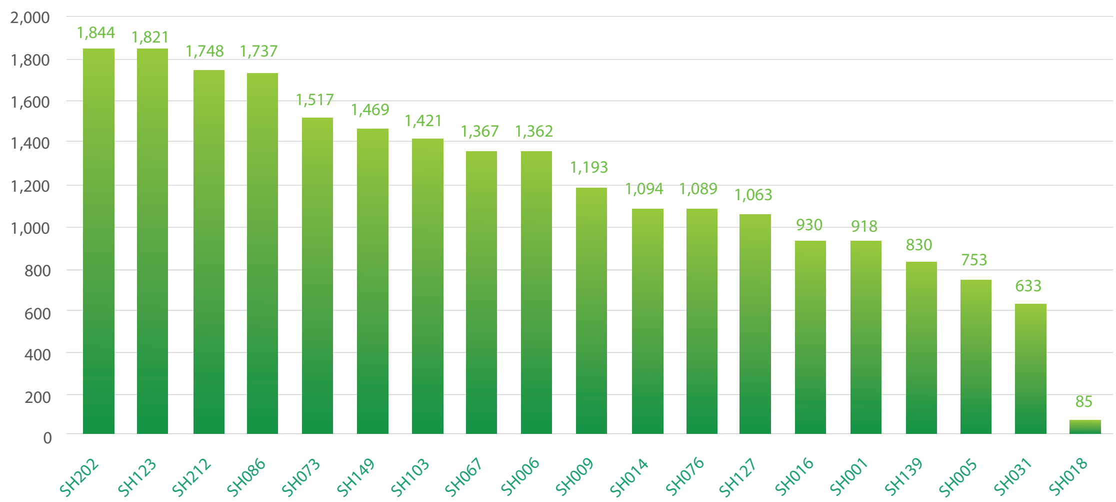


Figure 2.1: Defrost events per Superhome 2017/18 heating season

2.2.2.Energy Penalty due to Ice build-up

Figure 2.2 presents an example of the effect of evaporator ice build-up on the ASHP's heat output and consequently on COP for SH006. At minute 1, the AHSP commences operating in heating mode having come out of a defrost cycle. The Evaporator is free of ice and so the operation from minutes 1 to 14 can be regarded as normal operation, where the ASHP goes through a start-up period, adjustment of modulation and then a gradual increase in heat output to the point where output begins to level off. From minutes 14 to 31, due to ice build-up, the COP and heat output gradually fall while the electrical consumption remains steady at approx. 52 Wh/minute. During this period, the COP falls from 3.52 to 2.82. Between minutes 31 and 33, the fall in output is even more severe with the COP dropping from 2.82 to 2.22. At this point, heat output drops to zero signifying the beginning of the defrost period. Analysis of several defrost cycles has shown that the average reduction in COP for the part of the cycle where ice is forming can be in the region of 12.5% representing an average energy penalty per event in the region of 120Wh/event. Using SH006 as an example, over the 2018-2019 heating season, this penalty represents 6.2% of overall electrical energy consumed over the heating season. This loss in performance would be in addition to the energy consumed during the defrost process as outlined in the previous Section.

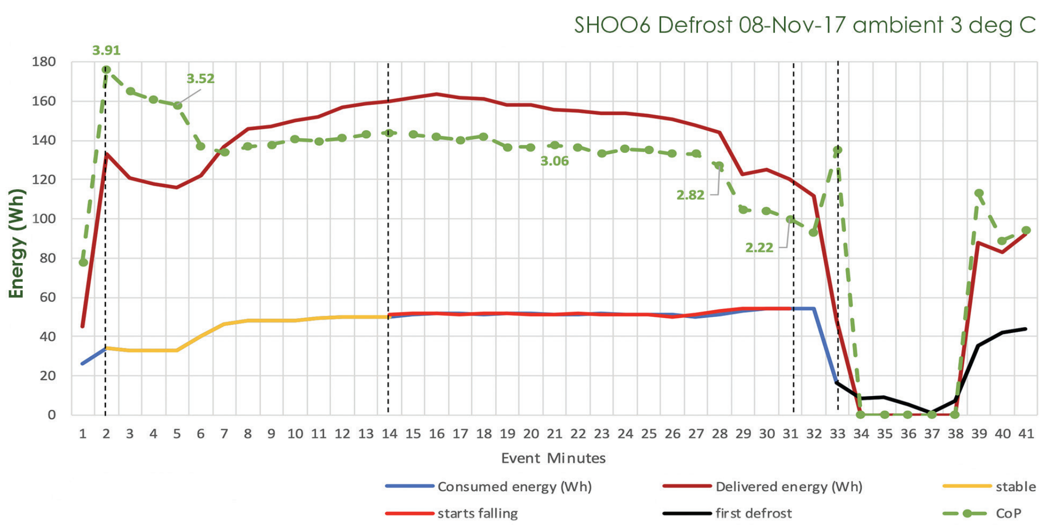


Figure 2.2: Typical defrost cycle

2.2.3. Effect of Defrost operations on Ability to achieve target flow temperature

Figure 2.3 presents a sequence of de-frost operations for a system that periodically falls slightly short of meeting the maximum demand of the house load. As *Tout* falls over a 12 hour period at night-time from 1˚C to -3˚C the load on the house rises while at the same time the potential heat output of the heat pump falls slightly. The graph shows that heat output and consequently COP, reduce as each heating operation progresses which is a consequence of ice build-up on the evaporator. Thus the heating system is challenged to achieve the internal temperatures required by the homeowner and the overall efficiency of the system is affected.

Consideration should therefore be given to slightly over-sizing an ASHP to allow for this performance reduction that results from ice build-up on the evaporator. Factors which might affect this decision are the scale of the deviation of internal temperature from target resulting from this output reduction and the likelihood of occurrence of outdoor temperature ranges. This point is further explored in Section 2.3.

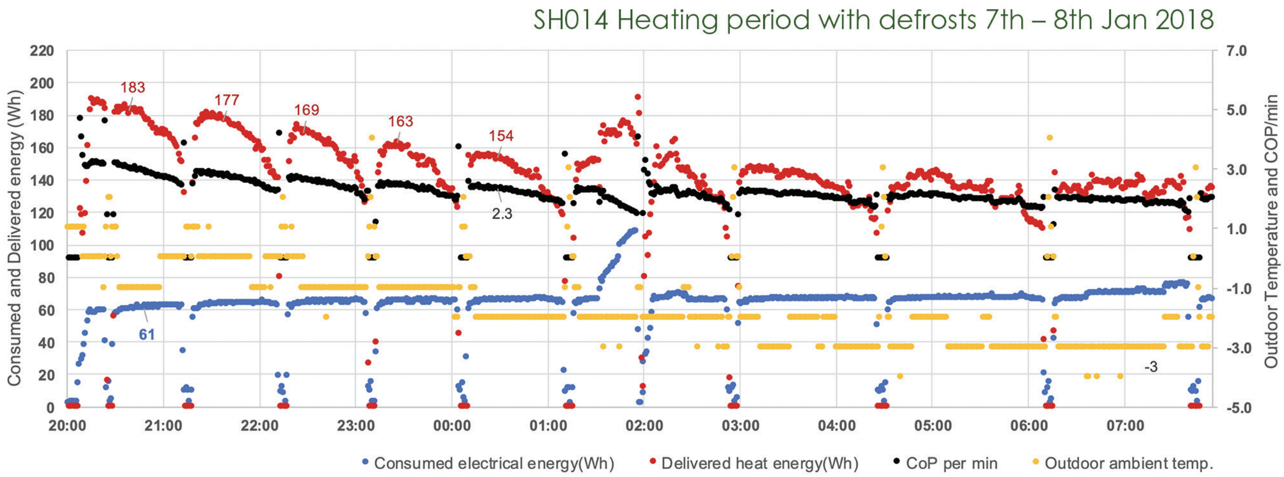


Figure 2.3: Defrost operations disrupting heat output and CoP

2.3 Over and under-sizing

Selecting a *Tdo* that is too low can result in a heat pump being oversized, assuming all other inputs to the load calculation are correct. These inputs include building fabric element u-values, building element areas, air tightness level etc. In periods of lower heating demand or in part-load scenarios, despite the presence of a variable speed compressor, the minimum output of the oversized ASHP will be too high compared to the heat load of the house and the heating distribution system output potential, thus resulting in problems such as high CC and reduced COP.

If *Tdo* is too high, the ASHP could be undersized and so may not maintain the required comfort levels in the building during cold weather. This will be further affected by defrost events in cold weather with high levels of relative humidity as the operation of the heat pump is interrupted, in some instances as frequently as every 45 minutes. Where a heat pump is sized too closely to the maximum house load, it will not have sufficient capacity to quickly make up the drop in flow temperature. Over time, this situation could potentially lead to a reduction in indoor temperatures until *Tout* increases and frost build-up ceases.

Figure 2.4 compares the achieved flow temperature to the target flow temperature for a system during a period of frequent defrost evets. Each time Tf drops from high to low represents a defrost event. Excluding the DHW spike in the middle of the graph, actual flow temperature is constantly below *Tsf* . The increase in *Tsf* after mid night is caused by a drop in *Tout* . As a result the differential between *Tsf* and *Tflow* increases. The temperature differential can be assumed to be caused by a combination of the ASHP being slightly undersized, the effects of the de-frost interruptions and the reduction in output during each heating cycle due to ice build-up.

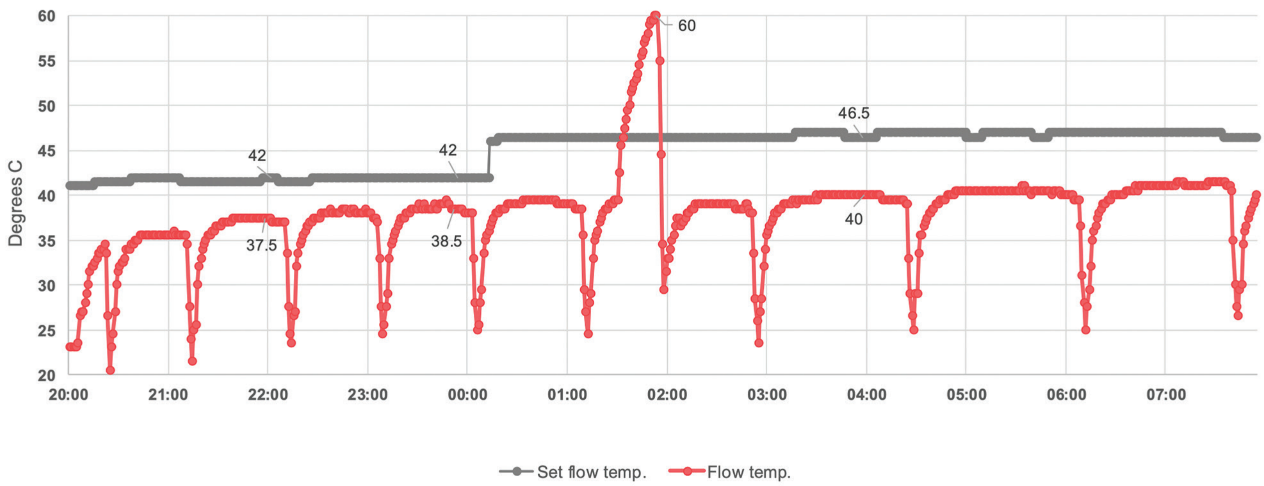


Figure 2.4: Defrost causing failure to achieve target flow temperature

Table 2.2 presents 2 of the SH2.0 systems in terms of maximum ASHP capacity, predicted maximum instantaneous house load as calculated by the Dwelling Energy Assessment Procedure (DEAP) tool and the amount of time whilst in heating mode that internal temperature was below target for each zone. The information presented includes all minutes during the heating season from October 2017 through to April 2018 when the ASHP was operating in space heating mode. The analysis demonstrated that the system with a small % of spare capacity (SH127) i.e. slightly oversized has significantly lower periods when the zone temperatures are below target when compared to the system which appears to be slightly undersized (SH014). This points to the difficulty in accurately predicting the u-values and for existing floor screeds for example.

Some heat pump manufacturers publish heating capacity tables with both peak and integrated outputs are presented, where integrated output takes into account the reduction in output associated with the defrosting process. In Table 2.3, there is a reduction in output for *Tout* <7°C whereas above this *Tout*, where very little de-frosting occurs the output is unchanged.

In this case, the integrated output and not the peak output figures should be used when selecting a heat pump to meet the design heat load of the house. This suggests that in the absence of such integrated outputs and in order to avoid a drop-off in performance in the air temperature range where defrosting happens, the catalogue peak output of the ASHP in this range of *Tout* should be discounted by 15-20% when selecting the appropriate heat pump.

Category	SH014	SH127
DEAP W/K	515	345
House Load (kW) at -3°C Tout, Tin = 20°C	11.85	7.94
Heat Pump capacity (kW) at -3°C Tout (Tf = 50°C)	11.03	8.22
Spare capacity (kW)	-0.82	0.28
Spare capacity %	-6.9%	3.5%
Zone 1 Minutes below internal target temperature	47,381	77
Zone 2 Minutes below internal target temperature	34,387	2,675

Table 2.2: Effect of undersized heat pump

Tflow = 45°C			
Tout	Peak Output (kW)	Integrated Output (kW)	Reduc-tion %
-7	9.83	8.16	17.0
-2	10.70	8.56	20.0
2	10.69	8.87	17.0
7	11.00	11.00	0
12	12.02	12.02	0
15	13.07	13.07	0

Table 2.3: Daikin catalogue data for 11kW split ASHP

Section 2.1 introduced the concept of the RHF which is intended to enable the ASHP to re-heat the house from set-back temperature to comfort level in a reasonable amount of time. The RHF is of similar magnitude to the defrost allowance demonstrated in Table 2.3 and so there is only a requirement to allow for this factor once when sizing the ASHP, i.e. heat pump capacity = maximum instantaneous heat demand + 15 to 20%.

2.2.4. Variable Speed Compressors

All of the ASHPs in the SH2.0 study have variable speed or modulating compressors. This technology enables the heat pump to regulate its heat output in response to varying heat demand from the house. The use of the Heating Compensation Curve (HCC) method of heat pump control means that at milder *Tout*, the target flow temperature (*Tft*) reduces. Modulation allows the heat pump's output to vary in response to changing house load or due to heating zones switching on or off. The designer should be conscious of the minimum output that the heat pump is rated for, and design the heat distribution system so as to prevent the heat pump from being connected to a heat load that is less than this minimum output.

Heating distribution and emission system design is critical to the efficient modulation and operation of the heat pump. While datasheets show that the COP at the minimum output is higher for the same Air/Water temperatures than at higher outputs, some instances have been found where extended periods of constantly modulating operation at low outputs actually yielded COPs that were well below the expected level.

The pattern of operation for a heat pump with a variable speed compressor should be as follows:

1. Start up and ramp up output aiming to achieve the target flow temperature

2. As target flow temperature is approached, modulate the compressor down. The degree to which it modulates back is calculated based on the difference between flow and return temperatures as well as what happens to the flow temp after each modulation. If the Tf continues to rise, further downward modulation is required; if Tf starts to fall, upwards modulation is required.

3. The system should continue to modulate to maintain a steady Tf until the internal room temperature (*Troom*) target is achieved, the level of output maintained dependent on *Tout*.

4. If there are problems with heat emission or distribution systems, a different pattern of modulation could be evident. One example would be where the output modulates immediately from max to min without any expended period of operation in the mid output range.

3. Heat Emission System

3.1. Heat Emission and its effect on Compressor Cycling and COP

Heat Pump Heating Systems fall under the category of Low Temperature Heating Systems (BRE, 2013). The primary objectives of designers should be to achieve the client's desired comfort level, aim to maximise the system's SPF while ensuring the heat pump is allowed to work within certain operating guidelines which prevent the heat pump burning or wearing out prematurely. This is achieved by operating the system at flow temperatures that are as low as possible while achieving target comfort levels as well as maximising the time that the system operates in steady state conditions thus avoiding compressor cycling (CC).

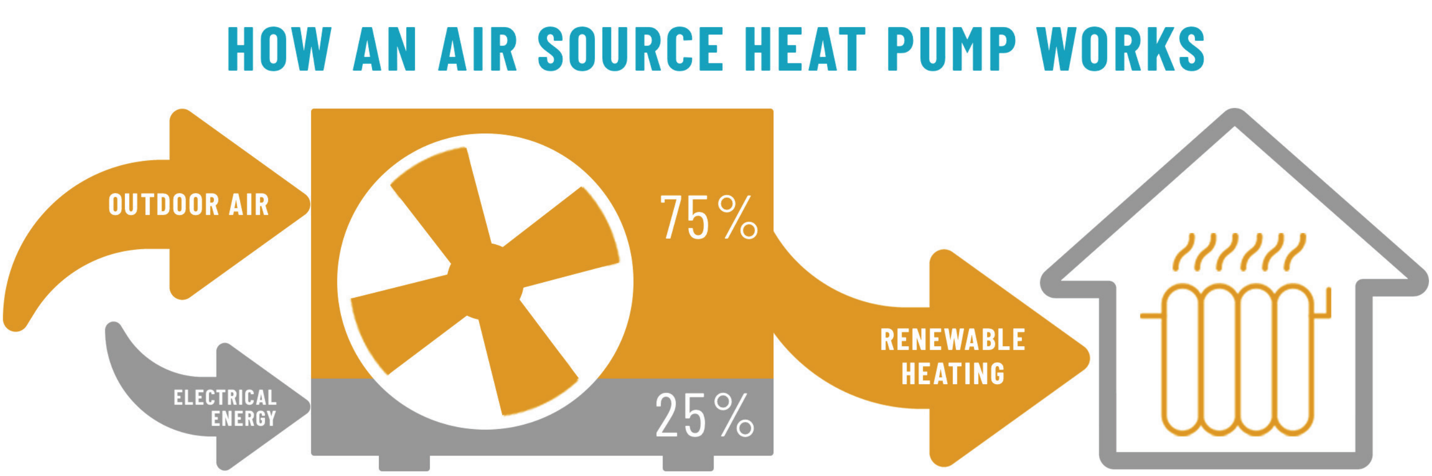
High CC can lead to:

- **Reduction in COP and thus an increase in running costs**
- **Burning out of electrical components**
- **Reduction in the lifespan of the compressor**

CC needs to be taken into account at the design stage. As noted in EN15450, "In order to minimize cycling, it shall be assured that the heating capacity delivered by the heat pump is completely transferred to the heating system". This standard goes on to suggest a target maximum of three compressor starts per hour.

This can be achieved by ensuring the following:

- **that the heat emission system design and target flow temperature (as per HCC) result in a system emission capacity (kW) that is as close as possible to the minimum output of the ASHP when** ***Tout*** **is in the range 10-15°C.**
- **that the pipework system linking the heat pump to the emission system is correctly designed so as to ensure the correct range of water flow rates can be achieved**
- **that the heating system is zoned so as to establish a minimum emission system output that the heat pump will always be connected to, and that consideration is given to maximising the water volume of the system. This point will be dealt with further in Section 4.**



A balance must be struck between the competing objectives of maximising SPF and achieving adequate system responsiveness. For retrofits with ASHPs, EN15450 sets a target SPF (Space Heating + DHW) of 2.8 and a minimum SPF of 2.5. For new buildings, the target and minimum SPFs are 3.0 and 2.7 respectively. It should be noted the SPF figures in EN15450 are for continental Europe, and so with Ireland's milder climate, it would be reasonable to expect that higher target and minimum SPF should apply. However, there is no guidance document publicly available for Ireland at present as to what these SPFs figures should be. The breakdown by type of emission systems within the SH2.0 project is shown in Table 3.1. Note that a mixture of steel and aluminium radiators were used depending on the specific design consideration for each house.

Zone 1	Zone 2	No. of Houses
Radiators	Radiators	12
Underfloor	Underfloor	3
Underfloor	Radiators	5

Table 3.1: Heat emission system per zone

3.2. Radiator Systems

EN442 is the standard which governs the rating of thermal output of radiators. The rated thermal output is given for the standard operating conditions of 75/65/20, i.e. flow temperature 75°C, return temperature 65°C and room temperature 20°C. This is presented as ΔT50, the difference, or delta (Δ) between the mean water temperature (MWT) and the room temperature. In addition, EN442 provides guidelines for a standard low temperature output at ΔT30.

For the SH2.0 homes equipped with radiators, the design flow temperature was generally 48°C. If the ΔT across the radiator is 7°C at maximum output, then the radiator MWT is 44.5°C for an external design temperature of -3°C, which when taken with a target room temperature of 20°C gives a radiator/room ΔT of 24.5°C. When selecting radiators to suit the room by room heat loss, as discussed in Section 2.4, a correction factor is applied to the catalogue radiator outputs listed at ΔT50. This correction factor is calculated from $F = (\Delta T/50)^n$ where n is approximately 1.3 for steel and aluminium radiators. Therefore, for the SH2.0 systems, the correction factor $F = (24.5/50)^{1.3} = 0.396$. If a room requires 1000W, this is corrected by dividing by the correction factor, 1000/0.396 = 2525W. From the ΔT50 outputs in the radiator catalogue, a radiator with this output is selected. When operated at ΔT24.5, this radiator will emit 1000W.

Designers need to take care to ensure that appropriate sizing calculations are completed and appropriate selection of radiators to match the heat load demands. This should also be balanced with the ability to install the relevant radiators sizes in the rooms.

3.3. Radiator Connections

When connecting flow and return pipes to radiators, Bottom Bottom Opposite End (BBOE) connection is the most commonly used configuration in Irish dwellings. As discussed in Section 3.2, radiators are selected based on the EN442 DT50 catalogue outputs. These outputs are based on Top Bottom Same End connections (TBSE). The results of a study carried out by the European radiator manufacturer Rettig is shown in Figure 3.1.

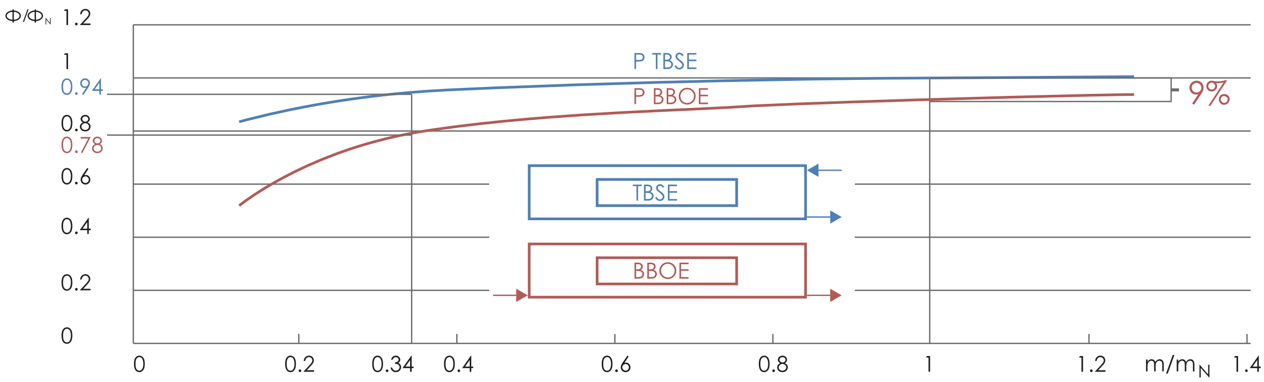


Figure 3.1: Variation of radiator heat output with flow rate

Figure 3.1 presents two issues which should be borne in mind by designers of ASHP systems connected to radiator systems. The first is that regardless of flow rate, at best, the thermal output of a radiator with BOE connections will be approx. 9% less than the catalogue, and so with reference to Section 3.2, the target output of 1000W used to select the radiator should be increased by 9% to 1090W. The second point relates to flow rate. Nominal flow rate is the flow rate that exists when the flow/return ΔT across the radiator is 10°C as per the standard excess temperature (ΔT50) test for measuring radiator heat output. Note that the nominal ΔT across the heat pump's condenser should be 5°C . In Figure 3.1, this represents m/mn =1. If this ratio is <1, the output reduces gradually until m/mn =0.34 after which, the output drops severely.

If m/mn is >1, there is a negligible increase in output. Higher flow rate will cause a reduction in flow/return ΔT across the radiator and lead to a higher MWT. However, due to the emissivity characteristics of the radiator, the increase in MWT does not lead to a significant increase in output once flow rate exceeds the standard water flow rate, i.e. the water flow rate relating to standard test conditions.
In order to gain an understanding of how water flow rate might affect the performance of the SH2.0 systems, a study was carried out into the overall ΔT as measured at the condenser of the ASHPs connected to radiators systems only, with the results presented in Figure 3.2.

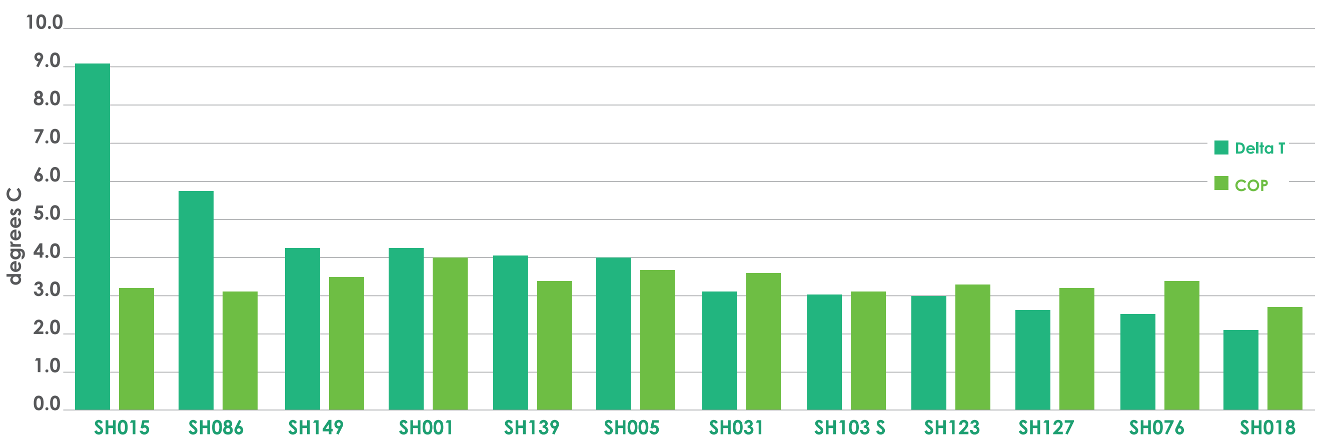


Figure 32: Average flow/return ΔT for SH2.0 systems

Noting that the manufacturer's requirement is a nominal 5°C ΔT across the condenser, 5 of the 12 systems have average ΔT within +/- 1°C of this requirement. Comparing the COP of these 5 systems (SH005, 139, 001, 149, 086) to the average space heating COP for all systems in November 2018 (3.35, calculated from SH2.0 Datasets), 4 of the 5 had higher than average COP. Of the 6 systems with ΔT between 2.1 and 3.1°C (SH018, 076, 127, 123, 103, 031), 5 of the 6 recorded COP below the average.

SH016 can be considered as an outlier. While it also had a below average COP the ΔT and COP for this system are comparable to the other systems as the ASHP is connected to a large buffer vessel located in a poorly insulated out-building.

3.4. ASHP and Radiator performance at low heat demand

In order to investigate system performance at low ASHP output, an analysis was carried out on SH086 which is fitted with an 8.5kW ASHP with radiators both downstairs and upstairs. The analysis was conducted for the period from 1/10/18 – 20/12/18.The data is grouped by CC per heating hour (CC/HH) per day, ordered from left to right according to increasing CC/hour. The data presented for each point on the CC/HH (x-axis) are manufactuer's COP (Specsheet COP) for the average *Tout* and *Tflow* for that day, actual COP, average output (kW) and *Tout*.

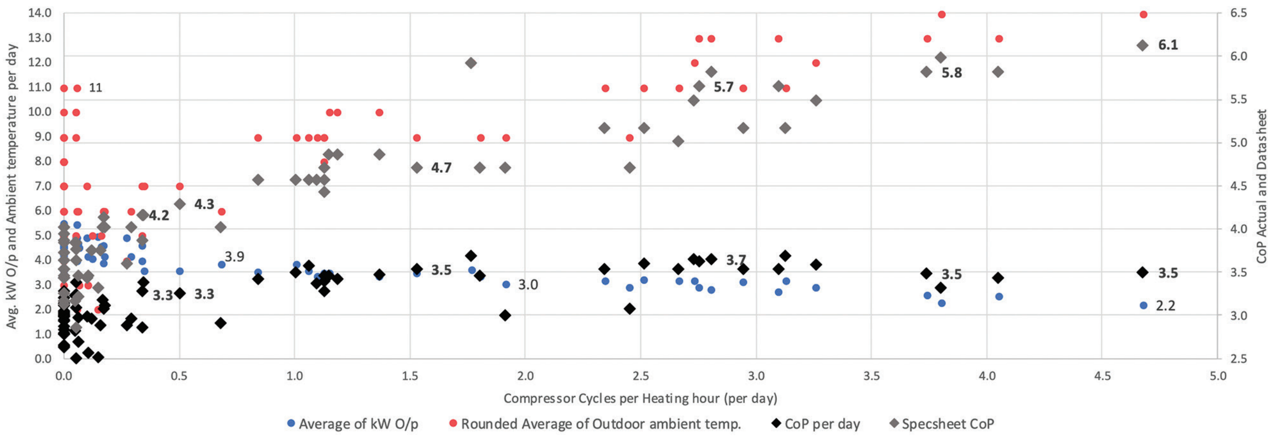


Figure 3.3: Effect of increasing CC/HH on average ASHP kW output and COP

The data presented in Figure 3.3 can be read in conjunction with the minimum heat output data presented in table 3.2. Table 3.2 shows that the lowest heat output achieved by the heat pump should be 3.2kW for the conditions A/W +2/45 and +2/50, and that the range of minimum outputs is from 3.2 to 4.9kW for the operating conditions presented. In Figure 3.3, the horizontal axis presents the daily totals of compressor cycles per hour. In the range between 0 and approx. 1.8 CC/HH, the average output ranges between 3.3kW and 5.1kW which is within the range of Table 3.2. As CC/HH increases above 1.8, the average output falls to between 3kW and 2.2 kW, which according Table 3.2., should not be possible. The graph presents average output which means that heat outputs took place which were even lower than the average figures and so such outputs should not be possible. These confused heat putput figures are the result of excessive stop/start operation where the heat pump fails to achieve steady-state operation as a result of excessive CC.

CC increases proportionally to increases in *Tout*. As *Tout* increases, the HCC generates a lower *Tsf*. If only one zone of the heating system is operating at the lower *Tflow*, the output of the ASHP could be much greater than the output of the radiator system thus leading to higher CC.

Tflow	35	40	45	50
T air	Mitsubishi W85 minimum heat output (kW)			
-2	3.3	3.4	3.4	3.5
+2	3.3	3.3	3.2	3.2
7	3.9	3.9	3.8	3.8
12	4.5	4.5	4.4	4.4
15	4.9	4.8	4.8	4.8

Table 32: Mitsubishi W85 minimum heat outputs

The reduction in *Tflow*, together with increased *Tout* should lead to higher COP, as indicated by the "specsheet COP" trend. However, the "COP/day" trend shows that the actual COP levelled off at around 3.5, despite the potential for COP to surpass 5.0. As the CC/HH figure increases, the gap between theoretical and actual COP can be seen to increase and the average heat output reduces indicating the negative impact of CC on ASHP heating system performance.

These themes are further explored in Section 3.5 where the performance of the SH086 system was assessed with three HCC settings.

3.5. Matching Heat Pump and Radiator System outputs – HCC Adjustments

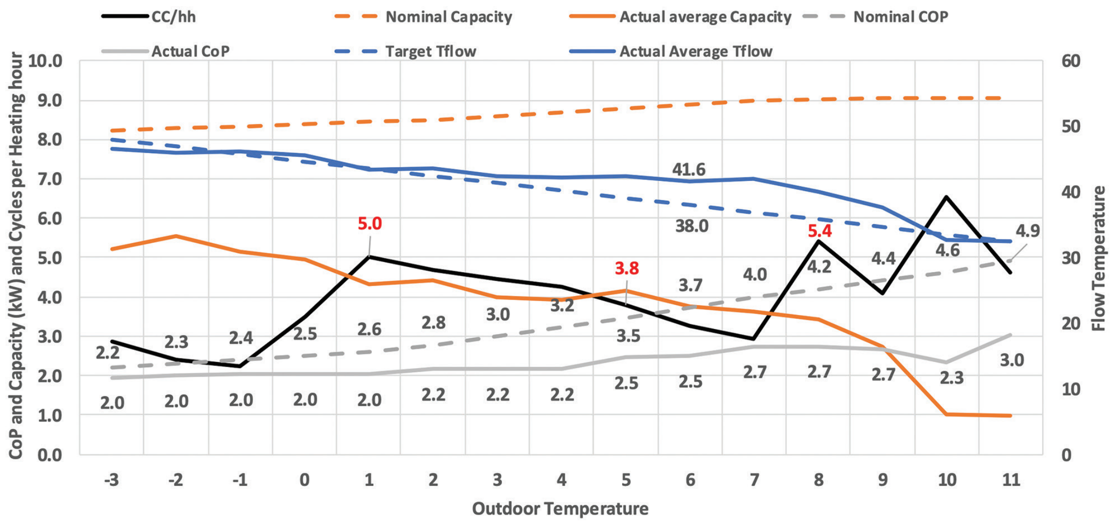


Figure 34: SH086 – low HCC

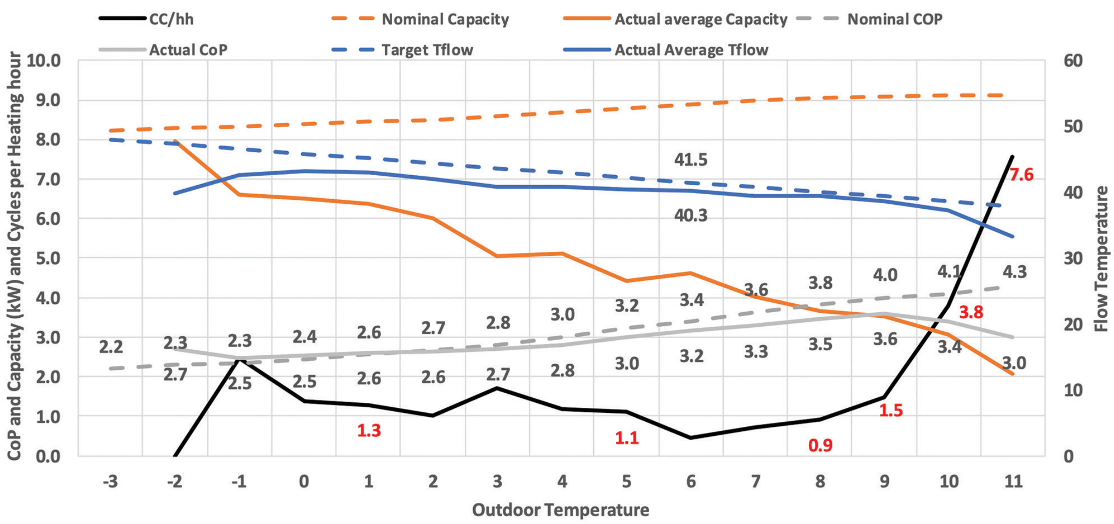


Figure 35: SH086 – medium HCC

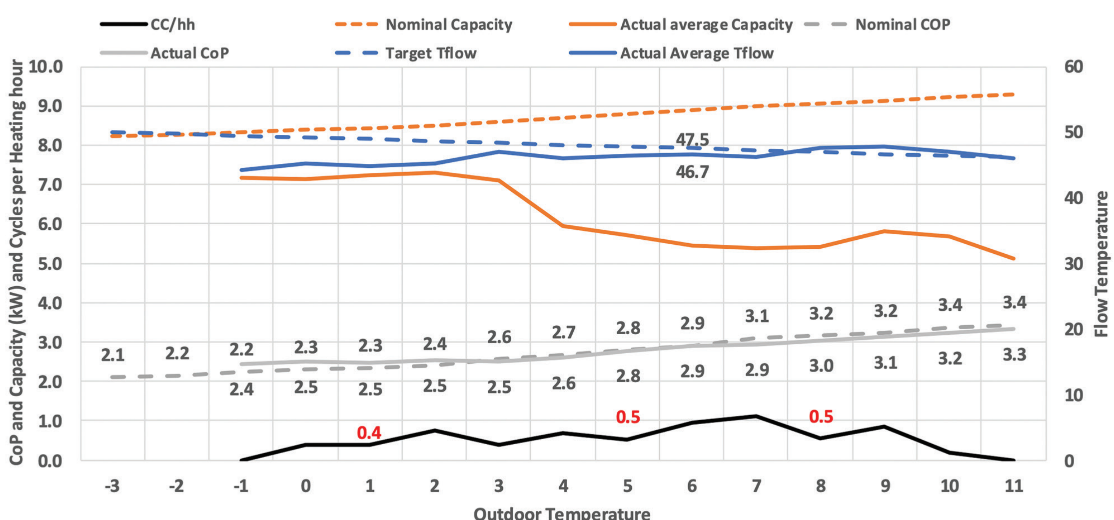


Figure 36: SH086 – high HCC

Anecdotal evidence from homeowners suggested that some systems, particularly those with radiators throughout, struggled to achieve target room temperature when *Tout* was in the range 8-12°C. Figures 3-4, 3.5, and 3.6 present three, 14-day periods where the HDD were very similar (approx. 155). For the range of *Tout*, the graphs compare the datasheet nominal heat pump output to the average actual heat output, datasheet COP to actual COP and the HCC target flow temperature to the actual average temperature.

Comparing the 45°C and 35°C settings, for the periods in question, the bulk of the heating took place when *Tout* was between 1-6°C. In this range, the actual COP at the higher setting was between 0.1 and 0.2 lower than the datasheet COP. In the *Tout* range of 6 - 9°C, the actual COP at the higher setting was 0.3 – 0.5 lower than the datasheet COP.

For these systems where the heat pumps are directly coupled to radiator systems, higher target flow temperature allows the heat pump to operate for longer before having to switch off. The benefits arising from this are a more predictable COP, much less CC and the fact that the higher flow temperature will lead to a more responsive radiator system that is better able to satisfy room temperature target. In order to balance these benefits against the need for maximising COP, it is perhaps not necessary to increase the lower end of the HCC as far as 45°C. Section 3.6 will show that 40°C should be sufficiently high.

3.6. COP Ratio

Heat Pump manufacturers publish performance data including heat output and COP which is based on tests carried out in accordance with EN14511. This standard defines a range of A/W operating conditions in terms of source and sink temperatures (Air and Water in the case of Air-to-Water heat pumps) for which heat output and energy input are measured. These conditions are carefully managed so as to ensure a constant ΔT across the heat pump condenser. For low and medium temperature applications (*Tflow* = 35 & 45°C), ΔT = 5°C, for high temperature (*Tflow* = 55°C) ΔT = 8°C and for very high temperature applications, (*Tflow* = 65°C) ΔT = 10°C. Measurements are only taken when the heat pump is operating in what can be called steady-state at the various operating conditions.

In real-world installations, ASHP system design should try to maximise the amount of time that the ASHP experiences steady-state operation as this will create the best chance of operational COPs being close to datasheet COP for any given A/W condition. Otherwise, predicting performance on the basis of datasheet COP will yield inaccurate COPs which will lead to higher than predicted energy consumption by the heat pump and thus lower than anticipated CO2 reductions.

SH #	Tout	Avg. Tf	Avg. cycles/ day	Avg. COP	Data sheet COP	COP ratio
SH127	12.5	29	124	2.6	6.30	41.3%
SH103	12.2	25	78	3.2	6.58	48.6%
SH123	11.9	31	35	3.5	5.87	59.6%
SH149	12.6	32	15	4.0	5.92	67.6%
SH001	11.6	27	28	4.3	6.34	67.8%
SH086	11.9	34	39	3.4	4.84	70.2%

Table 3.3: COP ratio in shoulder heating month (Oct-17)

The concept of a COP ratio was introduced during the SH2.0 project as a means of checking how close actual COP is to the manufacturer's data sheet for a given set of conditions. The ratio expresses the actual COP as a % of the datasheet COP. Ratios in excess of 90% have been calculated for systems operating for periods with no CC and very little compressor modulation while ratios below 60% were encountered for systems experiencing interruptions caused by high CC and excessive modulation. Tables 3-3- and 3-4 present data from 6 homes fitted with ASHP & radiators for what can be considered a shoulder heating month i.e. October and a main heating month i.e. Jan.

From Table 3.3 (shoulder heating month) a general trend can be observed where the lower flow temperatures generated by the HCC in the shoulder month led to higher CC which in turn led to lower COP ratios, with the ratio generally improving with reducing CC.

The same trend was evident for the main heating month (Table 3-4) but COP ratios were generally much improved. The actual COP figures were slightly lower than during the shoulder month due to lower air and higher water flow temperatures. However, cycles are reduced and the COP ratio has improved for all systems indicating that systems are operating closer to predicted/designed values.

SH #	Tout	Avg. Tf	Avg. cycles/ day	Avg. COP	Data sheet COP	COP ratio
SH127	6.3	34	46	2.7	4.59	58.8%
SH086	5.6	41	39	2.7	3.88	69.6%
SH103	6.2	36	32	3.1	4.4	70.5%
SH001	5.7	32	10	3.4	4.75	71.6%
SH123	5.9	36	8	3.2	4.4	72.7%
SH149	6.4	36	2	3.3	4.4	75.0%

Table 3.4: COP ratio, core heating month, Jan-18

The application of the COP Ratio is a useful tool to assess actual COP performance against predicated (manufacturers test data) performance. This tool can then be used to highlight operational problems with the system e.g. low flow temperature causing high compressor cycling.

Table 3.3 and Table 3.4, together with the data presented in Figures 3.4, 3.5 and 3.6 from Section 3.5 demonstrate that it is possible for systems to display COPs in the region of 3 to 4.

Considering that the minimum heat output of the ASHP at SH127 is approximately 4.5kW, it is clear that cycling will start to occur with *Tflow* < approx. 40°C with all radiators working, i.e. not zoned off. In reality this situation (both zones working together) is not always present as evidenced from data from January 2018, when both zones were open 20% of the time, with 66% of the time spent heating zone 1 only and 14% heating zone 2 only.
A trial was commenced in April 2018 whereby the HCC was changed to -2/48, 15/40. Table 3-7 presents the key findings from a comparison of the period 1st October to 28th February 2018 & 2019. The table shows that the higher HCC yielded a significant 97% reduction in CC, a 100% increase in the number of occasions where the zone target temperatures were achieved, a 15% increase in COP. All of this was achieved with no increase in energy consumed (EnCon/Degree Day measured in kWh/HDD).
This demonstrates that appropriate commissioning can result in improved performance of the ASHP with improved comfort levels.

3.7. HCC Adjustment and Zone Stat Satisfaction

Data recorded from SH127 during the 2017-2018 heating season highlighted this system as having high CC and low COP. This system was compared with a system which was performing well (SH001) in Table 3.5. The data used for this comparison is from the period October 2017 and May 2018. As can be seen CC in SH127 was close to 4 times that of SH001 and COP (Space heating) was >20% below SH001. This significant difference in COP performance prompted in-depth analysis of the SH127 data.

The HCC setting for SH127 at the start of this study was -2/48, 15/28 and it was found that the 20°C range in flow temperature yielded very different types of ASHP operation at either end of the range. Paradoxically, the system worked better at colder *Tout* as evidenced by frequent zone stat satisfaction events (ZSS) while it was unable to achieve the same result for milder *Tout*. At higher *Tflow*, the radiator system output was better matched to the range of ASHP heat output, and so CC disappeared. At lower *Tflow*, the radiators couldn't dissipate the heat being generated by the heat pump even as it modulated down to minimum output. In such circumstances, *Tflow* begins to overshoot its target and so the heat pump has no choice but to switch off. To further complicate the matter, the AHSP in question did not have a safeguard within its control system to limit the number of compressor starts as evidenced by the compressor re-starting after only 1 minute in most cases.

Table 3.6 provides a sample of the range of heat outputs for a radiator system sized to deliver a peak load of 8kW with Tflow set to 48°C, at design conditions.

3.8. Underfloor Heating vs Radiator Systems

In retrofit scenarios it should be borne in mind that existing installations may have been installed where the building regulations at the time of construction would have required lower levels of insulation than the current regulations. Indeed, the Part L 2011 calls for the u-value of a floor with underfloor heating to be better than a standard non-underfloor heating floor – 0.15W/m2K vs 0.21W/m2K. Accurate information regarding the build-up of existing underfloor heating systems is critical to enable accurate prediction of running costs and if under sizing is to be avoided, however this is not always possible in a retrofit scenario.

Underfloor heating systems are designed to operate at lower temperatures than radiator systems (30-40°C vs 40-55°C) and so could be expected to deliver higher SPFs. Figure 3.7 compares the average of the October and November 2018 COPs of 14 of the SH2.0 homes where the hatched bars represent homes where the majority of the space heating is provided by underfloor heating and the solid bars represent radiator systems.

System	Total CC's	Space Heat COP
SH001	2,893	3.6
SH127	10,385	2.8
Underfloor	Radiators	5

Table 3.5: Example relationship of COP with CC

Tflow (°C)	Tret (°C)	MWT (°C)	ΔT (°C) – Rad-Ro-om	o/p (W)
48	41	44.5	24.5	8,000
45	40	42.5	22.5	7,162
40	35	37.5	17.5	5,166
35	30	32.5	12.5	3,335
28	23	25.5	5.5	1,147

Table 3.6: Range of radiator system output with varying flow temperature

SH127	1/10/17 – 28/2/18	1/10/18 – 28/2/19
HCC	-2/48, 15/28	-2/48, 15/40
HDD	1365	1208
Total Heating time (hours)	2509	1197
Energy Consumed Space Heating (kWh)	2284	2021
Running Cost (€)	457	404
Total CC	9087	200
Total occurrences of zone stat satisfied	416	822
Total heat delivered in SH (kWh)	6058	6331
SH COP	2.7	3.1
EnCon/Degree Day (kWh/HDD)	1.7	1.7

Table 3.7: SH127 HCC Trial Results

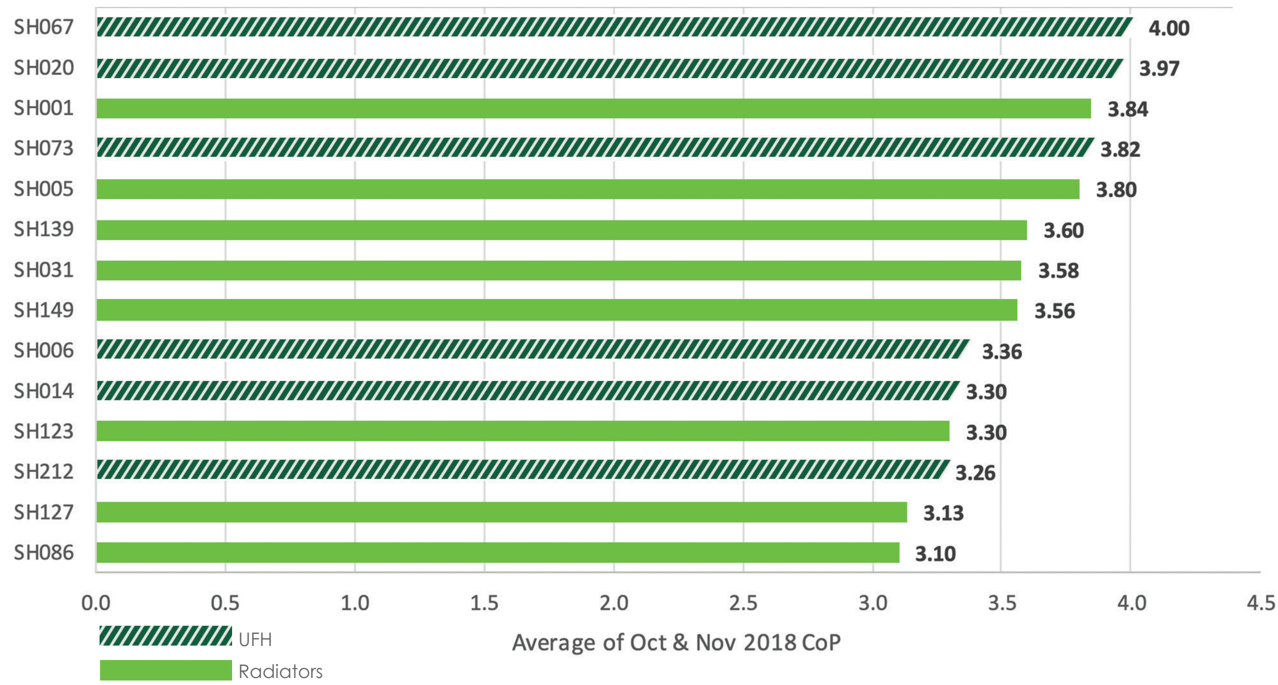


Figure 3 7: Comparison of COP for Radiator and Underfloor systems

Figure 3-7 shows that three of the underfloor systems are among the highest performers but that the other three perform less well, with a number of radiator system returning very high COPs. It has been the experience of the SH2.0 team that a large variety of factors are at play when considering the range of COPs for the different systems. Two of the underfloor systems pre-dated the Superhomes upgrade and so it might be expected that these would compare unfavourably. However, the best performer, SH067 was one of these, the other being SH014.

With regard to the better performing radiator systems; frequently systems were found to operate with high COPs (>3.5) but closer examination highlighted that the system may be only just maintaining internal temperatures, and so these high COPs might be deemed unrealistic.

In these cases, a sudden drop in *Tout* often led to significant periods where internal temperature was below target. A number of tests were carried out where the HCC was adjusted to increase the system operating temperature. Significant improvements to internal comfort levels resulted from relatively small increases in energy consumption and reduction in COP. Examples of these tests have been provided in Section 3.6.

Underfloor systems designed for use with heat pumps generally have pipes spaced 100-150mm apart – greater pipe density allows the screed to operate at lower temperatures. Where mixed systems (UFH + radiators) are used in a two storey dwelling, it is advisable that radiators are only used in first floor sleeping areas where the target temperature is lower, and where there is a tendency for significant heat gain from the lower floor. In such cases, the heat pump can take full advantage of the lower operating temperature afforded by good underfloor design. Problems can arise if mixed systems involve radiators in core living areas of the house. Radiators in these locations would have more work to do than those in first floor bedrooms and so would need flow temperatures higher than the underfloor design requires thus reducing COP.

Care should also be taken where ASHPs are connected to existing manifolds which were designed for boiler temperatures. These systems were often designed such that the water is circulated through the floor circuits by one pump while a second pump circulates between the ASHP and the manifolds with a 2 or 3-port mixing valve introducing high temperature water as required, while limiting the flow temperature to the floor to below 50°C. It is the recommendation from the SH2.0 project a better alternative solution is to have the full flow from the heat pump pumped directly into the underfloor flow pipes so as to ensure full engagement of the ASHP with the heating load. However, this was not always the case for the underfloor systems in SH2.0. The negative effect of CC on COP has been discussed in Sections 3.1, 3.4 & 3.5 in the context of radiator systems where operating radiators at low temperature can cause a mismatch between ASHP and emission system outputs. High levels of CC were also encountered in underfloor heating systems, with the main cause of this being the over-use of room temperature controls. Most of the systems had room thermostats in each room controlling electric actuators turning UF heating zones on/off. In these systems, if only one room calls for heat on its own, there is typically a relatively low water volume, the benefit of the large thermal inertia is greatly diminished and the mismatch between ASHP and emission system outputs is created thus leading to high CC. This can be addressed by reducing the number of thermostats to the minimum required to ensure even room temperatures in the zone, making sure that overheating and under heating is avoided. If this is not possible due to house design or orientation, a buffer vessel should be considered. (see Section 4.4).

In addition to the benefit of potentially lower flow temperature, underfloor heating has the advantage of having a far greater thermal inertia compared to radiator systems. This is ideal for heat pumps, especially those fitted with variable speed compressors as it should be very easy for the heat pump to continue to operate without any CC until the room temperature target is achieved.

4. Heat Distribution System

In the context of heating emission system design, Section 3.1 referred to EN15450, where the importance of ensuring that the heating capacity delivered by the heat pump is completely transferred to the heating system is noted. Two important ways where the heating distribution system design can affect this transfer are:

1.The extent to which the emission system output can be reduced through zoning

2.The effect of the distribution system hydraulic design on water flow rate, and the effect of flow rate on heat pump performance, especially compressor modulation.

4.1. Zoning

As far as zoning of a heating system is concerned, aside from stipulating that space and hot water heating should be zoned separately, a supplemental guidance document to Part L of the 2011 building regulations technical guidance document (TGD) also mentions that:

- **The water distribution system should be arranged for reverse return operation to maximise efficiency and ease commissioning and future maintenance**
- **Constant water flow should be maintained through the heat pump**
- **Pipe sizes should be in accordance with the manufacturer's recommendations**

The experience of the SH2.0 project was that heating system zoning design has a very significant effect on how the system performs. 18 of the 20 systems in SH2.0 were designed so that the heat pump is connected directly to the heating distribution system with 2 being connected to pre-existing buffer vessels. All systems were installed as 2-zone space heating + DHW. In the absence of a buffer, 2-zone heating systems regularly create situations where the heat pump is operating with approx. 50% of the heat emission system volume and emission capability available to the heat pump. The impact of this is considered here.

In Table 3.1, eight of the SH2.0 homes can be seen to have underfloor heating in the living area zone. One of these, SH014, also had underfloor in the upstairs sleeping zone. SH014 and SH067 differed from the other underfloor homes in that they did not have further sub-divisions of control within each zone. When the zone called for heat, all circuits within that zone were active. The other underfloor systems had individual room controls whereby it could be possible for the heat pump to be controlled to operate with only one room calling for heat. With a minimum heat output of approx. 5kW, and a possible room load of perhaps only 1.25kW this control set-up has the potential to generate high CC as was found on occasion in SH073 and SH212. Building design, solar gain and temperature settings play significant roles in determining how best to design the control system for an underfloor heating system, but situations where only one room can call the ASHP to operate should be avoided.

With the radiator systems in the group, living and sleeping zones all had the potential to be further subdivided through the use of thermostatic radiator valves (TRV). However, in all cases, the TRV's were set to maximum and so the room temperature would have to exceed 25°C before switching off a particular radiator, hence TRV's (set to maximum) were not found to cause interruptions to system output.

When zoning the heat emission system, the minimum emission capacity that the heat pump can be connected to should ideally be equal to or slightly greater than the minimum output of the heat pump at all times. This can be achieved by grouping all living and circulation areas and bathrooms so that they all operate together, and by controlling bedrooms separately using TRVs or room thermostats. The provision of a buffer vessel also warrants consideration. This would provide a thermal inertia to ensure the heat pump can operate a minimum run-time appropriate to the system. This together with control of how long the compressor stays off is a common heat pump control strategy for minimising CC.

6 Heating and Domestic Hot Water Systems for dwellings – Achieving compliance with Part L 2008

As well as providing a buffer-volume for sustain heat pump operation in times of low demand, the buffer vessel also provides the heating system with a thermal inertia which will maintain temperatures in the distribution system when the heat pump is off. If for space or cost reasons it is decided that a buffer is to be avoided, the following steps could be considered in order to help avoid CC:

- **Minimise zoning**
- **Oversize radiators even beyond the design ΔT of 24.5°C**
- **Provide extra radiators within zones**
- **Control strategies that switch the system off for periods so that when it becomes active again, the heat pump must run for longer**

Given that the SH2.0 homes were designed as conventional 2-zone space heating systems, the closest scenario to the ideal design described above would be when both zones were operating together. This could be described as full-load operation. The comprehensive data collection system installed on the SH2.0 systems meant that it was possible to identify when the systems operated in full or part load. Figure 4.1 presents a comparison of AHSP COP in both scenarios for a selection of homes. The general trend in Figure 4.1 is that the COP is higher when the ASHP operates in full load vs part load. This is due to the fact that with a larger heat load relative to the minimum output of the heat pump, there is a much better chance for the system to operate in steady-state conditions, i.e. reduced amount of modulation and of CC. This enables the ASHP to spend more time operating closer to the test-house conditions which are the basis of published COP figures. Maximising the minimum load that is available to the ASHP is a key step in optimising ASHP retro-fit system design and therefore designing should take careful consideration of the heating control solutions and set up when designing ASHP retrofits.

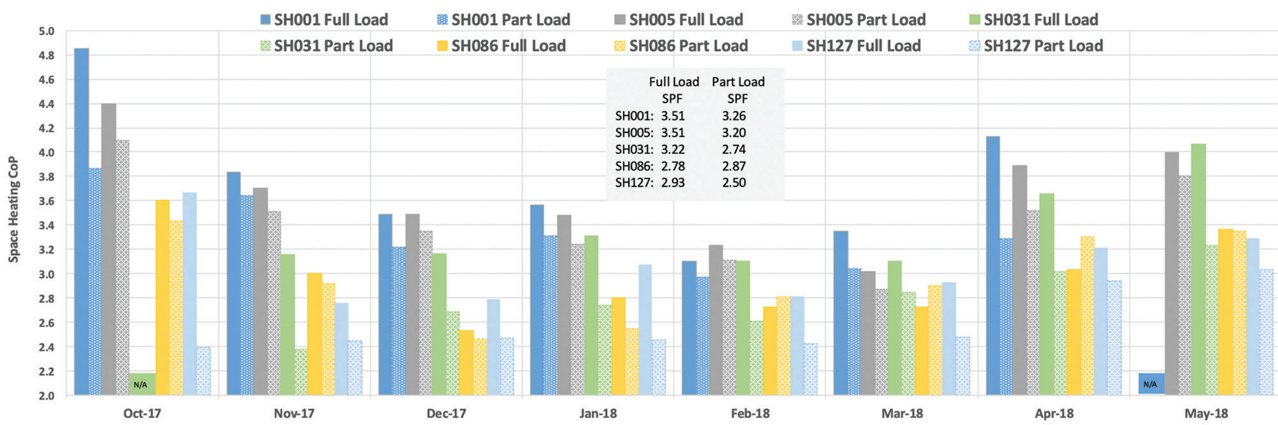


Figure 4.1: Full load vs Part load COP

4.2. Water flow Rate

As noted in Section 3.3, the nominal heat pump flow rate is such that the ΔT at the condenser is 5°C while design ΔT for radiators is 10°C. This means that at full load, the flow rate through the heat pump should be twice that through the radiator system. The ASHPs in 18 of the 20 SH2.0 systems were directly coupled to their heating systems meaning that the flow rate through the heat pump and emission system was the same which led to overall ΔT's in the range 0.5 - 12°C during normal heating operation. Anecdotal evidence from heat pump manufacturers suggests that designers allow for systems to operate with condenser ΔT in the range 5-8°C. However, ΔT above 5-10°C will lead to lower heating system mean water temperature thus reducing the emission system's output leading to the heat pump switching off prematurely, increasing CC and room temperature target not being satisfied. On the other hand, if flow rate through the heat pump is too high leading to ΔT <5°C, a reduction in potential COP can be expected.

It has been shown in Section 3.3 that the output of radiators with flow rate greater than nominal flow rate does not change significantly and so it could be argued that it is ok to set the system flow rate to the higher flow required to maintain a ΔT of 5°C.

In some systems, such as SH127, the existing distribution pipework allowed this to happen. However, an impact of this setting was a high level of noise at the radiator valves due to water circulation. While the homeowner in SH127 was not concerned by this noise, other homeowners did raise concerns about similar noise levels. By contrast, the configuration of pump and pipework in SH086 yielded lower flow rates. While this setup resulted in lower noise levels at radiator valves, the lower flow rate makes it easier for the heat pump to achieve its target flow temperature resulting in it operating at minimum output for longer periods than desired. This leads to long run times before the zone stat is satisfied.

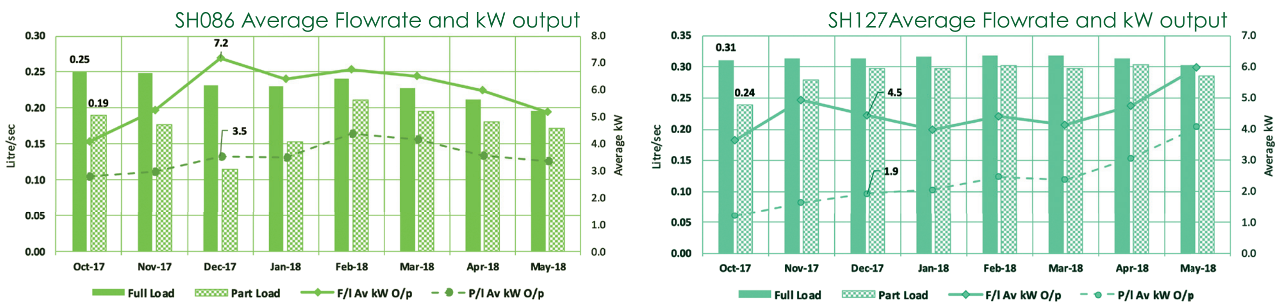


Figure 42: (a) SH086 and (b) SH127 Full and Part load

For both systems, the nominal full and part load AHSP flow rates should be 0.39 L/s and 0.22L/s, respectively. From Figure 4-2, the average actual flow rates were 0.23 and 0.17 L/s for SH086 and 0.3 and 0.28L/s for SH127. For the radiator systems, the ideal flow rates would be half the heat pump flow rates as the ideal radiator flow/return ΔT is 10°C as opposed to 5°C for the heat pump and so ideal radiator system flow rates would be 0.19 and 0.11 L/s for full and part load. In both examples presented, the full and part load flow rates are higher than that required by the radiator system but lower than that required by the heat pumps. With reference to Section 3.3, the radiator system output could theoretically be slightly higher as m/mn is >1 (at full load, 1.2 for SH086 and 1.6 for SH127) but experience from site visits has highlighted issues with noisy radiator valves due to high flow rate. From the heat pump point of view, the flow rates are lower than ideal and the concern here is that the heat pump would tend to modulate down its heat output prematurely leading to longer than necessary run times before zone target temperature is achieved.

In systems operating at relatively high flow rates, balancing of radiators can be difficult. Instead of achieving ΔTs of 10°C, it was more common in the SH2.0 homes to see ΔTs of 4-7°C.

In some cases, radiators close to the circulating pump were operating with ΔTs of 1°C, effectively acting as a bypass and thus prematurely warming up the return water to the heat pump. This would give the heat pump a false impression of the house heating load and begin to reduce its output. Some work was done in one particular building in an attempt to address this situation. However, while improved ΔT of 2°C was achieved by closing the lock-shield value the associated noise was unacceptable.

In commercial radiator systems, the lock-shield valve is not used to regulate the flow through the radiator. Instead it is only used as a service valve to enable a radiator to be removed. In such cases, the system is designed to generate a specific pressure differential (ΔP) across each radiator. With knowledge of this ΔP and the flow rate for each radiator, pre-set TRVs are used where the valve setting is selected from a chart. The TRV sets the flow rate and it switches the radiator on and off in response to room temperature. This approach is taken as it is a more accurate method of setting flow rate – lock-shield valves are not refined enough to give this level of control. However, these valves will only work if the system hydraulic design allows for sufficient pressure differential between the flow and return connections of the radiators. If in an effort to satisfy the flow requirements of the heat pump the flow rate is much higher than that required by the radiator system, even these more advanced control valves may not be capable of producing the required results. A potential solution to this issue is presented in Section 4.4.

4.3. Variable Speed Pumps

As noted above, all the SH2.0 ASHPs are fitted with variable speed compressors. In order for a constant ΔT to exist across the condenser, the condenser pump should vary its speed in conjunction with the compressor. This would require the pump to be capable of speed control under the command of the heat pump's controller. In 18 of the 20 SH2.0 systems, external condenser pumps were fitted. While these are modern pumps with their own variable speed drive, the external condenser pump cannot be controlled by the heat pump. Furthermore, attempts to set such pumps to modulating mode only caused them to ramp down to very low speeds due to the relatively low back pressure in the systems tested. The other two homes were fitted with split units where the internal unit contained the condenser pump which is variable speed and controlled by the heat pump controller. As a result, where split systems are installed there are opportunities for great control of the condenser pump to match demands.

4.4. Hydraulic De-coupling

The conflict between the flow rate required by the heat pump and the emission system causes the following issues:

- **Unacceptably high water-noise levels in the distribution and emission systems, especially with aluminium radiators which seem to amplify noise more than steel radiators**
- **Difficulty in balancing radiators leading to uncertainty regarding their true heat output**
- **The modulation of the heat pump's output is being done based on a compromised flow rate causing sub-optimal operation in terms of heat output and COP.**

Two solutions exist to help with this situation:

- **Buffer in parallel as per the schematic in Figure 4-3**
- **Automatic bypass valve**

The buffer volume as proposed is that required to enable the heat pump to operate for 10 minutes at minimum output at a Flow/Return ΔT of 5°C. For the 8kW ASHPs in this study, that would equate to a tank volume of 140L. In the schematic, double regulating valves (DRV) are shown. These would be commissioned to set the maximum flow rate to each zone of the heating system so that, the radiator system would only receive the flow rate required to achieve a ΔT of 10°C. In the scenario that the heat pump controls the speed of the condenser pump, when the flow rate leaving the heat pump is higher than is possible for the DRVs to allow through, excess flow bypasses through the buffer tank. Configured as shown, the flow from the heat pump is capable of getting directly to the radiators without being mixed in the buffer tank and reduced in temperature.

The provision of a buffer would also enable the HCC to be set slightly lower for milder Tout as the buffer would act as a deterrent to CC.

Automatic bypass valves (ABV) are used by some designers. In such cases, the condenser pump speed is controlled by the heat pump. As zones or TRVs close down, less flow is required by the emission system than by the heat pump. The pump will continue to try to achieve the heat pump's desired flow rate but because part of the emission system has closed off, there will be an increase in pressure in the distribution system which causes the valve to open automatically and allow some of the flow to bypass and return back to the heat pump.
In order to maximise the volume of water available to the heat pump when the ABV operates, it should be connected to the main flow and return pipes as far as possible from the ASHP.

Both situations can be considered when design is being completed. The buffer solution has the advantage of increasing the system volume thereby increasing the amount of time that the heat pump can run when emission and output are mismatched. The trade-off is between optimal performance and space/additional cost implications of installing a buffer vessel.

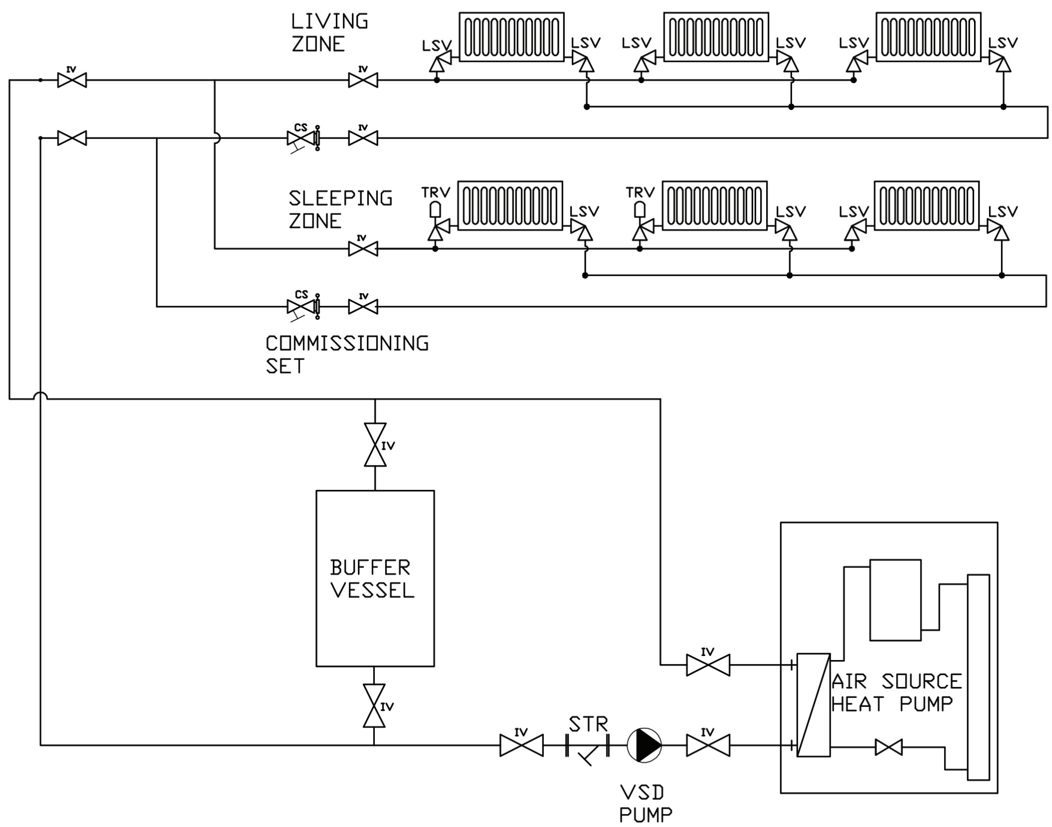


Figure 4 3: Buffer vessel in parallel – design by SH2.0 team

5. Domestic Hot Water (DHW)

EN15450 stipulates that the DHW system shall be designed based on the maximum daily hot water demand and relevant tapping pattern. This will enable the designer to determine the correct volume for the DHW storage tank, and the required heat pump capacity required to re-heat the tank within a specified time period. In the SH2.0 homes, the storage tanks ranged in volume from 200-400L with the majority (14/20) at 300L. In all cases, DHW heating received priority so that when there is a DHW demand, the heat pump is directed to only heat DHW so that all of the ASHP's heat output and temperature potential is available for hot water heating. The SH2.0 ASHPs ranged in capacity from 7.5-16kW with a majority of them being rated at 8.5kW (14/20).

Typical target tank temperatures for the SH2.0 homes were in the range 50-52°C whereby the heat pump would maintain this target without any electric immersion assistance. Approx. once a fortnight, a Legionella programme is intended to operate so as to heat the tank to 60°C or more, through operation of the heat pump up to 52°C with the final 8°C lift achieved by the electric immersion.

The tank temperature at the start of a DHW cycle depended on factors such as time schedule, demand and the presence of a solar thermal or PV with immersion dump. Typical daily DHW cycles saw the ASHP commencing DHW operation when tank temperatures ranged from 10°C to 35°C and so with the average target temperature being 50.4°C, the required DHW tank temperature increase required would take between 0.6 and 1.65 hours if the heat exchanger or coil in the tank was 100% efficient. Figure 5.1 shows, 15 of the homes which had regular DHW use, the number of DHW events and the total time spent by the ASHP in DHW mode for the period Oct-17 to May-18, a period of 243 days.

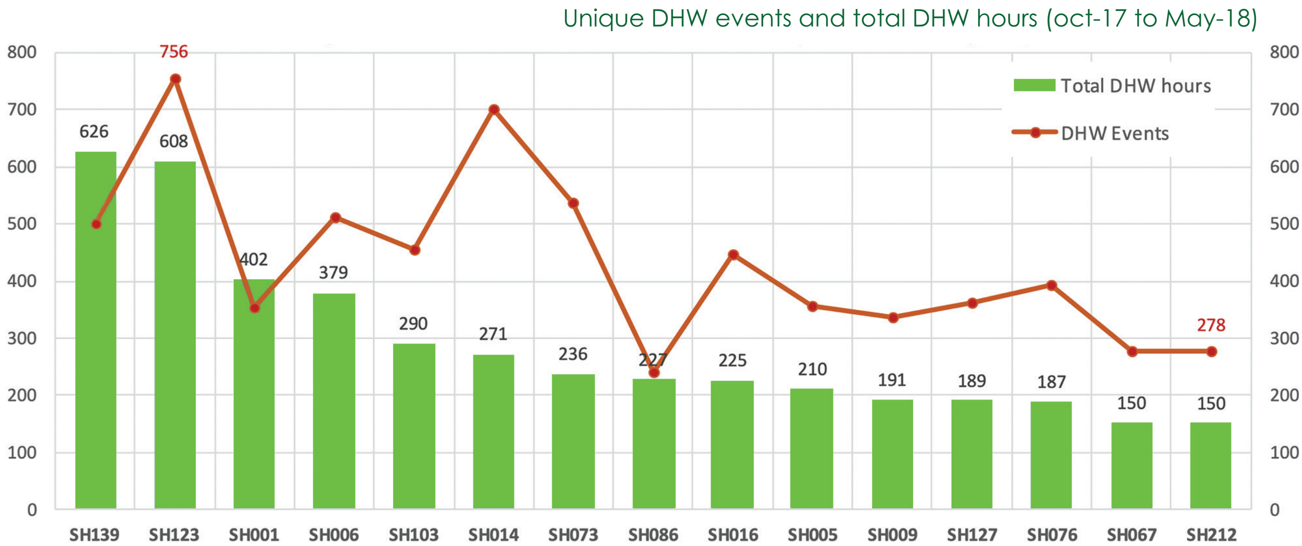


Figure 5.1: Unique DHW events and total DHW hours (oct-17 to May-18)

The graph is ordered left to right according to highest to lowest total hours in DHW mode with the number of DHW events generally following the same trend. The homes with the lower total hours tended to have less than 400 events which would equate to approx. 1.6 events per day, a reasonable figure given that the average storage volume was 300L and so generally, one DHW cycle per day should suffice for most days. Higher numbers of DHW events for the remaining systems can be explained by the lack of time schedule, time schedules with 3 operating periods due to high demand and problems relating to the heat exchanged in the DHW tank. In some cases, existing tanks were retained because they were part of a solar thermal system – these tanks did not have coils sufficiently large to dissipate the heat pump output and so the heat pump was forced to modulate to minimum output, thus extending the time spent in DHW mode. The 4 homes with the highest total hours in DHW mode all had average DHW event times in excess of 44 minutes, with SH139 = 75minutes and SH001 = 68 minutes whereas the average times for SH076, SH067 and SH212 were between 29 & 32 minutes.

6. Controls

6.1. Room Temperature Control

Irish Building regulations Technical Guidance Document part L, 2011 – Conservation of Fuel and Energy, Dwellings, Section 1.4.3 stipulates a requirement for the "automatic control of space heating on the basis of room temperature" by the use of "room thermostats, thermostatic radiator valve (TRV), or other form of sensing device". For the SH2.0 homes, combinations of all 3 methods were used. The most common system occurred when both living and sleeping zones were heated by radiators. In these cases, each zone was controlled by a wireless master room temperature sensor, with individual rooms provided with TRVs. In practice, the ASHP performs best when TRVs do not introduce interruptions to the heat emission system and so homeowners were advised to set the TRVs to their maximum setting so that they would not close down, except in the case of bedrooms where it is necessary to maintain a lower temperature than in living areas.

The other room temperature control system encountered occurred with underfloor heating systems, where each room was controlled individually by room thermostats which are wired back to a wiring centre where they cause electric actuators to open and close the pipe circuits to their respective rooms. Once the wiring centre receives a demand from any individual room, it sends a signal to the heat pump to commence heating. While this system is designed to ensure that each room can be independently controlled, in reality, the rooms in a zone will generally tend to arrive at the same temperature over time. Variations can occur due to different solar gain levels e.g. significant solar gain on the one hand and perhaps significant amounts of North facing glazing on the other. The SH2.0 homes fitted with UF Heating systems had 4 to 9 individual room thermostats which led to periods where the heat pump was sometimes found to be operating with only one room circuit, leading to high levels of CC. In such cases, it consideration should be given to minimising the number of room thermostats by grouping stats into zones e.g. rooms grouped together on the basis of orientation or, as in the case of bedrooms, different temperature requirements.

The position of the thermostat is very important.Where only one stat is used to control a zone, a wireless stat is best so that it can be moved around to find the most representative location. The thermostat should be located in an area that is affected by the convective air currents created by the heat emission system but must be far enough away from any individual heat emitter so as not to switch off to soon, to the detriment of cooler parts of the zone.

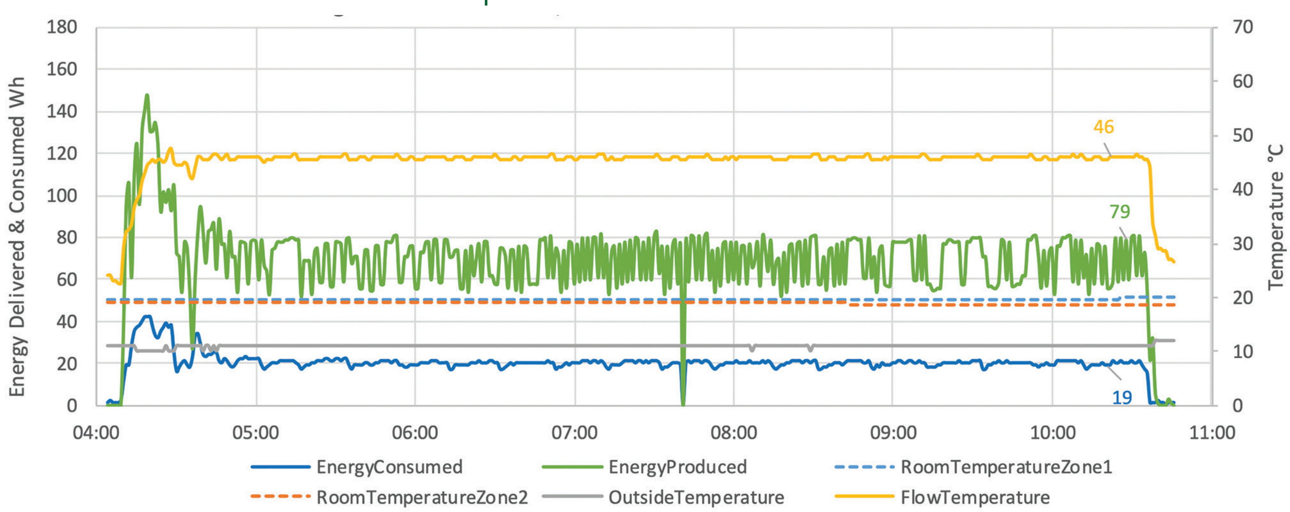


Figure 6.1: Poorly positioned room sensor – SH086

A case in point was SH086. The wireless thermostat was positioned on a countertop in the kitchen which was completely shielded from the radiator in that room by a full height kitchen unit. Figure 6.1 below demonstrates how the ASHP maintained a flow temperature of approx. 45°C from 04:22 until 10:37 without the ASHP switching off.
At 10:37, the zone 1 room sensor detected the 0.5°C increase from 19.5 to 20°C required for the zone target to be satisfied. As *Tout* was 11°C, the house load was well within the capacity of the 8.5kW ASHP and so it should not have taken over 6 hours to raise the zone temperature by 1°C.

At 4:30pm on the 22/2/2019, the wireless room sensor was moved to a different location within the same room, one that would be influenced to a greater degree by the radiator in the room. Figure 6-2 shows the ASHP switching off after periods of operation of 1 to 2 hours and remaining off for periods of 1 to 2 hours. Over the course of a week's trial, the homeowner did not notice any reduction in comfort level, despite the obvious reduction in ASHP operating hours. In Figure 6.1, the average heat output was approx. 4.5kW whereas in Figure 6-2, the average heat output was closer to 7.0kW.

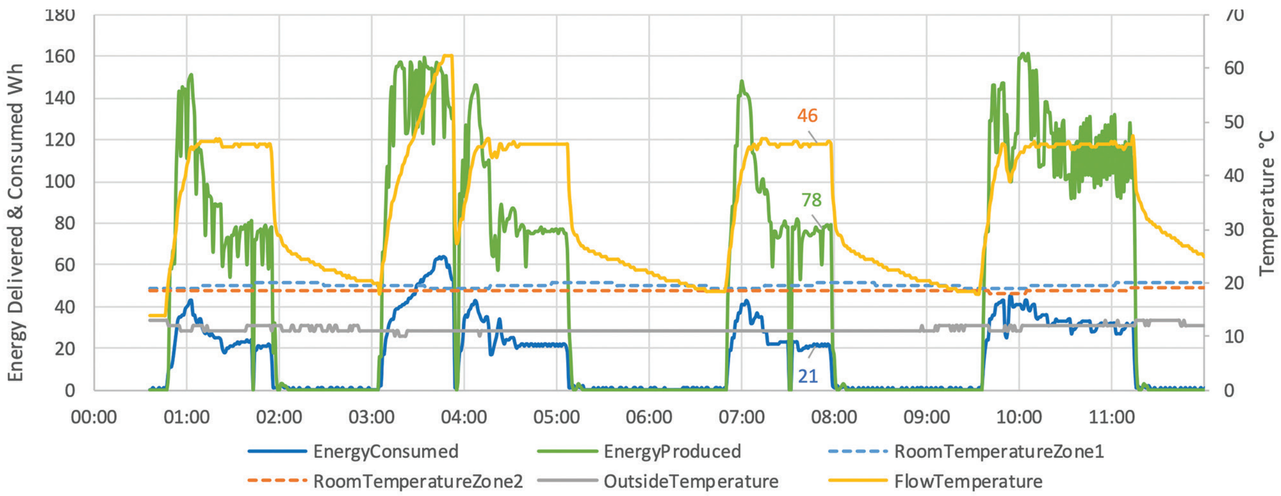


Figure 62: Correctly positioned room sensor – SH086

A question arises as to whether it is better to produce low heat output during continuous operation vs shorter bursts of higher heat output. Does heat produced at low output get absorbed into the heavy construction elements of the house while not contributing in the short term to raising internal temperature? Is there an energy penalty to be paid for using shorter bursts of higher output to satisfy the zone target temperature?

Taking the first 388 minutes of each graph, Table 6.1 presents the comparison. While there is a slight reduction in COP for the second scenario (high output, shorter run times), due perhaps to transient operation after each start-up and the fact that the datasheet COP for higher outputs is slightly lower than that of lower outputs, there is a 21% reduction in energy consumed, and the room sensor is satisfied for 56% of the time, whereas it is only satisfied at the end of the continuous cycle in Figure 6.1.

	Continuous Low Output	Short Burst, Higher Output
Graph	Fig 6-1	Fig 6-2
Sample time (mins)	388	388
Energy Consumed (Wh)	8,006	6,303
Heat Delivered (Wh)	27,707	20,467
Tout average	10.9	11.1
COP	3.46	3.25
Room Sensor Satisfied (mins)	0	218

Table 61: Room sensor position comparison – SH086

With reference to Section 4.4, a proposed use of Room Temperature control devices to compliment the zoning and hydraulic de-coupling proposed would involve a master thermostat / sensor for the living/ core zone, TRVs in the bedrooms to allow for cooler room temperatures and a second temperature sensor/thermostat in the coldest bedroom which would be capable of turning on the heat pump if required, in the knowledge that the living zone would receive heat until the bedroom sensor/thermostat was satisfied.

6.2. Heating Compensation Curve

The Heating Compensation Curve (HCC) is a control strategy which aims to reduce the flow temperature required by the heat pump in line with increasing Tout and falling heating load. This reduces the thermodynamic lift, the difference between evaporating and condensing temperatures, required of the heat pump and so there is an expected increase in COP when compared to systems operating at a fixed Tsf. In setting an HCC, the Tsf is selected for design conditions (Tout =-3°C) and for minimum load conditions, initially Tout =+15°C was used. Typical initial target flow temperatures for these limits were -3/48°C, +15/28-32°C.

For the homes within the SH2.0 study, 15 of the homes were initially commissioned with a HCC, 2 with fixed flow temperature and 3 using a control method called Auto Adapt which made decisions on what *Tsf* to use based on internal as well as external temperatures. Of the two systems employing fixed flow temperature, one homeowner was very familiar with the control system and so was in a position to manually adjust the set point upwards as required during colder weather. This house was fully heated by radiators and so such interventions were necessary due to the low thermal inertia of the heating emission system. The other fixed flow temperature system had underfloor for the main living zone. In both cases, evidence was found where internal target temperatures were not achieved during colder weather, a fact that was masked in the case of the underfloor system due to the high thermal inertia/ storage of the underfloor screed.

The Auto Adapt system presented a different analysis challenge. It was impossible to predict what the *Tsf* the control algorithm would select, sometimes allowing the systems to operate for long periods of time without achieving internal target and then running for short periods of time with *Tsf* as high as 60°C. As an example, near the end of the SH2.0 project, SH076, which had been operating on auto-adapt was changed to HCC. This system regularly struggled to achieve room temperature set points and was consequently running for very extended periods without switching off but within less than 24 hours of the change to HCC (-3/50, 9/40), the system switched off and stayed off for over 2 hours. Table 6.2 presents an analysis of the 10 days before and after changing from Auto Adapt to HCC.

While there was a 31% reduction in HDD for the 10 days after the change, when normalised for HDD, energy consumption reduced by 20%. It had been observed that Auto Adapt created conditions that led to long operating hours, high CC and poor attainment of zone target temperatures. The result of the change was to increase average Tflow by 5°C to 39°C which led large reductions in heating hours, cycling and minutes where zone 1 target temperature was not met.

Therefore, of the 3 control methods, HCC was deemed to be the preferred method, as long as the system was commissioned in keeping with the capabilities of the heating emission system. As discussed in Section 3, it is imperative that the heating emission system is capable of dissipating all of the heat produced by the heat pump. The setting of the HCC plays a fundamental role in achieving this.

SH076	Auto	HCC	% Difference
Data Period	2-12 March 19	13-23 March 19	
Degree Days (DD)	101	70	31%
Energy Consumed (kWh) per DD	2.2	1.7	20%
Heating Hours	227	102	55%
Compressor Cycles	289	72	75%
Minutes below Z1 target temp (all modes)	5039	83	98%
Average Tflow (°C)	34	39	-16%v

Table 6.2: 10 days pre-post change from Auto Adapt to HCC

Sections 3.5 & 3.6 of this report have presented methodology and analysis into the effects of HCC settings on ASHP performance. It was found that for radiator systems, the optimal flow temperature range is 40 – 50°C. An analysis was also carried out to determine if +15°C was the best *Tout* to use for setting the lower *Tsf*. Appendix A presents a calculation which suggests that +9°C is more appropriate. In other words, *Tsf* stays at +9°C for all *Tout* above +9°C and *Tsf* increases from 40-50°C as *Tout* falls from +9 to -3°C. The shift from 15°C to 9°C takes into account the fact that the compressor is capable of modulating. For example, a house requiring an 8.5kW system, the minimum output of the heat pump is approximately 3.9kW which would be comparable to the overall house load when *Tout* = 9°C and the target indoor temperature is 20°C where the DEAP instantaneous heat loss factor is 360W/K.

For *Tout* above 9°C, the radiator system should be capable of emitting a greater amount of heat than the instantaneous house load and so the zone thermostats should be satisfied without the need for the compressor to cycle off. In practice, the % of time that the SH2.0 homes operated with both zones together varied greatly, from 20-80% of all heating hours, depending on user settings. The suggested zoning & hydraulic de-coupling arrangement in Section 4.4 would maximise the benefit of this HCC approach.

As *Tout* falls below 9°C, *Tsf* rises from 40°C to a maximum of 50°C so that the radiator system output increases to meet the house load and the compressor now has the freedom to ramp up its output in order to maintain *Tsf* without the risk of CC.

A similar approach can be taken with underfloor heating systems but using lower Tsf. The SH2.0 homes fitted with underfloor heating were found to perform best with *Tsf* ranging from 32-45°C for new (2015 onwards) underfloor heating systems while older systems required *Tflow* of 48-50°C to maintain internal target temperatures when *Tout* was below zero.

It should be noted that the primary objective of these guidelines for *Tsf* is intended to ensure smooth operation of the ASHP while satisfying internal target temperatures in a timely fashion, which is taken to be the capability of raising the internal temperature of a space heating zone by 1.0°C in the space of 3-4 hours, and then switching off, using the enhanced thermal storage properties of the deep retrofitted home to prevent the temperature from dropping by 1.0°C in less than 2-3 hours.

Once it is proven that the deep retrofit system is capable of such performance, adjustments can be made by the commissioning engineer over the course of the first and second heating seasons in order to optimise the system by running with the lowest possible *Tflow* without introducing CC, excessively long run times or poorly maintained internal temperatures.

6.3. Time Control

Referring again to Part L 2011 of the building regulations, there is a requirement for "separate and independent automatic time control of space heating and hot water". In the experience of the SH2.0 team, the application of time control varied considerably across the homes studied. Most home employed time schedules for DHW, making use of night rate tariff also in the case of the Time of Use (TOU) pilot, avoiding the peak tariff between 6 and 9pm. For space heating, 8 of the 20 homes regularly used time schedules, with the logic behind each time schedule varying from house to house.

Time control should be used in conjunction with temperature set-back. This system was available to all of the SH2.0 homes although none were using it at the start of the SH2.0 project. Set-back is a control feature which allows the heating system work to the desired comfort levels when the home is occupied (e.g. living zone 21°C, sleeping zone 18°C) and allowing these temperatures to drop by a degree or two when house is unoccupied. Time schedules should be such that comfort levels are achieved before occupancy resumes. As heat pumps are sized close to the maximum instantaneous house heat load, set-back temp of no more than 2 degrees below target is suggested.

In conjunction with discussion relating to the correct setting of the HCC, examples were found where the combination of a low HCC and a time schedule can cause the ASHP to switch off before the zone target temperatures are achieved. Figure 6-3 presents an example of this happening at SH139. The ASHP is switched off at 08:00 as indicated by the ASHP output dropping from 5kW to 0kW. In this instance, the room thermostat required the room temperature to rise to 20°C in order to be satisfied, but instead, the temperature at switch-off was 19.5°C. In such cases, it is important that the HCC settings are verified before a time schedule is applied.

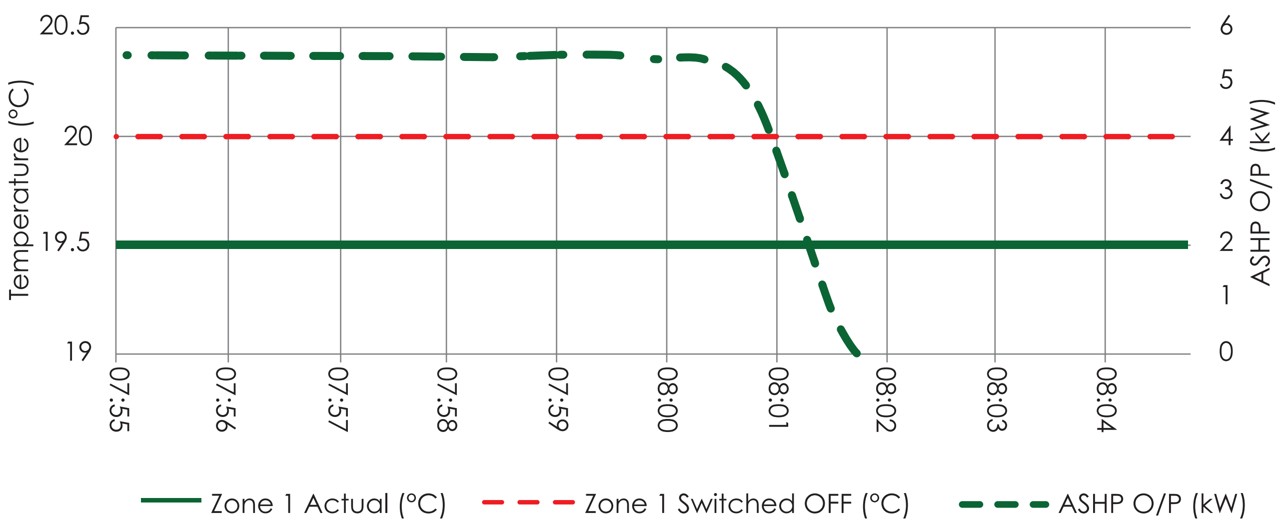


Figure 6 3: Time schedule switching off ASHP before target room temperature achieved

The SH2.0 applied a ToU Tarriff for a range of homes where specific tariffs were set to influence load shifting.

6.4 Compressor Control

Many heat pumps employ a layer of control that decides whether or not conditions are suitable to allow the compressor to start. This decision will depend on factors such as flow temperature and time since the compressor last switched on/off.

In any case, there is the intention that there are periods where the heat generator is not generating heat and that the building fabric is capable of retaining heat for a period of time before the internal temperature drops by a set amount, normally 0.5 to 1.0°C. The length of time that the compressor stays off will be proportional to the outside temperature. Once the heat pump re-starts, the design and commissioning setup of the system will dictate how long it takes to re-achieve the target internal temperature.

6.5 Commissioning

The preceding Sections of this report set out a design plan aimed at ensuring that the retrofit ASHP installation achieves the optimum balance between the following 3 objectives:

1. **Maintain required Home and DHW temperature**
2. **Maximise COP and achieve expected running costs**
3. **Ensure long life of equipment (15 years minimum)**

The correct design approach, subsequent communication of this to installers and then strict adherence to these design concepts coupled with high quality workmanship are critical to ensuring repeatable successful retrofit ASHP installations. This being the case, the importance of the final step of commissioning cannot be understated. It is absolutely of equal importance to the preceding design and installation stages.

From the experience of the SH2.0 project, the following headings require careful consideration at the commissioning stage:

1. **Confirm that condenser water flow rate is sufficient to maintain the manufacturer's stated nominal condenser ΔT when operating at maximum output**
2. **Ensure the heating system is fully purged of air**
3. **Ensure that radiator or underfloor circuits are properly balanced**
4. **Note the minimum heat emission capacity to which the ASHP may be connected. While this is mainly dictated by design, ensure that all living and circulation area TRVs are set to max, or in the case of underfloor heating, if not stipulated in the design, remove actuators from bathrooms/toilets and circulation areas**
5. **Each heat pump manufacturer has a specific checklist for controller settings. Ensure these are fully completed.**
6. **Where an HCC is used, ensure that the settings are not too low so as to cause excessive cycling.**
7. **Set appropriate DHW settings**
8. **Set up the legionella protection plan as per manufacturer's instructions**

Ideally, it is recommended that the installer should monitor the operation of the system periodically for the first heating season to ensure that the homeowner has sufficient knowledge to manage the ASHP system as well as ensuring that CC is minimised while COP is optimised. The experience of the SH2.0 team was that the best way to achieve this is by providing 1-2 follow-up site visits during the first heating season after commissioning.

7. Discussion

Developments in Air Source Heat Pump technology has meant that even in the coldest of weather, nominal heat outputs are maintained without the need for supplemental back-up heaters. Evaporator design and defrost control strategies are such that for properly designed systems, defrost cycles are almost unnoticed. And the use of inverter driven compressors means that less space and expense needs to be dedicated to creating hydraulic separation between the ASHP and the heating system so as to ensure optimum system operation.

System design is fundamental to the success of the ASHP retrofit installation. This involves a combination of heat pump sizing, emission and distribution and DHW system designs and the application and correct commissioning of the most appropriate control strategy. The key objective of the process from design to commissioning should be to ensure that the heat pump is allowed to operate in steady-state conditions as often as possible. This will give the system the best chance to achieve COPs that are as close as possible to the datasheet COP for the given A/W conditions that exist on any given day. This has been expressed as the COP ratio in Section 3.6
The performance of heat pump systems is most often described in terms of COP. However, COP alone does not provide a complete picture of how the retrofit ASHP system is performing. A heat pump that is slightly undersized will have the opportunity to operate for significant portions of time in steady state conditions, as its minimum output will generally be lower than the house demand thus leading to high COP figures. However, in cold weather, these systems have been found to struggle to fully achieve internal target temperature and switching off, a situation that is often compounded on nights when defrost cycles can take place every 45 minutes, further interrupting the ability to generate heat.

The heat emission systems in the SH2.0 study were all well designed with regard to the intended flow temperature range. Initially HCC settings tended to have a wide range with *Tsf* generally close to 30°C at the lower end of the curve when *Tout* = 15°C. A narrower range at slightly higher temperatures provided better performance by reducing CC and increasing the responsiveness of radiator systems thus leading to much higher frequency of zone thermostat satisfaction. For underfloor heating systems, older systems with less insulation under the screed required similar curves to radiator systems while newer systems were successful at lower temperatures.

The use of TRVs on radiator systems and actuators on underfloor manifolds has a major bearing on the potential for the heat pump operation to be interrupted. In radiator heated living areas, TRVs can generally be set to maximum with a zone thermostat taking control whereas, it is important that bedrooms can be controlled to a lower temperature and that heat does not migrate from warmer surrounding areas leading to overheating. Underfloor heating systems without buffer tanks and where every circuit is fitted with an actuator can work without excessive CC, but this depends on the house layout. If the underfloor rooms have generally the same aspect, they can be timed to work together, and in most cases they arrive at their target temperature together as was experienced in SH006.

However, the experience was different in SH073 and SH212 where there was a distinct difference in aspect between the front and back of the houses leading to occasions where the heat pump only had a circuit or two to work on which lead to increased CC.

There is much debate within heat pump circles about hydraulic decoupling. This stems from the fact that the heat pump and emission systems have different design flow rates. Decoupling can be used to allow the systems to work at their optimum flow rates while also providing addition thermal inertia which helps to minimise CC. Decoupling can be achieved to some extent through the use of automatic bypass valves, but the lack of additional volume means that the heat pump will inevitably switch off very soon after a bypass valve has operated, and so they only serve to avoid a sudden loss of flow which might lead to a refrigerant high pressure fault. Direct coupling of the heat pump to the heating system has been shown to cause noise in radiators systems while heat pumps may not have sufficient flow rate to maintain required levels of heat output.

The final settings of control parameters takes place at the commissioning stage of the project. Having verified basic parameters such as water flow rate through the heat pump condenser, the suggested approach to control settings and commissioning is to first ensure that the heating system as a whole is capable of generating and delivering the peak output required, and that the heat emission system is capable of achieving internal temperature targets in a timely fashion. Once this has been achieved, steps towards optimising energy consumption and COP should be taken, such as fine tuning the HCC and the implementation of time schedules coupled with set-back temperature.

8. Conclusions

1. **For radiator systems, flow temperature should be in the range 35-50°C, but where HCC allows for *Tflow* <40°C, commissioning tests should be carried out to ensure that excessive CC does not occur at *Tflow* between 35 & 40°C.**

2. **Flow temperature range for newly installed underfloor heating systems can be 30-40°C provided the emission system is designed to operate at these temperatures, and that thermal inertia is maximised by minimising the number of circuits which have individual control actuators.**

3. **Older (pre 2011 part L) underfloor systems may require a flow temperature of 35 – 50°C, with 50°C being the maximum flow temperature that should be supplied to a manifold.**

4. **Maximise the minimum heat emission capacity available to the heat pump by grouping all living and circulating areas and bathrooms together with bedrooms controlled separately to allow for controlling to lower room temperatures**

5. **Provide hydraulically de-coupling of the heat pump from the heating systems to allow adequate flow through the condenser when house load is low. This can be achieved using a small buffer vessel in parallel.**

6. **Allow for a 9% reduction in output of radiators connected BOE when sizing radiators to ensure sufficient heat emission capacity.**

7. **In the absence of a buffer vessel, minimise the number of actuators on underfloor circuits.**

8. **For heat pumps with VSD compressor, the condenser pump should ideally be controlled by VSD by the HP to maintain a constant condenser ΔT.**

9. **DHW is regularly heated to 50°C using the ASHP compressor only, leading to *Tflow* of 55-64°C on the primary side of the DHW coil/heat-exchanger. As DHW accounts for a significant proportion of the energy used by the ASHP, correct tank and heat-exchanger design is essential to maximise the COP and reduce energy consumed by this operation.**

10. **Above all else, the design, installation and commissioning of the retrofit ASHP system must ensure that interruptions to the operation of the heat pump are minimised, and that opportunities for operation in steady-state, as close as possible to EN14511 conditions, are maximised.**

9. References

CIBSE (2016) ***Environmental Design – CIBSE Guide A.*** CIBSE, London, UK

NSAI (2003) ***Heating systems in buildings - Method for calculation of the design heat load,***
I.S. EN 12831, NSAI, Dublin, Ireland

NSAI (2012) ***Heating systems in buildings - Design for water-based heating systems,***
I.S. EN 12828, NSAI, Dublin, Ireland

NSAI (2007) ***Heating systems in buildings – Design of heat pump heating systems,***
I.S. EN 15450, NSAI, Dublin, Ireland

NSAI (2014) ***Radiators and convectors – Part 2: Test Methods and Rating,***
I.S. EN 442-2:2014, NSAI, Dublin, Ireland

SEAI (2018) ***Heating and Domestic Hot Water Systems for dwellings – Achieving compliance with Part L 2008,*** SEAI, Dublin, Ireland

BRE, (2013) ***Design of low temperature domestic heating systems,*** BRE, Bracknell, UK

Department of Housing, Planning, Community and Local Government (2011)
Building Regulations (2011) Part L, Conservation of Fuel and Energy – Dwellings,
Department of Housing, Planning, Community and Local Government, Dublin, Ireland

Daikin (2017) ***Heating Technical Data, ERLQ-CAV3,*** Daikin Europe NV, Dublin, Ireland

Mitsubishi Electric (2017) ***Ecodan Databook,*** Mitsubishi Electric, Dublin, Ireland

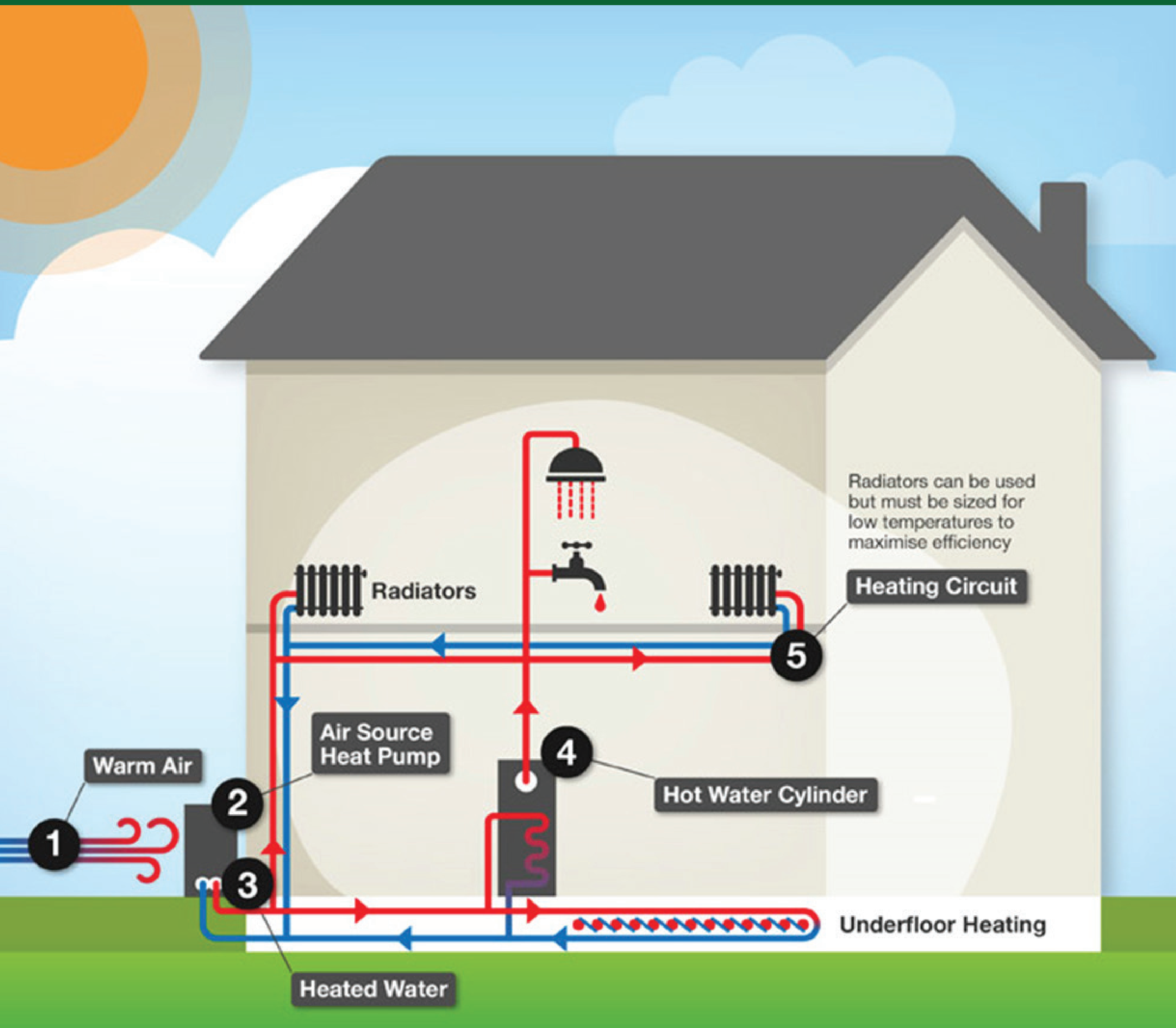
10. Appendices

Appendix A

From EN12831 - HEATING SYSTEMS IN BUILDINGS – METHOD FOR CALCULATION OF THE DESIGN HEAT LOAD

Reheat time hours	f_{RH} W/m^2		
	Assumed internal temperature drop during setback[3]		
	1 K	2 K	3 K
	building mass high	building mass high	building mass high
1	11	22	45
2	6	11	22
3	4	9	16
4	2	7	13

[3]In well insulated and airtight buildings, an assumed internal temperature drop during set back of more than 2 to 3 K is not very likely. It will depend on the climated conditions and the thermal mass of the building.


Radiators can be used but must be sized for low temperatures to maximise efficiency
Radiators
Heating Circuit
5
Air Source Heat Pump
4
Warm Air
2
Hot Water Cylinder
1
3
Underfloor Heating
Heated Water

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