DesignforIU: Comparison of certified versus operational performance of energy efficient technologies

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I. Summary for Policy Makers

Several key energy efficient technologies are set to be the dominant design choice in Ireland for the foreseeable future. Two of these technologies are Mechanical Ventilation Systems with Heat Recovery (MVHR) and Air Source Heat Pumps (ASHPs). Heat pumps are likely to play a significant role in decarbonising the residential heat sector, with the current Climate Action Plan calling for the installation of over 600,000 heat pumps in Irish buildings by 2030, which, given current grant aid investment levels, could lead to an investment of €2.1 billion in state funded grant aid for a single technology. As such it is critical that these systems deliver on this significant public investment and perform according to design expectations. Existing literature on the performance of heat pumps and MVHR systems has indicated that there are a significant number of systems that underperform with respect to design expectations. This has led to the use of an in-use factor (IUF) to correct for the differences between design and reality. The goal of this project was to monitor several residential and non-residential MVHR and ASHP systems and determine if an IUF is required to reflect differences between certified performance and actual performance levels. In addition to this, this work was also contextualised by assessing if systems comply as renewables (for heat pumps only) and by way of comparison with Irelands Building Energy Rating (BER) database and relevant literature.

The following is a summary of findings from this study:

- Previous international literature has indicated that heat pump and MVHR systems under-perform with respect to design expectations, with in-use SPF values of between 1.2 and 4.5 (average = 2.8) for heat pump systems and HRE values of between 50% and 82% for MVHR systems.
- Previous studies in Ireland [1], [2] have indicated that heat pumps underperform with respect to manufacturers values by between 3% and 59%, with large differences observed seasonally.
- Most of the heat pumps studied in this report underperformed by between 23% and 49% depending on the heat pump considered (one heat pump performed well with respect to design expectations).

- Current literature and calculation methodologies for heat pump systems are likely to lead to most heat pumps complying with up-to-date minimum thresholds for renewable compliance from an Irish (1.93) and EU perspective (2.31). Most of the heat pumps studied were considered renewable under these definitions.
- Given the rate of change of electricity grid efficiency and primary energy factor (PEF) the minimum SPF requirements from an EU and Irish perspective are likely to decrease in the coming years.
- Of the MVHR systems studied the average HRE was 80%, two of the three systems studied performed close to design expectations, one did not.
- The average difference between standardised SPFs used to represent heat pumps in the BER database (4.4) and that of literature (2.8) and this study (3.4) for space heating (SH) mode is between 100% and 160%. The performance of heat pumps in domestic hot water (DHW) mode was found to be comparable with expectations.
- Current data does not reveal conclusively what factors influence the underperformance of systems and what IUF could be applied in fair manner. However, over-sizing or underutilisation of heat pump systems is likely to lead to underperformance with shorter run times and more compressor cycling.

Based on this the following is recommended:

- A national high-resolution study with a larger, statistically representative (n > 12), sample of heat pumps and MVHR systems is required to conclude on the magnitude of gaps between design and reality in the Irish context.
- More work is needed in assessing heat pumps and MVHR systems over their lifecycle to assess the potential degradation of their efficiency and performance.
- Heat pumps should be considered for regular independent testing to ensure renewable compliance is maintained and manufacturers quoted efficiencies are confirmed.
- More work is needed to determine what effect different boundaries have on efficiency calculations.

II. Dissemination

Three articles relating to aspects of the DesignforIU project have been presented at international scientific conferences. The first two articles presented some of the findings from research on EAHPs and AWHPs at the HVAC 2021 Conference in Tallinn, Estonia. The third article focused on a detailed analysis of a non-residential MVHR system and was presented at the ASHRAE Winter Conference 2021 in Chicago, USA. Details regarding all papers [3]–[5] are indicated below. All final versions of these papers can be found on ResearchGate. For further information please contact nbertresearch@gmail.com directly.

- P. D. O'Sullivan, S. Colclough, J. Morehead, and A. O'Donovan, "Evaluation of the theoretical and in-use performance of Exhaust Air Heat Pumps," E3S Web Conf., vol. 246, 2021, doi: 10.1051/e3sconf/202124606003.
- O'Donovan and P. O'Sullivan, "In-Use Performance of Air-to-Water Heat Pumps: Are the Standards robust?," E3S Web Conf., vol. 246, pp. 1–6, 2021, doi: 10.1051/e3sconf/202124606002.
- O Donovan, M. Cantillon Murphy, E. Charles, N. Baker, and P. D. O Sullivan, "Technical evaluation of a retrofitted MVHR system : design and in-use performance," 2021.

In addition to the above dissemination documents, iterations of the work (as the work was developed) associated with the DesignforIU project were shared and presented at MaREI symposia.

- A. O' Donovan, J. Randrianasolo, and P. D. O Sullivan, "In-use performance of air source heat pumps and mechanical ventilation with heat recovery systems in Ireland:
  A review and Sensitivity Analysis," presented at the MaREI Symposium, November 2019.
- A. O' Donovan and P. D. O Sullivan, "In-Use Energy Performance of Air Source Heat Pumps: Connecting Predictions to Reality", presented at the MaREI Symposium, November 2020.
- P. D. O Sullivan and A. O' Donovan, "Comparison of certified versus operational performance of energy efficient technologies", presented at the SEAI National Energy Research Conference, November 2020.

III. Introduction

a. Project Background

The DesignforIU project was funded by the Sustainable Energy Authority of Ireland (SEAI) under the 2018 RD&D call and began in 2019. The project was funded in response to the thematic strand of the RD&D that year and focused on addressing some aspects of *"Topic 6: The In Use Factor: Comparison of the certified versus operational performance of energy efficiency technologies"*. The goal of this topic was to monitor a number of technologies including heat pump systems and a number of other energy efficient technologies (e.g. MVHR). The aim of the project was to monitor the operational performance of technologies in comparison to the certified performance for one heating season and determine if an "in-use" factor is required for the technology, and if this can be incorporated in the Non-Domestic Energy Assessment Procedure (NEAP) or Dwelling Energy Assessment Procedure (DEAP) methodologies. The DesignforIU project focused on the monitoring and analysis of four residential case studies that incorporated ASHPs and MVHR systems in their designs. Two non-residential systems were assessed, one ASHP and one MVHR. Monitoring on this project took place from December of 2019 to December of 2020 for the most part and the informed consent of participants was received prior to monitoring or for any extensions or amendments to monitoring periods.

b. Aims and Objectives

The aims of the DesignforIU project more generally were to assess if a gap exists between measured and actual energy performance for ASHPs and MVHR systems. Alongside this was the question of whether an IUF can be used to reflect gaps if they exist, and if this IUF would be appropriate in its adjustment of efficiency metrics in DEAP. Data was gathered for multiple purposes in aligning with the above aim of the DesignforIU project and more generally:

1) To review existing literature and identify performance levels that exist for MVHR and ASHP systems.

2) To measure the in-use performance of ASHP systems and compare this with manufacturers and standardised values.
3) To measure the in-use MVHR systems and compare this to manufacturer and standardised values.
4) To identify causes of underperformance if they existed for both types of systems.
5) To gather additional data for future systems model calibration and validation.

c. Disclaimer

The contents of this report were presented based on the best available information at the time of publication. All results should be taken in the context of the literature and other relevant work, errors and omissions included, including updates to the work presented. The work presented here, and the recommendations thereafter is to be interpreted in this context where and MeSSO Research nor MTU do not accept any responsibility for actions or consequences of actions that follow this publication, as the readers of the publication have been given the full knowledge as to the limitations and challenges of the work (see section X) and how the results should be interpreted in the broader context.

d. Acknowledgements

The authors of this report would like to thank all participants in the study including those who contributed data, information, and time to the project, including homeowners, facilities managers, architects, building energy assessors, design engineers and the technical support teams for various manufacturers. We would like to give particular thanks to John Morehead for his contributions regarding case study buildings and his knowledge of the design industry in Ireland. We would also like to thank Dr. Shane Colclough for collaborating with us on a non-residential case study building. We would like to thank those who worked with the project directly as researchers, specifically Alienor Fort, Joel Randrianasolo, Emeline Charles and Michael Cantillon-Murphy, as well as contributions from researchers in University College Dublin (UCD). We would like to thank MaREI Centre

and the Residential Energy Research Group for their collective work on filtering the National BER database, all figures that refer to this database use the filters that they have proposed for robust analyses of BER databases, and we kindly thank them for this work. We would also like to thank Brian McIntyre of the SEAI for his support in assisting with acquiring one of the retrofit case study buildings.

e. Report Contents and Structure

The findings in the following report refer to the findings from the DesignforIU project which include outcomes from an extensive performance-based literature review and an in-use monitoring of two residential exhaust air heat pumps (EAHPs), three conventional residential air to water heat pumps (AWHPs), two residential MVHR systems, one non-residential heat pump and one non-residential MVHR unit. The following report is broken into seven key sections. Initially, we present a state-of-the-art for renewable heat (section IV), then the sample or case study buildings and their respective energy systems will be presented (Section V and VI). In section VII we will present the types of data gathered in each case study, section VIII will indicate the assessment metrics used to represent in-use efficiencies for MVHR and ASHP heat pump systems and will also comment on the renewable compliance metrics used for heat pumps specifically. In Section IX we will present the results in four sections: a) a performance mapping of the in-use performance of MVHR and ASHP systems from national and international studies, b) the measured performance of case study buildings and their energy systems, as well as the renewable compliance of heat pump systems and c) the context of these results with respect to relevant databases and literature. Section X will indicate some observations, recommendations, proposed future work as well as challenges and limitations. The main findings and recommendations of this report are summarised in section 0. The main dissemination activities are summarised in section II.

IV. Context and State-Of-The-Art For Renewable Heat

There are strict carbon emissions targets in place in the European Union (EU) and there is a regulatory requirement for the Irish building stock to be nearly zero energy. Buildings are expected to reduce carbon emissions in the range of 80-90% by 2050 from 1990 levels [6]. In 2019, the residential sector accounted for 24% of Ireland's primary energy demand and total energy related CO_2 emissions [7]. Fossil fuels still dominate as fuel sources for Irish residential buildings and account for 73.2% of the CO_2 emissions in the residential energy sector, where oil accounted for 40.3% and gas accounted for 20.5% of CO_2 emissions respectively [7]. In the most recent analysis of the residential energy sector in 2016 [8], SH accounted for 61% of Ireland's residential final energy demand, while SH and water heating combined account for 80% of household energy use [8]. Additionally, the largest contributor to Irish heat related emissions in 2018 was seen to be SH demand for residential buildings, which accounted for 47% of heat emissions [9]. To tackle this dependence on fossil fuels for residential SH and hot water the strategy at national level is to promote the retrofit of the existing housing stock to B2 levels or higher [10]. Heat pumps have been identified as an integral energy efficient technology in providing renewable heating purposes [10]. Even in newly designed, or retrofit buildings, with state-of-the-art heating systems (such as heat pump systems), a gap between actual and predicted performance, commonly referred to as "the performance gap" has been reported [11]. The root causes for this performance gap - reported in the literature to be in the range of 1.3–1.9 times the designed value [12]–[16] - can be related to a range of complex and diverse factors associated with buildings. Increasingly, however, it is recognised that these state-of-the-art heating systems are underperforming [17]–[20]. This is true of heat pumps, whose component output energy is often claimed by manufacturers to be as high as 4 or 5 times the input energy, but more often reported in the field in the region of 2-2.5 [21], [22]. Although there is near unanimous agreement, across the building community, that heat pump technology has the potential to considerably improve the environmental performance of delivering heat and hot water, and thereby mitigate building related greenhouse gas emissions, there is a paucity of studies that monitor in detail the actual in-use performance of heat pumps in operation. Similarly, there is a paucity of studies that compares the performance of heat pumps with the pre-retrofitted boiler-based heating system, or quantify the carbon and costing of heat pumps over their full life cycle [23]–

[26]. Hence, for these reasons, the lofty claims for heat pumps building integrated performance remain largely invalidated. This matter was discussed at a recent Engineers Ireland meeting [27] and is reported by multiple authors and commentators [28], [29]. Only a handful of studies of significant scale have been undertaken to date. In 2012, the Energy Savings Trust analysed the in-use performance of 80 heat pumps; COPs of between 1.6 and 2.8 were reported with a median of 2.2. In 2017 an even more comprehensive study conducted by UCL monitored 699 heat pumps [18]. The median COP for ASHP was 2.65, while ground source heat pumps reported a median of 2.81. Ireland, and the world, are embarking on a mass market implementation of heat pump technology. According to the European Heat Pump Association (EHPA) statistics there was a confirmed stock of 13.24 million heat pumps in Europe in 2019 [30]. Heat pump sales in Europe grew by 17.5% in 2019, where over 1 million heat pumps were reported to be installed each year in Europe between 2017 and 2019 [30]. The *"total value of the 2019 market volume was almost 9.54 billion Euro"* [30]. Data for Ireland in the EHPA market report suggests that over 30,000 heat pumps were in operation in 2019 and over 40,000 in 2020 [30]. The EHPA statistics dashboard suggests that there was a stock of 44,000 heat pumps for the same year in Ireland [31]. Of the heat pumps sold in Ireland the vast majority that were sold year-on-year since 2015 were AWHPs. In 2019, 8,430 heat pumps were installed in Ireland which was a 47.2% increase in sales when compared to 2018, where 83% of the heat pumps in Ireland installed in this year were AWHPs used for heating purposes only (heating and hot water). The same EHPA report identified that heat pumps had a 6.2% market share for SH in Ireland [30]. The Climate Action Plan calls for the implementation of circa 600,000 heat pumps by 2030 [10]. 400,000 are to be installed in retrofit buildings [10]. That is more than 100 a day between now and 2030. In Europe, around 91% of heat pumps are ASHPs [32]. In Ireland the SEAI report this number to be as high as 95% [33]. Research on ASHPs has highlighted a marked decrease in the actual performance when compared to the theoretical performance [34]–[37].

V. Case Study Buildings

The performance of over six systems in six different buildings was analysed in this project. Four residential buildings were examined, each had at least an MVHR or an ASHP, some had both, or had an EAHP, MVHR and ASHP. Table 1 describes the residential buildings that had these types of systems, while Table 2 describes the non-residential buildings that had these systems. Two of the residential buildings (R1 and R2) were detached "one-off" passive house certified buildings and were exemplary examples of high-performance buildings with three different technologies in each building as well as solar photovoltaics.

Table 1: Summary of residential buildings assessed in the DesignforIU project

Metric	Units	R1	R2	R3	R4
Year of construction	-	2015^	2014^	2018	2015-2017*
Location	-	Cork, Urban	Cork, Rural	Tipperary, Rural	Cork, Suburban
Archetype	-	Detached	Detached	Detached	Semi Detached
No. Storeys	-	2	2	1	3
Building Type	-	New Build	New Build	Retrofit	New Build
Floor Area	m^2	287^	256^	183	170
Energy rating	-	A1	A1	A3	A3/A2*
Heating delivery	-	Underfloor/air	Underfloor/air	Underfloor	Radiators
Ventilation type	-	MV with NV purge ventilation	MV with NV purge ventilation	MV, with some NV openings	NV only
Wall u-value	$W/m^2 K$	0.11^	0.12^	0.2 to 0.23	*<0.16
Roof u-value	$W/m^2 K$	0.12 to 0.14^	0.12^	0.13	*<0.14
Floor u-value	$W/m^2 K$	0.07^	0.10^	0.12 to 0.13	*<0.13
Window u-value	$W/m^2 K$	< 0.81^	< 0.85^	1.4	*<1.53
Air tightness (@50pa)	1/h	0.6^	0.6^	3.81	-

*Values are estimated based on search of BER database for buildings with relevant floor area and number of stories

^ Values taken from Passive House Planning Package

Of these two case studies one building had a ground source heat pump (GSHP) and an EAHP. The GSHP was excluded from this report due to a focus on ASHP. One residential building was an example of a deep-retrofit and had both MVHR and ASHP systems. R4 was a typical new build in a housing development where the heat pump was the main energy efficient technology. Both non-residential examples examined either an (NR1) or MVHR (NR2) separately. The ASHP that was tested was already being monitored in MTUs retrofit testbed (e.g. NBERT). The MVHR system examined was a shallow retrofit of an existing open plan office, the system was a single zone unit.

Table 2: Summary of non-residential buildings assessed in the DesignforIU study

Metric	Units	NR1	NR2
Year of construction	-	2013	2019
Location	-	Cork, Suburban	Cork, Suburban
Building Type	-	Retrofit	Retrofit
Floor Area	m^2	223	165
Energy rating	-	A3	NA
Heating delivery	-	Radiators	Radiators
Ventilation type	-	NV	MV
Wall u-value	$W/m^2 K$	0.09 to 0.16	-
Roof u-value	$W/m^2 K$	0.09	-
Floor u-value	$W/m^2 K$	-	-
Window u-value	$W/m^2 K$	<1.2	-
Air tightness (@50pa)	1/h	1.6	-

Overall, for the most part the sample of buildings in the project represented typical technologies in A-rated buildings in Ireland, where most buildings were modern, air-tight, highly insulated and used MVHR and/or ASHP technologies for SH or DHW.

VI. Energy Systems

a. Manufacturers Quoted Performance

All heat pumps and MVHR systems investigated had quoted performance with respect to relevant standards, as well as values used for energy performance certificates. Five residential heat pump systems and two MVHR systems (named as EAHP-1 and EAHP-2 for simplicity) were evaluated, both systems were the same. Different standards are used in the representation of heat pumps and MVHR systems. Standardised approaches to reflecting the performance of heat pumps is typically broken into standards reporting SH efficiency in different climatic conditions (EN 14511 [38] and EN 14825 [39]) and standards used for DHW (EN 16147 [40]). EAHPs typically report on the energy recovered through heat recovery also with two different capacities and COPs reported which combine outputs from EN14511 and EN13141-7 [41].

Table 3: Quoted manufacturer performance of relevant energy systems in residential case study buildings

Metric	Units	R1	R2	R3	R4
Manufacturer	_	B*^	B*^	C*, E^	D*
Heat pump/system name	_	AWHP-1, EAHP-1	EAHP-2	AWHP-2	AWHP-3
AWHP system type	_	Mono	-	Mono	Split
Refrigerant	_	R410A, R134A	R134A	R410A	R410A
AWHP rated capacity	kW	5.2	-	8.5	5.3
EAHP rated capacity	kW	2.4	2.4	-	-
DHW AEC	-/kWh	1081	1081	1031	-
AWHP SCOP/AEC	-/kWh	5.15/1464	-	4.35/3903	3.22/3369
Heat exchanger	_	Counterflow (Polystyrene)	Counterflow (Polystyrene)	Plate (Polystyrene)	NA
MVHR quoted HRE	%	88	88	86	-
MVHR quoted SFP	W/l/s; Wh/m^3	_;0.4	_;0.4	1.37+;0.38+	-

*Heat pump manufacturer |^MVHR manufacturer|+ Values indicated for home with seven rooms

For residential MVHR systems, EN308 [42] and EN13141-7 [41] are the standards that are typically used for reporting the heat recovery or temperature efficiencies. The DEAP manual in Ireland refers to EN13141-6 [43] and EN13141-7 for specific fan power (SFP), this also refers assessors to the UK Product Characteristics Database (formerly known as SAP Appendix Q) [44]. In addition to this, different metrics are reported by the Passive House Institute. For non-residential MVHR systems the thermal efficiency is typically reported using different standards namely EN 308 and JIS B 8628 [45], fan power is reported with different units than residential units and referenced commission regulation (EU) No 1253/2014 [46]. Table 3 and 4 summarise the values reported by manufacturers datasheets for all case study systems.

Table 4: Quoted manufacturer performance of relevant energy systems in non-residential case study buildings

Metric	Units	NR1	NR2
Manufacturer	_	A	D
Heat pump/system name	_	AWHP-4	MVHR-3
AWHP system type	_	Mono	NA
Refrigerant		R410A	NA
AWHP rated capacity	kW	28	NA
AWHP SCOP/AEC	-/kWh	-	NA
Heat exchanger	_	NA	Crossflow Total Heat (paper)
MVHR quoted HRE	%	NA	83-88
MVHR quoted SFP	W/(m^3/s)	NA	757

Figures 1 and 2 indicate the quoted performance of all heat pumps studied in this project. For most residential units (excluding the EAHPs) the capacity and compressor frequency are variable, where data provided by manufacturers is not only with reference to different source (i.e., air) temperatures but also refers to variable sink temperatures (e.g., water or air). Both EAHP-1 and 2 as well as AWHP-4 were fixed speed units, as such their capacity varied linearly with external conditions or supplied temperatures.

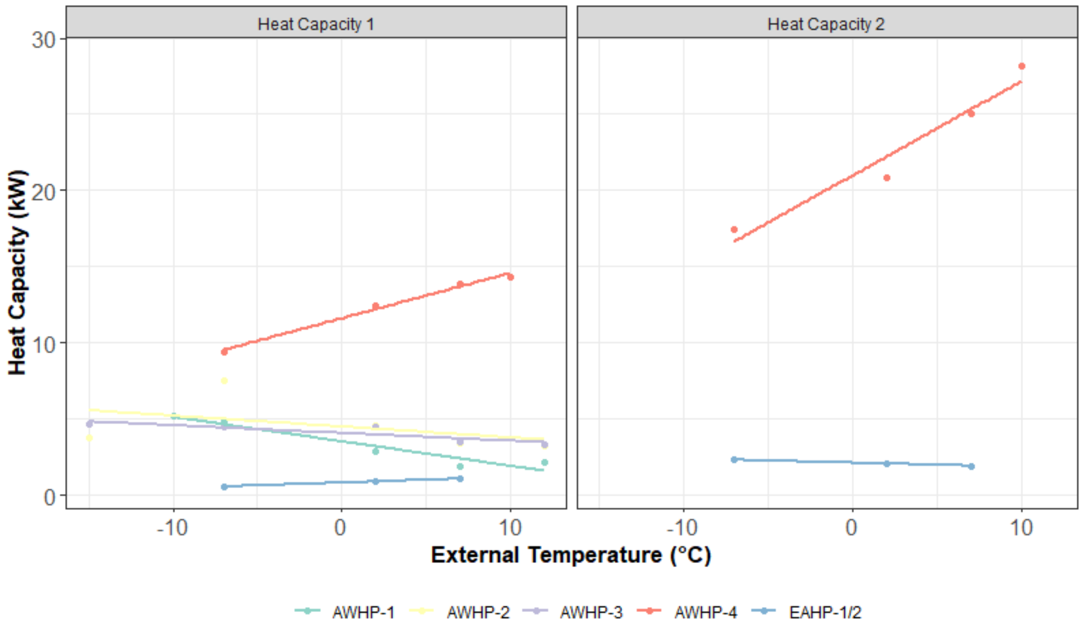

Figure 1: Heat capacity with respect to EN 14511 or equivalent standards for EAHPs (EN 308 [42]) or older models (EN 255) (Heat Capacity 1 refers to the heat capacity for EAHPs that is closest to the EN 14511 definition and to a single compressor for AWHP-4, Heat capacity 2 refers to the inclusion of recovered heat in EAHPs and the performance of AWHP-4 with two compressors)

The COPs reported by manufacturers also reflects the improvement in COP with higher external air temperatures, the only difference being where heat recovery is included (for EAHPs), an inverse relationship is observed, albeit with very high system level COPs when recovered heat is included. Evidently, there is a large variance in the COP between different heat pumps where some of these differences can be explained by the temperature application of the heat pump (i.e., medium, or low temperature application). AWHP-1, AWHP-2, and AWHP-4 are low temperature application heat pumps, AWHP-3 is a medium temperature application heat pump.

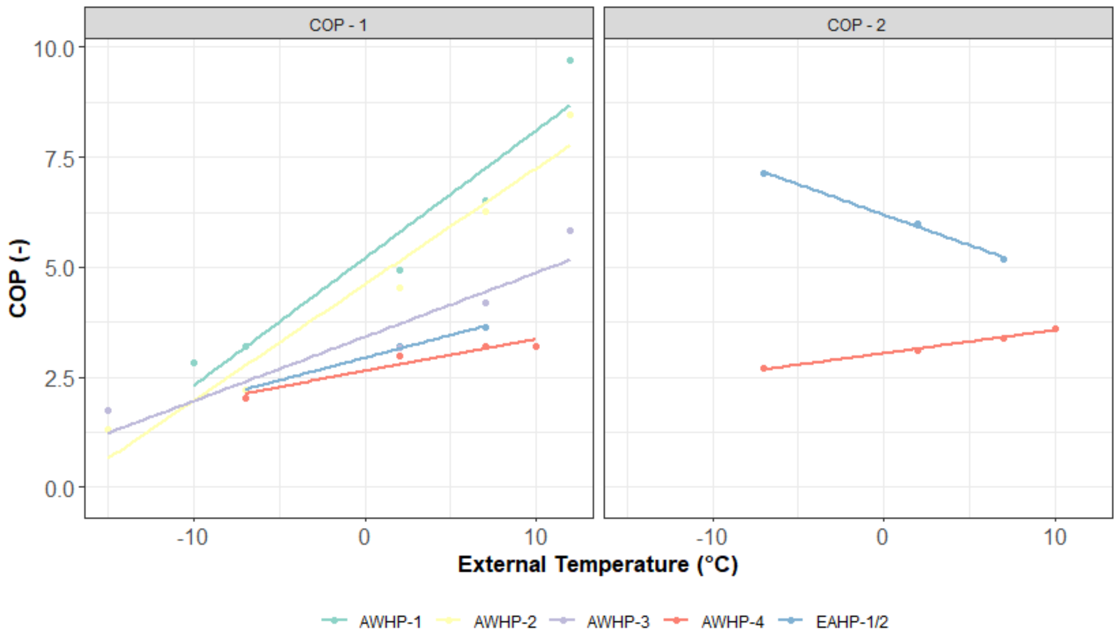

Figure 2: COP with respect to EN 14511 or equivalent standards for EAHPs (EN 308 [42]) or older models (EN 255) (COP-1 refers to the heat capacity for EAHPs that is closest to the EN 14511 definition and to a single compressor for AWHP-4, COP-2 refers to the inclusion of recovered heat in EAHPs and the performance of AWHP-4 with two compressors)

b. Requirements and Entry into DEAP and NEAP

Typically, the entry of residential heat pumps uses the test points in EN 14511 and EN 16147 or equivalent standards amongst other information to determine what efficiency values are used in DEAP. EN 15316 [47] is the standard used to incorporate this into DEAP, which makes adjustments for these test points and accounts for system level efficiencies and adjusting the test points for Irish climatic conditions (e.g. Dublin). Current technical guidance documents regarding heat pumps state that minimum efficiency requirements for heat pumps should be based on requirements from Ecodesign regulations [48]. The most up-to-date DEAP manual [49] states that the efficiency of heat pumps should be taken from Ecodesign, HARP or certified data. The default values that are in Table 4a of the DEAP 4.2.2 manual are 250% for both air-to-water heat pumps (AWHP) and exhaust air heat pumps (EAHPs). It should be noted that EAHPs are treated differently to AWHPs according to EU regulations [50] as exhaust air is not considered the same as ambient air under EU definitions. The minimum efficiency requirements referenced in EU regulation 813/2013 [51] are that SH efficiencies beyond 2015 should be greater than 100% for all heat pumps and greater than 115% for low temperature heat pumps. From 2017 onwards this increased to 110% for all heat pumps and 125% for low temperature heat pumps. In 2018, Directive 2018/2001 [50] stipulated that only heat pumps with a seasonal performance factor (SPF) greater than 1.15*1/η will be taken into account (where η is the ratio of total gross production of electricity and the primary energy consumption for the production of electricity). For non-default values a heat pump tool was developed for entry of data from relevant SH and DHW standards into DEAP, this has been superseded by an online tool that has this integrated. For the non-domestic heat pump studied, the same standard for SH was used only an earlier version of this which only used four test points instead of five (i.e., EN 14511). The older EU directives are cited for non-domestic systems, no minimums or defaults outside of this are stipulated in NEAP modelling guidance [52] or iSBEM documentation [53]. It is however indicated that the seasonal co-efficient of performance (SCOP) be used for SH in NEAP. For MVHR systems used in residential buildings (where new builds apply) a minimum heat recovery efficiency (HRE) of 70% and a maximum SFP of 1.2W/l/s, 0.26W/l/s is the default for EAHPs in Table 4g [48]. The default SFP indicated by the most recent DEAP manual is 2W/l/s, the default HRE indicated is 66% [49]. The same standards (as indicated in section VI (a)) are mentioned, however, there is a focus on the database by the British Research Establishment (BRE) [54]. For the non-residential system studied in this report (i.e. *Zonal supply and extract ventilation units, such as ceiling void or roof units serving single area with heat recovery*) the maximum SFP is 1.9W/l/s according to the relevant building regulations, the minimum HRE values required by EU regulations for these types of units is 73% [46]. The default efficiencies in iSBEM (or SBEMie) were 65% (according to Table 17) for a plate heat exchanger or recuperator type heat recovery systems like the one studied [53], no defaults for fan power were found.

VII. Measurement and Monitoring

a. System Configuration and Setup

Of the ASHP systems monitored the two predominant types were a monobloc and a split system. Figures 3 and 4 indicate the typical configuration of such systems which was based on available manufacturers schematics.

Although the monobloc systems used by AWHP-1 and AWHP-4 had a buffer vessel and no heat exchanger for DHW demand (i.e. SH only), the same principle of the system and how it was configured applies.

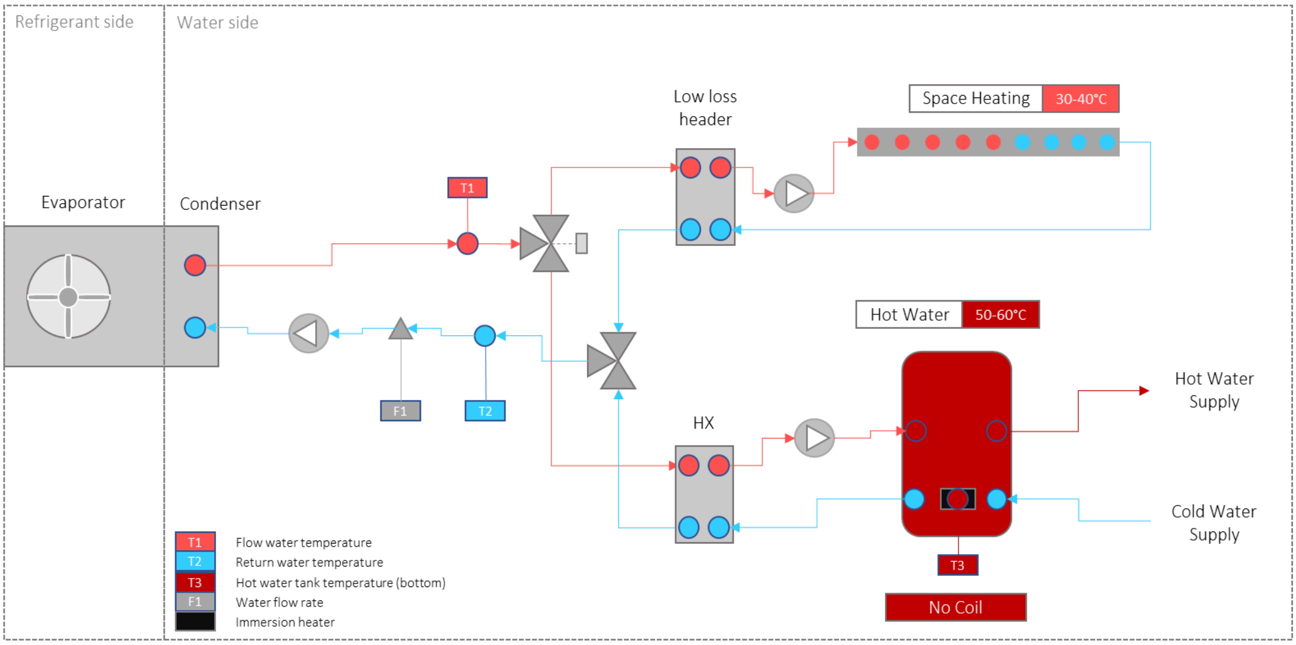

Figure 3: Illustrative system level schematic of monobloc system used in AWHP-2 of case study R3

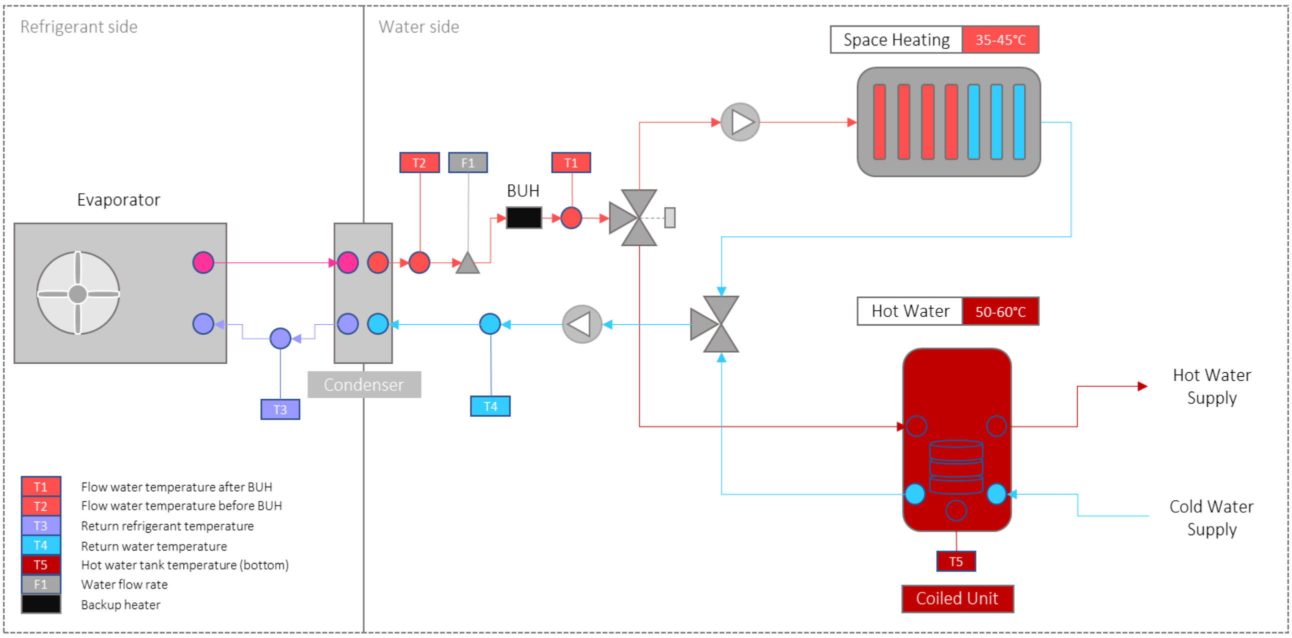

Figure 4: System level schematic of split system used in AWHP-3 of case study R4

For AWHP-2 (and other monobloc systems that are used in domestic settings typically) the refrigerant side of the system is located outside of the house where there is a primary loop between a three-way valve and the outside unit. This typically has anti-freeze of a certain percentage within it to prevent freezing of pipework between the outside and inside unit of the system. Typically, like with AWHP-2 there is an inside unit which has a three-way valve to send primary loop heated water/antifreeze to either an SH system (typically low temperature underfloor heating or low temperature radiators) or a DHW hot water tank. In the case of AWHP-2 a heat exchanger is used to separate hot water from the primary loop. As is common with these systems a back-up immersion heater is installed to aid when DHW demand is high or when legionella mode is activated. For the SH side of the system, a buffer or low loss header is used to separate thermal flows. AWHP-1 uses a 50-litre buffer vessel, the system supporting AWHP-4 uses a larger buffer not as a heat exchanger but to add mass to the system. The difference between the system in Figure 4 and Figure 3 is that the refrigerant side of the system is based outside of the outside unit, where refrigerant is brought inside the house. Unlike with the monobloc system the split system does not require antifreeze in the primary loop. The AWHP-3 system is configured similarly only no buffer vessels are used and the back-up heater is not used only for both SH and DHW loads. This is likely to be a characteristic of the medium temperature application meaning higher supply temperatures for SH. Again, a three-way valve is used to regulate flow in one direction or another. The hot water tank in this case is a coiled tank which differs from the uncoiled example in Figure 3. Both systems prioritise DHW over SH when DHW and SH are required at the same time. Both systems had an integrated flow measurement device and Negative Temperature Co-efficient (NTC) temperature sensors installed in various locations. Figure 5 indicates the configuration of the EAHPs studied. Not dissimilar to the other systems the EAHP system can supply heat to an entire house, in this case by an air-to-air based system which supplies DHW to the house also. The EAHP has an integrated MVHR system, where the air used in the EAHP is supplied after the Counterflow heat exchanger. The unit can be in heat exchange mode, and bypass mode can also be operated in conditions where the temperature in the system is above a temperature limit. For the most part heat is required during the heating season and so the temperature (T_4) that is used in the EAHP is pre-cooled extract air. The system itself has many operational modes which also include active cooling. For this report heating only will be considered.

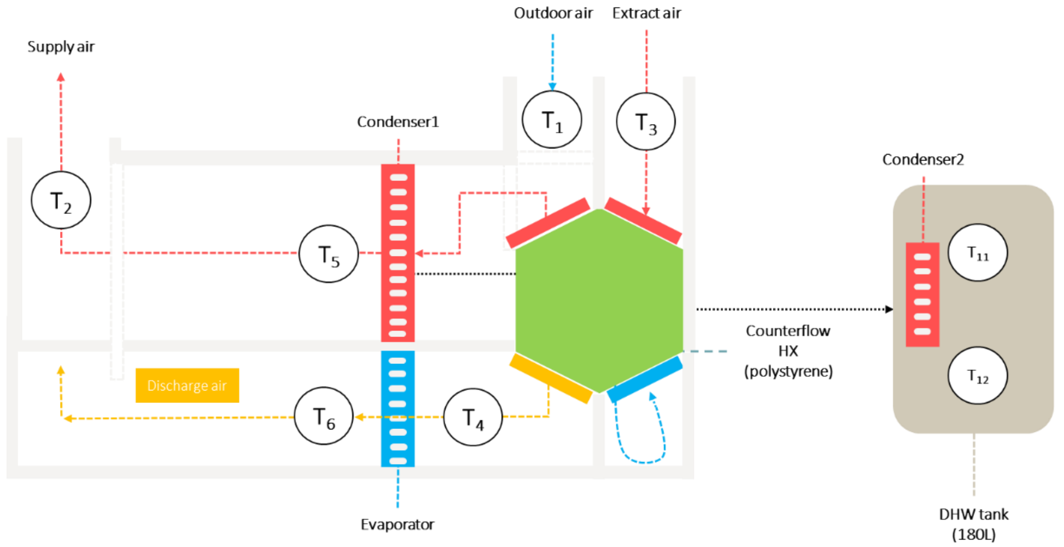

Figure 5: Illustrative schematic of EAHP systems (EAHP1/2) studied in R1 and R2

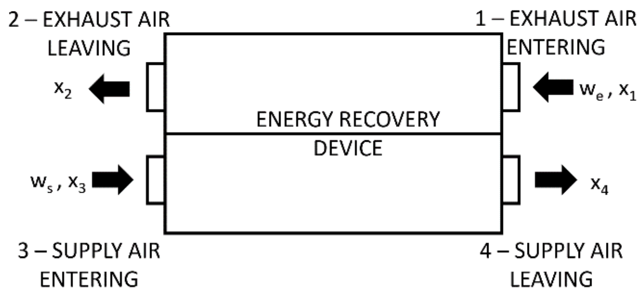

Figure 6: Illustrative schematic of MVHR system studied in NR2

Unlike the other systems the EAHP did not have an integrated flow measurement device in the system. In-situ field measurements were made to obtain flowrates on the supply and extract sides of the system. Figure 6 indicates a schematic of the MVHR system assessed in case study NR2. Like the MVHR system in R3, this system did not have a manufacturers data logging system. A series of field measurements were made in-situ before and after the heat exchanger for the MVHR in NR2. All MVHR systems and their ducted systems except the non-residential system removed air from all wet rooms and supplied air to non-wet rooms (or zones) in each house. The MVHR in NR2 had supply and extract branches in a single zone.

b. Instruments, Data, and Cost

A combination of manufacturers dedicated data-logging systems, low-cost energy monitoring equipment, low-cost indoor environmental monitoring (IEQ) equipment and in-situ measurements were used to assess the performance of the systems studied in this project. Table 5 summarises the different measurements taken for each system or case study, as well as the minimum measurement interval and the accuracy of the instruments. It should be noted that most manufacturers of residential heat pumps gave accuracies in percentages which may be misleading. Further analysis of temperature resistance curves from some manufacturers revealed that most NTC sensors used were more accurate in the typical temperature range they were measuring in. The accuracy of most of these sensors is likely to between ±0.2°C and ±0.7°C in their respective typical operating ranges. All electrical energy monitors used in residential buildings from were the same and used current transducers (CTs) to measure current. Data was sent via a Wi-Fi base station and stored online and then downloaded for analysis purposes. All IEQ monitors in residential buildings were the same and again Wi-Fi pairing was required with the home Wi-Fi network, data was stored online and then downloaded for analysis purposes. Data on IEQ performance was excluded from this report. There were large differences between manufacturer data logging systems in residential buildings. Two systems (EAHP-1/2, AWHP-2) had remote data logging capabilities although one was limited at a remote level to a smaller number of variables, an SD card was used to store data locally. Data from this card was downloaded using manufacturers bespoke software. For AWHP-3, a device supplied by the manufacturer was connected to the inside unit. Data with same level of detail could be downloaded, however, a PC was required as an interface between the indoor unit and the local Wifi network. Again, bespoke software was required for this, and was supplied by the manufacturer.

Table 5: Specifications of instruments used to measure performance in all case studies and respective systems

Name	System or Case Study	System/Location	Units	Measurement Type	Minimum Logging interval	Accuracy [Ref.]
T1 - Inlet	EAHP1/2, MVHR-3	EAHP	°C	Continuous	1min	±3%*, ±0.1°C [55]
T2 - Supply	EAHP1/2, MVHR-3	EAHP	°C	Continuous	1min	±3%*,±0.1°C [55]
T3 - Extract	EAHP1/2, MVHR-3	EAHP	°C	Continuous	1min	±3%*,±0.1°C [55]
T4 - Before Evap	EAHP1/2, MVHR-3	EAHP	°C	Continuous	1min	±3%*,±0.1°C [55]
T5 -After Cond	EAHP1/2	EAHP	°C	Continuous	1min	±3%*
T6 - After Evap	EAHP1/2	EAHP	°C	Continuous	1min	±3%*
T11 - Tank top	EAHP1/2	EAHP	°C	Continuous	1min	±3%*
T12 - Tank bottom	EAHP1/2	EAHP	°C	Continuous	1min	±3%*
Airflow rate (Supply)	EAHP1/2, MVHR-3	EAHP	m^3/h	In-situ	-	±10%^
Airflow rate (Extract)	EAHP1/2, MVHR-3	EAHP	m^3/h	In-situ	-	±10%^
T1 - Flow temp (cond)	AWHP-1,2,3,4	ASHP	°C	Continuous	1min	±3%* ±0.5°C [56]
T2 - Return (cond)	AWHP-1,2,3,4	ASHP	°C	Continuous	1min	±3%* ±0.5°C [56]
Flowrate (primary)	AWHP-2,3,4	ASHP	l/min	Continuous	1min	±2% [57] ±2% or ±5% [58]
Energy EAHP	EAHP1/2	EAHP	Wh	Continuous	1min	±2% [57]
Energy MVHR	MVHR-2,3	MVHR	Wh	Continuous	1min	±2% [59] 1.2%+0.005 [60]
Energy PV	R2	PV	Wh	Continuous	1min	±2% [59]
Energy Whole house	R1,R2,R3,R4	House	Wh	Continuous	1min	±2% [59]
Backup Heater	AWHP-2, 3	ASHP	Wh	Continuous	1min	±2% [59]
Energy GSHP	R2	GSHP	Wh	Continuous	1min	±2% [59]
Energy ASHP Total	AWHP-2,3	ASHP	Wh	Continuous	1min	±2% [59]
Energy Inside Unit	AWHP-1	ASHP	Wh	Continuous	1min	±2% [59]
Energy Outside Unit	AWHP-1	ASHP	Wh	Continuous	1min	±2% [59]
Energy HP	AWHP-4	ASHP	Wh	Continuous	1min	0.2% [61]
Energy pump	AWHP-4	ASHP	Wh	Continuous	1min	0.2% [61]
Bedroom	R1,R2,R3,R4	House	°C/%/ppm	Continuous	5min	±0.3°C/±3%/±50ppm [62]
Kitchen or Living	R1,R2,R3,R4	House	°C/%/ppm	Continuous	5min	±0.3°C/±3%/±50ppm [62]
Attic	R3	House	°C/%/ppm	Continuous	5min	±0.3°C/±3%/±50ppm [62]
Outside	R1,R2,R3,R4	House	°C/%	Continuous	5min	±0.3°C/±3% [62]
Supply	NR2	House	°C/%/ppm	Continuous	5min	±0.1°C/±3%/±50ppm [63]
Extract	NR2	House	°C/%/ppm	Continuous	5min	±0.1°C/±3%/±50ppm [63]

*Value supplied by manufacturer | ^Quoted by service technician |

. For AWHP-4 a data logging system was in place since 2013 which was gathering parameters from a local BMS system, data logging was enabled locally through a MODBUS connection and a reporting software gathered data every 15-minutes. For this heat pump there were many limitations (not least the intermittency of the heat pump), however, there were also situations where the reporting software was turned off or there were power outages. Data was also stored locally which presented some challenges. Data for AWHP-4 was taken from December 2019 to March 2020. The fourth source of data for this study was data taken from the nearest national weather stations and was accessed from Met Éireann's website [64]. In addition to the parameters indicated above, there was also the mixed capability with most systems to get data on the compressor frequency (AWHP-1,2) as well as the mode of operation (AWHP-1,3), both EAHPs systems indicated mode of operation for Bypass (0% to 100% as a percentage and indicated the compressor frequency as 0% or 100% when the heat pump was operational). Efforts were made to get more information from various manufacturers on the compressor frequency, but this was seldom made available. Most data-sets were analysed in RStudio [65] and specific details relevant to each will be described in **section IX**. Table 6 indicates the cost of different solutions that was investigated in this project. In our case a low-cost option was utilised. However, many options were presented including third party solutions which would be used in one case study for one year, and other portable options that could be used at any site. The cost to measure per system can be an issue when system level parameters are part of measurement procedures or protocols. None of the costs indicated in Table 6 include labour or man hours for installation, upgrading or interfacing, maintenance cost or the cost to analyse the data. The experience of the project team is that if/when these systems are to be monitored consideration should be given as to the cost benefit of different measurement and monitoring setups. If heat pumps are to be tested as part of any future legislative requirements, high specification remote solutions are likely to be cost beneficial and sufficiently accurate if in-situ or intermittent testing is preferred for multiple heat pumps over their lifecycle. The cost benefit of third-party setups is likely to be marginal over the lifecycle of the products themselves at current costing levels. A solution that requires manufacturers to provide some of the instruments and the third-party providing others may offer the best cost benefit but over 20 years with calibration, maintenance and data hosting costs may not be the most reliable or satisfactory.

Table 6: Costs of different monitoring approaches at different accuracy and monitoring levels

Instrument/System	Manufacturer or Company	Cost/item	Description
Remote access board	Man A	€ 344.40	Used to enable remote data logging
Remote access board	Man B	€ 639.60	Used to enable remote data logging
SD card/remote access	Man C	€ -	Onboard storage remote was no cost
USB interface for DL	Man D	€ 260.51	Used to interface
Electrical monitoring	Comp1	€ 174.00	Basic 3CTs for heat pump, house and backup heater
Heat meter 1	Comp 2	€ 1,129.00	Replacement heat meter for AWHP-4
Remote setup 1	Comp 3	€ 3,000.00	Suggested by manufacturer D as third party for **one year**
Remote setup 2	Comp 4	€ 885.00	Costed remote solution CTs and heat meter inclusive for **one year**
Remote setup 3	Comp 5	€ 19,970.00	Costed portable remote solution high specification, all system types, internal/external monitoring (any site any location)

However, there is a need for independent testing equipment which is done typically before the sale of these products. The same logic would be favoured for in-use testing, where an independent authority would test each system with their testing apparatus.

VIII. Assessment Metrics

a. Energy Efficiency

There are numerous metrics used to assess the performance of energy efficient technologies and different interpretations of these vary depending on the time scale and the type of system level phenomena investigated.

In this study, we examine performance using metrics relevant to MVHR and ASHP systems. Equation 1 and 2 describe the HRE for supply and exhaust sides of the system as defined by I.S. EN 13141-7:2010 [41].

$$\eta_{supply} = \frac{T_2 - T_1}{T_3 - T_1} \times \frac{q_s}{q_{ex}} \quad [-] \tag{1}$$

$$\eta_{exhaust} = \frac{T_3 - T_4}{T_3 - T_1} \times \frac{q_{ex}}{q_s} \quad [-] \tag{2}$$

Where, T_1 is the outdoor air temperature, T_2 is the supply air temperature, T_3 is the extract air temperature, T_4 is the exhaust air temperature and q_s and q_{ex} are the volumetric flowrates, in m^3/s, of both supply and extract sides of the system respectively. Equation 3 describes the bulk SPF for MVHR systems.

$$SPF_{bulk} = \frac{E_{MVHR}}{Q_s} \tag{3}$$

Where, E_{MVHR} is the overall energy consumption of the MVHR unit in specific modes (heat exchange or by-pass), and Q_s is the airflow rate of the supply side. The SFP described here was a bulk value for the entire unit, which was the most comparable with manufacturer's datasheets given the available data. Efforts were made to measure the fan power of each fan; however, this was not possible. The theoretical performance of all AWHPs can be calculated in a very simple manner by using the Carnot Co-efficient Of Performance (COP) shown in Equation 4.

$$COP_{Carnot} = \frac{(T_{src} + 273.15)}{\Delta T_{s-s}} \quad [-] \tag{4}$$

Where, T_{src} is the source temperature (which is the external ambient air temperature for AWHPs) and ΔT_{s-s} is the difference between source (T_{src}) and sink (T_{sk}) temperatures, known as the lift temperature, described in Equation 5.

$$\Delta T_{s-s} = T_{sk} - T_{src} \;\; [°K] \tag{5}$$

The sink temperatures we refer to here depends on the system adopted and the set points observed, these can be fixed, or weather compensated. Typically, manufacturers report the COP at fixed flow temperatures or for a temperature application (e.g. low, medium, high temperature application), calculated at the test points indicated in EN 14511 [38]. Equation 6 describes the heating capacity, Q_{HP}, in Watts, W, according EN 14511-3.

$$Q_{HP} = q\rho C_P \Delta T_{i-o} \quad [W] \tag{6}$$

Where, q is the volumetric flow rate in m^3/s, ρ is the density of water or air in kg/m^3, C_p is the specific heat capacity of the heat transfer medium at constant pressure in J/kg K and ΔT_{i-o} is the difference between inlet and outlet temperatures (which is applicable for waterside or airside calculations), expressed in Kelvin, K, which depends on the system. Equation 7 describes the calculated heat capacity for EAHPs (Q_{HP}) according to EN 14511-3 [66] using the same theory as above with reference to the EAHP examined specifically (see Figure 5).

$$Q_{HP} = q\rho C_P \Delta T_{4-6} \quad [W] \tag{7}$$

Where, q is the volumetric flow rate in m^3/s, ρ is the density of water or air in kg/m^3, C_p is the specific heat capacity of the heat transfer medium at constant pressure in J/kg K and ΔT_{4-6} is the difference in temperature (in this case we refer to temperature across the evaporator), expressed in Kelvin, K. Equation 8 describes the COP. The COP is defined in EN14511-1 [67] as the ratio of heating capacity to effective power input.

$$COP = \frac{Q_{HP}}{P_E} \quad [-] \tag{8}$$

Q_{HP} is the heating capacity expressed in Watts and P_E is the effective power input, also expressed in Watts, W. The Carnot COP is a useful benchmark in that it gives an indication of the maximum COP that a system can achieve for given boundary conditions. It also relates well to manufacturers test points. The metrics used to assess heat pumps in-use performance can vary, though, the most reported metrics are either time-averaged COPs or SPFs. The SPF and monitoring for heat pump systems in the building sector (SEPEMO-Build), more commonly known as the SEPEMO project, defined a series of system boundaries that can be used to calculate system efficiency over a defined time interval [68]. Outside of presenting individual test points according to EN 14511, manufacturers also quote a Seasonal COP (SCOP) according to EN 14825 [39] if the heat pump is used for SH and may also quote a seasonal value for hot water performance according the EN 16147 [40]. The SCOP is a weighted COP, which includes different heat pump operational modes as well as backup heating requirements and reflects the annualised performance of a heat pump at reference climates. The SCOP is often calculated with the assumption that the heat pump is designed for the load it is tested at, which often neglects the use of backup heating. However, most published field trials or studies report a SPF to reflect annualised performance.

Table 7: Comparison of approaches in standards (Taken from [68]]). (SPF H4 and EN 15316 are highlighted in blue)

Component	SPF H_1	SPF H_2	SPF H_3	SPF H_4	EN 14511	EN 15316-4-2	EN 14825	EN 16147	EuP, Lot 1: 2012	EuP Lot 10:2012
Compressor	x	x	x	**x**	x	**x**	x	x	x	x
Brine fan/pump		x	x	**x**		**x**		x		
Back-up heater			x	**x**		**x**	x	x	x	x
Buffer tank/pump				**x**		**x**		x	x	
SHW fans/pumps				**x**		**x**		x		
Final or Primary	F	F	F	**F**	F		F	F	P	P

The SPF can be quoted at different boundaries (see Figure 7 and Table 7). Different standards or approaches result in performance being reported at different boundaries which includes or excludes different types of supporting systems such as pumps or fans. Table 7, Table 8 and Figure 7 indicate what standards or design stage tools (Passive House Planning Package (PHPP) and DEAP) are relevant to which boundaries and what is included from this perspective.

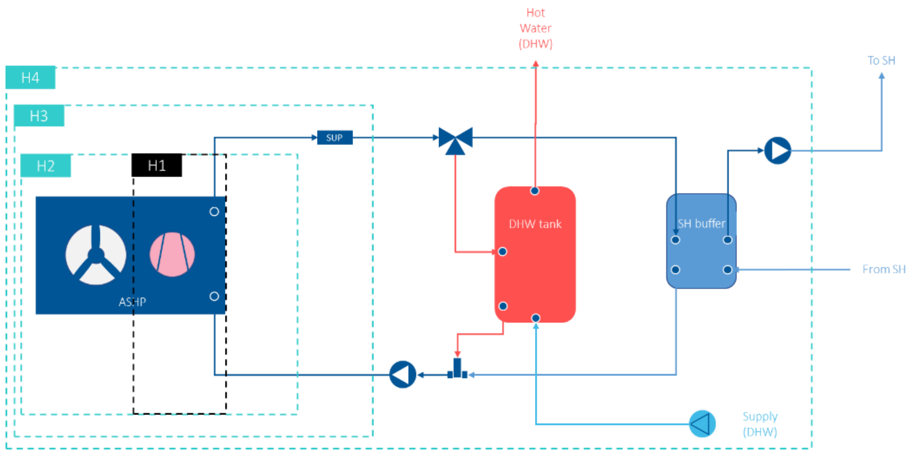

Figure 7: System boundaries defined by SEPEMO project and RHPP project for calculation of seasonal or system performance factor.

Table 8: Comparison of in-use measurement boundaries from SEPEMO, this study (SPF-H3,ex) and relevant standards and standardised values used in Ireland (HX refers to 'heat exchanger')

Component or heat source	SPF H_1	SPF H_2	SPF H_3	SPF $H_{3,ex}$	SPF H_4	EN 14511	EN 16147	PHPP V9.0	DEAP 3.2.1
Compressor	x	x	x	x	x	x	x	x	x
Brine fan/pump		x	x	x	x		x	x	x
Back-up heater			x	x	x		x	x	x
Buffer tank/pump					x		x		x
SHW fans/pumps					x		x		x
Heat on condenser side	x[1]	x[1]	x[1]		x[1]	x[1]	x[1,2]		x[1,2]
ΔQ evaporator side (ΔT_{4-6})	x[1,2]	x[1,2]	x[1,2]	x[2]		x[2]		x[2]	
ΔQ inlet air and discharge air (ΔT_{1-6})								x[2]	
Heat from HX (ΔT_{3-4})				x[2]		x[2]			

[1] Used in typical AWHPs in-use

[2] Used for EAHPs

Table 8 describes these boundaries and their relevance to EAHP's, it should be noted that some standardised approaches (e.g. EN 16147 DHW standard for HPs) find EAHPs difficult to interpret because there are questions over the inclusion of the heat recovered from associated system level heat exchangers. It is also worth noting that the EAHPs unit studied in this project were not covered under EcoDesign standards such as EN 14825 and so no SCOP is calculated for this type of system. Furthermore EU directives do not consider EAHPs as a renewable source [50] as this air is not a “naturally occurring thermal energy”, surface or sewage water is excluded also. It should also be noted that Table 8 is an interpretation of what is applicable for EAHPs given the available data from manufacturers data. Equation 9 describes the calculation of SPF at the H_1 boundary while Equation 10 describes the calculation of SPF at the H4 boundary.

$$SPF\ (H_1)\ =\ \frac{Q_{HP}}{W_{HP}}\quad [-] \tag{9}$$

$$SPF\ (H_4)\ =\ \frac{Q_{HP+}Q_{BUH}}{W_{HP}+W_{se,p}+W_{buh}+W_{sk,p}}\quad [-] \tag{10}$$

Where, Q_{HP} is the heat output of the heat pump, Q_{BUH} is the heat output of backup heaters, W_{HP} is the work of the heat pumps compressor, $W_{se,p}$ is the work of the source pump or fan, W_{buh} is the work of the back-up heater and $W_{sk,p}$ is the work of the distribution pump(s). The work of SEPEMO proposed that this could be reflected in the form of a Carnot efficiency for the same conditions. In addition to instantaneous or continuous measurement of the SPF, a cumulative version of this was also calculated monthly using averaged hourly data (shown in Equations 11 and 12).

$$SPF\ (H_{1,\mathrm{m}})\ =\ \frac{\sum Q_{HP,m}}{\sum W_{HP,m}}\quad [-] \tag{11}$$

$$SPF\ (H_{4,\mathrm{m}})\ =\ \frac{\sum Q_{HP,m+}Q_{BUH,m}}{\sum W_{HP,m}+W_{se,p,m}+W_{buh,m}+W_{sk,p,m}}\quad [-] \tag{12}$$

Where, $Q_{HP,m}$ is the heat output of the heat pump accumulated over a month, $Q_{BUH,m}$ is the heat output of backup heaters accumulated over a month, $W_{HP,m}$ is the work of the heat pumps compressor accumulated over a month, $W_{se,p,m}$ is the work of the source pump or fan accumulate over a month, $W_{buh,m}$ is the work of the back-up heater accumulated over a month and $W_{sk,p,m}$ is the work of the distribution pump(s) accumulated over a month. Equation 13 describes the Carnot efficiency for SH only ($\varepsilon_{C,SPF}$). Equation 14 describes the proposed in-use Carnot efficiency, which is the Carnot efficiency calculated at the H_4 boundary.

$$\varepsilon_{C,SPF} = \frac{SPF_{H1}}{SPF_C} \tag{13}$$

$$\varepsilon_{C,\,SPF\ H_4} = \frac{SPF_{H4}}{SPF_C} \tag{14}$$

Where, SPF_{H1} is the calculated SPF at the H_1 boundary for heating only and SPF_C refers to the Carnot SPF, which is the Carnot COP (from Equation 4) calculated using in-use data. The Carnot efficiency described by SEPEMO was found to vary between 0.3 to 0.5 for small electric heat pumps, and 0.5 to 0.7 for large very efficient electric heat pumps [68]. The efficiency metric proposed in this paper is an adjustment of the Carnot efficiency to reflect in-use performance at a different boundary. Where, SPF_{H4} is the calculated SPF at the H_4 boundary for heating only and SPF_C refers to the Carnot SPF, which is the Carnot COP calculated using in-use data. By including additional energy from components other than the compressor, this efficiency metric will attempt to reflect system performance as a function of the theoretical maximum.

b. Energy Performance and Renewable Compliance

In addition to efficiency criteria, the final energy consumption can be calculated for each end-user. In addition to this, the primary energy consumption ($E_{primary}$) can be calculated using Equation 15.

$$E_{primary} = E_{final} \text{ x PEF} \tag{15}$$

Where, E_{final} is the final energy consumption for each end-user or device (heat pump, whole house, etc) and PEF is the primary energy factor which accounts for the energy consumed by the energy sector supplying each end-user or device. The PEF indicated by the SEAI in 2020 was 1.830257 [69]. It should be noted that current evidence on PEFs for electricity suggests that the PEF for electricity is decreasing at a rate faster than was projected (see Figure 8). As such, the primary energy consumption for all case studies and its associated appliances and devices (being primarily electrical devices) is likely to improve over time.

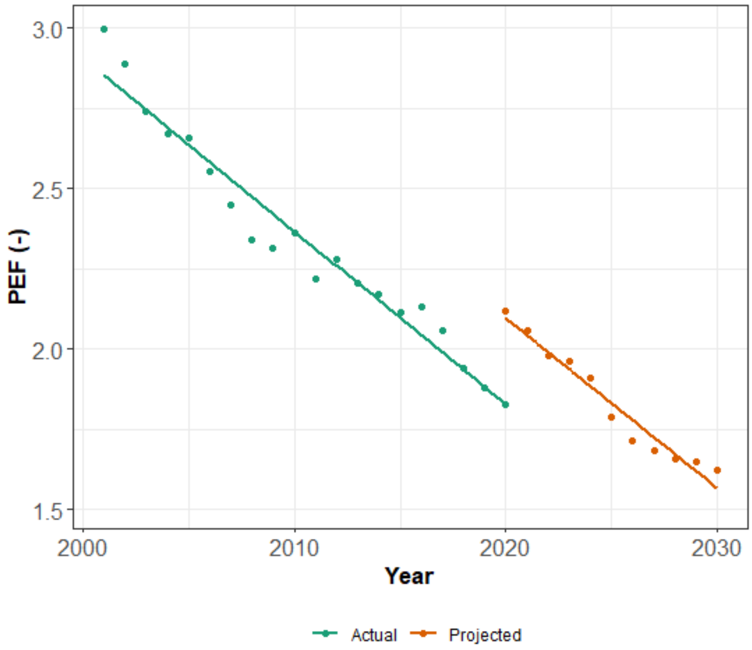

Figure 8: Actual and projected primary energy factors for electricity in Ireland provided by the SEAI [69], [70]

In terms of renewable contributions this is determined according to relevant EU regulations in accordance with Equations 16 and 17 (mentioned in **section VIII b**).

$$E_{RES} = Q_{usable} * (1 - \frac{1}{SPF}) \quad (16)$$

$$SPF_{min} = SPF > 1.15 * \frac{1}{\eta} \quad (17)$$

Where, E_{RES} is the amount of energy delivered by the heat pump that can be considered renewable, Q_{usable} is the estimated usable heat delivered by heat pumps, which is only considered permissible if the average SPF of the heat pump is greater than SPF_{min} indicated in Equation 16 and η , refers to the ratio between the total gross production of electricity and the primary energy consumption of electricity, and is calculated based on EU averages [50]. Previous work on renewable definitions has identified that the minimum SPF that heat pumps need to achieve is around 2.88 [71], however, more work is needed on update to date definitions as this is likely to vary from member state to member state. To illustrate this, Figure 9 indicates the likely minimum SPF required at EU level and in Ireland (IE) for heat pumps to be considered renewable since 1990.

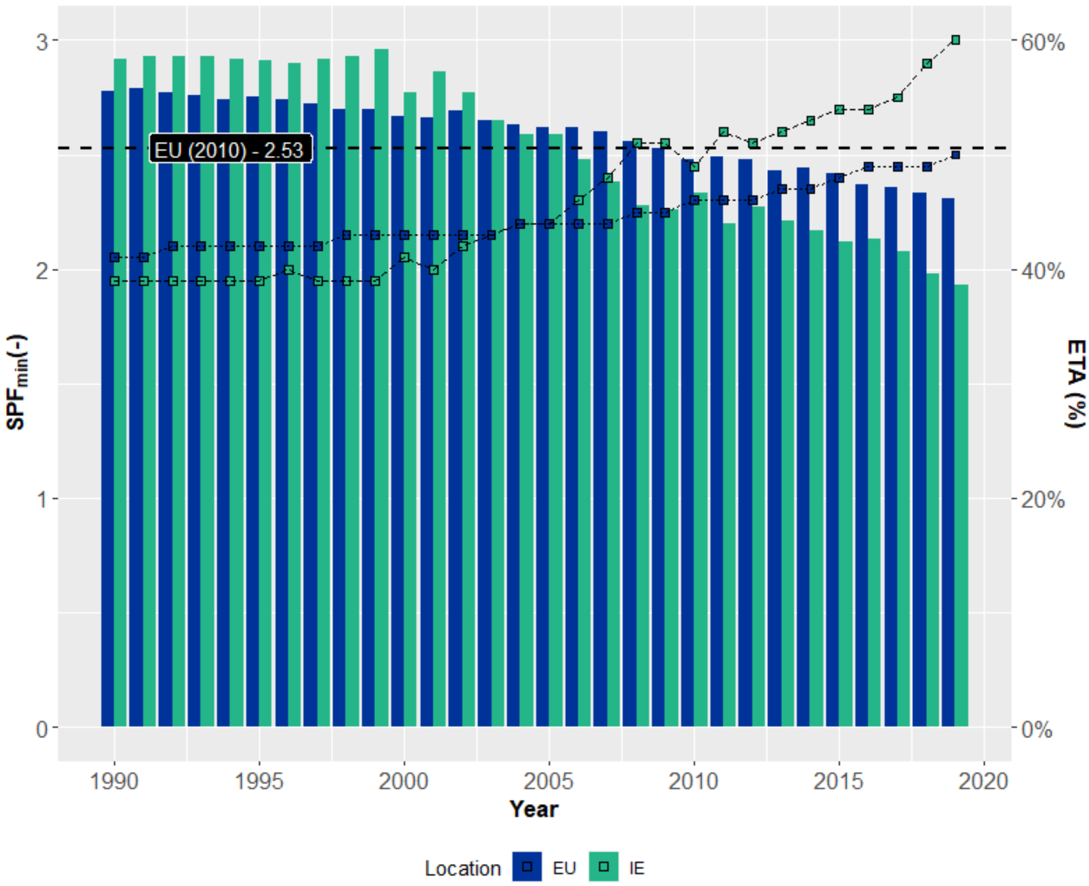

Figure 9: Calculated minimum SPF requirements in Ireland and in EU from 1990 to 2020 based on data taken from Eurostat [72]. (Bars indicate minimum SPF requirements, lines and points indicate eta values, minimum SPF for 2010 quoted by [1] is indicated in black)

Based on this figure we can see that the improvement in the efficiency in the Irish electricity generation market has led to grid efficiency (eta) values that are greater than the EU average year on year since 2003. The effect of this is that in Ireland, heat pumps could be considered as renewable heat with average SPF values that are lower than the EU average, where in 2019 the minimum SPF was 1.93 in Ireland and 2.31 in the EU. These indicates that minimums considered for renewable compliance are likely to be updated in the coming years [1], [3]. This development and likely reduction in the PEF (shown in Figure 9) is likely to lead to a reduction in requirements for heat pumps over time in terms of minimum performance criteria for renewable status. It should also be noted that the criteria determining the renewable energy contribution (shown in Equation 16) limit the renewable contribution of heat pumps to always be less than 100%. This is shown in Figure 10 below.

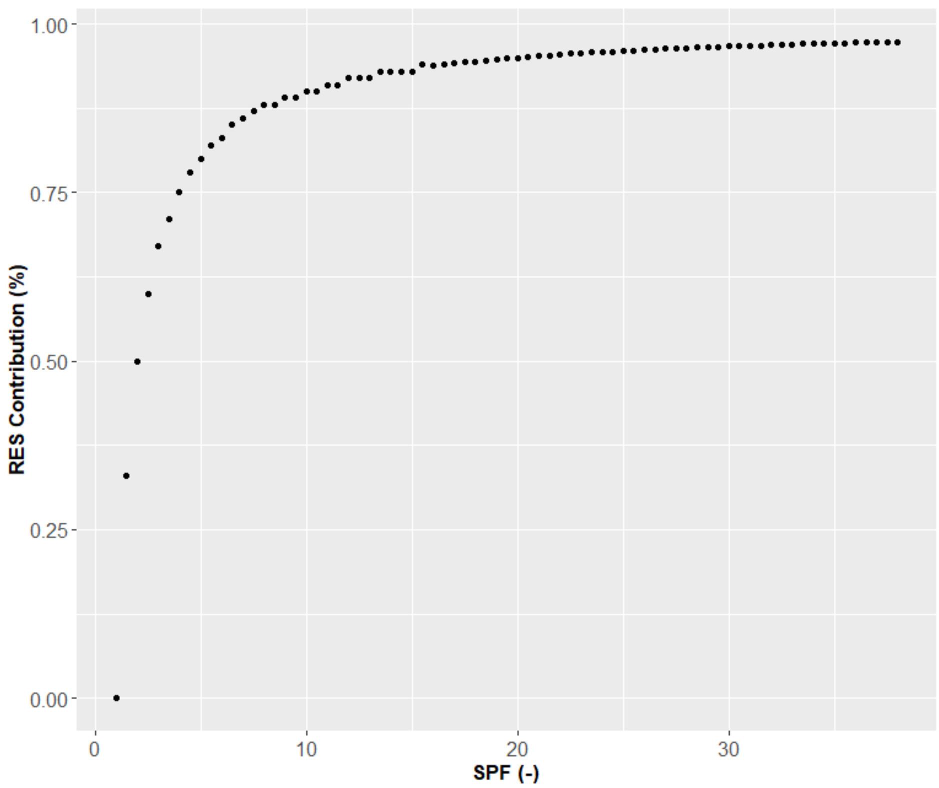

Figure 10: RES contribution of heat pumps according to EU directive 2018/2001

It should also be noted that there tends to be a large spread in performance for these systems whether they are EAHPs or ASHPs, which means some systems may not comply with minimum requirements. In general terms with upper SPF values of around 5 or 6 it is likely that around 60% to 85% of the heat generated by heat pumps will be counted as renewable. For heat pumps to cover the electrical energy they use to generate heat minimum SPFs of greater than 2 are required.

IX. Results and Discussion

a. Performance Mapping Study

The following mapping study summaries the likely performance of AWHPs, EAHPs and MVHR systems as installed in residential and non-residential buildings.

AWHP systems

Table 9 provides some detail as the performance from different studies in Europe and further afield. The main drawback of using a heat pump is that the PEF for each location can affect the renewable status of the device itself (as indicated earlier). Nowak et al. reported that seasonal performance factors (SPFs) needed to be around 2.875 at a minimum if heat pumps were to be counted as renewables [71]. The EHPA state 2.53 as a requirement [73], while a value of 2.5 is a requirement in some countries [22]. There is a clear goal for operating AWHPs to exceed these average SPFs in-use. Over the past ten years an extensive library of published AWHP in-use performance studies has emerged in the literature [1], [2], [18], [74]–[78]. Table 9 provides some detail as the performance from different studies in Europe and further afield. The average SPFs reported in these studies is around 2.8 with a large range of reported values of between 1.2 to 4.5 depending on the system boundary and location. Overall, this indicates a spread in SPF values reported in literature where several heat pumps are not likely to comply with minimum renewable requirements.

Table 9: Examples of in-use performance of air source heat pumps from field studies, according to location and calculation method

Author or Study [Ref]	Year	Location	No HPs	SH	DHW	System Boundary	In-Use SPF (%)		
							Mean	Min	Max
Kelly and Cockroft et al. [74]	2011	UK	8	x	x	-	2.7	-	-
Fraunhofer (PH1&PH2) [79]	2011	DE	18	x	x	H2	2.9	2.6	3.5
EST Phase 1 [19]	2012	UK	22	x	x	H4	1.8	1.2	2.2
SEPEMO [68]	2012	AT	2	x	-	H3	3.6	3.5	3.7
	2012	DE	3	x	x	H3	3.3	3.2	3.5
	2012	NL	4	x	x	H3	3	1.7	3.7
	2012	FR	1	x		H3	2.5	-	-
	2012	SE	2	x	x	H3	2.5	2	3
DTI [80] in [81]	2013	DK	11	x	-	H3	2.7	1.9	3.2
Fraunhofer ISH [78] in [81]	2014	DE	35	x	-	H3	3.2	2.3	4.3
Enova [82] in [81]	2015	NO	5	x	x	H4	1.8	1.2	2.3
IEA HPT Project [77]	2016	UK	2	x	x	H3	3.5	3.3	3.7
	2016	SE	1	-	x	H3	3.1	-	-
	2016	CH	3	x	x	H3	2.9	2.6	3.1
	2016	CH	5	x	x	H3	3.2	3	3.5
	2016	UK	15	x	x	H4	2.4	2	3.6
RHPP [83]	2017	UK	292	x	x	H2, H4	2.7,2.4	1.5	4.5
O'Reilly et al. [2]	2019	IE	20	x	x	-	3.3	2.6	4.3
Chesser et al. [1]	2021	IE	12	x	x	H4 (H, A)	3.1,3.0	2.5	3.6

EAHP systems

With the transition to low energy housing across Europe integrated units that combine EAHPs with Heat Recovery Ventilation (HRV) and, in some instances, DHW storage, are becoming increasingly popular in the domestic market with over 320,000 EAHP systems installed to date [73], and 24,000 of these purchased in 2017 alone [73], (less than 1000 of the total EU EAHP stock is in Ireland [73]). Research on the performance of EAHPs has been well documented over the past 40 years. Table 10 summarises this work in tabular format. Limb et al.

documents much of the early work in this area in the mid 90's [84]. Limb highlighted that residential air-to-water EAHP systems could achieve COP's of between 2.0 to 3.5 on average and up to 5.0 in "extremely favourable cases" and COP's of between 2.0 and 5.0 for air-to-air systems [84]. Studies contained within their review identified considerable energy savings with EAHP systems when compared to other conventional systems at the time. Average energy savings of between 25% and 50% were reported and upwards of 60% in some cases [84]. More recent field studies have indicated that EAHPs have generally underperformed when compared to design expectations [17], [85]–[88]. Charlick and Summerfield et al. [85] reported COP's of between 1.4 and 2.8 for EAHPs in the UK. Mikola and Kõiv et al. [86], [87] reported COP's of between 2.9 and 3.4 depending on the outside conditions in Estonia. Littlewood and Smallwood et al. [17] reported COP's between 0.4 and 1.7 for EAHP's in the UK. Rämä et al. reported COP's of between 2.0 and 4.0 for EAHP's depending on conditions in Finland [88].

Table 10: Results from literature review on the performance of EAHPs by various methods

Author or Study [Ref]	Year	Study Type	Location	COP, SPF, SCOP (%)		
				Mean	Min	Max
Limb et al. [84]	1996	Literature review	Multiple	2.0 to 3.5	2	5
La Francastoro and Serraino et al. [89]	2010	Simulation calibrated	Italy	-	4.5	6
Charlick and Summerfield et al. [85]	2012	Field study	UK	-	1.4	2.8
Mikola and Kõiv et al. [86], [87]	2014	Field study	Estonia	-	2.9	3.4
Littlewood and Smallwood et al. [17]	2016	Field study	UK	-	0.4	1.7
Rämä et al. [88]	2015	Field study	Finland	-	2	4
Thalfeldt, Kurnitski and Latõšov [90]	2018	Simulation	Estonia	3.6		

However, simulations of EAHPs have indicated that EAHPs can achieve average seasonal values of between 3.6 and 6.0 [89], [90]. Thalfeldt, Kurnitski and Latõšov et al. [90] modelled the performance of EAHP systems in Tallinn, Estonia. This work indicated that EAHP's can achieve SCOP values of 3.6. La Francastoro and Serraino et al. [89] validated and simulated an EAHP model for conditions in Italy. This work highlighted that EAHP systems can achieve average seasonal values of between 4.5 and 6.0 [89].

MVHR system performance

The multi-objective expectations on MVHR systems are evident from other work in the field. A limited number of studies have been identified which focus on the performance of heat recovery ventilation in non-residential buildings [91], with this work focusing largely on the design of systems. Residential studies indicate an under-performance of MVHR systems in reality when compared to nominal values or design expectations, despite the vast majority reporting IAQ benefits [92]–[95] and energy savings [95]. Most studies report differences between reported and actual efficiencies. Nominal or rated values of between 75% and 94% [96]–[100] were observed, with actual or in-use values of between 50% and 82% [99]–[101]. However, some example systems within studies indicate improvements in HRE when compared with nameplate values [95]. Although there is a lack of nominal SFP values, the expectation with most domestic systems is for a value of less than $0.45W/m^3/h$ [95]. Reported values of SFP in-use are in the range of $0.23W/m^3/h$ and $1.90W/m^3/h$ [95], [100]. Outside of these metrics, studies in the literature have identified several in-use issues that lead to underperformance. The most common issues reported are:

1) Flow deviations from design [93], [95], [102],
2) Poor installation and commissioning of systems [95],
3) Inadequate maintenance [95], [103],
4) Lack of training or information for users during handover [102], [104],
5) Issues with flexible ducting or ducting generally [93], [104],
6) Unbalanced systems [93], [95], [99], [102], [103], and,
7) Air permeability deviating from design values [95].

Based on this review, there is lack of detailed in-use field studies of MVHR systems with non-residential application, as well as the limited number of studies, which conduct post-design evaluations in tandem with post occupancy evaluations.

In-use factors used in current literature

To date there have been a number of IUFs proposed for mechanical ventilation systems to reflect in-use performance and differences in system design in reality [105]. A series of IUFs are used for the SFP and the HRE depending on the type of data available (measured or other), the type ducting (flexible, rigid, or no duct) and the difference in insulated ducting as it enters dwellings. The use of these types of IUFs are mostly used to discourage poor design and are based on observations and laboratory measurements of in-use issues, where HRE values were reported to be 90% but in-use efficiency was 50% in some cases [106]. This was particularly the case when systems were installed outside of the thermal envelope [107]. Factors of between 0.25 and 0.9 are used to adjust the performance data from manufacturers of MVHR systems for different reasons or observations [107]. A limited amount of IUFs are used for ASHPs, however, the BRE proposed a value of 0.95 for hot water vessels [108].

b. Measured Performance and Renewable Compliance

Air-to-water heat pumps - Residential

Energy measurements for all residential heat pumps (AWHP-1,2,3) were conducted in continuous period between February 2020 to December 2020. However, system level data was extracted from some systems in small one-week periods (e.g., AWHP-3) or was limited by visiting restrictions for each case study. At least five weeks of shoulder and winter season data was collected for each system. SPFs were calculated using an hourly averaged dataset that was statistically aggregated from an empirical dataset that sampled at 1-minute intervals. In addition to this, cumulative monthly values were also reported to provide a more holistic view of seasonal variations for heat pumps that had sufficient data (i.e., at least one month). SH and DHW modes were also investigated separately. Figure 11 indicates energy or power maps for each heat pump where the colour indicates the consumption for that hour (which was down sampled from 1-minute data). All three heat pumps have different intensities by virtue of their size but also because of their system configuration. Both AWHP-1 and 2 were continuously operated but it should be noted that AWHP-1 was deliberately turned on and off by homeowners over the summer, it was also used for SH only.

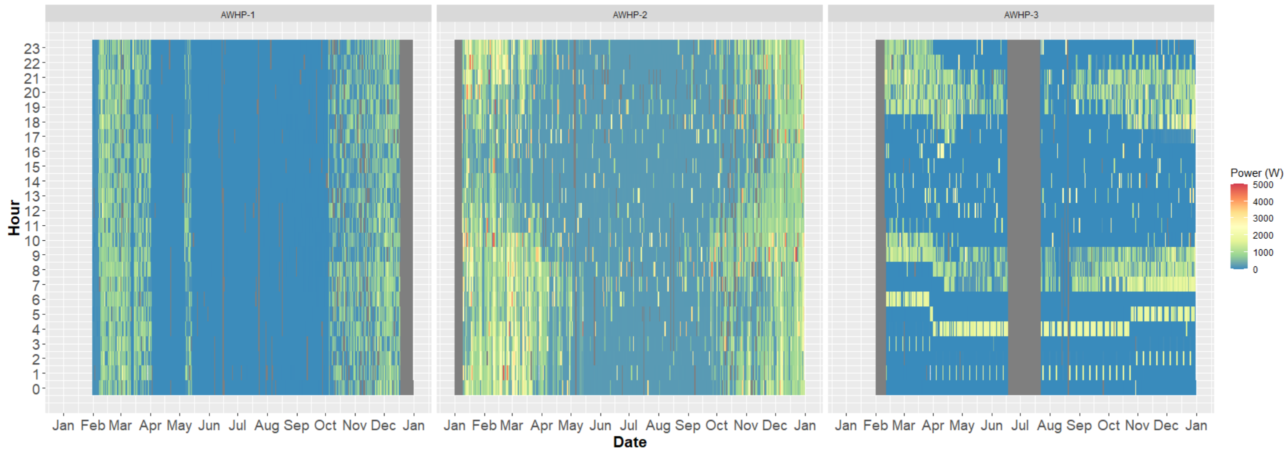

Figure 11: Energy/power maps for all residential AWHPs in 2020 (Grey indicates periods where data logging systems dropped out, colour is the hourly Wh as indicated on Efergy systems)

Figure 12: Compressor frequency maps for AWHP-1,2 in 2020 (Low frequency for AWHP-1 <= 30%, Low frequency for AWHP-2 = <=25% (red), Mid-range is between either of these lower thresholds and 90% (nude), high frequency is greater than 90% (blue))

AWHP-2 shows the greatest diversity of power as it satisfies DHW and SH but also appears to vary in intensity and frequency in the shoulder seasons. AWHP-3 is the only system that has a schedule for heating in the morning and evening for several hours. The activation of legionella mode is also indicated at 2am or 3am depending on the season. It's also clear when DHW is operated intermittently between these schedules. Figure 12 indicates compressor frequency maps for AWHP-1 and AWHP-2 respectively. Based on this, different utilisation of variable speed drives is visible. Both vary frequency; however, AWHP-2 appears to be on more with fewer off (i.e., red) cycles and more periods of continuous operation in the mid-range more than AWHP-1. AWHP-1 on the other hand appears to be operating intermittently each hour with several low frequency or off periods which is likely to lead to greater on off cycling for this unit.

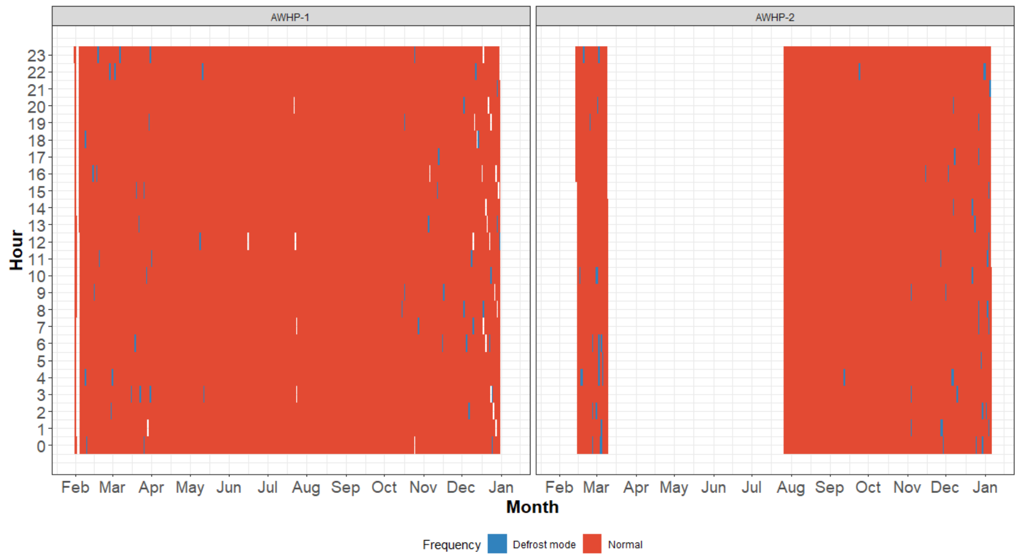

Figure 13: Defrost mode maps for residential AWHP-1,2 in 2020

Figure 13 indicates the defrost status of the same heat pumps. Very few defrost cycles are evident for each heat pump (<1% of the time). Defrost cycles appear to occur during heat demand periods (i.e. winter) and less so in shoulder seasons. Figure 14 indicates the average hourly performance of each AWHP with respect to expected average values from EN 15316. Given the lack of data on mass flow rate on the condenser side of the system for AWHP-1 an evaporator side calculation was conducted which related the airflow rate of the outside evaporator fan to the fan speed indicated on the manufacturers data logging system. Details on airflow measurements can be found in Appendix A. For comparative purposes it was assumed that mass flow rate of the pump was the maximum as delivered in manufacturers testing. Both approaches (evaporator and condenser side approaches) yield similar results.

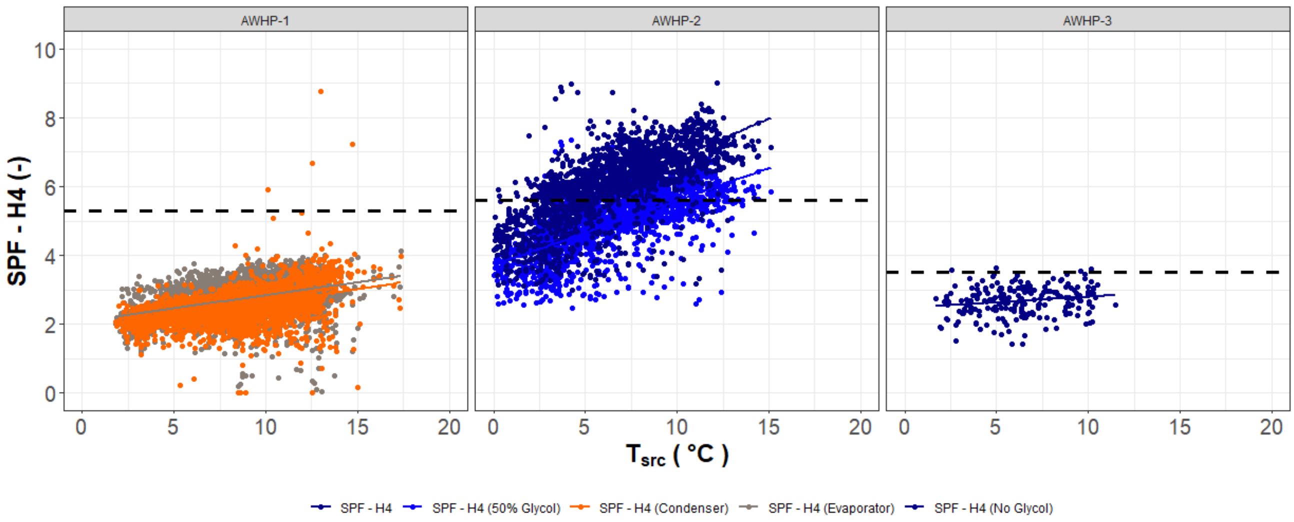

Figure 14: Relationship between SPF and external air temperature for AWHPs in SH mode (Fill indicates the types of data used for calculations, dashed black line indicates the values entered in energy rating systems or for EN 15316)

Differences presented in AWHP-2 are for differing assumptions around glycol percentage. Larger differences are observed for this assumption. However, both datasets have average values that are close to the value proposed in EN 15316. Overall, AWHP-1 and AWHP-3 are seen to underperform with respect to expected values for EN 15316. The causes of these are not fully clear, however, for AWHP-1 it is likely to be over-sizing or under-utilisation of heat pump capacity (see Figure 12). For AWHP-3 it is also likely that on off cycling is more prevalent given the fact that there a distinct on and off periods scheduled in this systems control system (see Figure 11 for reference to scheduled operation). However, more data is required to confirm this although it is likely that by design on off cycling is like to be cause.

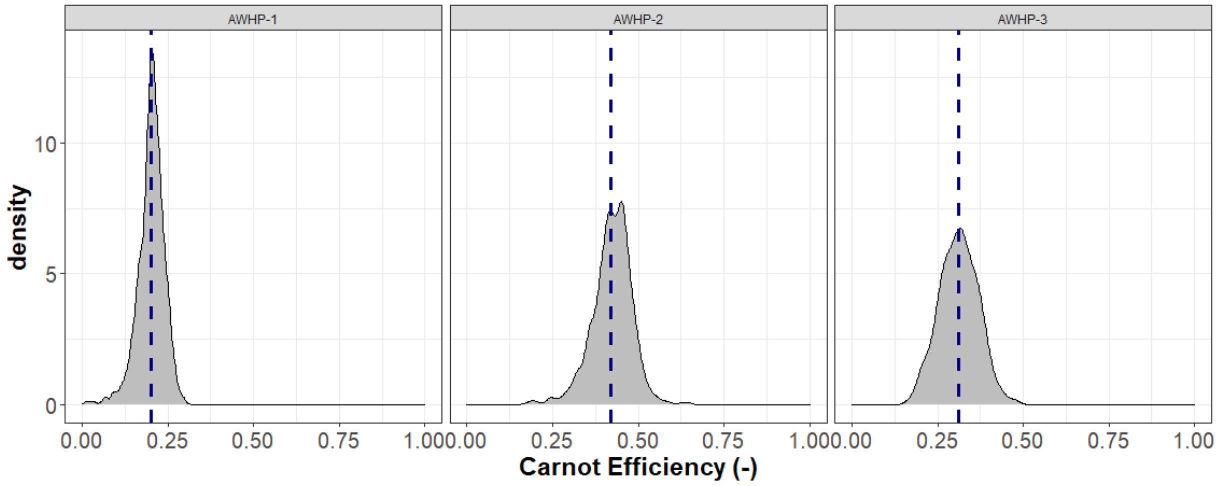

Figure 15: Relative density of in-use Carnot efficiencies for all three AWHPs for SH mode only (Dashed vertical navy line indicates the mean, efficiency is calculated based on evaporator side for AWHP-1 and a 50% glycol assumption for AWHP-3)

Figure 15 indicates the differences in in-use Carnot efficiency (see equations 12 and 13) for all AWHPs mentioned. As a ratio of maximum theoretical to actual, this metric provides some insight for comparison purposes between heat pumps. This metric gives some insight into whether each heat pump is under/over performing however despite this it is difficult to attribute under-performance to one factor based on this, given the fact that there is a distribution of efficiencies.

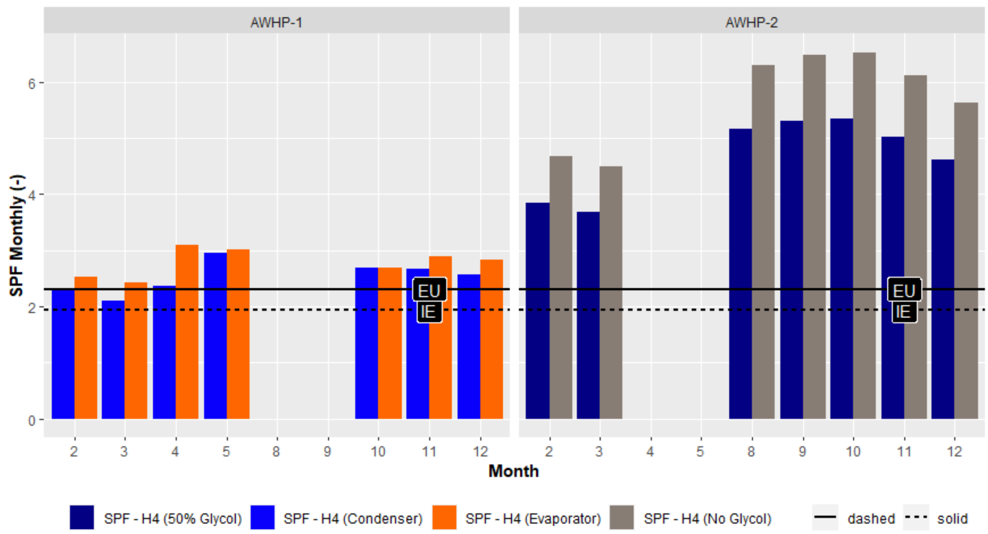

Figure 16: Monthly SPF at H4 boundary for two residential AWHPs in SH mode (Dashed and solid lines indicate the renewable status threshold indicated by EU directives)

It does however serve as a diagnostic for other studies. AWHP-1 has the lowest Carnot efficiency and is outside the range proposed by SEPEMO for small electric heat pumps (i.e. 0.3 to 0.5), AWHP-3 performs better in this regard, while AWHP-2 has a mean that is close to the upper end proposed. Figure 16 indicates the monthly SPF (see Equations 11 and 12) for both AWHP-1 and AWHP-2. Lower seasonal variance was observed for AWHP-1 in SH mode and all monthly values (using an evaporator side calculation) comply with minimum SPF requirements over a monthly time horizon. Larger seasonal variance was observed for AWHP-2 in SH mode. Most calculated monthly values are greater than 3.5 with high values of between 5 and 6 in September and October. Unsurprisingly this heat pump also demonstrates compliance with minimum SPF values according to EU directives. In addition to SH mode, AWHP-2 and AWHP-3 also satisfied DHW demand in their respective dwellings. Figure 17 indicates scatterplots of SPF (at the H4 boundary) and the source temperature (i.e. air) for both systems in DHW mode only. Figure 18 indicates the cumulative monthly performance for AWHP-2 with reference to EU and IE thresholds described earlier.

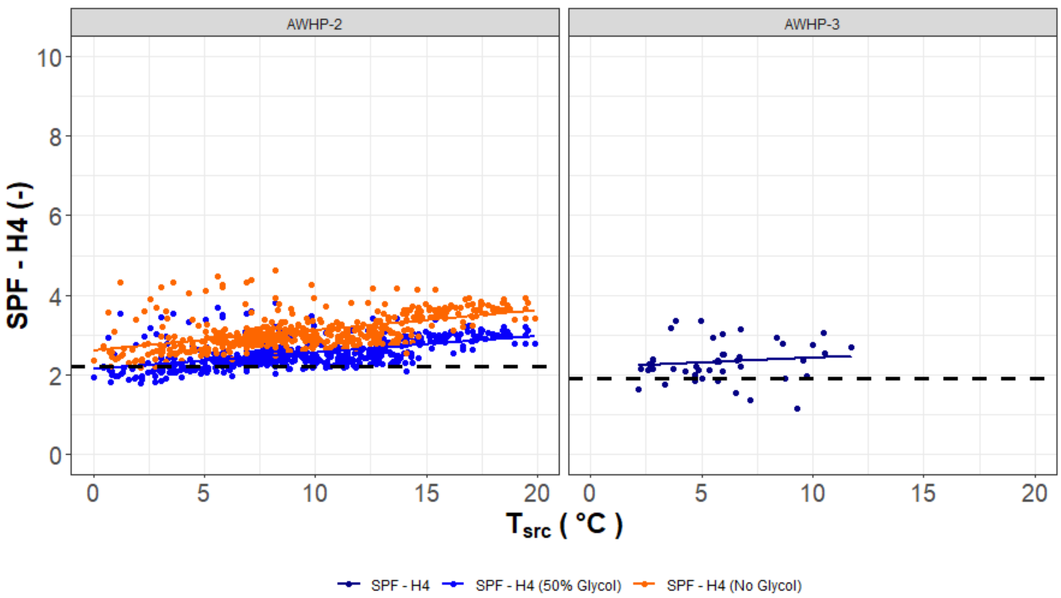

Figure 17: Relationship between SPF and external air temperature for AWHPs in DHW mode (Fill indicates data used for calculations, dashed black line indicates the values entered intro energy rating systems)

Both figures highlight that overall AWHP-2 and 3 outperform the standardised values that are used to represent their performance in energy performance certificates for DHW. Figure 17 highlights a relationship between

DHW SPF values and external conditions which is less variant than the same external conditions for SH. This is likely due to the higher flow temperatures required for DHW.

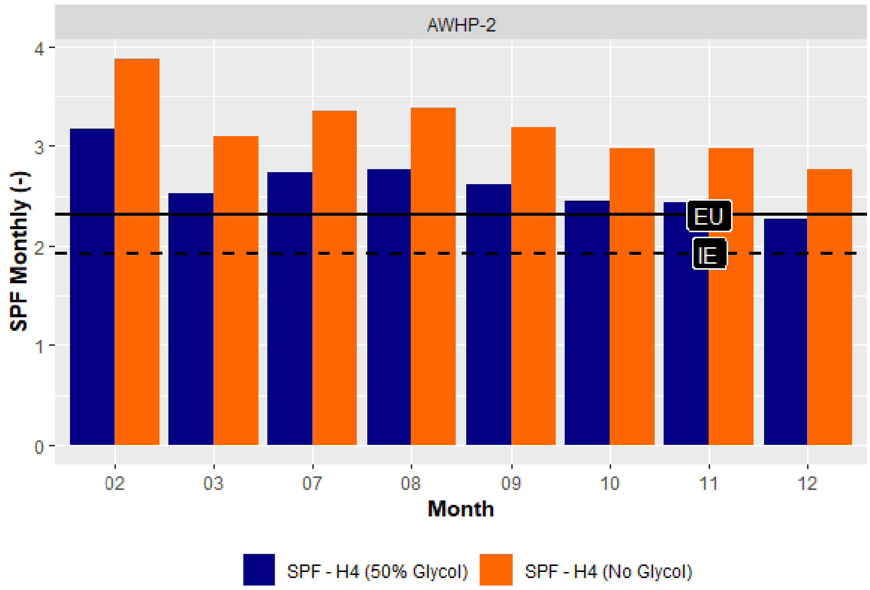

Figure 18: Monthly SPF at H4 boundary for AWHP-2 in DHW mode (Dashed and solid lines indicate the renewable status threshold indicated by EU directives)

In addition to this, the accumulation of energy over a month highlights no significant seasonal variation (if anything higher SPF monthly values in winter).

Table 11: Hourly averaged performance of all residential AWHPs in SH and DHW respectively compared with values entered into DEAP

HP	EN 14825 (SH)	EN 15316 (SH)	SPF_{H4ave} (SH)	EN 15316 (DHW)	SPF_{H4ave} (DHW)
AWHP-1	5.1	5.3	2.7	NA	NA
AWHP-2	4.4	5.6	4.9^{+} to $6.0^{\wedge}$	2.2	2.5^{+} to $3.1^{\wedge}$
AWHP-3	3.2	3.5*	$2.7^{\neg}$	2.0	$2.3^{\neg}$
* Value taken for similar building in the same housing development | + 50% glycol assumption | ^ 0% Glycol assumption | ¬ data for heat pump was limited					

This may be counter intuitive; however, it is likely that less cycling is occurring in colder months and as such the heat pumps efficiency is not suffering as much. During summer months the external conditions may be favourable, but due to the lack of SH load, and consistent heat pump operation, it is likely that summer SPFs are similar to winter months due to cycling. The intermittence of operation is likely to cause substantially reduced COPs overall. Previous work has highlighted that cycling can cause a 25% reduction in seasonal efficiency [109], where COPs on start-up can be 50% less than when the heat pump is fully operational [110]. For the most part AWHP-2 meets minimum SPF criteria for DHW also. Table 11 indicates the average performance of all AWHP's compared to their standardised performance. These results indicate that overall, two of the three heat pumps investigated under-performed with regard to standardised SH performance (49% for AWHP-1 and 23% for AWHP-3). One heat pump (AWHP-2) is likely performing in line with its expectations on SH performance. Regarding DHW performance, both heat pumps that had sufficient data appear to perform close to expectations with one appearing to out-perform standardised DHW values. Regarding DHW, it is likely that the values used in DEAP are lower also because of the required tank temperature of 60°C, which penalises the performance indicated in EN 16147 by including additional back-up heater operation to lift water (by often 10°C) from that standardised average hot water tank temperatures.

Exhaust Air Heat Pumps – Residential (R1/R2 - EAHP1/2)

There were two EAHPs (EAHP-1, EAHP-2) that were analysed in this study. Both had integrated MVHR units also (MVHR-1, MVHR-2), this section will describe the performance of all four systems. One EAHP and MVHR was in an urban location the other in a rural location. For simplicity when referring to both the MVHR and EAHP, the EAHP term will be used, however, when discussed separately specific acronyms will be used. Figure 19 indicates the power or energy (Wh) consumed at each hour of 2020 for each EAHP system. From this there are two distinct energy levels for this unit, one in blue (HX mode) and one in green (HP mode). There appears to be no mode that is between these. It should be noted that EAHP-2 was in a different fan setting during February before it was changed to a lower fan speed. Both energy maps indicate no patterns of HP consumption that show no considerable seasonality in the data (i.e. winter summer). Patterns appear random which is indicative of DHW usage for the most part.

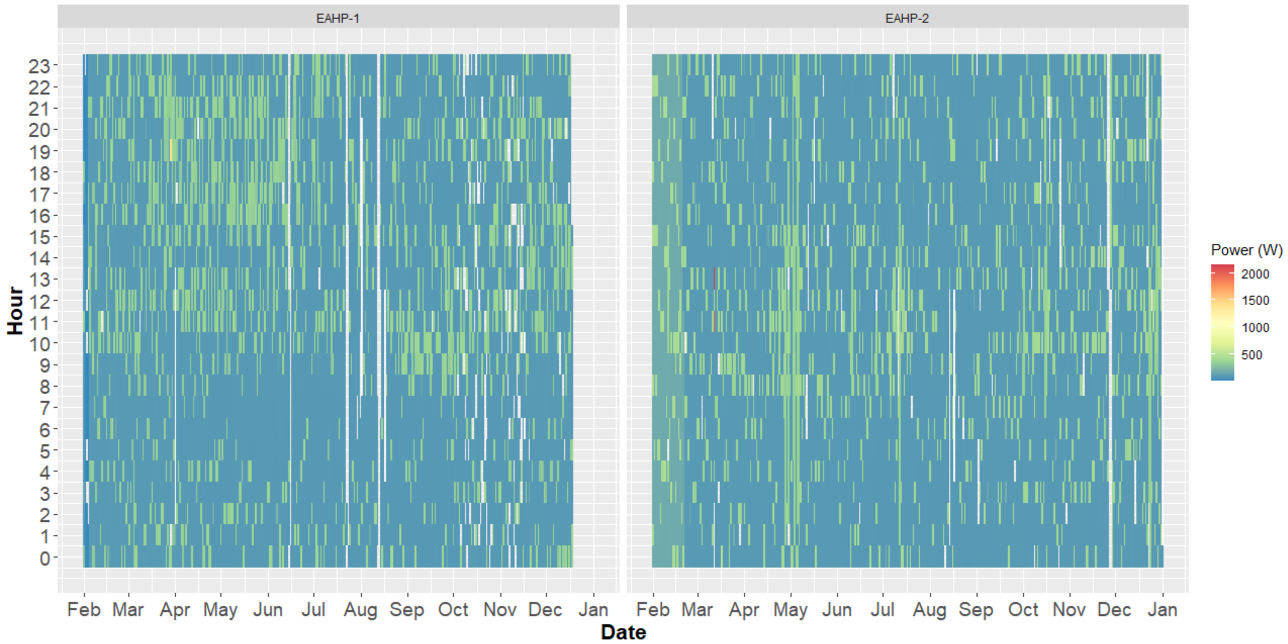

Figure 19: Hourly Energy maps for both EAHP systems in 2020.

However, it is likely with long periods of operation for both systems that SH mode is activated, but this cannot be conclusively identified. Figure 20 indicates the modes of operation extracted from the manufacturers dedicated data logging system. It should be noted that none of the data received allowed for an SH or DHW comparison in HP mode. The same pattern of usage is evident in this map (which was also indicated in Figure 19). Overall, EAHP-1 was in HX mode 77% of the time studied, in HP mode for 22% of the time with the remainder indicating the operation of the bypass. EAHP-2 was in HX mode 84% of the time and in HP mode 16% of the time (bypass operation was not operated). In reference to both MVHR systems, the bulk SFP on the supply side for both units was 0.41W/m^3/h (~1.5W/(l/s)) and 0.36 W/m^3/h (~1.3W/(l/s)) for MVHR-1 and MVHR-2 respectively. Both systems are likely to be performing efficiently in this regard as this value overestimates the fan power. Both systems are likely to be matching or outperforming their designed values (1.2 W/l/s) on a per fan basis.

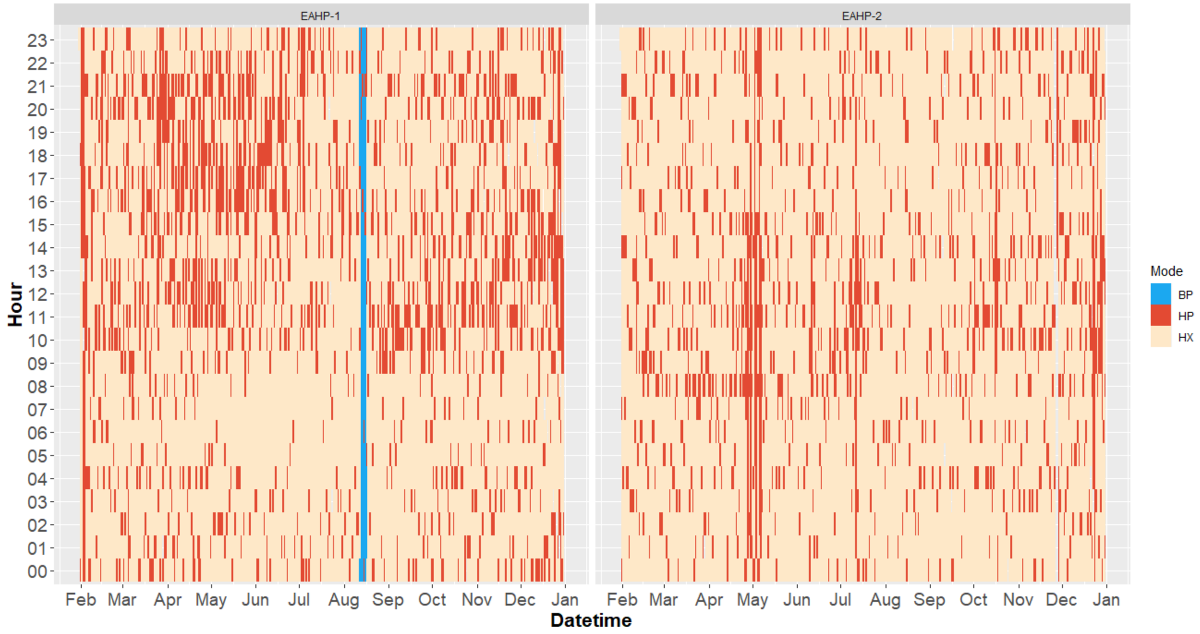

Figure 20: Mode map hourly for both EAHPs in 2020 (BP = Bypass mode, HP = Heat pump mode, HX= Heat exchange mode)

Table 12 gives an indication as to the measured HRE for both units compared with standardised values. MVHR-1 appears to be unbalanced with a supply side HRE of 64% on average, MVHR-2 has a supply side efficiency of 92%. MVHR-2 appears to outperform its standardised value and manufacturer values. MVHR-1 underperforms by 24% on average. Examining the extract side of the system and comparing with PHPP leads to different comparative performance. Both systems perform well when compared to design values in PHPP (1-2% difference) when extract side HREs are used. The under-performance in the supply side of the system is likely a balancing issue, based on observations of the extract side of the system, this side is more consistent in these types of systems. Table 13 indicates the performance of the EAHPs at different types of boundaries (see **section VII (a)**) for more detail on boundaries). There is a considerable difference in reported SPFs depending on which boundary is used to report efficiency. The average in-use efficiency is 3.6 and 2.3 for EAHP-1 and EAHP-2 respectively at the H_1 boundary. Overall there is more stability in the $H_{3,ex}$ boundary and higher efficiency values on average as would be expected due to the inclusion of recovered heat (EAHP-1 =3.7 EAHP-2 =2.9).

Table 12: Calculated values of heat recovery efficiency (HRE) for supply and extract sides of both MVHR systems of both EAHPs compared with standardised values. (Mean values have been indicated in bold)

Source	MVHR-1			MVHR-2		
	Min	Max	Mean	Min	Max	Mean
Measured HRE (supply)	28%	75%	**64%**	75%	99%	**92%**
Measured HRE (extract)	70%	88%	**81%**	70%	88%	**79%**
Manufacturer*	-	94%	**88%**	-	94%	**88%**
PHPP	-	-	**79%**	-	-	**78%**
DEAP	-	-	**88%**	-	-	**84%**
*Estimated based on manufacturer supplied information and typical unit flowrates						

As no modal information was available for each dataset it is difficult to identify the difference between both heat pumps. It is likely that there is predominately DHW mode data based on energy and mode maps (see Figures 19 and 20). Figure 21 highlights some of the differences that exist.

Table 13: Calculated SPFs for both EAHPs during monitoring from February to December 2020 according to system boundary compared to manufacturer reported and standardised values. (Equiv. Boundary refers to the equivalent in-use boundary, bold indicates mean values)

Source	Equiv. Boundary	EAHP-1			EAHP-2		
		Min	Max	Mean	Min	Max	Mean
SPF – H_1	-	0.0	6.0	**3.6**	0.0	6.0	**2.3**
SPF – H_3	-	0.1	3.6	**2.2**	0.1	3.4	**1.5**
SPF – $H_{3,ex}$	-	1.4	5.2	**3.7**	1.4	5.2	**2.9**
Manufacturer (DHW)[1]	H_4	-	-	**2.2**	-	-	**2.2**
PHPP (DHW+SH)	$H_{3,ex}$	-	-	**2.0**	-	-	**2.4**
DEAP (DHW)	H_4	-	-	**2.1**	-	-	**2.1**
Manufacturer (SH)[2]	-	1.7	5.7	**4.0**	2.1	5.9	**4.1**

[1] Value was calculated using the relationship between fan speed and airflow rate defined in Appendix A

[2] Calculated using a linear fit of COP and inlet air temperature, based manufacturer data for flowrate of 220m^3/h

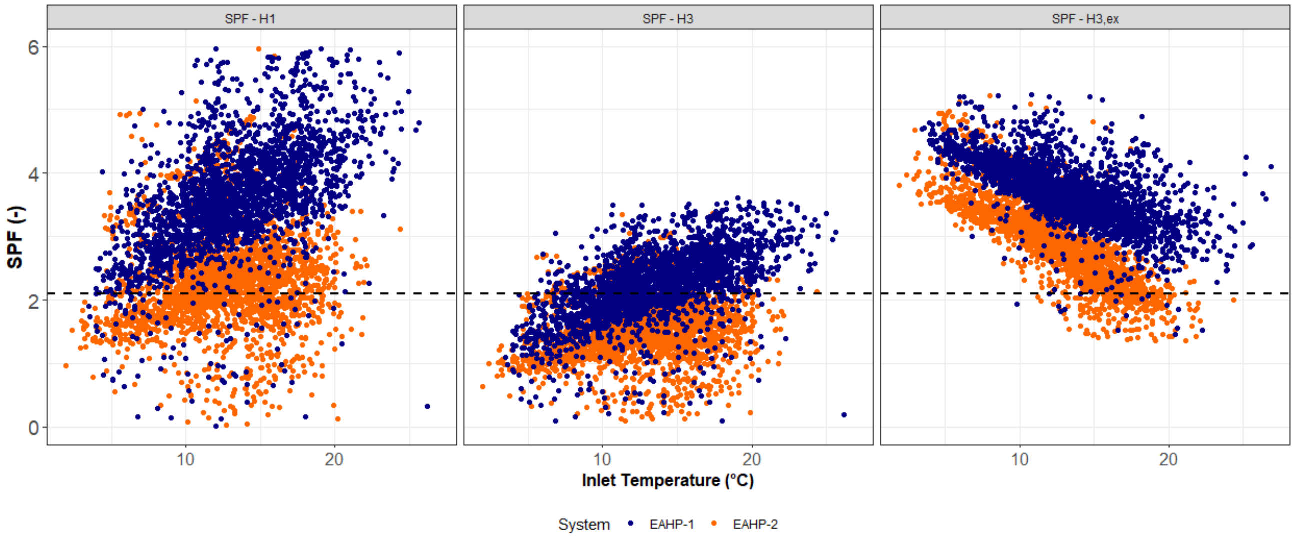

Figure 21: Scatterplots of SPF with respect to inlet temperature (T_1) for three different boundaries (Dashed line indicates DHW values used for EAHPs as entered into DEAP)

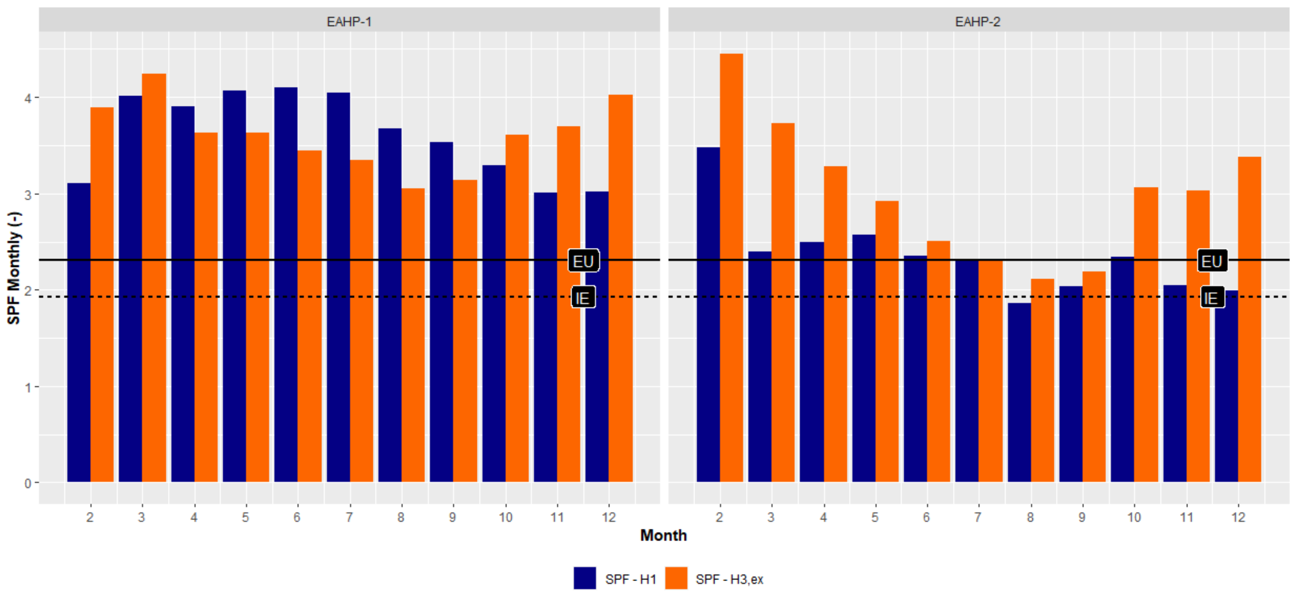

Figure 22: Monthly SPF at H4 boundary for EAHP-1/EAHP-2 (Dashed and solid lines indicate the renewable status threshold indicated by EU directives)

Given that both EAHPs have fixed speed compressors there is tends to be a linear relationship between capacity and inlet air temperature (and post-HX air), the urban local of EAHP-1 is there likely to experience higher external air temperatures on average (which is illustrate in Figure 21). Aside from this there is also a clear difference in the shape of both scatterplots (for EAHP-1 and 2). Some SH effects or more prevalent in EAHP-1 compared with EAHP-2, however, confirmation of this is needed. Clearly both systems perform well when compared with values in DEAP. Figure 21 also highlights the consistency of SPF values when the heat recovered in the HX is included as part of efficiency calculations, but also the inverse relationship with inlet temperature that is indicated in manufacturers data (see **Section VI**). Figure 22 indicates the cumulative monthly performance of both EAHPS using a similar metric to other work assessing EAHPs [111] with respect calculated EU thresholds and IE threshold for renewable compliance. This indicates that EAHP-1 complies with EU minimum SPF requirements consistently monthly irrespective of boundary considerations. The under-performance of EAHP-2 (in 4 of 11 months) is likely due to DHW data, however more information is required to confirm this.

Air-to-water heat pump – Non-residential (NR1 – AWHP-4)

A limited amount of data (for one heating season) between December 2019 and March 2020 was gathered at 15-minute intervals for the non-residential heat pump (AWHP-4).

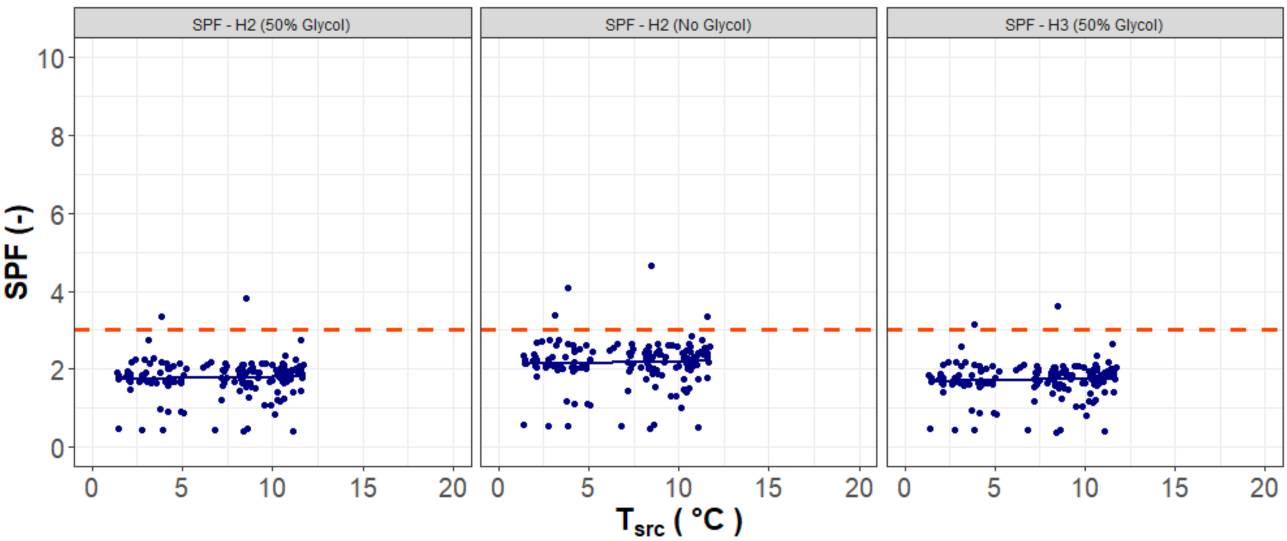

Figure 23: Scatterplots of SPF with respect source temperature (air) (Dashed line indicates reference COP of 3)

The data logging system was connected to a BMS system and so data was easily available. H2 and H3 boundaries were also possible with this system. There was no equivalent EN 15316 standard used to represent the systems performance, however, a COP of 3 was taken as a reference value as manufacturers data suggested. Figure 23 indicates the performance of AWHP-4 based on 15-minute data. Overall, the average performance of the system was 1.7 or 2.2 (depending on assumptions for glycol percentage). It is likely however that an SPF of 2 is realistic given the data. The lack of temperature relationship suggests that the unit is oversized for its application. The unit is a 28kW unit (14kW when using one compressor) and the non-residential office space has a floor area of a medium sized house (223m^2). It is likely that an 8 to 9kW unit will be sufficient for this building which means that this ASHP system was designed for a building nearly three times the size of the one it is serving. Despite this, the unit is on the margins of being a renewable in Irish terms but would not be a renewable if EU averages are considered. The unit is likely to be underperforming by 27% to 43% compared with design expectations.

Mechanical ventilation with heat recovery – Non-residential (NR2- MVHR-3)

The system performance of MVHR-3 was assessed via in-situ testing of the bulk SFP and HRE. The unit itself was new and was installed into a retrofitted office space. No fabric upgrades were completed as part of the retrofit; however, new radiators and lighting were installed in the open plan office. The performance of the unit was assessed in Heat Exchanger (HX) mode and Bypass (BP) mode during winter conditions in January of 2020. The external mean temperature during testing was 6.9°C. The MVHR consumed around 10W in standby or off mode which was the energy used for the controller. At high speeds the MVHR unit consumed between 215W and 236W (including 10W mentioned for controls). The unit consumed between 103W and 121W in low-speed mode. Table 14 compares manufacturer (Manu) and in-use performance (In-use) for different energy efficiency and airflow metrics under different operating modes. Overall, the unit consumed over 25% less energy at similar settings to those proposed by manufacturers. The bulk SFP was consistently lower than expected and there was a 1% to 3% improvement in HRE compared with manufacturers values.

Table 14: Summary of energy and airflow parameters calculated during a series of tests on MVHR-3

Operation Mode	Airflow rate (m³/hr)		Energy consumption (W)		SFP W/(m³/s)		HRE (supply) (%)	
	Manu	In-use	Manu	In-use	Manu	In-use	Manu	In-use
Low flow - HX	440	540	103	121	843	805	88	89
High flow – HX	800	828	303	236	1364	1026	83	86
Low flow – BP	440	540	86	103	704	690	NA	NA
High flow - BP	800	828	289	215	1301	934	NA	NA

A detailed subjective thermal comfort evaluation was also completed. This involved evaluating thermal comfort under a range of different flow speeds, different number of hours in operation and different time of day/week. Table 15 below presents the findings for this. The internal environmental performance as perceived by occupants of the space varied considerably depending on the setting of the MVHR as well as the, schedule used. As occupants were required to be seated prior to voting, this led to lower numbers than was possible in the office space (n max = 18). This was representative of the office space, as it was transiently occupied. Table 3 summarises the qualitative votes of occupants during Week 2 and Week 3. Overall, 73responses to the thermal environmental questionnaire were gathered during Week 1 and Week 2. The results indicate that the high flow setting was unfavourable amongst occupants. This is likely due to high air velocities at occupant work stations. This led to some occupants leaving the room entirely and not using the space, and led to a low sample size for morning voting on the 2nd of March (n=6). When switched to the low setting this improved, however, some of this improvement is also likely to be due to the room warming up over the course of the day. Following the 2nd of March, the MVHR schedule was adjusted to operate for shorter periods until it was turned off on Friday the 6th of March. Improvements in acceptability were seen over the course of Week 2, however, mean thermal sensation votes(MTSV) for each period (daily in most cases) were found to swing from negative to positive and back again. Although acceptability levels were highest on Friday, the lowest calculated daily MTSV was on Wednesday. The schedule on Wednesday was considered the most comfortable schedule. When occupant votes were correlated with indoor air temperatures in the middle of the office space, temperatures of between 19°C (68°F) and 22°C (72°F) were seen as satisfactory. During Week 3 very low MTSV values were indicated when the optimized schedule was used, which supports its adoption. However, some variability between acceptability and those dissatisfied needs further investigation.

Table 15: Summary of subjective votes from occupants during different days of the week

Week	Date (flow setting)	n	MTSV	PD (%)	Comfortable (%)	Acceptable (%)
2	02/03 (high)	6	-2.83	100%	0%	0%
	02/03 (low)	8	-0.875	25%	38%	63%
	03/03 (low)	10	0.60	20%	50%	80%
	04/03 (low)	11	0.55	9%	82%	82%
	05/03 (low)	8	0.75	25%	63%	75%
	06/03 (low)	10	-0.60	20%	60%	90%
3	08/03 (low)	10	0	12.5%	60%	60%
	09/03 (low)	10	0.3	0%	90%	100%

More details are available regarding energy and air quality performance below:

https://www.researchgate.net/publication/349252002_Technical_evaluation_of_a_retrofitted_MVHR_system_design_and_in-use_performance

c. Context and General Discussion

ASHPs and MVHR systems are set to be a dominant technology in Irish buildings in the foreseeable future. There has been and there is likely to be a marked uptake in the use of mechanical ventilation (MV) systems in all buildings. Additionally, retrofit and new build strategies at national level are pinning their renewable heating hopes on heat pumps [112] and the reduction in grid PEF and emissions.

ASHP systems

Figure 24 indicates the progression of Irelands energy sector (in residential terms) towards an electrified heat source. The percentage of new buildings that have electricity (i.e. standard, night-rate etc.) as the main fuel for SH is now outweighing oil, gas and other fuel sources. This monumental shift is stark where in 2010 around 6% of the buildings that were entered into the BER database were using electricity as the main fuel source. In 2019, the percentage of buildings using electricity as the main fuel source for heating was around 71%, it is likely that this percentage will increase in the future. Considering this context, it is likely that (in energy terms) ASHPs are likely to have a greater effect on renewable targets where deviations from standardised or reported values could have major effects on primary energy consumption [114]. It is also likely that manufacturers reported values could be misleading where the likelihood of high COPs is unlikely [115].

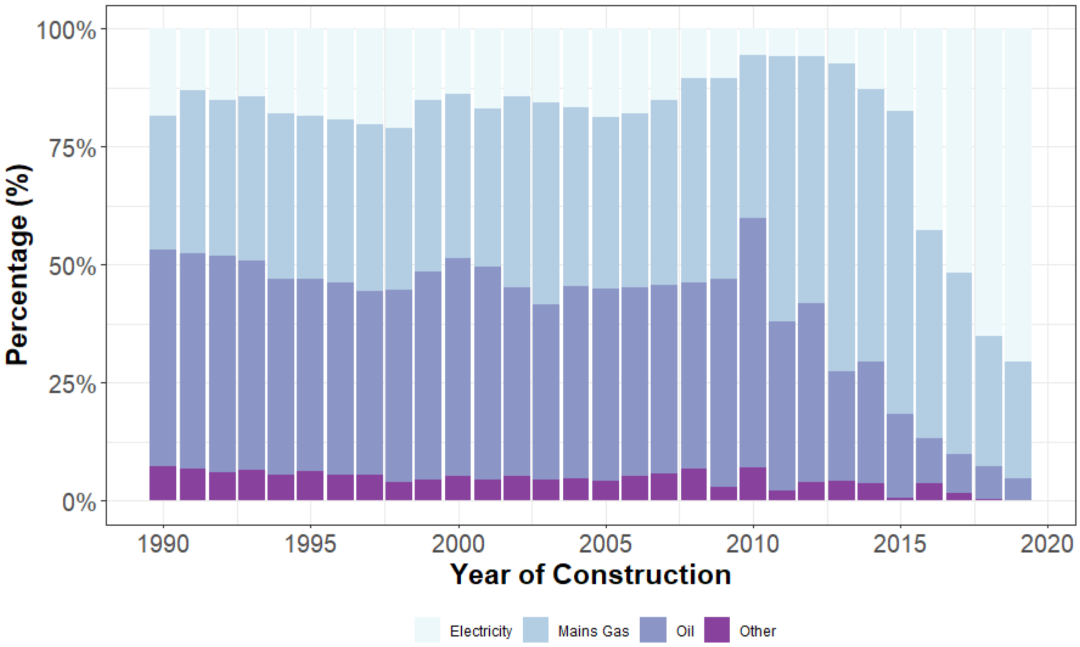

Figure 24: Stacked bar chart of percentage of buildings in BER database use different fuels as their main heating source as entered the national BER database for buildings from 1990 to 2020 [113]

Figure 25: Bar charts of average efficiency of SH and water heating systems as entered the national BER database for buildings from 1990 to 2020 (cross data points indicate the maximum efficiency reported)

It is therefore critical that designers are urged to not overestimate the performance of water heating and SH systems. Figure 25 illustrates the average efficiency of systems for SH and water heating over time according the NBER database (for residential buildings only). There is clear change in system level efficiency reported in this database overtime, with large changes occurring in the past decade but particularly over the last five years. This increase in mean SH and water heating efficiency indicates a clear increase in the use of HP technologies for both SH and DHW. Previous studies of ASHPs have indicated average values of 2.8 (280%) generally, with maximums of 4.5 overall (see **section IX (a)**), SPF values from recent studies in Ireland were on average 3.3 [2] (330%) or 3.1 [1] (310%) (depending on the sample size) with maximums of 4.2 (420%) (considering data for individual shoulder and winter season months) [2] or 3.6 (360%) (for individual systems over the entire heating season) [1] respectively. A recent systematic review by Carroll and Chesser et al. identified examples of minimum performance values that are as low as 1.4 (140%) in Northern Ireland [22], [116]. They also highlighted a lack of available data to calibrate or train data driven models for ASHPs [22]. Despite the body work, that has been completed to date, clearly more data on heat pump performance is required particularly in the Irish context and particularly given the quantities of heat pumps that are envisaged to be installed by 2030. More studies are required which span different system configurations, manufacturers, and temperature applications. Sample sizes (with a confidence interval of 95% and 5% margin of error) of between 340 and 380 (population from 30,000 to 600,000) would be sufficient going forward which is similar to studies in the UK [83]. The average performance of the residential heat pumps in this study was 3.4 (340%) and 2.4 (240%) for SH and DHW respectively, but also variable depending on the heat pump the system, the configuration, and the building it was serving. Figure 26 indicates the distribution of efficiencies for systems with an efficiency greater than 100% in the context of previous studies and the average of the BER database itself. The average difference between the values entered DEAP for SH and measured in-use values from literature, previous studies and this study is between 100% and 160% depending on the average taken for comparison purposes. The gap for DHW is less at about 30% to 60%, however, is less of an issue as the average SPFs in DHW mode appear to be higher than those in the BER database. The scale of these differences and the variance in results between the BER database and that of literature more generally is cause for concern.

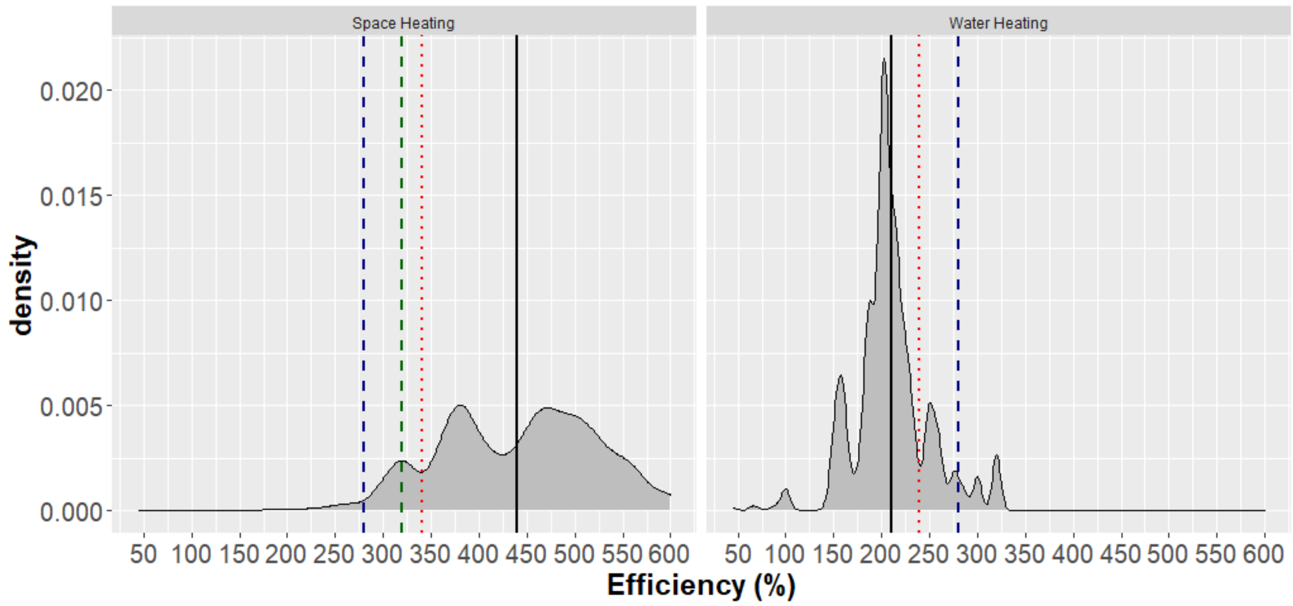

Figure 26: Relative density of efficiencies in domestic BER database compared with different studies. (Vertical lines indicate different average values from different studies, where solid black is average from BER database, dashed dark green is average of two recent studies in Ireland [2] (this is taken for SH only given the lack of evidence differentiating modes), dashed navy is average in literature overall (not considering DHW or SH separately) and dotted red is averages in this study for DHW and SH respectively.

There is a need to measure the performance of a larger sample of heat pump systems to fully determine the extent of differences and gaps that exist. However, at this early stage with a limited number of field studies of ASHPs in Irish residential buildings (as indicated in **section IX (a)**) the average efficiency values are higher than the literature in general (3.4 vs. 2.8) for SH. There is a gap between standardised efficiencies used to represent heat pumps, but this gap is difficult to express as IUF in a conclusive manner. Particularly as current IUFs for MV systems are based off laboratory studies where factors influencing performance can be controlled. Previous work by O'Reilly et al. has indicated COP ratios of between 41.3% and 75.0% depending on the season and the heat pump [2]. This indicates heat pumps underperforming in efficiency terms by between 25% and 59%. A recent study by Chesser et al. indicated an underperformance also however with smaller performance gaps between actual and predicted performance of 3% to 24% on average [1]. The measured results in this study indicate similar levels of underperformance of between 23% and 49% for residential heat pumps. The causes of underperformance are likely to be multi-facetted. In this study, the most obvious of causes for underperformance would appear to be over-sizing of heat pumps for their application. Systems that are oversized are likely to cycle more (even those that have variable speed compressors). When systems operate more intermittently or for short periods of time (in DHW mode or in swing or summer seasons) they are more likely to cycle on and off which can lead to average COPs that are 50% less on start-up [110]. The results presented in this document also indicates a decline in seasonal performance due to the same phenomena indicated by Madonna et al. [109]. Regarding the non-residential system studied (AWHP-4), the same over-sizing phenomena is observed where the system is underperforming by as much as 43% compared with design expectations. Similarly, there is a considerable need to measure non-residential systems in more detail and particularly those with different load profiles (i.e. process loads, office heating loads, data centre loads etc.) or those that are reversible or for cooling purposes. Unfortunately, no data on HP efficiency was available for non-residential buildings at the time of this publication to contextualise these results. Almost all heat pumps were considered renewables under EU regulations. The only system that could be excluded from this was AWHP-4 which was borderline in this regard.

MVHR systems

Regarding MVHR systems, there is also an increasing trend towards MV systems as a means to ventilate our buildings in Ireland. Figure 27 indicates the percentage of buildings in the BER database that use different ventilation systems. Similarly, to the trend in fuel type use, it is evident that MVHR system (and mechanical systems more generally) are seeing an increase in use over time, where in 2005 there were less than 1% of buildings that used an MVHR, in 2019 19% of buildings that were constructed used MVHR systems. The uptake in MV systems more generally seems to be slower than the electrification of heat, however, the changes in ventilation system choices are still remarkable.

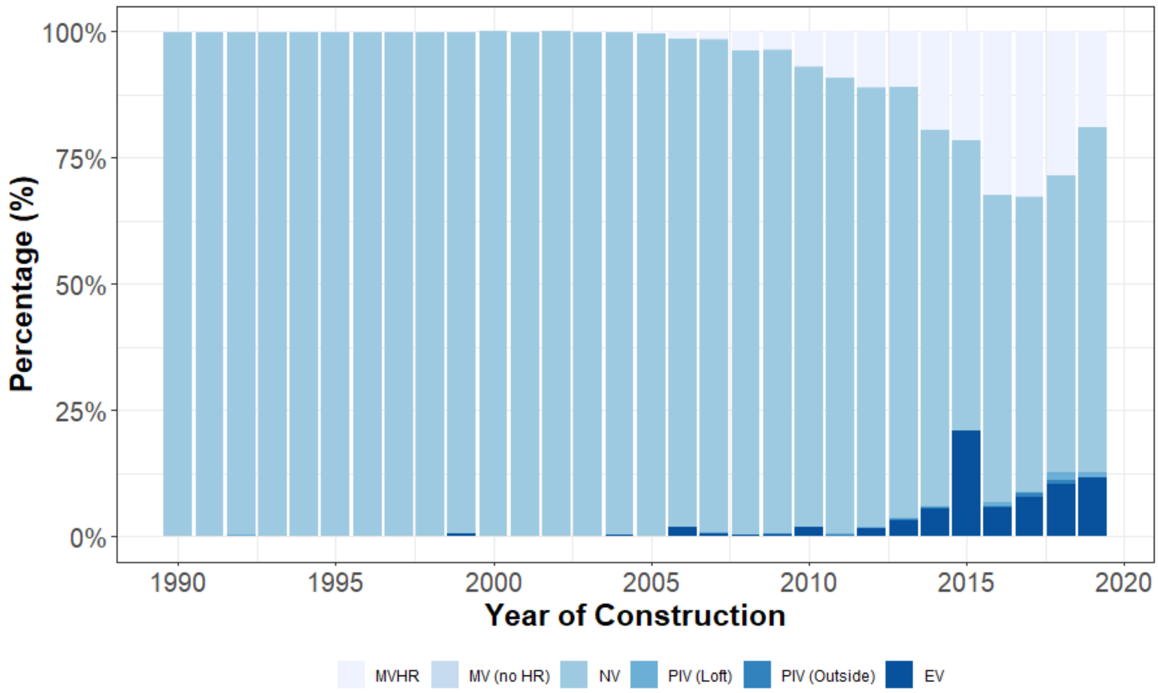

Figure 27: Percentage of ventilation system types in NBER database as entered the national BER database for buildings from 1990 to 2020.

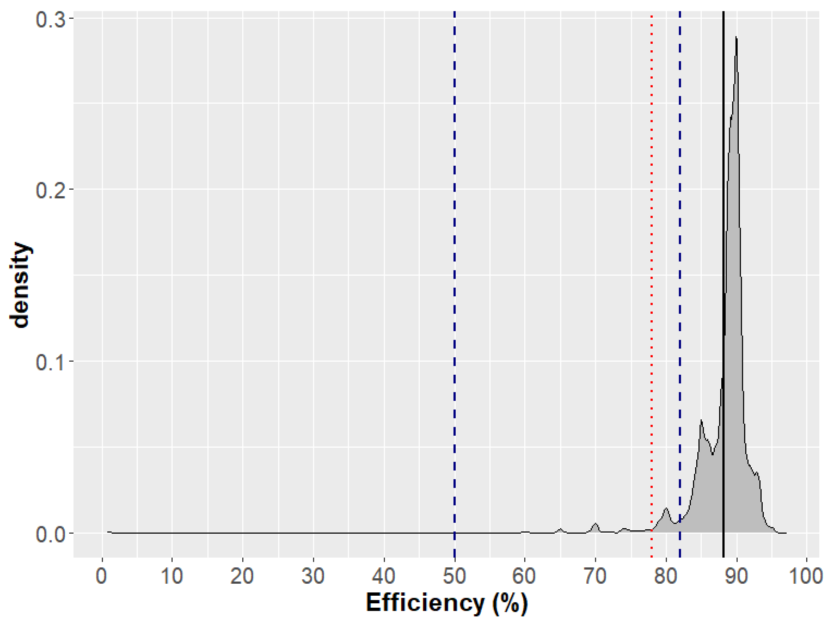

Figure 28: Relative density of reported heat exchanger efficiencies in NBER database compared with literature and this study (Navy lines indicate minimum and maximum values reported from mapping study, dotted red is average of the two domestic MVHR systems, black is average of NBER database with default values of 66% removed.)

In efficiency terms there would appear to be a remarkable difference between the manufacturers specified HRE and in-use HRE as indicated in Figure 28. The average results of the residential systems studied in this project would appear to be nearer the upper end of those studied in literature. The non-residential system achieved a high HRE compared with the minimum EU requirements and outperformed its nameplate value which is welcome. Similarly, to HPs there was no available database nationally to compare the work in this study with reported values for non-residential systems. As it was not possible to measure SFP on supply or extract sides separately discussion of these results could not be contextualised and compared adequately with national databases. The systems studied appear to perform close to their expected values. It should be noted that larger scale studies are required in Ireland for MVHR systems. Those that are a similar scale to those by Merzkirch et al. [100], Sharpe and McGill et al. [95] would yield results that are more conclusive. In addition to this, the performance of extract only systems should also be considered given the amount of these systems that are installed in new dwellings.

X. Observations, Recommendations, Future Work, Challenges and Limitations

The following section will detail observations, proposed recommendations and future work suggested by the MeSSO research team. These will be discussed in the context of the system type (EAHPs, AWHPs, MVHRs) and the building type (residential and non-residential) as well as general recommendations.

Exhaust Air Heat Pumps

Two EAHPs were studied, both were had small heating capacities (maximum capacity of 2.4kW) and had an additional heat pump (of at least 5kW in capacity), were in PassivHaus dwellings, and were used for SH and

DHW. These systems potentially suffered from an efficiency perspective because of having two heat pumps for SH as such their full potential was not realised. EAHPs are treated differently to other heat pumps in the EU at domestic level despite having similar levels of system level performance in literature (EAHPs ~ 3.0, AWHPs ~ 2.8, see Tables 9 and 10 for reference). A more comprehensive study of these systems is needed with a larger sample size. There is evidence from low energy buildings that these systems perform consistently well in colder climates [111]. Work completed by the MeSSO Research group has indicated that there is some difficulty in considering EAHPs in the same way as AWHPs, where their system level performance can be determined at different boundaries [3]. With this some major international studies have excluded EAHPs from their studies which is disappointing [68]. More work is needed in determining the correct boundaries that are appropriate in representing the performance of EAHPs that are agreed by all stakeholders. The renewable status of these systems is likely to be on average compliant even at the H1 boundary (average = 3.0), overall (considering the PEF and ETA of the EU grid in 2020 see **section IX (b)**). Some manufacturers data for these systems may not be sufficient to perform and adequate analysis of the system regarding its rated performance with reference to international standards that these products were tested under, without additional measurements of airflow rates.

Air to Water Heat Pumps

Three domestic and one non-domestic AWHPs were studied with capacities varying from 5kW to 28kW, all served A-rated buildings, some were used for SH only and others were used for SH and DHW. Varying levels of performance were seen with these systems, average SPFs of 3.4 were observed for SH and 2.4 for DHW in residential systems, and for the most part systems were considered renewable. Some systems did not meet design expectations (AWHP-1 and AWHP-3) and one system did (AWHP-2). The non-residential heat pump studied (AWHP-4) was likely to not be renewable and was considerably oversized for its application. Oversizing or under-utilisation was considered the main issue with AWHP-1 also. Under-performance of AWHPs is likely to lead to considerable differences in energy use when contextualised with assumed performance from national databases. Currently existing databases of actual performance are limited with small numbers of heat pumps being studied (n = 6 to 12) a larger study is required to determine to what extent the HP stock is underperforming. A suitable sample of non-residential buildings is also required for different applications. It should also be noted that there is a need to measure new heat pumps with new and emerging refrigerants (i.e. R32, CO_2) as companies implement the Montreal Protocol and other HFC reduction protocols [117]. The variation of HP performance over time should be considered also as there is some evidence that HP performance may degrade over-time [118], but the performance of systems and their components may improve. This is critical as the heat pump stock nationally is likely to have a mixture of refrigerants with early adopters having higher global warming potential (GWP) refrigerants and older systems. As such the average and cumulative energy performance of the ASHP stock is likely to vary year-on-year for various reasons outside of the likelihood that future warming may lead to more favourable conditions but potentially poorer performance due to intermittency in systems (i.e. more shoulder season weather) that were designed for a climate that was the typical at the time of design but is likely to change over the lifecycle of the product (i.e. 20 years). This issue may not be the case with brine to water or ground source heat pumps due to the consistency in ground temperatures.

Mechanical Ventilation with Heat Recovery

Two domestic MVHR systems and one non-domestic system was studied, all had quoted HRE values that were above 80% (supply side). Two of the three systems were found to outperform their rated values. One system was likely unbalanced from a supply side perspective. More work is needed to confirm the state of the Irish MVHR stock and their performance with a larger sample, like the other systems few large studies exist for measurements of these systems in Ireland. Difficulties were found in that not all manufacturers had performance data and not systems were easily measured for their SFP in-situ.

General

Acquiring adequate and reliable data from energy efficient systems is a major challenge if systems are to be assessed, but is a need for many research stakeholders not least those who endeavour to model HP systems [22]. The MeSSO team found engagement with some manufacturers to be difficult, where simple requests for sensor specifications or explanations of variables from manufacturers systems required multiple engagements and some cases access to vital important (e.g. test results from EN 16147) was not made available. More engagement and consultation are needed between government authorities and manufacturers to facilitate researchers in measuring these systems in open and transparent manner. Given the scale of investment in HPs there is a need for independent verification of system level performance which is mandatory for all HP manufacturers. All tests require a harmonised testing protocol that makes comparisons with manufacturers and standardised tests fair and appropriate boundaries need to be considered (see Tables 7 and 8 in **section VIII a**). Many studies use various boundaries to report results and averaging methods as well as seasonal variations need to be considered also (i.e. monthly, yearly etc). A testing protocol that is harmonised worldwide or at EU level is needed which is similar to those used in for vehicles [119] to ensure systems meet expectations as products are being sold on SCOP values that are often not achieved. Additionally, DHW should be more in focus over-time with A-rated buildings as the proportion of demand for DHW is much higher in these buildings. In some countries the proportion of energy demand for DHW as a function of both SH and DHW can be 50% for new buildings [120]. Consideration should be given to regular testing of HP systems to ensure they are maintained and operated as renewable devices. In addition to this, there is a need for researchers to assess the difference when reporting efficiencies at different boundaries or considering the inclusion and exclusion of heat recovery in certain systems. An example of this would be that current minimum thresholds for ASHP are reported at the H2 boundary [1], products are usually quoted at the H3 boundary and DEAP reports at the H4 boundary. For MVHR systems there is a focus on the supply side of the system for DEAP, however PHPP uses the extract side of the system when reporting efficiencies. Examinations of both sides of the system are required as they may lead to different results. Overall, there is a need to integrate in-situ measurements of in-use performance with performance values that are reported at design stage. Regarding the reflection of underperformance with an IUF, the MeSSO team could not attribute anything that could be easily specified at design stage that could reflect gaps that do exist. Finally, further work is required to simulate the effect of underperformance using calibrated tools such as TRNSYS [1] to isolate the causes of differences between design and reality.

Challenges and limitations

The following analysis and recommendations were conducted with available manufacturers data, energy measurement data and engineering judgement. There were several challenges and limitations in this study:

- The work presented considers a small sample of heat pumps and MVHR systems and examines them in some detail and results should be considered in that context. However, the work is contextualised to add to the discussion with data from the BER database and that of literature.
- The work considered low-cost measurement options to measure performance, this led to challenges regarding data acquisition and therefore not all boundaries could be considered for all heat pumps and as some variables were not measured, and some manufacturer systems did not measure compressor percentage or the status of the heat pump (e.g. DHW or SH)
- For MVHR systems the HRE was the only efficiency metric that could be considered comparable, issues in measuring power for each fan made SFP measurement prohibitive in a comparable manner to those reported in standards.

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XII. Appendix A

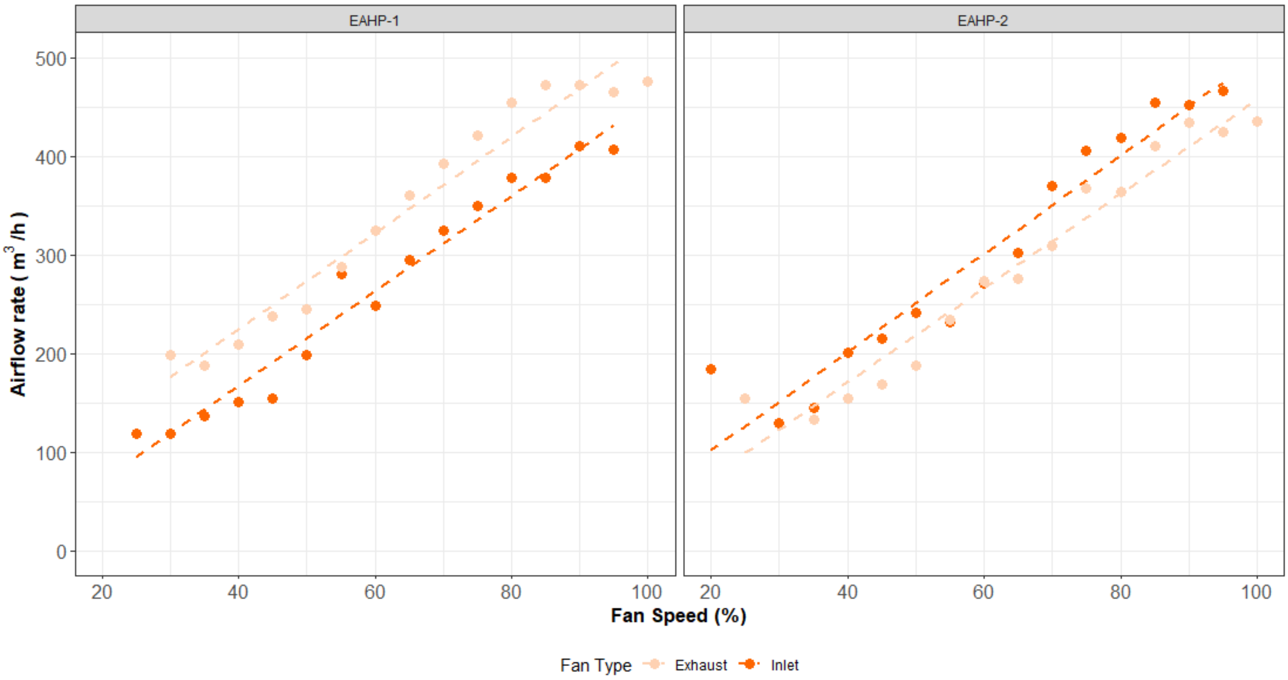

Figure 29: Empirical field measurement data for inlet and exhaust airflow rates with respect to fan speed conducted for EAHP-1 and EAHP-2

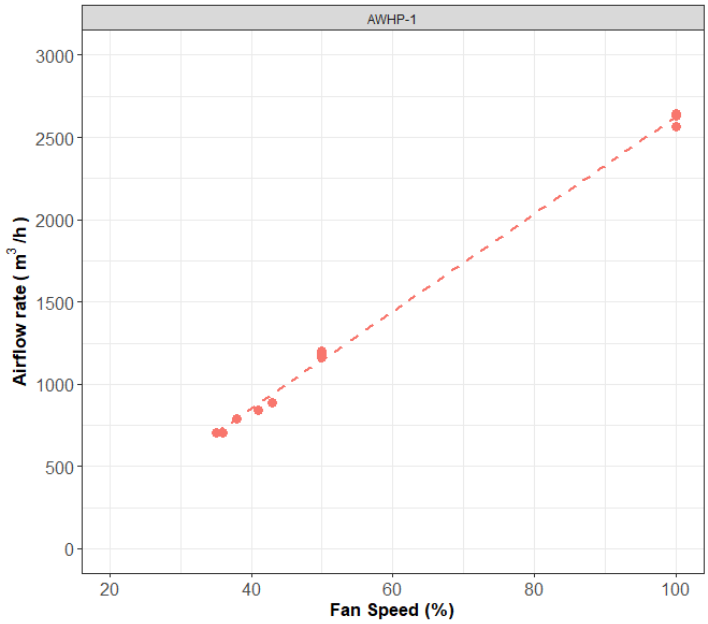

Figure 30: Empirical field measurement of outside unit evaporator fan in-situ for AWHP-1