Variable Speed Heat Pump Compressor for Demand Side Management and Network Stability

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I confirm that the word count of this thesis is less than 100,000 words

Abstract

Decarbonization of the UK residential heating sector is crucial to meet the Committee on Climate Change recommendations, 2019 of net zero greenhouse gases emissions by 2050. The current progress with residential building sector carbon neutrality is slow and hence acceleration in action is required. The electric vapour compression-based heat pump (HP) system is a mature technology to meet the residential heat load demands but needs performance improvement with better control for end users satisfactions and meeting grid demands. The HP co-efficient of performance (COP) could be improved with variable speed compressor-based system for capacity control due to match between heat supplied and demand by installing additional control devices as a demand side management tool. This also assist in peak clipping and reduces overall energy demand for load shaping. Capacity control with variable speed mode (VSM) performs better than the capacity control with fixed speed mode (FSM) specifically under varying heat load conditions. Hence, this becomes more beneficial in domestic retrofit due to poor insulation characteristics of old housing stock. However little information is available in the literature regarding the retrofit applications of the developed variable speed compressor HP system specifically in UK context.

The contribution to the knowledge from this research work was a) the successful development of the novel variable speed compressor based domestic Air Source Heat Pump (ASHP) system for seasonal performance improvement, energy savings, that allows flexible network operations due to match between heat supply/and demand by installing additional control devices/heating distribution system, b) Evaluation of the HP that is able to capitalize upon the range of heating capacities that can be generated across the full range of speeds associated with its compressor's variable speed drive- can respond to heating supply temperature needs as well as any constraints placed on its operations by electricity costs of electrical network capacity, c) The HP retrofit assessment via annual running cost, and carbon emissions savings in comparison to oil/gas boilers, electric heater and very high heat supply temperature ASHP for UK housing stock using Archetype approach. The HP in VSM annual performance improves by 27% at 35 ^{o}C heat supply temperature in comparison to FSM. Furthermore, the HP(VSM) annual performance degrades by 51% by increasing the supply temperature from 35 ^{o}C to 55 ^{o}C. The HP annual running cost was higher compared to the advanced GB (90% efficiency) but proved to be advantageous in terms of carbon emissions.

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Publication's List

Some of the main results from this research work have been published in peer-reviewed conferences and Journal with the following details;

CONFERENCE PAPERS

Muhammad Abid, Neil Hewitt, Ming Jun Huang, Christopher Wilson, Donal Cotter. Experimental Study of the Heat Pump with Variable Speed Compressor for Domestic Heat Load Applications, "25th Compressor Engineering, Refrigeration, Air-Conditioning, and High-Performance Buildings" 24-28 May 2021 at Perdue, USA.

https://docs.lib.purdue.edu/cgi/viewcontent.cgi?article=3677&context=icec

Muhammad Abid, Neil Hewitt, Ming Jun Huang, Christopher Wilson Donal Cotter. Heat supply temperature impact on the seasonal cost of low carbon domestic heat pump technology. International Conference on Applied Energy (ICAE); 2020.

https://www.enerarxiv.org/page/thesis.html?id=2685

Muhammad Abid, Neil Hewitt, Ming Jun Huang. Variable Speed Heat Pump Compressor for Demand Side Management and Network Stability. Energy Technology Partnership (ETP) Sustainable Energy 9th Annual Conference 2020. https://www.etp-scotland.ac.uk/NewsandEvents/Events/ETP9thAnnualConference2020.aspx

JOURNAL PAPERS

Muhammad Abid, Neil Hewitt, Ming Jun Huang, Christopher Wilson Donal Cotter. Performance analysis of the developed air source heat pump system at low to medium and high supply temperature for Irish housing stock heat load applications. Article No. 1392258 *Sustainability* **2021**, *13*(21), 11753. https://doi.org/10.3390/su132111753

Muhammad Abid, Neil Hewitt, Ming Jun Huang, Christopher Wilson Donal Cotter. Domestic retrofit assessment of the heat pump system considering the impact of heat supply temperature -a case study. Article No. 1397418. Sustainability **2021**, *13*(19), 10857; https://doi.org/10.3390/su131910857 .

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Nomenclature/abbreviations	
ASHP	air-source heat pump
C1	case study 1 (35 ^{o}C fixed heat supply temperature)
C2	case study 2 (45 ^{o}C fixed heat supply temperature)
C3	case study 3 (55 ^{o}C fixed heat supply temperature)
CF	corrections factor (-)
CO_2	carbon di-oxide
COP	co-efficient of performance (-)
DB	dry bulb temperature (^{o} C)
E	electrical energy consumption (KWh)
EH	electric heater requirements (KWh)
EWT	entering water temperature (^{o}C)
EIR	electric input ratio (-)
FSM	fixed speed mode (-)
GB	gas boiler
GHGs	greenhouse gas emissions
HDDs	heating degree days
HD	heat demand(KW)

HC	heating capacity (KW)
mtoe	mega tonnes of oil equivalent
OB	oil boilers
P	electric power (KW)
Pr	pressure ratio (-)
Q	heat pump useful heat output (KWh)
PLF	part load factor (-)/ ratio of part load to full load efficiency
PLR	part load ratio (-)/ratio of heating capacity to maximum HP capacity
PID	proportional Integral Derivative
RH	relative humidity (-)
RPM	revolution per minute (-)
SCOP	seasonal co-efficient of performance
Ta	ambient temperature (°C)
VSM	variable speed mode (-)
VHTHP	very high temperature heat pump
V_{meas}	measured value (-)
V_{pred}	predicated value (-)
WB	wet bulb temperature (° C)
WST	water supply temperature-heat supply temperature (°C)
2A35W	ambient temperature of 2 °C and heat supply temperature of 35 °C
Symbols	**Use/units**
%	percentage (-)
φ	relative humidity (-)
ε_{is}	isentropic efficiency (%)
ε_{vol}	volumetric efficiency (%)
P_{total}	total electric power consumption(kW)
p_{comp}	electric power consumption(kW)
p_{aux}	auxiliaries power consumption(kW)
£	pound
ω	frequency (Hz)
ρ	density(kg/m^3)
f_{COP}	COP correction factor (-)
C_c	degradation co-efficient
C_p	specific heat capacity (kJ/kg. K)
$\dot{m}_{ref}$	refrigerant mass flow rate (kg/s)
$\dot{m}_w$	water mass flow rates(kg/s)
ΔT	delta T (difference between heat supplied and returning temperature to the HP)
Δh	enthalpy difference(kJ/kg)
Subscripts	
a	air
hp	heat pump
full	full load
i	bin number
w	water

Chapter 1: Introduction

1.1 Domestic heating sector decarbonization in UK and air to water heat pump (HP)

Building's sector is one of the major contributors to the global emissions and was responsible for 38% of greenhouse gas (GHG) emissions in 2019 [1]. In UK, the residential sector was responsible for the second largest energy consumption, with 29% of the total final energy consumption and accounted for 17% of the total carbon dioxide emissions [2]. The energy consumption(mtoe) by sectors is depicted in Figure 1.1. Overall energy consumptions got reduced in UK since 2000 by 11% due to the utilization of most efficient devices, enhanced housing insulation, average ambient temperature rise, and deindustrialization. All sectors except transport have shown fall in energy consumptions with drop in Industry and domestic sector energy consumption by 39% and 14% respectively during the period (2000 -2019). The domestic energy consumption is more responsive to ambient temperature variations and number of heating degree days, as the great proportion of the consumption goes to space heating. Rise in average temperature due to greenhouse gas emission causes lower heat demand effecting the consumption behaviour. The recent report by International Energy Agency (IEA) recommend that the sale of new fossil fuel boilers should stop from 2025 to achieve net-zero emissions target by 2050 [3]. On the way to net zero emissions, multiple milestones are there and lags in any sector may results in failures to meet the target elsewhere. The energy and climate policy strengthening, and implementation is required by all government to achieve the global pathway to net-zero emissions. The UK as a signatory of the climate change act, 2008 legislated and set the target of net-zero carbon emission by 2050 [4, 5], has banned sale of fossil fuel-based boilers by 2025 [6]. The current measures are insufficient towards the committed pathways and hence acceleration in actions is required. In this regard the climate change conference of the parties, COP26 organized in Glasgow, urged signatory countries to speed up the action required to achieve the goals of Paris climate change agreement and there is no time for delay and no room for excuses [7,8]. Among the UK residential sector, space heating (SH) & domestic hot water (DHW) demand consumes 79% while cooking, lighting and other appliances consume the rest of the energy (Figure 1.2) [9]. This encourages renewable energy-based alternatives including greening gas or heat pump (HP) technology instead of fossil fuel-based heating options to decarbonize the domestic heat load demands. Domestic heating decarbonizations via the greening gas including biomethane or hydrogen appears attractive but seems to be more costly with missing convincing results [10]. The HP technology become attractive for the residential sector heat decarbonizations.

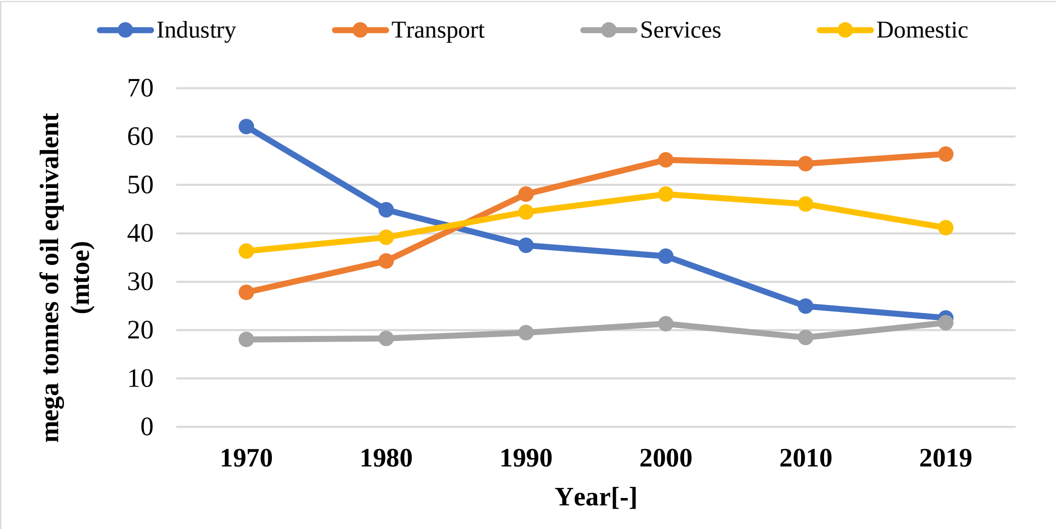


Figure 1.1UK energy consumption by sector during the period of 1970 to 2019 [2]

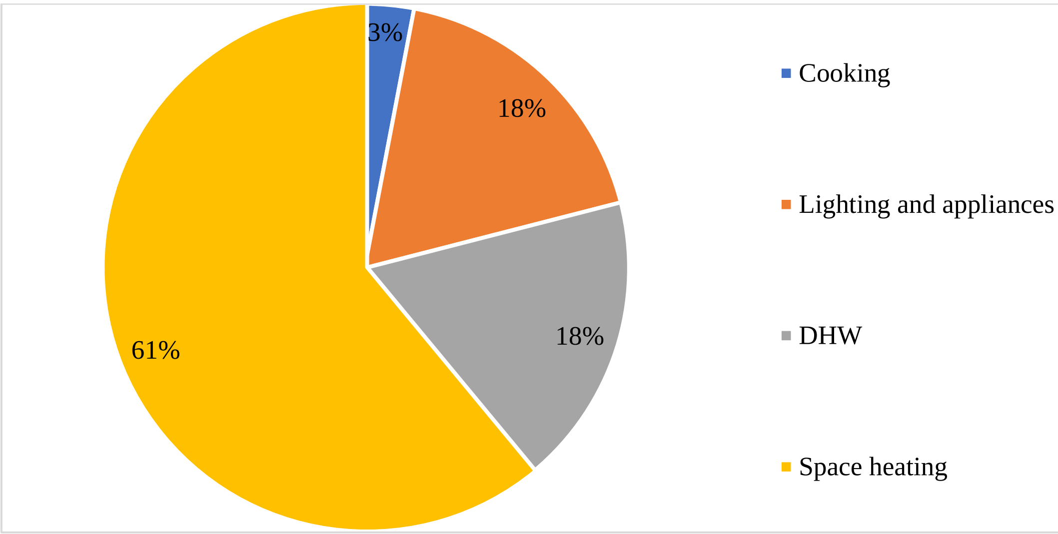


Figure 1.2 SH consume major part of energy in residential buildings [9]

One of the key milestones settled for the building sector is retrofitting measures and 50% of the heating demands supposed to be met by HP by 2045[3]. The electrification of the domestic heating sector via the HP technology become more feasible due to reduction in carbon footprints because of increasing renewable energy integration into the network grids [10, 11]. The carbon emissions factor reduced for UK grid electricity during the last two years period (2018 -2020) was 23% [12]. The ASHP appears more attractive for retrofit solution compared to the ground source heat pump (GSHP) technology due to the lower initial capital cost, labour work and space required, and less house disruption inside the existing housing stock [13]. Vapor compression-based heat pump technology has shown a great potential improvement with recent advances inside cold climatic conditions [14]. Recent review study on ASHP in the context of UK found it a promising technology in decarbonization of domestic heating sector[15]. However, the poor control restricted the technology uptake at large scale in UK compared to other European countries [16]. The varying load conditions, climatic conditions, lack of proper control, proper sizing, existing housing stock poor insulation, initial capital cost, and lack of incentives by the government were some of the other major challenges and barriers in this regard.

1.2 ASHP systems market demand as a domestic retrofit technology

The large number of old housing stock, poor insulation and the lack of incentives were some of the challenges for the HP installation inside UK [16]. The post 1980 domestic building represent around quarter in England, Scotland, Wales and 40% in Northern Ireland, while the remaining housing stock are old aged [17]. Limited number of houses with only about a quarter of houses were built in the last 40 years-except Northern Ireland (Figure 1.3). The percentgae of houses built in last 40 years was 40% in Northern Ireland. The HP retrofitting inside the very old period housing stock could not be neglected as the number of hard to heat homes in Northern Ireland represent least efficient of 14% [17, 18] and 19% in England [19]. A great portion of housing stock are with moderate level efficiency ratings acoss the UK with very small percentage of most efficeint buildings (Figure 1.4). The major portion of the housing stock are with moderate efficiency rating and retrofit measures are highly recommended to save energy and carbon emissions. The HP replacement instead of gas boilers requires the performance assessment and improvement potential investigation to compete with falling gas prices in UK. The existing running cost analysis allows the HP technology to compete in off-gas grid areas only [20,21]. To facilitate the HP system installations attractive at large scale inside the residential buildings its performance and control needs to be improved to compete with modern condensing gas boilers in UK. The variable speed compressor-based system was found a potential candidate, by producing range of heating capacity to match the load demand could improve the performance resulting in less carbon emissions with reduced annual running cost [22]. However limited information was available for the unit's performance inside domestic retrofit.

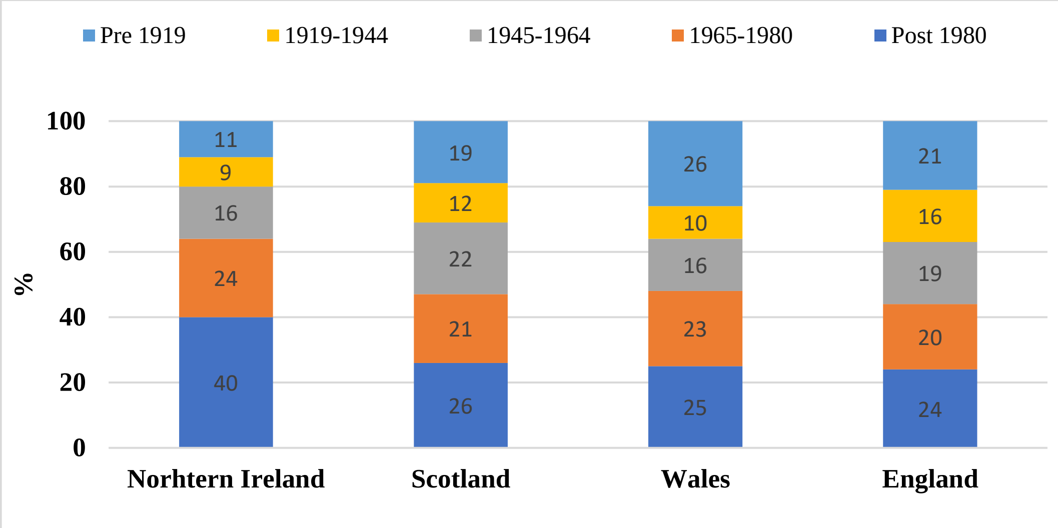


Figure 1.3 UK currently existing houses stock with construction dates [17]

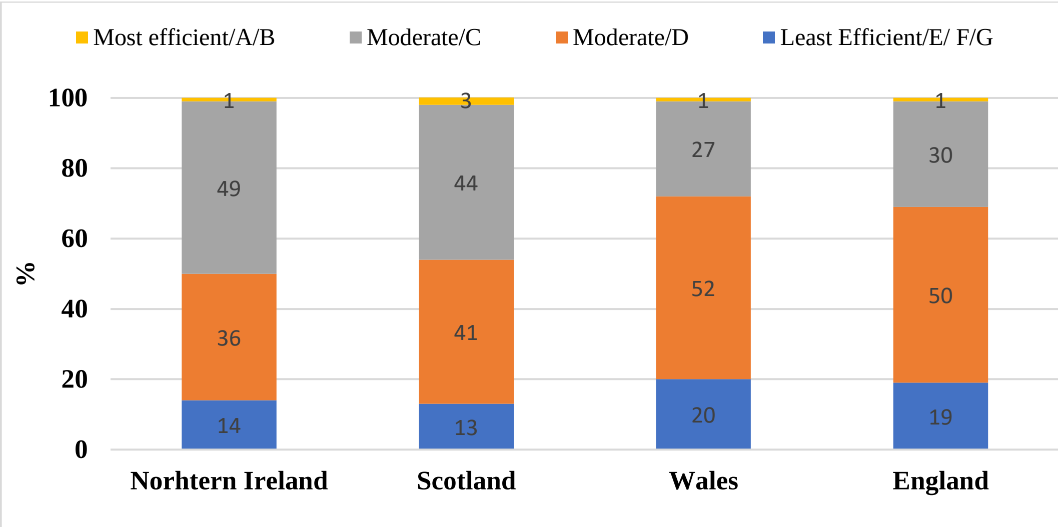


Figure 1.4 UK housing stock with efficiency ratings [17]

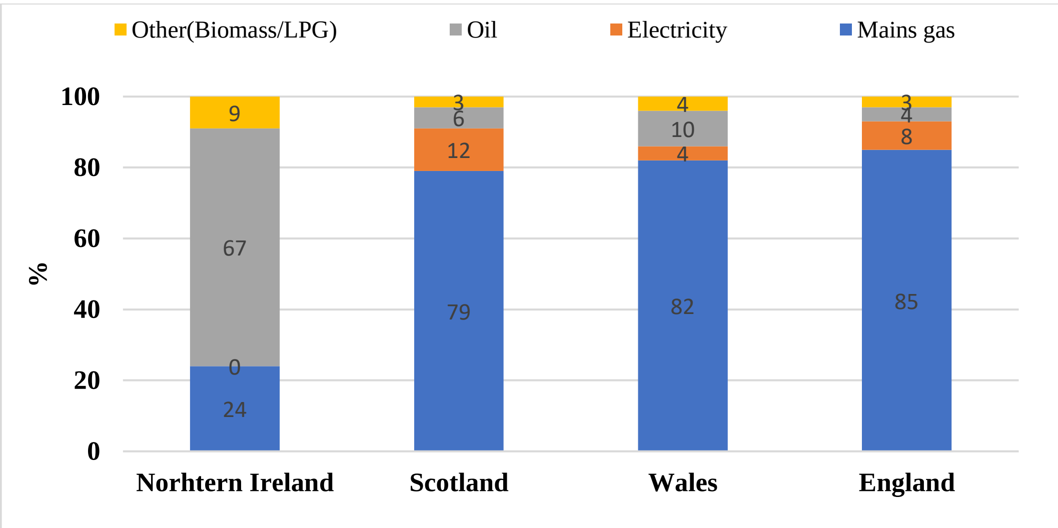


Figure 1.5 Northern Ireland heating system mainly rely on oil-based boilers [17]

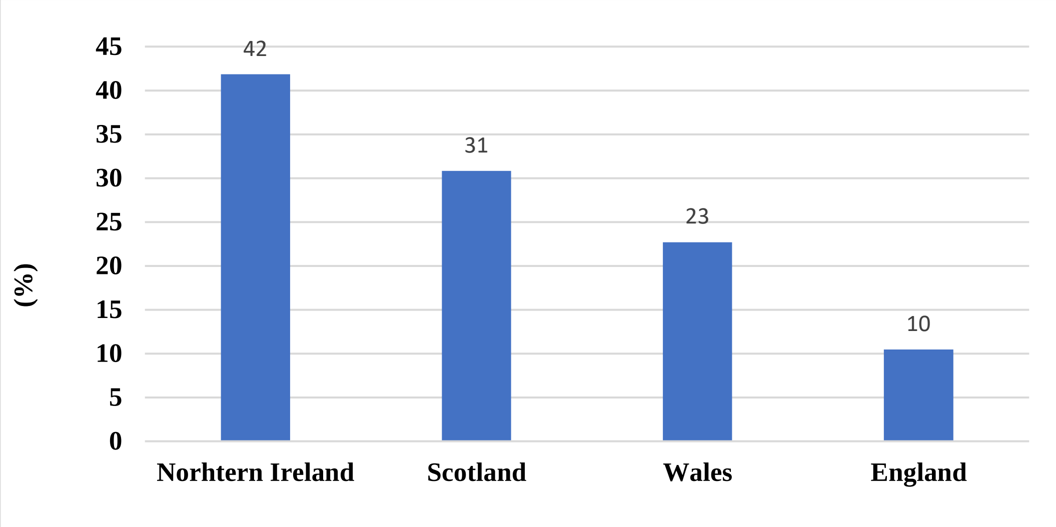


Figure 1.6 Percentage of UK houses in fuel poverty with NI at the top [17]

The major portion of domestic heating sector representing 67% of the total 80,000 dwellings were relying on oil boilers in Northern Ireland (NI) mainly because of off-gas grid areas as can be seen from Figure 1.5[17]. However, in England approximately 85% of the total 27.4 million dwellings utilizes gas boilers for heating purposes due to the network availability [23]. Northern Ireland leading in fuel poverty for households with percentage of 42% (Figure 1.6). Spending 10% and above of the total household income for maintaining the adequate temperature conditions inside the house is a fuel poor. The poor insulation in old housing stock was one of the reasons for extra spending on heating homes [17]. Thus reducing the heating cost for the households could reduce the fuel poverty.

1.3 Heat pump capacity control: variable speed compressor-based technology

The seasonal COP, defined as the ratio between the useful energy heat output to the electrical energy consumed during a specific time period could be improved with proper HP size and control approach selection according to the property type and age period. The HP technology utilizes environmental thermal heat energy from the air/ground, with some proportion of electrical energy mainly goes to compressor operations. The air source heat pump (ASHP) has lower initial capital cost and less disruption requirements with installations inside the existing buildings, compared to the ground source heat pump (GSHP) system, and hence more favourable for domestic retrofit and have great opportunity for market penetrations in the context of UK [15]. The high starting current, temperature control due to its intermittent operation was among the barriers in the promotion of the technology uptake for large scale installations in UK [24]. The restart of the unit with intermittent operation causes high current consumption because of compressor pressures re-establishment. The ON/OFF controlled systems synchronized operation at large scale can cause network failure. The risk of network instability increases due to additional current due to load demand with back-up electric heater requirements at peak times [25]. The issue with ASHP system is its performance degradation at lower ambient temperature conditions and more compressor work is required during peak heat load demand, causing extra pressure on networks. The smooth operations are essential due to the existing network's capabilities in UK. The HP selection to meet the house peak load demand requires larger components size, with increased cost, and negative impacts the overall system performance due to higher cycling losses during off-peak period. The peak load demand occurs only for the limited period of the year and the idle operation of the system for the remaining period causes losses, fluctuations, and network destabilizations. In seeking a system which can modulate its capacity according to the load requirements and operate over a wide range, a variable speed compressor-based HP system is a viable option. The control of the system at part loads conditions basically differentiates variable speed drive from the conventual fixed speed system [26]. The COP is improved using variable speed compressor due to heat supply match to that of demand at part loads, resulting in to lower cycling losses, back-up electric heater requirements. This also causes higher thermal comfort and better compressor life due to less deteriorating effect [27]. The building heat load demand and the system operation over the balance point for a great portion of building load is possible with variable speed control [28]. The variable speed compressor-based HP system can meet heating loads demand at lowest ambient temperature conditions by the speed modulation and avoids oversizing. However, there are certain challenges with the variable speed compressor-based system in addition to the benefits associated for demand side management. The variable speed capacity control savings depends on and varies according to the heat supply temperature in contrast to fixed speed mode of operations. The current standard BS EN14511 classify the HP system depending on the heat supply temperature as low (35 ^oC), medium (45 ^oC), high (55 ^oC), and very high temperature (above 65 ^oC) (BS EN14511 [29]. The heat supply temperature impacts the COP, and compressor efficiencies. The COP for ASHP ranges between 3.2 and 4.5, according to lab scale tests as per BS EN14511 at lower heat supply temperature of 35 ^oC [30]. The variable speed compressor operation has certain drawbacks with impact on compressor efficiencies and inverter losses. Therefore, potential performance improvement with the control mode of operation at low to medium and high heat supply temperature for the UK domestic retrofit in different age period with thermal characteristics and building types needs thorough investigations. At the same time the challenges and drawbacks need to be considered with the compressor while operating the system in variable speed mode as a part of demand side management programmes and large-scale deployment.

1.4 HP domestic integrations and control method

The heat supplied to the buildings needs to be matched to that of demand to enhance the thermal comfort. The varying load demand according to the ambient conditions poses challenges for the heat pump system control. The traditional approach used is open-loop curve control method commonly used for heat pump capacity control, where heat transfer medium supply temperature is based on outdoor air temperature, as can be seen from Figure 1.7. The curve number is chosen to get control system meet the heat load demand for the actual building by increasing or decreasing the supply temperature to control the heating capacity. Room temperature sensor is used as an additional complement to control the heat supplied and maintain the required temperature inside the built environment. The deviation of the room temperature from the set temperature is then utilized by controller to change curve. The internal heat gains compensation is also possible and considered by using a room temperature sensor. The heat pump is turned ON/ OFF based on the dead band difference, given by the heating curve. The value for dead band is usually kept smaller for heating system with higher thermal inertia i.e., underfloor heating with low heat supply temperature requirements and higher dead band value with low thermal inertia heating system i.e., radiator and fan coils heating distribution system at high heat supply temperature of 55 ^{o}C. The cyclic characteristics and the response time with the heating distribution system needs careful attention and consideration to end user's satisfaction. These challenges need to be addressed to make the technology acceptable and overcome the social barriers. Thermal inertia influences the cyclic properties of HP for space heating [31]. Variable speed mode (VSM) has less effect due to load matching and temperature of water inside the system is maintained at set-point. Response time at start up is also affected by thermal inertia of heat distribution type with the fastest response by radiators heating distribution system and slowest by underfloor heating options [31]. The building heat demand inverse relation with ambient temperature, mismatch between heat load demand and HP heating capacity was one of key challenge with ASHP [26]. The relationship with changing ambient temperature in case of fixed speed mode (FSM) of capacity control is illustrated in Figure 1.8 [26].

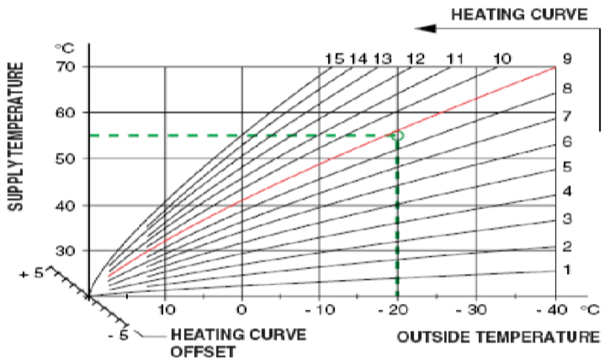


Figure 1.7 HP control heating curve to meet heat load demand [31]

The cycling losses increases with the increase of ambient temperature above the design point due to higher heat production than the load demand. The cycling losses could be reduced by system design at upper minimum expected conditions but during extreme conditions at minimum lower expected conditions-the system will be undersized and thermal comfort will be compromised leading towards the electric heater requirements. Traditional thermostat control approach regulates the supply temperature but the space temperature swings around a set point and thermal comfort is compromised, while in case of varying speed compressor the speed is continuously managed and set point temperature is maintained. The higher speed operation consumes more electricity but with improved thermal comfort to end user due to match between the load demand and heat supplied quickly at the start and once the set temperature of room is achieved the compressor speed is reduced.

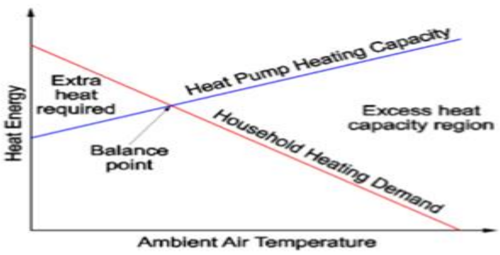


Figure 1.8 HP Heating capacity vs building load demand with fixed speed mode (FSM) [26]

In contrast VSM could operate for wide ambient temperature possible range at balance point with the heat demand from house matches to that heat supplied. The balance point is the design point in conventional system based on the minimum expected environmental temperature result in oversizing, and on/off cycles. While in case of variable speed compressor-based system the number of match points could be increased for a larger range of heating demand over the experienced ambient temperature conditions (Figure 1.9).

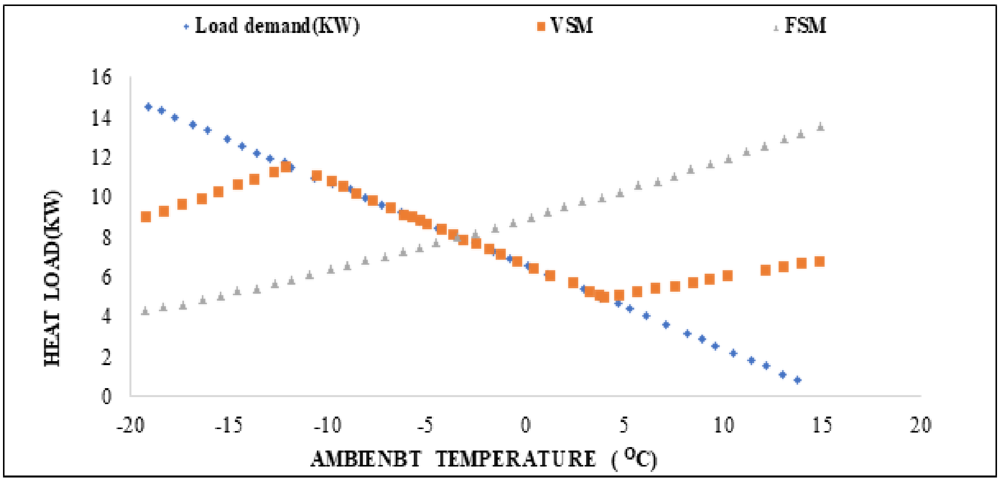


Figure 1.9 HP Heating capacity vs the building load demand variable speed mode (VSM) [28]

During variable speed mode of control (VSM) the refrigerant flow is controlled with reduced on/off cycles requirements, and improved compressor life. To achieve required pressure again and again with cycles needs higher start-up current to the motor and causes the compressor damage.

1.5 Variable speed heat pump compressor for demand-side management

Demand side management (DSM) programmes involves a) peak clipping by utilization of improved device performances or additional control devices, b) valley fillings, load shifting from peak period to off peak period by using energy storage, or simply all activities by utility aiming at load shaping. DSM with regards to heat pump weighted towards grid benefit is generally aimed to allow stable cost-efficient operating of the network. The role of the heat pump in smart grids were investigated in recent review study and highlighted the importance of the variable speed compressor-based HP system in smart grids for electrifications of heating sector [32]. The study recommends investigations of the internal happenings and challenges with the compressor for its domestic heat load application while utilizing this as a part of demand side management programme [32]. The performance and control with variable speed compressor-based system has value in this context and could provide services for the stable and economic operations of the heat pump system and has been thoroughly investigated in the literature but at the same time this has an impact on the compressor. The potential benefits towards the grid stability with the variable speed mode have been investigated in literature neglecting the issues and challenges. Providing the flexibility to power system and implementation of DSM strategy will affect the HP design, control, operation, and integration method at different level (Figure 1.10) [32]. The HP control hierarchy for the stable and economic operations of the network has three major boundaries levels; a) power system including grid signals, control and aggregations, b) building consisting of room thermostat control, HP control, etc c) HP unit level involves vapor compression cycle, compressor type and control, and other auxiliary's control. The three levels are interacted in a way where the power system levels send signals to the lower level (building/HP unit control) and provides feedback to adjust the systems operations according to the network demands.

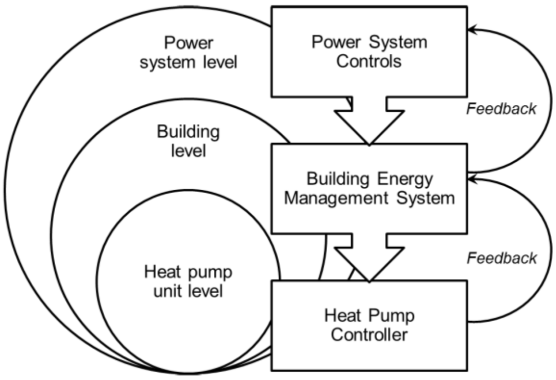


Figure 1.10 HP Integration level and control hierarchy [32]

The optimum design of HP unit, heating distribution system, building type/load, control approaches for an individual application needs to be developed to achieve stability to the power system. Demand side management (DSM) includes all activities, effecting end user's way of energy consumption for stable and economic operation of the power system, from the generation to transmissions. DSM programs includes installation of efficient devices on consumer side, additional devices, control systems to turn on/off the devices as needed, strengthening communication between network operators and consumers [33]. The electrification of heating sector and energy savings with variable speed compressor-based heat pump system could avoid network instability. A significant amount of energy could be saved with the efficient use of technologies but needs proper assessment. The HP could provide services to the power system at different levels of control. The load shaping could result in achieving multiple objectives of cost savings, reliability improvement, avoid the constructions of extra power plants. The DSM analysis and assessment could be conducted at different details level and time period. For example, the energy savings estimate on annual, seasonal, or monthly, daily, or hourly basis [33]. In the current research work the performance assessment was conducted for energy savings with the combination of these time periods and applications for various property types with the utilization of variable speed heat pump compressor for capacity control. The possible improvement potential assessment with VSM operation instead of FSM as a demand side management tool and the challenges/issues with the compressor efficiencies needs further investigations. The electrification of domestic heating sector via the heat pump with the current network capability increases the importance of energy efficient devices as a DSM tool.

1.6 Aim and objectives of the research work

Changing the energy systems towards 100% renewable energy sources is one of the main challenges of the century and hence the overall aim of the research work was the performance assessment of the UK domestic sector decarbonization and electrifications via experimental development of variable speed air source heat pump system and the role of HP in domestic heat sector electrifications. The following were the objectives of the research:

- ❖ to investigate the variable speed heat pump compressor for demand side management and network stability with challenges and potential benefits while using the system for range of domestic heat load application.
- ❖ and performance evaluation of the developed system under different conditions for domestic retrofit inside the UK housing stock and carbon emission savings in comparison to fossil fuel-based heating technologies.
- ❖ with COP improvement, energy savings potential for domestic retrofit with variable speed compressor technology. The performance investigations with challenges to the variable speed compressor-based HP system for capacity control and the impact of the parameters i.e. heat supply temperature, ambient temperature conditions at constant heat loads via experimental development and testing.
- ❖ the importance of heat pump sizing, match between heat load demand and supply on the performance improvement, energy, cost, carbon emissions savings.
- ❖ comparative performance improvement with variable speed mode (VSM) of control to that of fixed speed mode (FSM) operations in different climatic conditions and different property types at three level of heat supply temperature mentioned in this Thesis as as C1, C2, and C3.

The research hypothesis was the variable speed compressor-based heat pump system has the potential to decarbonize the UK domestic heat sector via electrifications by 2050 and that the variable speed compressor HP technology is valuable and more efficient in terms of energy & carbon emission savings, with improved performance for the domestic retrofit.

1.7 Layout of Thesis

The rest of the thesis have been divided into six chapters and can be summarised as follows: **Chapter 2** describes the literature review, research challenges/issues/gaps/motivations with the existing knowledge of the topic. **Chapter 3** explains the methodology used for problem solving and the reason for choosing this specific method to achieve the objectives. The heat pump development, testing, data collection approach, analysis, domestic heat loads, and retrofit for the Irish housing stock were briefly explained. **Chapter 4** presents the heat pump performance assessment with challenges for variable speed compressor-based system. The testing regime under the steady state laboratory conditions have been discussed. The research carried out on the testing results and its impact on the system performance, compressor efficiencies, inverter losses under the variety of ambient temperature and heat supply temperature conditions. The chapter summarizes the challenges with the compressor while using this for demand side management and network stability purposes. **Chapter 5** discusses the comparison between the FSM and VSM for the evaluation of the annual co-efficient of performance (COP) in UK housing stock at low to medium and high supply temperature. The numerical model developed and the importance of the proper design/selection/operating over balance point for different loads demand according to the building type were presented. The heating capacity over production and under production and the impact of the cycling losses on the annual COP are discussed. **Chapter 6** deals with the heat pump retrofit assessment inside the UK housing stock instead of the fossil fuel-based heating technologies/and/or high supply temperature ASHP system and the savings with the annual running cost and carbon emissions. **Chapter 7** presents the conclusions, answers to the research questions, limitations and future work recommendations.

1.8 Brief about SPIRE 2 Project

The project objective is to evaluate, develop and facilitation of mass energy storage deployment at large scale for operation in profitably in new market structures of UK and Ireland. This project is supported by the European Union's INTERREG VA Programme, managed by the Special EU Programmes Body (SEUPB).

Chapter 2: Literature review

2.1 Introduction

The literature review chapter on the variable speed heat pump compressor-based heat pump system is divided into three subsections leading towards research gaps/motivations. The literature review has been extended in the individual study chapters with further specific results and comparison. The first two sections of the literature review chapter deal with the retrofit assessment of the low to medium high and very high heat supply temperature. The earlier research work carried out in this context have been summarized. The other sections of the literature review cover the capacity control method for heat pump aiming at domestic heat load applications and the earlier work on the challenge posed by the variable speed compressor for domestic heat load applications was summarized. The research questions /gaps/motivations based on the current knowledge were presented in the last section 2.6.

2.2 Low to medium and high heat supply temperature HP retrofit studies

The standard BS EN14511 classify the HP system depending on the heat supply temperature, as low temperature (35 ^{o}C), medium temperature (45 ^{o}C), high temperature (55 ^{o}C), and very high supply temperature heat pump (above 65 ^{o}C) (BS EN14511, 2004) [29]. For convenience the HP studies as a domestic retrofit technology could mainly be divided into two categories in this study based on the water supply temperature (WST), i.e., a) low to medium (35-45 ^{o}C) & high supply temperature (55^{o}C), and b) very high supply temperature (60 ^{o}C and above). The seasonal co-efficient of performance (COP) was commonly used as the performance evaluations indicator for comparison purposes according to the locations. The analysis and calculations have been performed using different approaches in the literature at single level of heat supply temperature neglecting the heat supply temperature impact on its COP value. In this regard, Kinab et al. [34] formulated a detailed model based on lab-based testing results for variable speed ASHP seasonal performance optimization. The seasonal performance for Nice, Nancy, Macon and Trappers were of the order of 3.27, 2.76, 2.93, and 2.93 respectively at heat supply temperature of 45 ^{o}C. A simple but novel numerical model for the seasonal performance for Bologna city of Italy for three different commercially available ASHP types of system (i.e., mono/multi, and variable speed compressor) was developed [35]. The approach for the seasonal performance evaluation with different heat load demands buildings was applied to eleven different types [35]. The load demands in each bin and the corresponding impact of operating the system over balance point was communicated but without looking into the compressor challenges, heat supply temperature impact and retrofit assessment. The seasonal performance value of 3.8 was achieved at heat supply temperature of 35 ^{o}C in variable speed mode (VSM) of control. The frost challenge for the evaluation of the seasonal performance with ASHP system using the bin approach and its impact on the seasonal performance evaluations was studied in more details by the same authors in another work [36]. Other seasonal performance evaluation approaches including parametric model was developed by Underwood et al. [37] with the possibility of validation with lab scale/field trials, and manufacture results for the seasonal performance evaluation. The part load operation considering the HP part load factor was not modelled and were recommended for further investigations. The seasonal performance of HP and maximization of benefit with the substitution of gas boilers for high heat supply temperature level were investigated via modelling [38]. The gas boilers were found beneficial at high heat supply temperature. The existing installed heating distribution system inside the buildings could be one of the barriers in promotion of the heat pump technology [16]. The high heat supply temperature heat pump system could overcome the challenge but at the cost of lower COP values. The cost and benefits analysis needs to be used while choosing the heat pump type based on heat supply temperature. Around 27.5 million England residential buildings are old aged, installed with wet radiators [39]. The wet radiators are installed to work with gas/oil boiler efficiently [40]at very high heat supply temperature of 75 °C, as per BS-EN 442-2:201 standard 4 recommendations [41]. In contrast the HP system works efficiently at low to medium and high heat supply temperature. The replacement of heating distribution system inside the existing house will be a major disruption. However, the pros and cons including the efficiency improvement and carbon emissions savings comparison is important to provide solid information to the policy makers and homeowners/tenants prior to the installations of such systems and the replacement of the existing heating distribution system. The literature review in this category for domestic retrofit applications were mainly based on the predictions/and field trials of the commercially available units, manufacturer information was used and without controlling/considering the heat supply temperature impact, and the compressor speed. The nominal value was assumed for heat supply temperature and delta T values [42,43], and without considering the building insulations, climatic conditions simultaneous impact on the system performance and carbon emission savings. HP with variable speed compressor retrofit investigations with experimental development were recommended for further studies having lab tests results instead on relying on manufacturer data for heat pump characterizations [10]. Also, the field trials results could not provide the controlled delta T value and assumptions were required for seasonal performance evaluation at different part loads [10]. The climatic conditions impact considerations along with building insulation level and the extent to which it effects the environmental and economic feasibility for other types of HP (water to air, and air to air) was assessed [44]. The ASHP type needs extra attention due to inverse relationship of building heat load demand and heat production in winter season [45]. Therefore, the complete information become more critical and assessment studies are required prior to installation of such system in a specific building type, and locations. Earlier studies for different nationalities/locations were available in literature. For example, the seasonal performance factor (SPF) for retrofit applications of different building types in Italian climatic conditions were reported with the potential improvement of 19% via weather compensation strategy [42]. Nominal value of heat supply temperature at 45 ^oC and the radiators oversizing were assumed without looking into heating distribution system installations/upgradations requirements, heat supply temperature impact on the carbon emissions, cost. The feasibility of ASHP retrofit installation for the eligible Canadian housing stock for the combined SH & DHW demand was investigated in the research [46]. The final heat supply temperature of 55 ^oC were obtained in the second stage via auxiliary boilers and 50 ^oC was achieved by the HP in first stage. The HP retrofitting could save energy of 36%, with up to 23% reduction in greenhouse gas emissions in comparison to fossil fuel-based heating technologies. However, the study reported mainly the high-level impact on the carbon emission without looking into the technological depth and improvement. In Germany, the experimental data from field trials results of 21 ASHP system and 22 GSHP was assessed for domestic retrofit applications by Huchtemann [47]. The HP heat supplying temperature of 40 ^oC were utilized with underfloor heating distribution system and 55 ^oC when radiators were used with intermittent operations only (ON/OFF controlled). The VSM of control and the potential associated benefits for load matching were not considered. The ASHP replacement instead of gas boiler assessing the carbon-savings, annual running cost in the existing office considering the insulation improvement characteristics by 2030 scenario in UK context was investigated by [48]. The ASHP was found potential decarbonization technology and amount of carbon emission reduction was based on the future DHW supply and the grid carbon intensity. The ASHP retrofit study for residential buildings in Scotland was conducted by Kelly and Cockroft for the evaluation of annual running cost and carbon emission at supply temperature of 55 ^{o}C [11]. The HP retrofitting instead of gas boilers reduces 12% carbon emissions, but with increase of 10% annual running cost. Cabrol and Rowley [49] compared the HP simulation results in domestic buildings in UK climatic conditions at single heat supply temperature of 35 ^{o}C with no comparative results for other supply temperature and with limited lab-based tests. The impact of the speed variation on the system performance was based on assumptions instead of real tests. The ASHP was found effective both in terms of carbon emission and cost savings due to low supply temperature in contrast to the gas boilers. The UK Department of Energy and Climate Change (DECC) with the Energy Saving Trust (EST) investigated field trials results for AWHPs & GSHPs, at heat supply temperature ranging between 30 °C and 55 °C [50]. The study concluded that the poor HP performance in UK compared to other European countries were due to lack of match between heat production and house load demand, large floor surface area, HP sizing issue, lack of proper capacity control, and ignoring the house insulations characteristics with different age period. The HP needs to be sized properly according to individual property type looking into the age period and annual heat demand characteristics. The conclusions drawn were similar to other researchers [51]. Hybrid heat pump-gas boiler system combination was investigated with the aim of reduction the unfavourable behaviour of the ASHP during the coldest period of the year to investigate the impact on annual energy saving [52]. The importance of proper sizing of the HP was highlighted, with the hybrid system tuned out to be more economical in comparison to monovalent HP. The ASHP as a domestic retrofit technology instead of gas boiler in south of Italy for energy consumption while considering the heating system installations cost in combination with electricity and/or gas bills [53]. However, the focus was the performance improvement with building insulations instead of HP.

2.3 Very high heat supply temperature HP retrofit studies

The second category of the literature review on heat pump domestic retrofit were the very high heat supply temperature pump systems. The economized vapor injection (EVI), cascaded unit, diesel engine-based heat pump systems were investigated in this regard with the 1900s Mid terraced test house in UK [43,54-55]. The technology was found expensive in terms of annual running cost but with savings of carbon emissions. The additional heating distribution system installations could be avoided with this type due to the efficient operation of the old wet radiators at high supply temperature. The full potential of the variable speed compressor-based system in domestic retrofit was recommended for further investigations at low to medium and high heat supply temperature. The technoeconomic analysis with the ASHP retrofit inside was conducted at supply temperature of 75°C to compare the annual running costs to that of gas boiler at different percentage efficiencies [43]. The annual running cost for the HP was found higher than the gas boiler with a cut on carbon emissions in the range of 14% to 57% depending on the boiler's operating efficiency. The domestic retrofit study with low to medium, high and very high supply temperature variable speed diesel engine heat pump system development was conducted in [20,21]. The potential benefits with high temperature engine driven HP in remote areas (off gas/ electricity networks), while meeting the 1900s Mid terraced test house heat demand were investigated but without considering the climatic conditions impact, age period and different property types [20,21].

2.4 HP capacity control approaches (VSM vs. FSM) for domestic heat load applications

The network stability, and large-scale deployment of HP technology for domestic heat load applications needs careful consideration. The literature of variable speed control was limited to the specific technical and/or economic benefits of variable speed control technology, and energy saving potentials and for other application areas. The high starting current associated with ON/OFF control resulting in extra pressure on the network was found one of the barriers for HP technology uptake inside UK, compared to other European countries [24]. The installations of ON/OFF controlled heat pump systems at large scale and the simultaneous operation can result in network failure. The risk of network instability due to current load further increases with the requirements of back-up electric heater requirements when the ASHP is unable to meet the required load at very low ambient temperature. The variable speed compressor of the HP for the capacity control have proved more efficient in this context [56-57]. Conventional way of controlling the heating capacity is the intermittent operation of compressor and comparative study between the variable speed and on/off control was conducted with seasonal efficiency improvement in the range of 1-25% and efficiency improvement of 10-30% range at a single operating condition [58-64]. Dynamic performance of variable speed compressor-based heat pump system for domestic heat loads applications and comparison with the conventional way of controlling was investigated in [64]. The reason for performance improvement was better part load efficiency, match between the supply and demand, smaller number of on/off cycles, unloading of heat exchangers, less requirements for back up electric heater, and smaller frosting losses [65]. Experimental and theoretical analysis including the on/off control, variable speed compressor control method was proposed to match the GSHP heating capacity with the building load with the variable speed operation as more suitable approach [66]. Munari et al. [67] performed comparative study of the two modes of control for the energy performance in terms of heat supply temperature requirements, with consideration of compressor efficiency, and climatic conditions impact studied by Adhikari, et al., 2012 [68]. The three climatic conditions of Italy (Milan, Rome, Palermo, and significant energy savings with variable speed control specifically at Palermo and Rome during the heating seasons because of part load operation of the system for most of the time were investigated in [68]. The backup electric heating requirements and its impact on overall system performance was investigated with both control mode [69]. The heat supply temperature with ON/OFF control mode during the on-cycle needs to be higher compared to the continuous system operation which leads to lower performance and higher energy consumptions [70]. Limited number of studies combining the comparative study of heat supply temperature (low to medium), and control mode (VSM vs. FSM) with GSHP aiming at the system efficiency improvement was reported in [31,71], but without considering the retrofit assessment. The impact of hydronic heating distribution system on the HP performance with different supply/return temperature without considering the economic, insulation, property type, and climatic conditions impact [31,71]. The reduction in water supply/return temperature values from 55/45 ^{o}C to 35/28 ^{o}C results in increasing seasonal performance in the range of 30-35%. A couple of experimental studies comparing the two control modes aiming at industrial applications was conducted by [72,73] aiming at the performance improvement potential and energy saving potential. The study of [72] investigated the capacity control with a fuzzy logic control algorithm instead of the commonly used thermostatic control. The ASHP performance combined with economic aspects of VSM in comparison to FSM for domestic heat load applications was investigated in [58]. The aim was to study the payback period with variable speed control mode, for detached type building in UK. Other studies combining the performance and cost analysis investigation for the variable speed compressor technology were conducted [25, 28], but for other locations. The total cost of ownership analysis approach was used to investigate the economics of both the on/off and variable capacity control schemes for an ASHP in different climate zones [28]. The variable speed capacity control method in contrast to ON/OFF control was found more economical for colder climatic conditions, and savings of up to €5000 was reported for 15 years period. The warmer climatic conditions for SH with variable speed control was not found cost effective. In [25], the objectives were to investigate annual COP of a variable capacity GSHP system with changing climatic conditions, and the conditions under which variable speed control perform better than on/off controlled system, the removal of auxiliary electrical heater. The HP operates continuously at partial loads to follow the building load demand during VSM of control was found more efficient, in contrast to conventional way of controlling the heating capacity with the load adjustment [26, 74-75]. Faster temperature control, low starting current, noise, vibrations, and continuous control with rare transient losses were well-established advantages with VSM against FSM [76-80]. Comparative studies for the two mode of HP capacity control were made at different level including the annual performance, and at individual steady state tested conditions with optimal speed of operation. The annual performance comparison was found more valuable than individual conditions comparative performance analysis [31]. Annual performance improvement in the range of 10-25% was reported depending on HP type (ground/air source) compressor used, operating speed range, and design load [31]. The reason for performance improvement was better part load condenser, and evaporator efficiency, smaller number of cycles, back up electric heater requirements, lower losses due to defrosting, supply temperature [81]. Limited number of articles were found investigating the performance improvement and energy savings with control mode and considering heat supply temperature simultaneously but with GSHP [78, 71]. The reduction in heat supply temperature from high (55^oC) to low (35 ^oC) result in improvement seasonal performance factor by 30-35% [31]. Three types hydronic heating distribution systems, i.e., underfloor, fan assisted hydronic coil unit, and traditional wet radiator were considered in analysis for comparative study between VSM & FSM. The negative impact due to cycling get reduced with VSM of control because of load matching with heat supplied [31]. The overshoot and undershoot of the room temperature become a great concern during intermittent operation specifically with underfloor heating system due to high thermal inertia and thermal lag. During overshoot when the heat pump producing extra heat than the demand at higher ambient temperature conditions with higher thermal inertia heating distribution system store the extra thermal energy at higher dead band set point temperature causing the comfort issues. The heat is continued to release from the underfloor thermal mass materials even after turning OFF the HP. In contrast during overshooting when the HP producing less heat than the required demand at low ambient temperature conditions the thermal energy absorbed by the underfloor heating distribution system with higher thermal inertia make the situations worst causing thermal comfort issues.

2.5 Variable speed compressor-based HP challenges

The experimental investigation with variable speed compressor operation at 35 Hz, 40 Hz, 50 Hz, 60 Hz and 75Hz considering challenges with the compressor isentropic, and volumetric efficiencies, inverter losses, pressure ratio were investigated in [82]. It was concluded that the compressor efficiencies were strongly dependent on the operating frequency and pressure ratio. The high and low speed operation causes electromechanical losses and lubrication issues [82]. The maximum isentropic efficiency was 65% when the pressure ratio was 2.2, while the volumetric efficiency showed linear decrease from 98% to 83% with the increase of pressure ratio from 1.5 to 5.6. The inverter efficiency was in the range of 95-98% according to the changing compressor power. The compressor power was in the range of 1.5 to 6.5 KW. The isentropic efficineyc was highest at 65% at nominal speed with pressure ratio of 2.2, while the volumetric efficiency linearly degraded from 98% to 83% with pressure ratio variation range between 1.5 to 5.6 due to varying speed (35-75 Hz) [82]. The inverter losses due to change in compressor supplied frequency was analysed experimentally for GSHP and get reduced as a percentage of total electric power consumption with the increase of speed, while keeping constant load/source side temperature of 26 ^oC /4.5 ^oC with a frequency variation range was in between 30-90Hz [83]. The testing analysis with low to medium and high heat supply temperature of 35 ^oC, 45 ^oC, 50 ^oC and ambient temperature conditions of -15 ^oC, -7 ^oC, -2 ^oC, 2 ^oC, 7 ^oC on variable speed compressor based ASHP was conducted aiming for Irish housing stock retrofit applications, but with uncontrolled heating capacity [22]. The COP value increased from 2.43 to 4.26 with the increase of ambient temperature from -15 ^oC and 7 ^oC at heat supply temperature of 35 ^oC but with varying heating capacity. At higher heat supply temperature of 50 ^oC the COP value increased from 1.73 to 3.15 with the increase of ambient temperature from -15 ^oC and 7 ^oC. However, the heating capacity were not constant, and ambient temperature was not the only reason for the COP improvement, but also depending on the operating frequency. ASHP system with a designed nominal heating capacity of 9.8 KW was developed and tested at very high heat supply temperature of 60 ^oC with a range of ambient temperature conditions under the steady state laboratory condition according to British European test standard BS EN14511 [54]. The objective was retrofit assessment of the HP system and decarbonization of Irish domestic heating sector. The efficiency improvement potential with the capacity control method by varying compressor speed was investigated. The range of the tested compressor speed was limited due to issues with the compressor failures and the lack of dedicated inverter. The frequency modulation range was in between 37.5 - 75 HZ and HP were tested at four fixed frequencies of 37.5 Hz, 50Hz, 60Hz, and 75 Hz. The heating capacity variation range was 3.14- 4.87KW, 3.4-5.38KW, 3.43-5.57 KW, 3.49-5.86 KW, 3.59-6.03 KW, 3.73-6.59 KW due to frequency modulation from 37.5-75 Hz for ambient temperature conditions of -15 °C, -7 °C, -2 °C, 2 °C, 7 °C, and 15 °C respectively [54]. The highest performance was recorded for all tested conditions at nominal frequency of 60Hz due to high compressor isentropic and volumetric efficiencies, lower discharge line temperature, pressure ratios and highest inverter efficiencies. The overall annual co-efficient of performance (COP) calculated for Belfast climatic conditions was 2.15. Other study focusing on the inverter losses due to varying compressor supplied frequency was also analysed experimentally for ground source heat pump (GSHP), keeping constant load/source side temperature of 26 °C /4.5 °C respectively [83]. The results show that overall, 30% reduction in COP values was noticed according to the operating frequency. The pressure ratio variation was in the range of 2.7 to 5.8 according to the compressor frequency variation in the range of 30-90Hz. The isentropic efficiency was highest when the compressor frequency was near to the nominal value of 50 Hz. The inverter losses as a percentage of total electric power consumption reduces with the increase of compressor speed at constant ambient temperature conditions and heat supply temperature. The compressor supplied frequency have strong impact on the isentropic and volumetric efficiencies at lower and higher speed operation of the HP compressor results, because of lubrication issues and electromechanical losses due to internal leakages. A couple of experimental studies conducted on the variable speed compressor with application for industrial plant cold storage was conducted to investigate the energy saving potential by operating the compressor at optimal speed [72, 73]. The control of cooling capacity with the variable speed using a fuzzy logic control algorithm instead of the commonly used thermostatic control method working at constant nominal frequency of the compressor and the performance improvement was evaluated [72]. An increase in energy saving of up to 20% was reported for various conditions while using the variable speed control in comparison to the thermostatic control. The study [73] for finding optimal frequency was conducted for maintaining the required cold storage temperature, by keeping evaporating, condensing temperature, and heat load constant and varying frequency for power saving by matching the load demand and comparing the results with that of nominal frequency of 50 Hz. It was recommended that compressor speed regulation to devise an optimal control strategy at each separate working condition and determination of optimal frequency is very important. It was found that at 30 Hz frequency could save input power of 15% compared to nominal frequency for maintaining a cold store at 0°C under constant conditions for reciprocating compressor and 25% in case of scroll compressor. Also, the HP system was evaluated for optimal frequency comparing with nominal frequency for power savings at supply temperature of 34 °C and 42 °C and approximately 30% of energy saving could be achieved at frequency of 24 Hz and 36 Hz satisfying the loads at set conditions, compared to that nominal frequency. Based on the above literature review the ASHP experimental development, simulating constant heat loads under the controlled laboratory conditions to look into the inside happening with the compressor speed variations, and the annual performance evaluation at low to medium and high heat supply temperature in both modes for capacity control was missing to the best of author,s knowledge specfically in the context of UK. Some of the results for this research was presented earlier in authors work [84-88]. The earlier comparitive studies with capacity control were performed either under single steady state conditions for optimal frequency, individual component analysis (i.e compressor isentropic and volumetric efficiencies, inverter losses), single heat supply temperature. Based on the literature review, research gaps and recommendations for further investigations could be highlighted as follows: i) strong recommendations for the retrofit assessment and investigations into the performance evaluations of the variable speed compressor based ASHP via experimental development and testing at fixed heat loads at low to medium and high heat supply temperature and experienced ambient temperature conditions, ii) Numerical modeling of the HP performance based on the testing results and validations for the annual performance predictions, iii) The annual performance improvement and energy saving with VSM in comparison to FSM with low to medium and high supply temperature inside different property types with age factor considerations; iv) part load operation and the importance of balance point with different property types; v) optimal frequency for constant heat loads demands for the range of water supply temperature (WST), different ambient temperature conditions, vi) the impact of temperature differential on the HP performance, compressor life, vii) performance improvement potential (COP) values at these different tested conditions, viii) The heating capacity and COP value variation and dependency on the compressor operating frequency, viii) the challenges with the variable speed compressor while utilizing this for demand side management programs, ix) annual running cost and carbon emissions analysis, x) the heat supply temperature impact on energy savings with the simultaneous considerations of the control mode in the context of UK. Table 2.1 summarize the aim and objectives for the earlier research work.

Table 2.1: Literature review summary with variable speed capacity control approach

Reference	Article type	Objective	Category	Building Types	Locat ions	Retrofit analysis
Muir and Griffith 1979 [89]	Tech. report	Comparing the seasonal efficiency ratio with variable speed control and ON/OFF control for domestic refrigeration and A/C system. The analysis results show improvement at steady state conditions and huge amount energy savings with variable speed control due to reduction in cycling losses at part load conditions.	Refrigeration and A/C/Experimental development	N/A	N/A	N/A
Tassou et al., 1981 [90]	Research	. The potential savings with variable speed control in comparison to conventional system were reported 15% in energy efficiency conversion. The electronic expansion valves were recommended with variable speed control.	Domestic heat load /Experimental development	N/A	UK	N/A
Tassou et al., 1982 [91]	Research	Modulating capacity impact on the HP performance	Domestic heat load	N/A	UK	N/A
Tassou et al.,1983 [27]	Research	Conventional system comparative study with variable speed control system	Domestic heat load applications	N/A	UK	N/A
Tassou et al., 1984 [58]	Research	Cost analysis of VSM and FSM control	Domestic heat load	Detached	UK	N/A
Tassou, 1991 [64]	Research	Variable speed HP dynamic performance analysis	Domestic heat load//Experimental development	N/A	N/A	N/A
Tassou and Qureshi, 1994 [80]	Research	Variable speed rotary compressor with inverter was investigated. The impact of inverter on starting current, power factor and system efficiency.	Refrigeration system	N/A	N/A	N/A
Qureshi & Tassou, 1996 [26]	Review	Review study on capacity control approaches	Variable speed drive systems	N/A	N/A	N/A
Tassou and Qureshi 1998 [60]	Research	Different type variable speed compressor performance and its comparison with FSM counterpart. The potential benefit increases up to 24% energy savings in comparison to moderate climate of 12%.		N/A	N/A	N/A
Senshu et al. ,1985 [92]	Research	Domestic scale capacity-controlled HP system with different compressors investigation. Annual performance improvement of 30% was achieved in comparison to the conventional reciprocating compressor. The performance degrades for VSM at nominal load conditions in comparison to FSM because of inverter losses.	Domestic heat pump/Inverter losses	N/A	N/A	N/A
Cuevas and Lebrun, 2009 [82]	Research	Challenges/issues with variable speed compressors and inverter losses	Compressor efficiencies	N/A	N/A	N/A
Hewitt et al., 2011[22]	Research	Experimental development with different type heat pumps for domestic retrofit assessment with steady state testing results	Experimental development	N/A	UK	N/A
Quinn, M, 2012[22]	Thesis	Variable speed ASHP system retrofit assessment with very high heat supply temperature	Experimental development	Mid-terraced	UK	1900s
Shah et al., 2016[39]	Research	Experimental study of a diesel engine heat pump in heating mode for domestic retrofit application with very high heat supply temperature	Experimental development	Mid-terraced	UK	1900S
Madonna and Bazzocchi, 2013 [42]	Research	Variable speed ASHP annual performance analysis via modelling/ simulation for different building types, age period in Italian climatic conditions, while applying the weather compensation strategy	Residential buildings	Flats	Italy	1980s, 1990s, 1992/05
Lee et al., 2020[10]	Thesis	Retrofit assessment with Very high heat supply temperature cascaded ASHP simulation based on field trial results with commercial unit	Residential buildings	Mid terraced, Semi, and detached	UK	1900s, 1970, 1990

In summary, the following major research questions needs to be answered:

- ❖ HP system capability of effectively working in future smart grids-key factors influencing the flexibility provided to the power system?
- ❖ The performance investigations with challenges to the variable speed compressor-based heat pump system for capacity control at constant heat loads and its impact on heat supply temperature, ambient temperature conditions?
- ❖ The importance of heat pump sizing, match between heat load demand and supply in the energy savings and retrofit (cost, carbon emissions savings)?
- ❖ The comparative annual performance improvement with variable speed mode (VSM) of control to that of fixed speed mode (FSM) operations in different climatic conditions and different property types?

2.7 Research gaps/motivations

Based on the above literature review it is evidenced that some research topic needs further investigations. Based on the research gaps the goals for this research work were set. The challenges that lead to this research motivations could be summarized as follows:

a) no study in the open literature exists for the domestic retrofit assessment in UK based on the dedicated variable speed compressor-based drive system investigating the challenges, utilizing the compressor speed for capacity control, in different housing type, age period in fixed ambient and load conditions via experimental development
b) focusing on the single aspect of the HP system for domestic heat load applications and neglecting others could lead to misleading information. Therefore, the variable speed compressor based ASHP considering the simultaneous impacts of the heat load demand, building type, age period, climatic conditions, retrofit assessment. The HP testing with constant fixed heat load demands and simulations inside the controlled laboratory conditions in this context was missing from the literature.
c) the challenges with variable speed compressor for demand side management and network stability was recommended by the recent heat pump review study [32]. No study in the open literature considering the drawbacks/issues/challenges with the variable speed compressor efficiency and inverter losses while utilizing the HP with wide range of domestic heat loads applications under different conditions as a demand side management strategy. The only study considering the drawbacks with

the compressor was not used as a part of the heat pump system [82] for domestic heat load applications and demand side management.

d) the research question and investigations of optimum operating point for the variable speed compressor-based HP system for UK retrofit assessment covering majority of the housing stock. The house thermal characteristic, property type, climatic conditions, match between heating capacity and heat load demand, control approach, (VSM vs. FSM), at low to medium and high heat supply temperature.

e) no study in the open literature exists to the best of authors' knowledge on the ASHP as a retrofit technology considering annual running cost, carbon emissions, operating modes of control, in the context of Ireland, Northern Ireland, and Scotland at low to medium and high fixed heat supply temperature based on real system experimental development comparing with the other heating option of oil/gas boiler, electric heating option and very high temperature ASHP.

Chapter 3: Methodology

3.1 Introduction

The chapter describe the approach used to conduct the research in this thesis to answer research questions and overcome the research challenges associated with the variable speed compressor heat pump system. Therefore, the designed heat pump with 9KW nominal heating capacity development, installations inside the conditioning chamber, with proper data acquisitions system, and characterizations via testing was conducted under the range of ambient temperature conditions at three different heat supply temperature with a fixed heat loads commonly required in residential buildings. The performance was established under the steady state conditions inside the lab, aiming at domestic heat load applications. The property type insulation characteristics and loads demand were briefly explained. The economic aspects of the HP system with unit and operating cost in comparison to the other technologies investigated with domestic retrofit assessment was presented briefly in this chapter. The other climatic conditions for Ireland, Northern Ireland, and Scotland investigated in the study chapters with characteristics were also briefly explained.

3.2 HP characterizations and background information for domestic heat load applications

The reverse Rankine based vapor compression cycle is the most used for the domestic HP system applications with the major components of heat exchangers, compressor, expansion valve and refrigerant. Heat energy is transferred between source and sink through the refrigerant as a working fluid. The refrigerant changes its state by absorbing heat, from liquid to gaseous phase on lower pressure side and rejecting heat on higher pressure side changing its state from gas to liquid and hence named vapor compression cycle with the basic principle of operation shown in Figure 3.1. The heat is extracted from low temperature level heat source by adding compressor electric work, and the heat is rejected to the heat sink at higher level temperature. Carnot COP is the theoretical upper limit for the HP efficiency operating between the condensation and evaporation temperature and is based on reversible isentropic expansion and compression, heat transfer without any losses is expressed as follows:

$$COP_C = \frac{T_c}{T_c - T_e} \tag{3.1}$$

Heat and friction losses are existing in real applications in addition to electromechanical, and lubrication issues to the compressor. Also, the HP is working in between the two temperature $T_e - \Delta T_e$ & $T_c - \Delta T_c$ instead of $T_e \& T_c$ as the temperature differential is required between the heat source and sink side and refrigerant side for heat transfer.

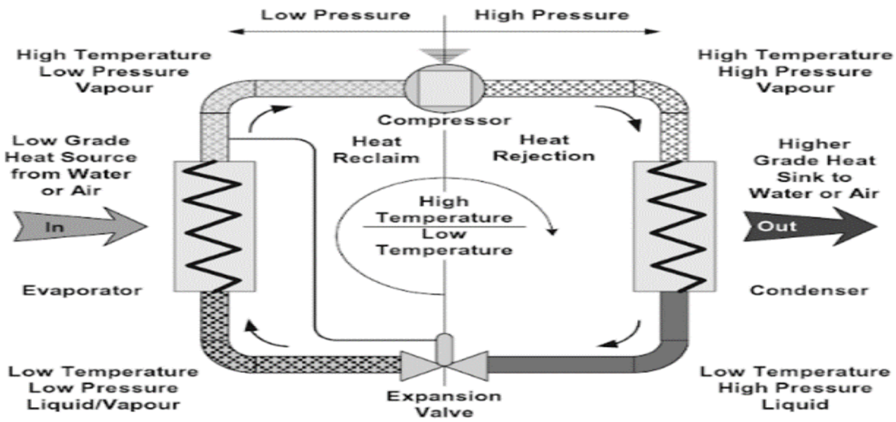


Figure 3.1 HP Basic principle of operation [22]

The COP and hence $T_e \& T_c$ values depend on the source side temperature and heating distribution system instead of the HP design itself. To improve the HP efficiency the real cycle should work as close as possible to Carnot cycle, and this requires the smaller temperature difference between the heat exchangers and source/sinks, reduction losses in compression/expansions. The heat exchanger components selection for this purpose is very crucial to maintain the lower temperature differential. Heat exchanger transfer heat from high temperature fluid to low temperature fluid and provide larger heat transfer surface area for the maximum energy transfer along the temperature gradient. The heat transfer ($\dot{Q}$) is equal the product fluid mass flow rate, specific heat capacity, and temperature difference between the hot and cold fluids expressed by the following equation 3.2:

$$\dot{Q} = \dot{m} c_p \Delta T \tag{3.2}$$

Higher heat is transferred in case of counter flow for the same transmission area. This is because of high temperature difference maintaining during the process. In case of parallel flow, the temperature difference is highest at the time both fluids entrance to the heat exchanger, and subsequently reduces with the transfer of heat. The HP uses two heat exchangers for evaporation and condensation process. Sensible and latent heat transfer proportionally occurs in both the evaporator and condenser. Subcooling occurs in condenser for the reduction of refrigerant quality before entering evaporator while superheat occurs in evaporator to make sure the dry gas entry into the compressor (for protection of its operation). Subcooling and superheating needs optimization in the range of 3- 5K as less sensible heat energy is transferred per unit heat exchanger area compared to latent heat. Operating pressure, range of temperature, desired transmission area, properties of fluid and cost are factors effecting each heat exchanger selection. Axial fan draws air through and over the tubes for the increase heat transfer through evaporator by increasing air flow rate. The shape of the evaporator tube plays important role in reduction of frost and increasing performance by increasing heat transfer area U & cylindrical shape. The evaporator utilized could produce 11kW at -2 ^{o}C. The shell and tube or plate type heat exchanger is usually preferred selection for condenser to transfer heat from refrigerant to water according to temperature range, pressure, and size of system. Brazed Plate type heat exchanger water cooled is mostly used for the condenser of the domestic heat pump due to its shape, weight, and size and was utilized in this research with the model number B80ASHx28/1P. Temperature and pressure reduction are required before the refrigerant entry into evaporator to enable heat absorption. The subcooled liquid refrigerant leaving the condenser is passed through a flow restrictor (fixed or variable diameter orifice) device for expansion process. This expansion device should also control the refrigerant flow through evaporator to provide the accurate superheat before it enters compressor. However higher superheat negatively affects the system performance (COP) as the evaporator capacity reduces and cause increase of work to compressor. Hence the role of properly selected expansion device is very important for accurate evaporation and control of superheat. Electronic expansion valve is more suitable for variable capacity and convers higher capacity range, where stepper motor controls the needle movement from the seat and controls the aperture diameter. Voltage or milliamp signals generated from temperature and pressure sensors at suction line are sent to controller and operates the motor accordingly. A comparison is made between the refrigerant measured temperature and saturation temperature for the measured pressure with any difference indicating superheat value. The motor accordingly arranges the needle position and hence the refrigerant rate of flow to get the accurate superheat value. This method of superheat control is possible through a wider capacity range with high accuracy and make sure accurate refrigerant quantity evaporates across the evaporator resulting in its higher performance and was utilized with the developed system. Based on the above considerations the equipment/components were utilized for testing to establish the system performance under different test conditions. The modulation range for the compressor was in between 15Hz and 120Hz. The controller for heating and cooling applications provided the function of map management and compressor speed with refrigerant R410a. The superheat set point was 10K during all the tests and it was maintained by using electronic expansion valve. The heat pump test facility mainly consists of the developed vapor compression type-based heat pump system, conditioning chamber for maintaining the required conditions, heat load generation and dumping rig, control devices, data acquisition system installed at Jordanstown campus, Ulster University. The test facility could provide range of ambient temperature, and humidity conditions on source side and the required set points values for EWT/ WST by using PID controllers. Compressor drives refrigerant in HP system and produces pressure difference among the high condenser pressure side and low evaporator pressure side. The power used in HP system is mainly used by compressor and effects the heat production and COP. The built-in inverter drive for variable speed compressor was chosen to reduce inverter losses and operate over a wide of operation to achieve the potential benefits with variable speed compressor producing range of heat demands. The hp components detail are mentioned in Table 3.1.

Table 3.1: List of the main components for the developed system

Component Name	Model Number/Manufacturer
Scroll compressor	XPV0302E-4X9/Copeland Ltd
Drive	ED3015B-H2XB – 3 Phase/ Copeland Ltd
Condenser- Plate heat exchanger	B80ASHx28/1P/SWEP
Evaporator	I/50294/01/ECO
Axial Fan	R13-5035-6M-50/Copeland Ltd
Accumulator	A12-507/ALCO
Sight glass	MIA-038 /ALCO
Mass flow meter	MASS2100 DI 15 /Siemens
Controller	Superheat and Envelope Controller – SEC MONO/ Copeland Ltd
Electronic expansion valve	EXL-BF1-Unipolar stepper motor valve/ Copeland Ltd
Converter	RS 485
Temperature sensor	NTC (ECN-EG30)/ Copeland Ltd
Temperature sensor	PT100
Pressure Transducer	PT4-30m/ ALCO
Communication cable	SEC2-ED3-3W
Refrigerant	R410A
Data logger	DT85

The heat pump system pictorial view with the conditioning chamber, and water entering the heat pump and heat supply temperature piping system is presented in Figure 3.2. The conditioning chamber seen in the picture have been installed in earlier work with the associated heat transfer characteristics analysis of the wall, floor, and roof materials and with overall U value calculated to be 0.25W/m^2. K [22]. The variable speed compressor could produce the optimal performance at 60 Hz according to bin distribution and the number of highest hours occurring at 7 ^{o}C in Belfast for the 1900s Mid terraced test house.

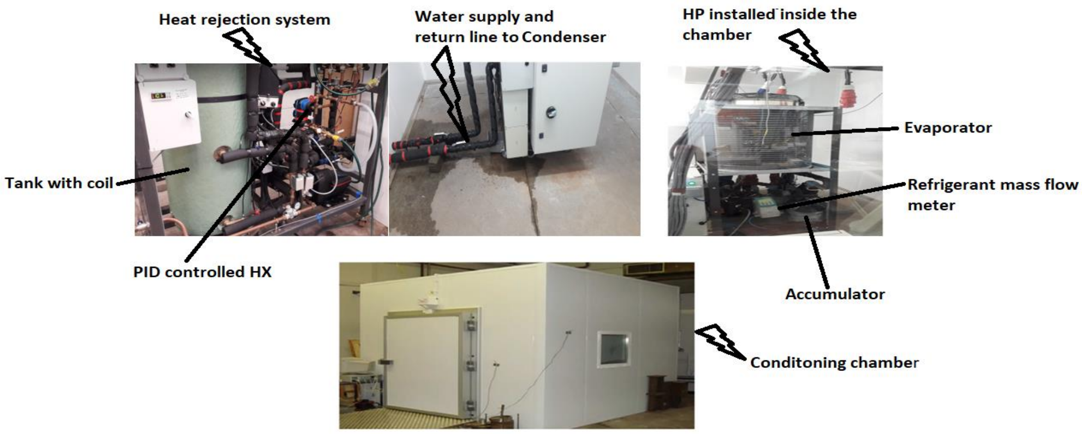


Figure 3.2 Tested heat pump in pictorial view

Simplified schematic diagram, with HP system details, and heat rejection system is shown in Figure 3.3, and consisted of three basic circuit, i.e., close loop water (shown red), fresh water supply (shown blue), and refrigerant circuit. The water circulating circuit including variable speed water pump for controlling flow rates inside the close loop circuit, water tank with heat exchanger inserted, valves for air venting, safety, expansion tank, 3-way PID valve. The difference between water/heat supply temperature (WST) and entering water temperature

(EWT), ΔT was controlled by PID controlled heat exchanger on the load side. The PID valve operated according to the supply temperature set points value. The heat load was dumped to the water tank via the inserted coil and/or secondary heat exchanger by supplying fresh water from the water supply tank, fitted at top of the roof, Block 6, Jordanstown campus, Ulster University.

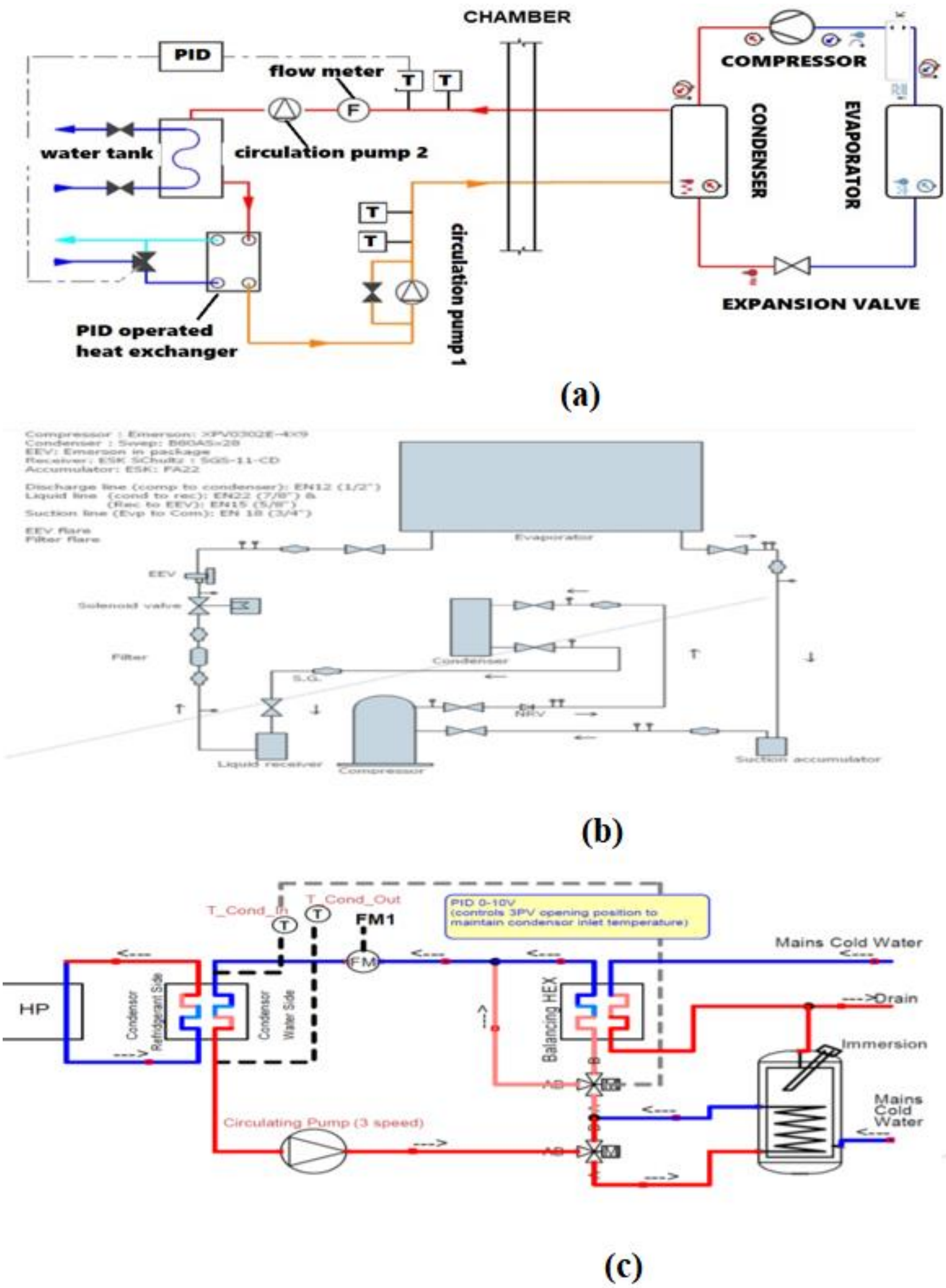


Figure 3.3 Schematic diagram a) Entire set-up with conditioning chamber, b) HP system, c) heat rejection system [88, 93]

The testing procedure mentioned in BS EN14511-3 [29] was followed in terms of methodology, sensor, quantity, & measurement accuracy requirements. The methodology and test conditions for water supply and return temperature, dry and wet bulb temperature is presented in the Table 3.2. The water mass flow rate mentioned in the standard were fixed according to the standard rating conditions, however in our case this was adapted to simulate the required fixed heat load demand, while maintaining constant delta T (difference between water entering and supply temperature).

Table 3.2: Air to water heat pump testing procedure mentioned in BS EN14511-3 [29]

Outdoor heat exchanger		**Indoor heat exchanger**	
Inlet dry bulb temperature(^oC)	Inlet wet bulb temperature (^oC)	Inlet temperature(^oC)	Outlet temperature (^oC)
2	1	30	35
-2	-3	30	35
7	6	30	35
2	1	40	45
-2	-3	40	45
7	6	40	45
-7	-8	50	55
7	6	50	55

3.3 The control devices and data acquisition system

The data logger instruments, measurement devices consisting of temperature, pressure, mass flow meters sensor and control mechanism inside the chamber were properly chosen and installed to adequately perform the tests with good confidence in the testing results. All the measured variables were monitored, and the data were recorded every ten (10) seconds with an average value considered for analysis purposes. The data utilized were obtained through recording and monitoring from; a) source side conditions, b) inlet and outlet of the main components c) load side quantities. On the source side the ambient temperature, humidity was measured and controlled by the heater, cooler, and humidifier. The PID humidifiers was able to meet the requirements mentioned in the standard for the relative humidity (RH). The device types measured quantity and the uncertainty associated with the measurement devices is shown in Table 3.3. On water side entering and supply temperature was measured using PT100 temperature sensor, water mass flow rates via flow meter, and pressure gauges for pressure measurement. The pressure, temperature at the inlet and outlets of compressor, condenser, evaporator, expansion valve was subsequently measured.

Table 3.3: Uncertainties ranges for measurement instruments

Air side	Measured quantity	Measurement device	Units	Uncertainties
	Relative Humidity(Ø)	Hygrometer	-	±0.8%
	Dry bulb temperature (DB)	Thermocouple(T-type)	°C	± 2
	Wet bulb Temperature (WB)	Thermocouple(T-type)	°C	±0.3
Refrigerant side	Mass flow rate	MASS 2100 DI15	Kg/s	± 1.3%
	Enthalpy (h)	Estimated from P, T measured values [94]	kJ/kg	1-1.76%
	Pressure (P)	PT5 Pressure transmitters	kPa	± 1 %
	Temperature (T)	NTC (ECN-EG30)	°C	± 0.5
Water side	Mass flow rate	Electromagnetic, Eltek, GC 62	kg/s	±1.5 %
	Pressure difference(static)	Pressure Gauge	Pa	± 5 %
	Temperature inlet /outlet (T)	PT100, Eltek GD 24	°C	± 0 .1
	Electric power meter	Landis and Gr P350	W	± 1 %
	Current	Transducers LEM AKR 50 C420L	A	± 0.5 %
	Voltage	Transducers (ABB CC-U/V	V	± 0.5 %

The refrigerant mass flow rate through the condenser and evaporator was measured through the flow meter installed in the liquid refrigerant line. Initially, energy balance analysis, and comparison with the manufacturer model SELECT software was completed to see the reliability of experimental results, shown in Figure 3.4. The co-efficient of performance (COP) shows comparative results with some smaller values in the electric power consumption resulting in improved COP values.

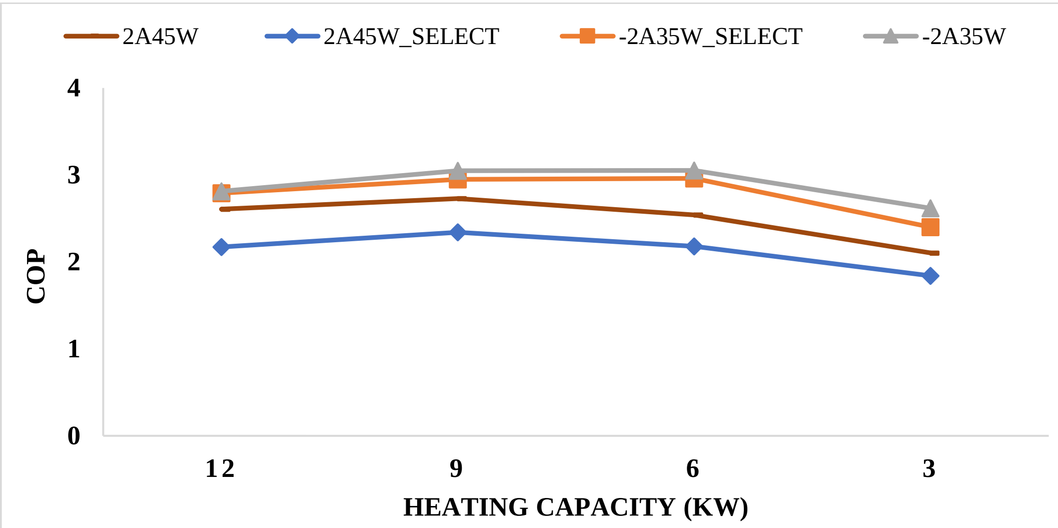


Figure 3.4 Experimental COP comparative results with SELECT software

The experimental results for COP of the HP system at two ambient conditions of -2 ^{o}C, and 2 ^{o}C with WST of 35 ^{o}C and 45 ^{o}C respectively were compared with SELECT [95] with a heating capacity of 12KW, 9KW, 6KW, and 3KW. The comparative testing results shows an average value of 16% higher performance at 2 ^{o}C ambient temperature conditions and heat supply temperature of 45 ^{o}C to that of SELECT model. The reason for the higher performance is the lack of defrost strategy and higher humidity at test conditions [22]. The humidity ratio considered with the manufacturer software was lower than the actual testing conditions.

3.4 The HP with testing facility

The data utilized were obtained from the data acquisition system through recording and monitoring; a) source side conditions, b) HP system main components at inlet and outlet, c) load side quantities. On the source side the ambient temperature, humidity was measured and controlled by the heater, cooler, and humidifier. The PID humidifiers was able to meet the requirements mentioned in the standard for the relative humidity (RH). The device types measured quantity and associated uncertainty with measurement devices were shown in Table 3.3. On water side entering and supply temperature was measured using PT100 temperature sensor, water mass flow rates via flow meter, and pressure gauges for pressure measurement. The pressure, temperature and at the compressor, condenser, evaporator, expansion valve, inlet, and outlet were subsequently measured and recorded. The refrigerant mass flow rate through the condenser and evaporator was measured through the flow meter (Model Number: MASS 2100 DI15) with measurement range between 0 to 1.56kg/s) installed in the liquid refrigerant line. At lowest refrigerant mass flow rates measurement range lower limit were experienced but with an accuracy of ± 1.3%.

3.5 Experimental procedure and test conditions

The ambient temperature, humidity, and WST values for the tested conditions, with definite constant heat loads have been shown in Table 3.4. The system performance was tested for the range of ambient temperature conditions at low (35 ^{o}C) to medium (45 ^{o}C) and high (55 ^{o}C) WST. The main testing regime was developed at constant value of delta T of 10 ^{o}C to make the result consistent and comparable because of most of the existing building installed heat emitters were designed for the flow outlet/return temperature difference of 10 ^{o}C [96]. The HP system performance was investigated with definite constant heat loads of 3 KW, 6 KW, 9 KW, 12 KW, 15 KW, 18KW commonly required in the domestic building for SH and DHW demand. Four different ambient conditions of -2 ^{o}C, 2 ^{o}C, 7 ^{o}C & 15 ^{o}C, and various delta T values was chosen to investigate the system performance for low to medium WST applications of 35 °C to 55 °C. The ambient temperature, humidity, and WST values for the tested conditions is shown in Table 3.4. Existing building have designed heat emitters, working more efficiently with higher WST), with a flow outlet/return temperature difference in the between 20 °C and 10 °C [41]. Therefore, the main testing regime was developed at $\Delta T = 10$ °C with some test for ΔT as 5 °C, and 20 °C as well for comparative study at single ambient conditions, and varying heating capacity and WST. Although, the ΔT value mentioned in the standard was 5 °C, which has been followed and extended the tests to 10 °C to see the impact on the HP performance as well. The tests were conducted by keeping the heat demands constant and the system adjusting the operating frequency according to the source/load side conditions based on the ambient temperature, relative humidity, water supply temperature, the difference between the water supply/return temperature to HP.

Table 3.4: Humidity requirements as per standard [97]

Heating capacities(kW)	Ambient Temp (° C)	DB (° C) / WB (° C)	RH (%)	RH (%) – upper limit	RH (%)- lower limit	WST (°C)	ΔT (°C)
12,9,6,3	-2	-2/-3	79.3	85.5	73.2	35, 45	10
12,9,6,3	2	2/1	83.9	88.7	79.1	35, 45, 55	10
15,12,9,6,3	7	7/6	86.9	90.8	83	30, 35, 40,45, 50,55,	10
18,15,12,9,6,3	15	15/14	90	93	87.1	30, 35, 40,45, 50, 55,	10
18,15,12,9,6,3	-	-	-	-	-	25, 30, 35, 40,45, 50,55	5
18,15,12,9,6,3	-	-	-	-	-	40, 45, 50,55	20

The HP performance results was evaluated with a range of heating capacity of 3-18 KW, with four ambient temperature conditions, and at low to medium and high heat supply temperature. The amount of heating capacity was controlled with the close loop water mass flow rates variation with variable speed pump. The existing radiators are designed for supply/return temperature difference between 10 °C and 20 °C but at the same time the differential have an impact on the COP, which has been very rarely investigated earlier based on the literature review, is being considered this study and the results are presented here. The changing difference between WST and return temperature have also an impact on the compressor life evidenced from the pressure ratio variations. The results show that for constant fixed heating capacity, supply temperature, and ambient temperature conditions the system works more efficiently at higher delta T values, because of comparatively higher compressor isentropic efficiencies, and lower pressure ratio and discharge line temperature

(DLT). The variation in differential (5 °C, 10 °C, 20 °C) could be achieved by changing the water mass flow rates inside the heating system close loop circuit. The water mass flow required, for achieving individual heating capacity with different delta T values was shown in Table 3.5. Higher water mass flow rates inside the heating system at constant heating capacity results in lower delta T values. Higher delta T value is specifically important at the HP start up time to achieve the radiators desired temperature more quickly to maintain the thermal comfort inside the room and at the same time improve the radiators efficiency (by having more difference between the room set point value and radiator temperature rejecting more heat).

Table 3.5: Set point mass flow rates(kg/s)

HC (kW)	ΔT=20 (° C)	ΔT =10 (° C)	ΔT =5 (° C)
18	0.22	0.43	0.86
15	0.18	0.36	0.72
12	0.14	0.29	0.57
9	0.11	0.22	0.43
6	0.07	0.14	0.29
3	0.04	0.07	0.14

3.6 Limitations

3.6.1 Conditioning chamber and HP operating limitations

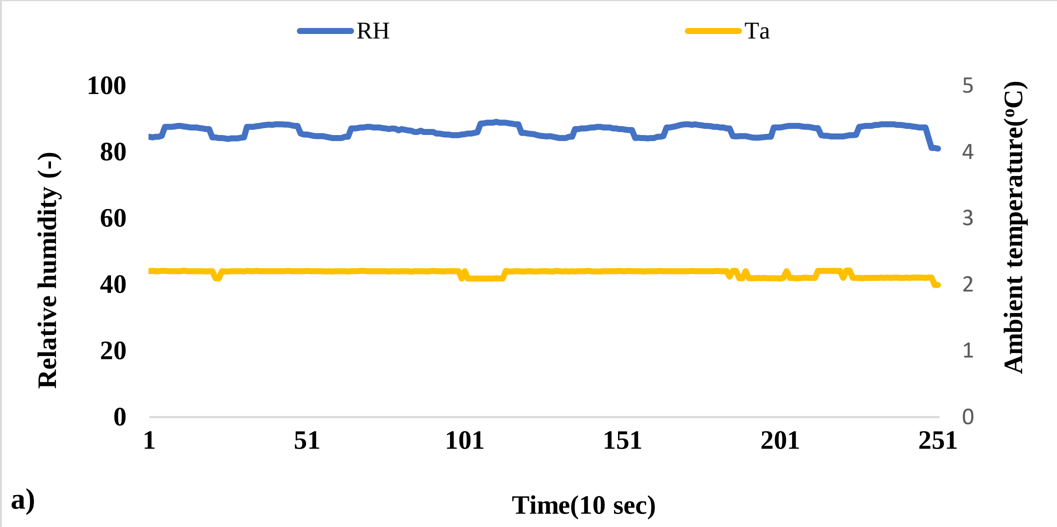
Due to compressor operating frequency range of 120-15Hz, climatic chamber limitations, certain heating capacity testing were not conducted at extreme conditions. The system was unable to achieve 18KW, heating demand at 7 °C, and 15KW at 2 °C conditions due to compressor speed upper limit. For example, at ambient conditions 7 °C with WST of 35 °C at frequency of 118.63 Hz a maximum heating capacity of 16.52KW was obtained when the EWT was 25 °C. Similarly, due to lower limit of 15Hz low heat demands of 2KW were not possible. The low heating demand consumes less power and most of the time lower heat is required by building. The tests for 2KW were conducted for 15 °C and 7 °C and the output heat to the load was higher. At 15A35W the speed reduced very slightly to 15.03HZ for a heat demand of 2KW and the heat rejected was 2.96 compared to 3KW at 15.21Hz for the same tested conditions. However, it was interesting to note the compressor jump to higher speed for maintaining proper lubrication after two (2) hours of operation at low speed & lower heating demand tests to avoid compressor failure due to improper lubrication. The low-speed system operation at 2KW test for 7 ^{o}C ambient conditions with WST of 35 ^{o}C results in 2.43KW heat rejection with operating frequency of 15.08Hz instead of 21.73Hz when the full heating demand of 3KW was meet for the same tested conditions. The heat capacity further reduced to 2.20KW at frequency of 15.09Hz with WST of 45 ^{o}C at the same ambient conditions of 7 ^{o}C with isentropic efficiency of 44.7%. Similarly, at ambient conditions of 2 ^{o}C & -2 ^{o}C a maximum of 12KW heating capacity tests were possible because of the chamber limitations instead of the speed upper limit and therefore at low ambient conditions and testing with full speed was found challenging. At 2A35W at heating demand of 15KW an average value of 13.75KW heating capacity were achieved with the frequency of 108.85Hz.

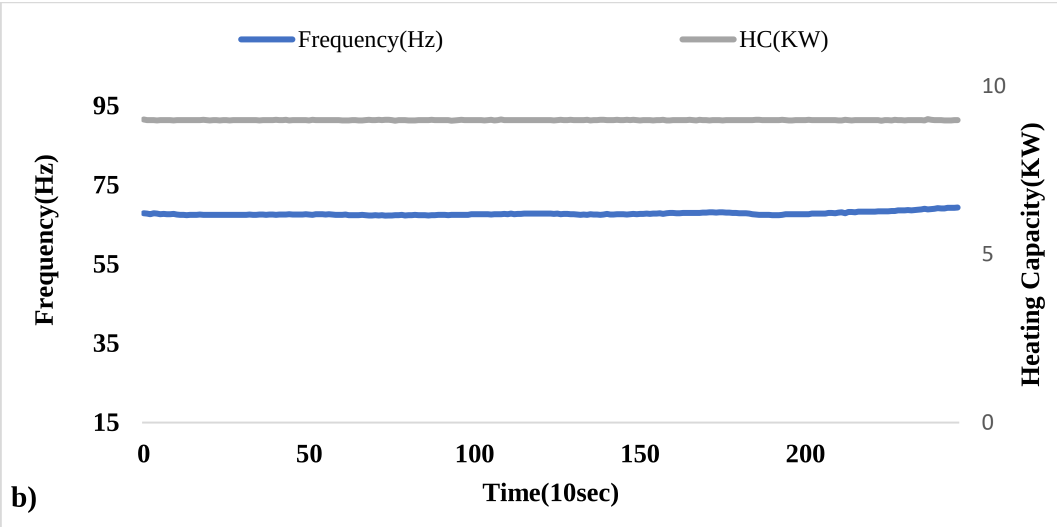
3.6.2 Issues /Challenges during the HP testing regime

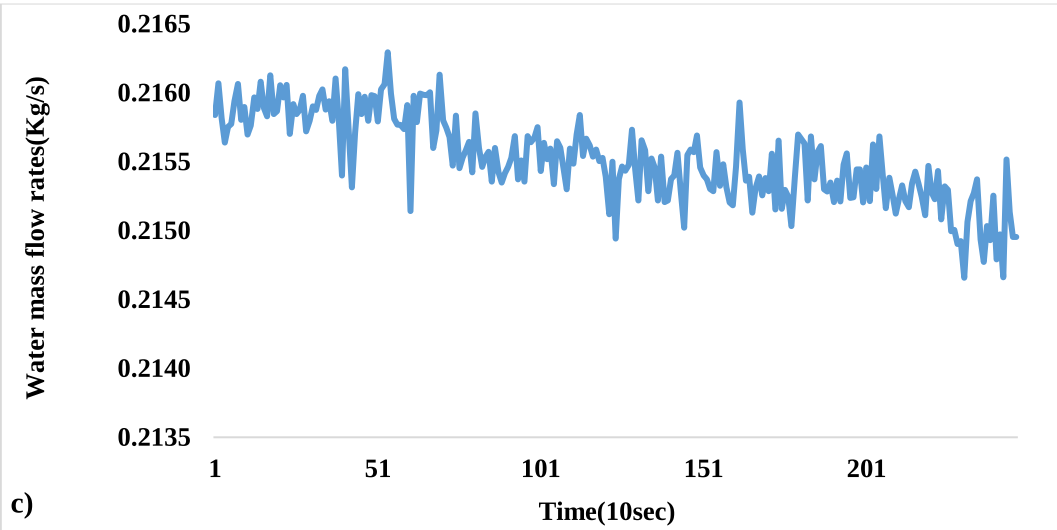
At the start of the testing development and execution certain challenges were faced with constant heat load generations and adjustments required on water mass flow rate inside the close loop circuit, and heat rejection to the fresh water. The heat rejection capacity was limited while using a single PID operated heat exchanger due to limited flow rates of the water inside the open loop circuit, and the tank with fitted heat exchanger coil. Therefore 2nd heat exchanger was utilized for fulfilling the needs for higher amount of heat rejection. The stable operation of the controller was the biggest challenge and instability were seen for the controller during certain tests due to humidity control, and PIDs working against each other, but were managed accordingly.

3.6.3 Set point control (temperatures, humidity, load demand)

The heat/source side conditions were managed very carefully with minimum disruption to the system. This result in steady state achievement while maintaining a good accuracy with the differential limits mentioned in the standards with the Figure 3.5, displaying EWT, LWT ambient temperature and superheat control for a single test during the test period. The superheat set point was 10 during all tests and it was maintained with high accuracy by using electronic expansion valve. The super heat loss its control during several test mainly because of sudden changes on the load and source side conditions. Whenever there was an abrupt change on the load side due to this PID controlled valve opening or closing then the controller respond mainly by changing the superheat to the new load conditions. On the load side, the heating capacity was balanced by the required mass flow rate, and delta T value. The heat load was adjusted very carefully so that the PID valve on the load side for controlling delta T should work very smoothly and avoid this disruption to the system.


RH
Ta
100
80
60
40
20
0
5
4
3
2
1
0
1
51
101
151
201
251
Relative humidity (-)
Ambient temperature(ºC)
a)
Time(10 sec)


Frequency(Hz)
HC(KW)
95
75
55
35
15
10
5
0
0
50
100
150
200
Frequency(Hz)
Heating Capacity(KW)
b)
Time(10sec)


0.2165
0.2160
0.2155
0.2150
0.2145
0.2140
0.2135
1
51
101
151
201
Water mass flow rates(Kg/s)
c)
Time(10sec)

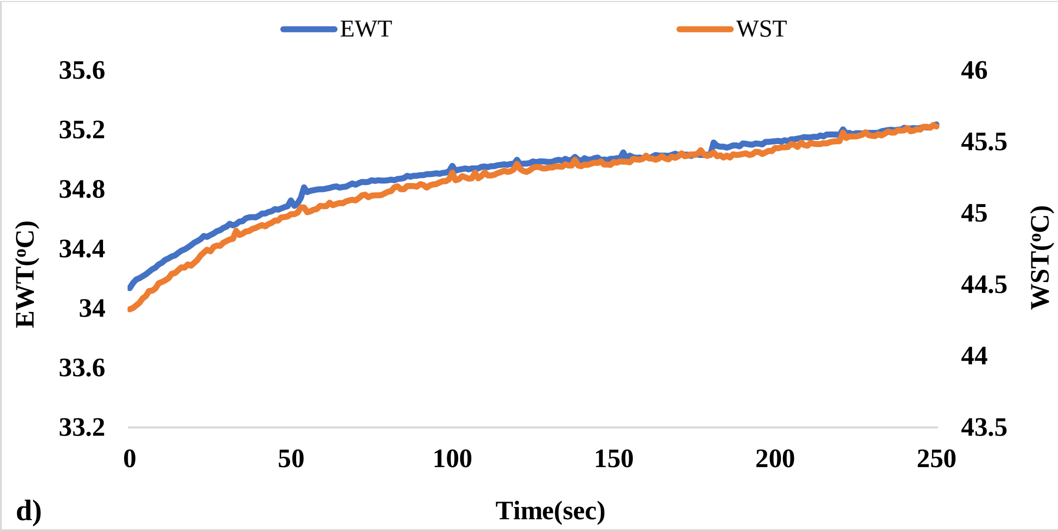


Figure 3.5 Set parameters accuracy within limits as per standard a) Source side, b) 9KW heating capacity and, c) Water mass flow rates (Kg/s) d) EWT, WST

Similarly, on source side ambient temperature and humidity was controlled by the PID heaters, coolers, and steamers and any abrupt changes to these variables resulted in loss of control for the controller.

3.6.4 COP reduction due to frost

Frost occurs at the evaporator as water vapor freezes on its surface because of removal of heat energy from air resulting in reduced thermal contact between the air and refrigerant and degrades heat transfer process. The compressor pressure ratio increases with increase of air and refrigerant temperature difference because of frost build up, also resulting in lower temperature and pressure at the evaporator exit cause an increase of the compressor work. Hence the heating capacity, COP is reducing, and compressor work increases, therefore the frost(ice) removal is necessary. Different defrost approaches are used including reverse cycle, hot-gas by-pass, utilizing the heat energy generated by heat pump system. The defrost cycle must be balanced and needs to continue until all the ice is removed and stopped after that to avoid extra losses. Other techniques including electric defrosting using heating element, built in evaporator and hot air blown across the evaporator and external energy is used for that purpose. During this type of defrosting, the heater element turns on to melt the ice and the compressor commonly is stopped to reduce the electricity consumption for running both items and is set to off during normal compressor operation. To make the process more efficient the hot air is blown with the heating element placed in front of evaporator, the axial fan draws heat across the evaporator and compressor is turned off to increase defrost and reduce power load. Defrost control is very crucial because of performance and capacity degradation due to frosting and extra energy required for defrosting. Proper control can reduce the defrost time and performance degradation due to frosting. Commonly used control methods by manufacturer are based on timings, or discharge temperature limits and the temperature difference across the evaporator and could be used a combination of these approaches to enhance the associated benefits. During the temperature difference control method, the cooling capacity and evaporator coil and air temperature difference is monitored and defrost can be triggered when difference reaches to a certain level when the ice builds. As an example, a defrost timer can start when the temperature difference reaches the certain limits. The timer pattern can be based on experiments for operating ranges and will results in a smaller number of defrost cycles, evaporator working on full capacity even if small amount of frost builds up specifically if the evaporator is oversized for the tested conditions. These parameters implemented in controller for the performance monitoring can be used to trigger defrost. However, the increase in control strategy complexity will result in higher unit cost.

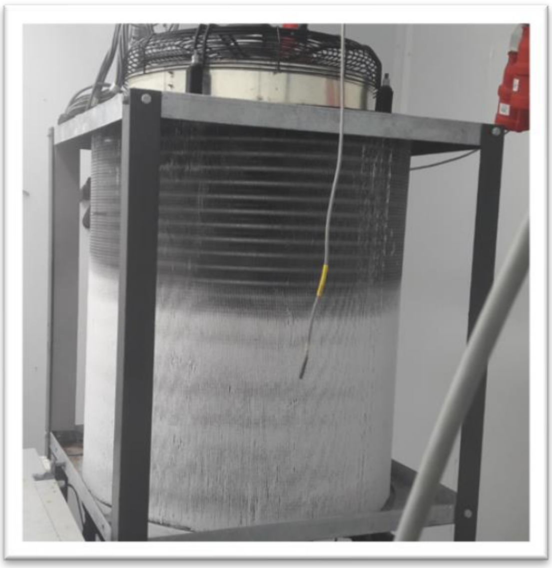


Figure 3.6 Frost build-up process over evaporator

Frost causes the COP reduction according to the ambient temperature and humidity conditions, and part load operations of the system. This performance degradation due to frosting have been considered in the analysis of annual COP calculation in chapter, and retrofit assessment in chapter 6. Tests were developed to calculate the reduction in COP values at different ambient temperature conditions and have been considered while doing the annual performance analysis (Figure 3.6).

3.7 Mathematical Model

The heating capacity ($\dot{Q}$) in kW on water side was calculated by equation 3.3a using measured water inlet and outlet temperature difference (ΔT) in K, the water specific heat capacity (kJ/kg. K) and the water mass flow rate in kg/s. The heating capacity on refrigerant side was calculated by equation 3.3b using the measured values of the refrigerant mass flow rate and the enthalpy difference between the inlet and outlet of the condenser.

$$\dot{Q_w} = \dot{m_w} \; C_P \Delta T \qquad (3.3 \text{ a})$$

$$\dot{Q}_r = \dot{m}_r \Delta h \qquad (3.3 \text{ b})$$

Water side heating capacity divided by enthalpy difference in (kJ/kg) across the condenser was also used for comparison with the measured refrigerant mass flow rate values, and by showing good agreement between the values confirming the measurement data accuracy. Measurement of the heat transfer on both side of the condenser, i.e. refrigerant & water side make sure an energy balance exists on the processed water side and high-pressure refrigerant side and the assumption of adiabatic heat transfer could be made neglecting loss of heat to the surrounding because of the system design and well insulated heat exchangers.

$$\dot{m}_r = \frac{\dot{m}_w * C_p * \Delta T}{\Delta h_r} \qquad (3.4)$$

The co-efficient of performance (COP) at each individual testing conditions, defined as the ratio between the heat output and total electric power consumption were calculated using equation 3.5:

$$COP = \frac{Q}{P_{total}} = \frac{C_p * \dot{m_w} * \Delta T}{p_{comp} + p_{aux}} \qquad (3.5)$$

whereas Q is the heating capacity in KW, P is the total electric power consumption in KW, c_p the water specific heat capacity, $\dot{m_w}$ is water mass flow rate (kg/s), and ΔT is the difference between the water supply temperature (WST) and entering water temperature (EWT).

The compressor isentropic efficiency, defined as the ratio between the isentropic thermodynamic work to the actual thermodynamic compressor work was calculated by equation 3.6;

$$\varepsilon_{is} = \frac{\dot{W}_{isen}}{\dot{W}_{comp\_actual}} = \frac{\dot{M}_{comp} * (h_{2is} - h_1)}{\dot{W}_{comp\_acutal}} \tag{3.6}$$

The volumetric efficiency of compressor; the ratio between the actual mass flow rates at the compressor suction to the theoretical value of the mass flow rates was calculated by equation 3.7;

$$\varepsilon_{vol} = \frac{\vartheta_{suction}}{V_{swept}} * \frac{\dot{m}_{suction}}{\omega} \tag{3.7}$$

Whereas ϑ and $\dot{m}$ represents the respective refrigerant specific volume in (m^3/kg) and mass flow rates(kg/s) at the compressor suction, and ω denotes the compressor speed (Hz). Experimental work has an associated uncertainty to a certain extent based on the independent variables measured, contributing to the final experimental results error. The error associated with the individual measuring devices mentioned in the methodology was utilized to find total error for the heating capacity, electric power consumption and COP during the steady state testing regime. The error analysis was performed using the approach developed by ASHRAE Guideline [98] with the following equations of 3.8, & 3.9.

$$Q_{Error} = Q * \sqrt{\left(\frac{\dot{m}_{w_{error}}}{\dot{m}_w}\right)^2 + \left(\frac{\Delta T_{Error}}{\Delta T}\right)^2} \tag{3.8}$$

$$CoP_{error} = CoP * \sqrt{\left(\frac{Q_{HP_{Error}}}{Q_{HP}}\right)^2 + \left(\frac{P_{Error}}{P}\right)^2} \tag{3.9}$$

The bin method was used for the evaluation of the annual performance for the heat load demands for different property types. The HP COP in each bin (i), and electrical energy consumption (E) for the heat pump only in KWh, and COP correction factor were calculated by equation 3.10, 3.11 and 3.12 respectively.

$$COP_{hp}(i) = COP_{hp}(i) * f_{COP}(i) \tag{3.10}$$

$$E_{hp}(i) = \frac{Q_{hp}(i)}{COP_{hp}(i)} \tag{3.11}$$

$$f_{COP}(i) = \frac{PLR(i)}{1 - C_c + C_c * PLR(i)} \tag{3.12}$$

The $f_{COP}(i)$ determines the COP correction factor and PLR represent the part load ratio [35,42]. The degradation co-efficient (C_c) value recommended in standard was 0.9[42].

3.7.1 Model validation

The HP simple numerical model was developed and calibrated at nominal heating capacity testing results using the approach suggested by other researchers [99] for parameter identifications. First, the model was calibrated with nominal heating capcity tests for finding the co-efficient values, followed by the model validation with other testing results. The detail for the model development, residuals values, co-efficients and validation with experimetnal results were completed. The developed numerical model with biquadratic, cubical form with two outputs parameter, i.e. coefficient of performance (COP) and electric power consumption (P) determined by equation 3.13, and 3.14 respectively.

$$COP_{model} = 3.64 - 0.000242 * \omega^2 + 0.000739 * \omega * T_W - 0.00097 * {T_w}^2 + 0.001825 * \omega * T_a + 0.0009 * T_w * T_a - 0.0000438 * \omega * T_w * T_a + 0.00237 * {T_a}^2$$

$$COP_{model} = 2.9 - 1.38 * 10^{-6} * \omega^3 + 0.00038 * \omega * T_w + 4.2 * 10^{-6} * \omega^2 * T_w - 9.9 * 10^{-6} * \omega * {T_w}^2 - 4.755 * 10^{-6} * {T_w}^3 + 0.0022 * \omega * T_a - 0.000025 * \omega^2 * T_a + 0.00127 * T_w * T_a + 0.0000233 * \omega * T_w * T_a - 0.000049 * {T_w}^2 * T_a - 0.000014 * \omega * {T_a}^2 0.00017 * \mathrm{T_a}^3$$

(3.13)

$$P_{model} = 0.385 + 0.00013 * \omega^2 + 0.00066 * \omega * T_w + 0.0001 * {T_w}^2 + 0.00036 * \omega * T_a + 0.000288 * T_w * T_a - 0.00000633 * \omega * T_w * T_a - 0.00094 * {T_a}^2$$

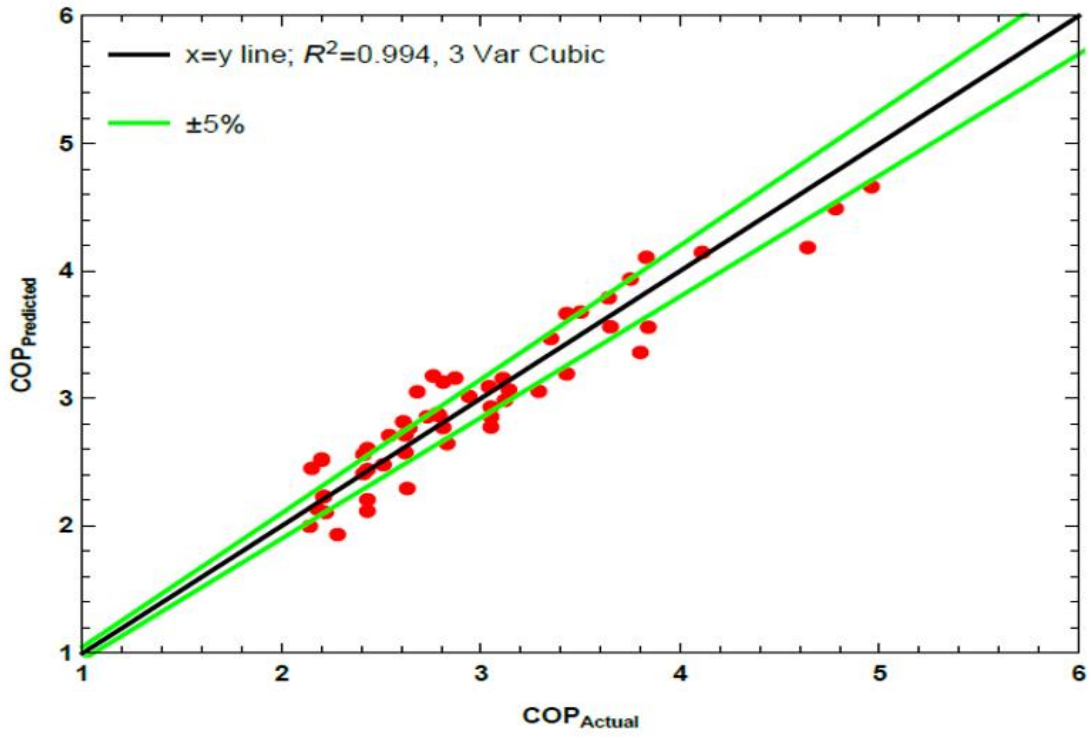
$$P_{model} = 0.506 + 0.00000312 * \omega^3 + 0.00088 * \omega * T_w - 0.0000117 * \omega^2 * T_w + 0.000015 * \omega * {T_w}^2 - 0.00000603 * {T_w}^3 + 0.00080 * \omega * T_a + 0.00000753 * \omega^2 * T_a - 0.00188 * T_w * T_a - 0.0000288 * \omega * T_w * T_a + 0.000046 * {T_w}^2 * T_a - 0.0000286 * \omega * {T_a}^2 + 0.000059 * {T_a}^3 \quad (3.14)$$

The outputs of the model were dependent on three variables, i.e., water supply temperature (WST), ambient temperature conditions, and operating speed. The model validation shown in Figure 3. 7 was performed with experimental test results using the MATLAB software [100] and presented in more detail in Appendix A. Comparison was conducted between measured experimental and predicted values to have an idea of the model error via the root mean square error (RMSEs) approach, defined by equation (3.15).

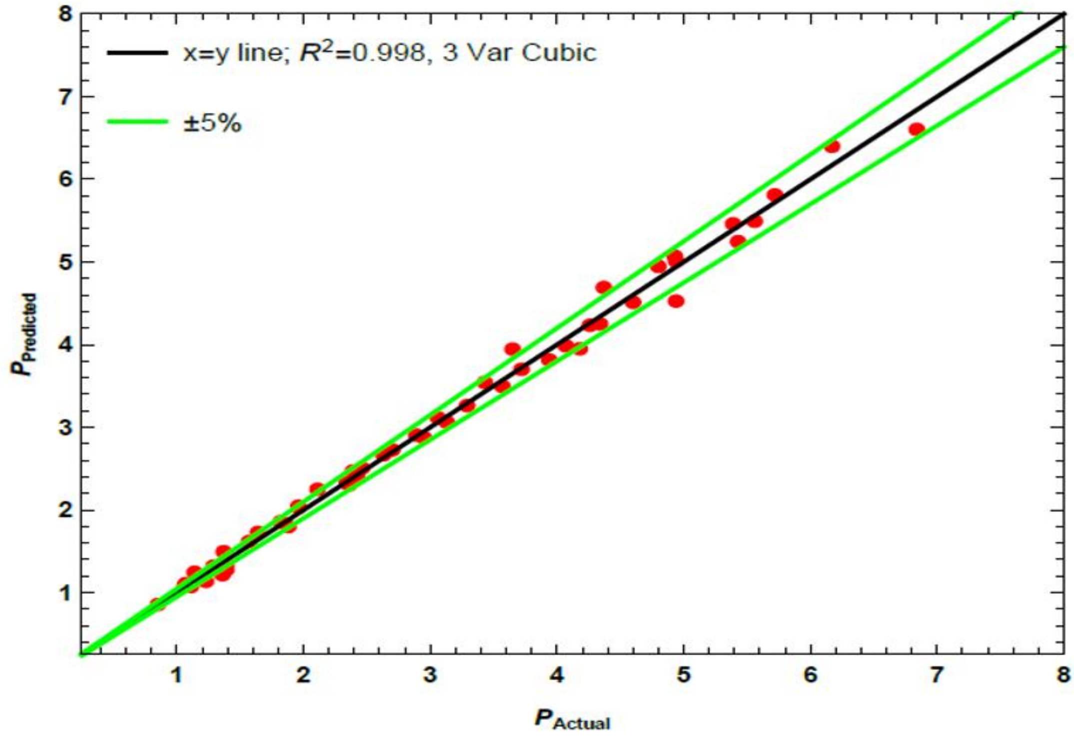
$$RMSE = \sqrt{\frac{\sum_{i=1}^{n}(V_{measured} - V_{sim})^2}{n}} \quad (3.15)$$

The validation procedure is performed as follows:

a. The model co-efficient was calculated by calibrating the model first using experimental data above 7 °C and a reference point from manufacturer data sheets.
b. The calibrated model was then used to calculate the COP and power consumptions(P) parameters at different steady-state points. The inputs to the model were kept constant and set like the measured values at the heat supply temperatures and ambient temperature as well as compressor speed.
c. The calculated values of the model were than compared to the measured experimental data. The model validation shown in Figure 3. 7 and it shows good agreement between the measured data and modelled values with relative errors in in the range of ±5 %. The compressor frequency modulation range was in between 15-120 Hz and range of ambient temperature conditions and at low to medium and high heat supply temperatures.



(a)



(b)

Figure 3.7 Model validation with experiments a) COP (-), b) Power P(KW)

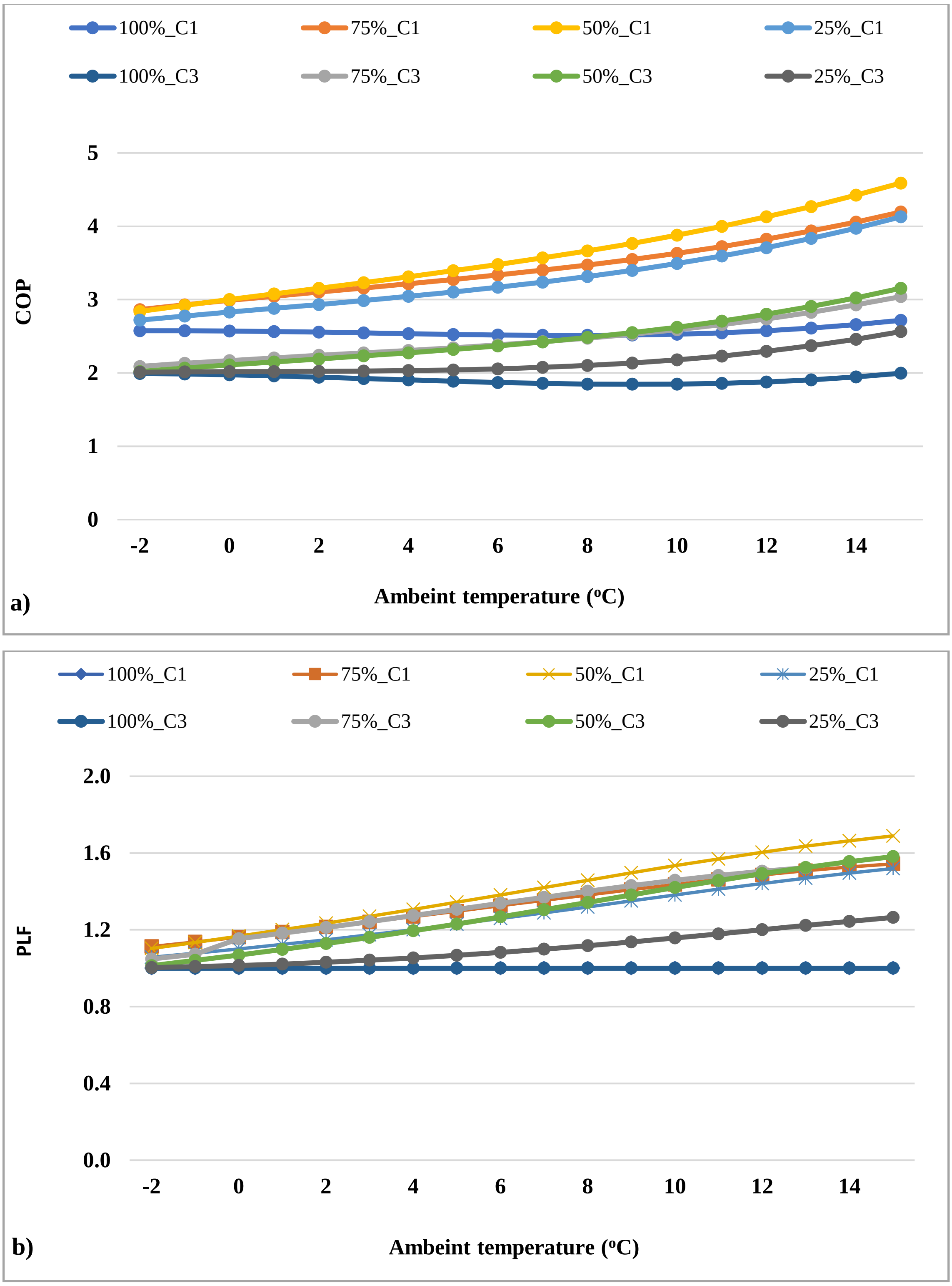
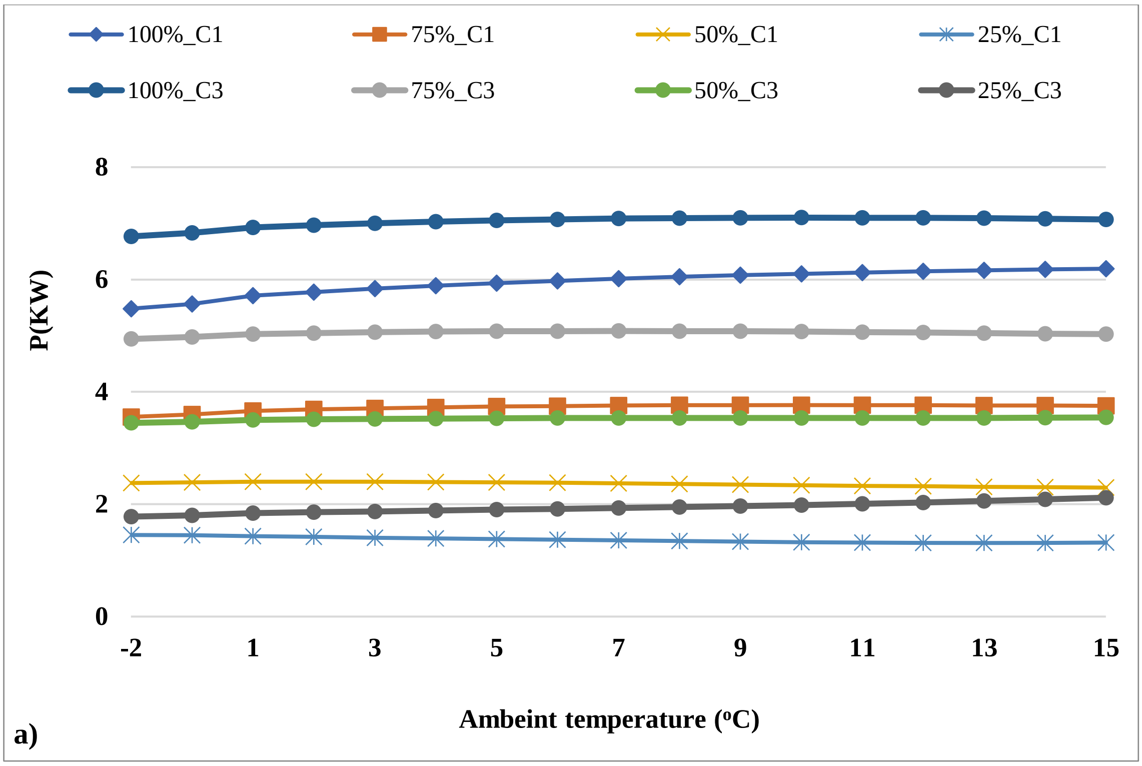


Figure 3.8 HP performance operating at different percentage (%) capacity a) COP, b) PLF (-)



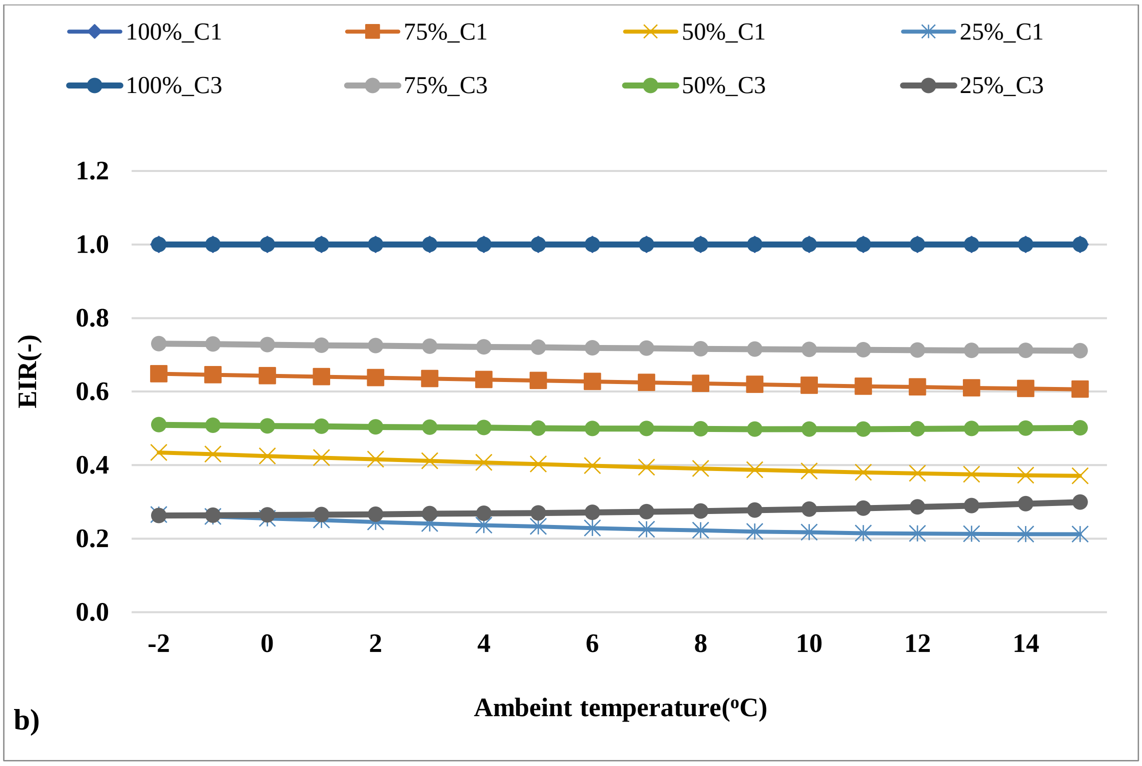


Figure 3.9 HP power consumptions at different percentage (%) capacity a) P(KW), b) EIR (-)

3.7.2 HP Part load performance

The HP characterization results at three constant heat supply temperature of 35 ºC, 45 ºC, 55 ºC named as C1, C2, and C3 respectively, four different percentage capacity, and ambient temperature conditions range could be seen from Figure 3.8 & 3.9. The 100% capacity meaning when the system is operated at 120 Hz while the 50% capacity operation represent the 60Hz operating frequency for the compressor. Figure 3.8 a & b shows the COP and the corresponding part load factor in each bin for the three cases studied at four percentage HP capacity values (25%, 50%, 75%, 100%) . Similarly, the electrical power consumption by the heat pump and the electric input ratio (EIR) were shown in Figure 3.9(a, b). The HP part load performance into different property types with varying load in each bin during VSM of operation and FSM of operation were evaluated using Equation 3.16 to 3.18, with the approach developed by other researchers [35, 42, 49].

$$PLR = \frac{Partial\ load\ capacity}{Full\ load\ capacity} = Q/Q_{full} \tag{3.16}$$

$$EIR = \frac{Consumption\ of\ power\ at\ partial\ load}{Consumption\ of\ power\ at\ full\ load\ performance} = P/P_{full} \tag{3.17}$$

$$PLF = \frac{Partial\ load\ performance}{Full\ load\ performance} = COP/COP_{full} \tag{3.18}$$

The annual COP, ratio between the total useful heat output divided by the total electric power consumption for the complete year was calculated by equation (3.19).

$$Annual\ COP = \frac{\mathrm{Q}_{total_{annual}}}{\mathrm{P}_{total_{annual}}} \tag{3.19}$$

The total electric power consumption (P) was the combination of the HP electrical energy plus the back-up electrical energy consumptions during the entire year.

3.8 Locations, property types and age factor considerations

The testing regime developed aiming to investigate the heat load applications inside the UK housng stock with different climatic conditions. The climatic conditions considered were from milder to severe conditions to predict the annual performance of the system as a domestic retrofit technology and demand side management strategy. HP performance assesment was conducted in four different climatic conditions of Valentia, Dublin, Belfast, Aviemore. The climatic conditions varies from milder to severe conditions with average hourly, maximum, and minimum,temperature parameter shown in Table 3.6. The parameters was calculated using TRNSYS 17 [101] database meteonorm weather data file. The heating degree days (HDDs) for individual locations was calculated using the base temeprature of 16 ^{o}C.

Table 3.6: Four (4) locations with climatic characteristic conditions

LOCATION	Valentia	Dublin	Belfast	Aviemore
Annual average hourly ambeint temperature (°C)	10.65	9.48	8.82	6.79
Max. hourly temperature (°C)	23.00	23.15	23.70	24.25
Min. hourly temperature (°C)	-1.55	-4.05	-4.70	-11.15
HDDs	1829	2284	2515	3252

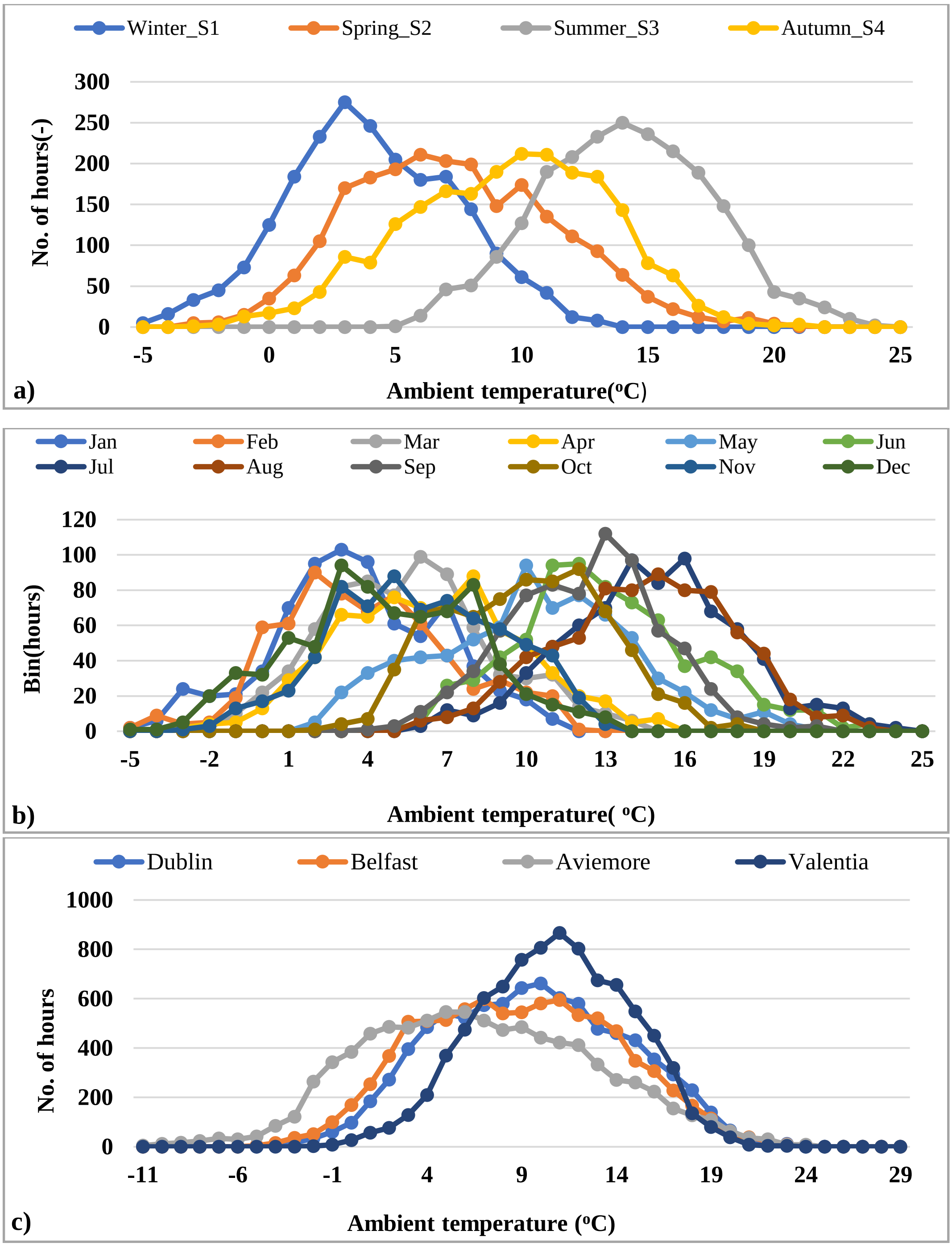


Figure 3.10 Bin distribution in Belfast climatic conditions a) seasonal, b) monthly, and annual distribution for c) all considered locations

The HP performance has been evaluated during twelve (12) months, four (4) seasons in addition to the annual performance for the Belfast climatic conditions. The seasonal (S1, S2, S3, and S4) bin distribution representing the respective seasons of Winter (Dec-Feb), Spring (Mar–May), Summer (Jun-Aug) and Autumn (Sep–Nov) and monthly bin distribution for Belfast climatic conditions is shown in Figure 3.10 (a). The annual bin distribution for the (4) other locations considered in the analysis were shown in Figure 3.10 (c).

3.8.1 Belfast monthly bin distribution

The performance of the HP in Belfast climatic conditions is investigated in more details and therefore the monthly and seasonal bin distribution was mentioned. The other climatic conditions analysis has been restricted to annual performance only. The HP performance have been evaluated during twelve (12) months and four (4) seasons to evaluate the monthly and seasonal HP performance in addition to the annual performance.

3.8.2 Property types, age period factor considerations

The impact on the HP performance due to building type, age period having different insulations and thermal characteristics is investigated by considering five property types with four age period, making the total number of Architypes twenty (20), as reported in Table 3.7**.** The five property types of Flats, Mid terrace, End terrace, Semi-detached, and Detached with age period of (1900-1949), (1950-1975), (1976-1990), (1991-2007 onwards) have been considered in the analysis because of significant percentage (approximately 90%) of these housing stock presence in Ireland, Northern Ireland, and Scotland [102].

Table 3.7: Twenty (20) building investigated according to building types & age period

	Building Type	**Building age period**
	Flats	1900-1949
	Mid terraced	1950-1975
Archetype	End terraced	1976-1990
	Semi-detached	1991-2007 onwards
	Detached	
No. of Archetype	5	4

The building regulations and the thermal characteristics (U- values) evaluation with age period have been present ed in Table 3.8 according to the age of the buildings. The building thermal characteristics have a significant impact on the annual load demands and the instantaneous power required by the HP in each bin. The benefit associated with the VSM in comparison to the FSM depends on the load demand in each bin and the over production and underproduction of heat depends on the individual load demands.

Table 3.8: Building regulation with thermal characteristics, U value (W/m2 K) evaluation with age period [103,104]

Property age	**Components**			
	Wall	Ceiling	Floor	Window/door
1900s	1.65	1.42	1.2	4.8
1970s	1	0.68	1.2	4.8
1980	1	0.68	1.2	4.8
1990s	0.6	0.35	0.51	4.8
2000s	0.45	0.35	0.51	3.1
2010s	0.30	0.20	0.22	2
2016 amendment	0.18	0.13	0.13	1.4

The average building loads demand based on experimental results [105], was adapted for the five property types according to age period duration**.** The property type with the area for different building type including Detached (142 m^2), Semi-detached (133 m^2), End terraced (124 m^2), Mid terraced (94 m^2), and Flats (75 m^2) have been considered in the analysis. The building thermal characteristics during different age periods causes the load demand variation for the same property type due to different insulation characteristics and building standards during the considered period. Based on results, detached type represents the highest heat load demand during all age period, while Flat's type representing the lowest heat demand. The building space heat (SH) loads demand have been adapted for the considered property types using building energy signature (BES) approach [35]. The heat load demand including both SH and DHW demand in each bin is depicted in Figure 3.11. The domestic hot water (DHW) demands weighted average value in percentage (%) demand were used according to the age period using the approach suggested by the researchers [106]. The additional DHW percentage (%) share of annual SH demand considered were 15%, 16%, 18%, and 27% according to the four respective age periods of (1900-1949), (1950-1975), (1976-1990), (1991-2007 onwards). The actual DHW could be impacted by many other factors including the occupancy pattern and the number of occupants, age, working class as well.

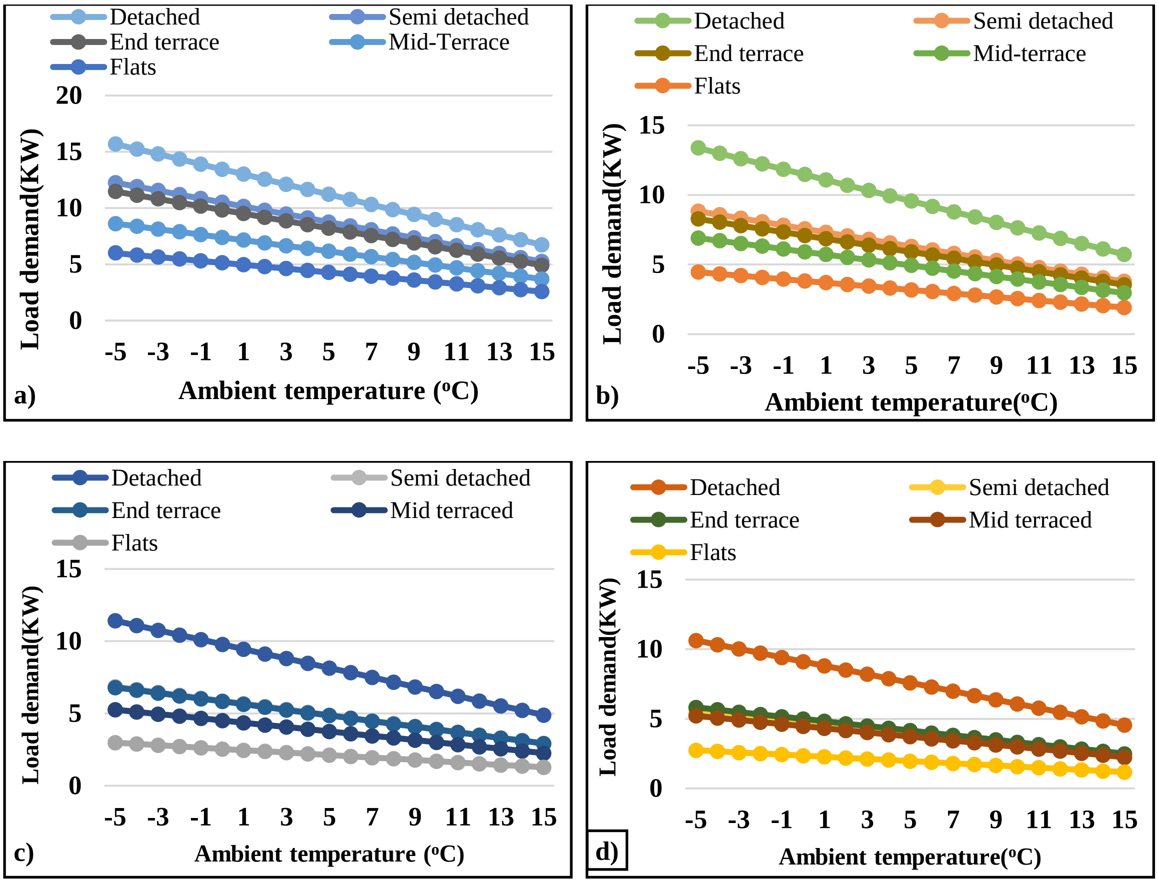


Figure 3.11 Combined (SH & DHW) heat load demand for five property types, with four age period, a)1900-1949, b) 1950-1975, c) 1976-1990, 1991-2007 onwards [87,88]

3.9 Methodology for the HP annual running cost analysis and payback period

The main issue with the HP retrofit is the HP initial capital and the payback period analysis becomes crucial in this context. The associated additional control devices cost in case of variable speed mode of control (VSM) in comparison to fixed speed mode of control (FSM), COP improvement due to control mode, energy and money savings needs investigations to justify the additional cost. Similarly, the additional heating distribution installation cost is required to assure the low heat supply temperature underfloor heating option viable in the current housing stock which are mostly installed with the high supply temperature radiators. Therefor in this research work the methodology adapted for the cost analysis and payback period is discussed here with the focus on the above mentioned three aspects; a) Initial capital cost for the HP system to replace the existing boilers, b) additional control devices cost, c) heating distribution cost. The additional cost associated with the control devices for the VSM based HP system against FSM was calculated to be £ 1000 (inverter and additional control devices) using approach suggested by [28,58], and the heating distribution installations cost as £6000 [102]. The equation used to calculate the payback period is as follows:

$$Pay\ back\ period = \frac{Additional\ cost\ due\ to\ control\ devices/heating\ distribution\ system(£)}{Annual\ cost\ savings(£)} \quad (3.20)$$

The unit capital with installation cost for 8 kW HP system was reported to be £ 8,750 with a 44% additional cost for the replacement of the existing heating distribution system. The only upgradation of existing old radiators system with advanced radiators is less expensive compared to the completed installations of the new heating distribution system, a combination of underfloor heating distribution system at the ground floor & advanced radiators at upper floor. An increase in the capacity of the existing heating distribution system is required according to the difference in the heat supply temperature from the heat pump system and the room temperature denoted as to ΔT . The higher the difference needs more relative increase in capacity to make fulfil the needs for the heat demand. Table 3.9 represent the capacity increase requirements for the heat emitters cost analysis relative to $\Delta T = 50$ at the base case of 70 ^{o}C heat supply temperature and the room temperature of 20 ^{o}C [102].

Table 3.9: Impact of heat emitter output comparison with ΔT [102]

Heat supply temperature (^{o}C)	**(ΔT) (^{o}C)**	**Heat output relative to ΔT=50 (%)**	**Capacity increases relative to $\Delta T = 50$ (^{o}C)**
55	35	63	1.6
45	25	41	2.4
35	15	21	4.8

Additionally, the annual running cost savings with the heat pump system retrofit instead of other heating technologies, i.e., oil/gas boilers, electric heater, have been considered in the analysis.

3.10 Chapter summary

In this chapter the approach used for carrying the research with the heat pump testing regime to establish the system performance map with limitations and applications for the domestic property types were presented. The numerical model validation with the experimental results and its applications for the UK housing stock considered in the analysis with the associated climatic characteristics of Ireland, Northern Ireland, and Scotland were briefly explained.

Chapter 4: Variable speed heat pump compressor performance with challenges for demand side management

4.1 Introduction

In this chapter the performance assessment and challenges with variable speed heat pump compressor for domestic heat load applications have been established under the laboratory conditions. The testing results for the developed variable speed compressor-based heat pump system under steady state conditions was presented. The study consists with four subsections: 1) Introduction, 2) testing results analysis and discussions, 3) impact of delta T on the system performance and compressor efficiencies, 4) Error analysis. The introduction highlights the importance of capacity control for the system performance improvement, with drawbacks for compressor efficiencies and inverter losses. Section 2 explain the performance results and compressor efficiencies variation with heat supply temperature and ambient temperature conditions for different heat loads demands. The HP annual COP can be improved by capacity modulation via varying compressor speed to match the heat supplied with the heat load demand inside domestic buildings due to lower cycling losses in comparison to intermittent operations. The reduction of ON/OFF cycles, continuous control over the room temperature, reduction of the extra electrical energy for compressor pressure build-up during start-up and less negative impact on the network stability were the well-established benefits associated with variable speed control. However, the speed modulation has an impact on the compressor efficiencies, pressure ratios, discharge line temperature, associated inverter losses and at only single point of the system operation highest performance could be achieved. The point where the internal compressor pressure ratios become equal to that of the external pressure ratio called the nominal point of operation has the highest compressor efficiencies, lowest discharge line temperature, while keeping the other tested conditions constant. The system performance is maximum at nominal value and degrades by operating the system above/below to that of nominal value of 60Hz. The system performance with heating capacity in the range of 3-18KW was measured, analyzed, and evaluated via the compressor isentropic & volumetric efficiencies, pressure ratio, inverter losses, discharge line temperature, electric power consumption. The compressor isentropic, volumetric efficiencies, pressure ratios, discharge line temperature, and inverter losses varies with changing steady state test conditions and contributes towards the system overall COP. The heat supply, ambient temperature conditions and the heating capacities are the major three factors causing variation to the system performance. The relationship between the varying parameters of ambient temperature, heat supply temperature, and heating capacities on electrical energy consumptions, operating speed requirements, and COP values have been established experimentally.

4.2 HP testing results at low- medium and high-water supply temperature

The HP testing results with heating capacity range of 3-18 KW, with experienced ambient temperature conditions, at low to medium and high heat supply temperature was measured and evaluated. The steady state testing results with ambient temperature conditions range from -2 ^{o}C to 15 ^{o}C at heat supply temperature of 35 ^{o}C, 45 ^{o}C, 55 ^{o}C was illustrated in Table 4.1,4.2, & 4.3 respectively. The constant heating capacity, COP values, compressor volumetric & isentropic efficiencies, inverter losses, discharge line temperature (DLT) and pressure ratios are significantly impacted due to heat supply temperature. The COP increases with lower heat supply temperature because of lower pressure ratio, discharge temperature, and improved compressor efficiencies. At fixed heat supply temperature, ambient temperature conditions the nominal speed operation at all tested conditions shows superior performance than the low and upper speed operation to that of nominal speed value. The reason for this is high compressor isentropic, volumetric efficiencies lower inverter losses, pressure ratios, and discharge line temperature. The linear degradation of the volumetric efficiencies was observed with decreasing pressure ratio for all tested conditions below/and above the nominal speed value was also evidenced from the literature [82]. The compressor isentropic, volumetric efficiency, pressure ratio, inverter losses, electric power consumption, contributes to the overall system performance. The compressor work and electrical energy requirements increases with the increase of speed of operation because of the system working on the higher amount of the refrigerant mass flow rate per volumetric displacement of compressor. The inverter losses as percentage of total electric power consumption get increased at lower heating capacities due to its poor performance at low-speed operation. The nominal frequency of 60Hz for a fixed source/sink side conditions provided higher COP was because of higher compressor isentropic, volumetric efficiencies, inverter efficiencies, and lower pressure ratios, discharge line temperature. The highest compressor isentropic efficiency calculated was 73.1%, and volumetric efficiency of 96.7% at test conditions of 15A30W (ambient temperature conditions of 15 ^{o}C and the heat/water supply temperature of 30 ^{o}C) for the heating capacity of 12KW when the operating frequency was 61 HZ. The inverter losses as a percentage of total power consumption and increases with the decrease of heating capacity due to lower overall power consumption by the compressor and increase in inverter losses at low-speed operations. Similarly, the discharge line temperature has optimal value at nominal specific test conditions.

Table 4.1: Heat Pump test results summary at lower heat supply temperature of 35 °C

Set Point T_a (°C)	HC (kW)	ΔT	$\dot{m}_w$ (kg/s)	RH (%)	T_a (°C)	ω (Hz)	P (kW)	Pr $(-)$	DLT (°C)	ε (%)	ϑ (%)	I_{loss} (%)	COP
15	18	10.06	0.43	90.25	14.91	101.35	4.94	3.62	74.08	66.56	93.58	2.83	3.64
	15	9.89	0.36	89.70	15.16	93.10	3.65	3.48	70.00	67.30	93.50	3.84	4.11
	12	9.95	0.29	91.18	15.19	62.87	2.42	2.71	59.07	72.13	95.03	5.80	4.96
	9	10.01	0.22	91.87	14.95	45.60	1.88	2.50	62.40	70.14	90.85	7.45	4.78
	6	10.08	0.14	89.34	15.11	30.04	1.29	2.39	58.83	67.88	84.78	10.86	4.64
	3	9.91	0.07	91.47	14.92	15.21	0.85	2.33	68.17	56.53	77.22	16.45	3.50
7	15	9.97	0.36	89.14	6.90	107.69	4.80	4.53	77.97	63.68	88.96	2.92	3.12
	12	9.99	0.29	88.37	6.82	85.27	3.57	4.14	71.98	65.86	89.03	3.92	3.35
	9	9.84	0.22	87.85	6.76	58.77	2.34	3.47	63.77	66.54	93.12	5.98	3.84
	6	9.86	0.14	87.91	6.79	37.78	1.57	3.19	65.60	68.08	83.61	8.89	3.80
	3	9.85	0.07	87.77	7.32	18.37	1.06	3.69	64.18	60.09	76.74	13.17	2.81
2	12	9.89	0.29	85.67	2.12	92.25	3.94	5.10	74.98	62.21	82.49	3.55	3.04
	9	9.98	0.22	87.04	2.13	67.19	2.63	4.10	67.42	66.22	89.96	5.33	3.43
	6	9.97	0.14	84.47	1.92	43.48	1.82	4.01	68.78	64.12	83.90	7.69	3.29
	3	9.85	0.07	86.10	2.19	21.10	1.07	3.72	63.81	59.39	70.00	13.07	2.79
-2	12	9.92	0.29	83.25	-2.33	102.84	4.26	5.54	78.64	59.71	75.38	3.29	2.81
	9	9.98	0.22	82.23	-2.23	74.32	2.95	4.86	72.73	63.14	83.93	4.75	3.05
	6	10.03	0.14	82.52	-2.22	48.64	1.96	4.60	70.27	63.04	79.23	7.13	3.05
	3	9.96	0.07	81.93	-1.85	23.80	1.14	4.42	67.02	57.70	78.66	12.26	2.62

Table 4.2: Heat Pump test results summary at medium water supply temperature of 45 °C

Set Point T_a (°C)	HC (kW)	ΔT	$\dot{m}_w$ (kg/s)	RH (%)	T_a (°C)	ω (Hz)	P (kW)	Pr $(-)$	DLT (°C)	ε (%)	ϑ (%)	I_{loss} (%)	COP
15	18	9.92	0.43	90.99	15.25	112.58	5.72	4.16	82.48	65.02	85.76	2.45	3.14
	15	9.98	0.36	90.12	14.87	97.02	4.37	3.87	80.86	70.76	95.57	3.20	3.43
	12	9.93	0.29	91.54	14.87	65.27	3.13	3.46	73.32	71.36	95.51	4.48	3.83
	9	9.92	0.22	90.56	14.88	47.46	2.40	3.26	73.57	70.98	93.88	5.85	3.75
	6	9.89	0.14	89.27	14.88	30.99	1.64	3.06	75.75	68.14	88.93	8.56	3.65
	3	9.96	0.07	91.54	14.87	15.52	1.11	3.08	86.73	52.12	74.86	12.58	2.68
7	15	9.85	0.36	88.07	6.82	108.33	5.56	5.42	87.90	61.65	88.69	2.52	2.64
	12	9.97	0.29	88.22	7.71	88.50	4.34	5.32	88.39	64.09	89.31	3.23	2.76
	9	10.05	0.22	88.42	6.86	59.69	2.89	4.43	78.64	70.81	92.01	4.84	3.11
	6	9.86	0.14	88.68	6.83	38.95	1.96	4.15	80.86	68.86	88.43	7.13	3.05
	3	10.08	0.07	88.81	6.95	19.15	1.23	4.02	79.36	55.43	81.31	11.39	2.43
2	12	9.97	0.29	86.46	2.10	93.84	4.60	5.78	88.89	61.24	93.61	3.04	2.61
	9	9.88	0.22	88.78	2.18	67.64	3.29	5.42	87.86	65.69	89.64	4.25	2.73
	6	9.95	0.14	85.08	2.04	44.39	2.36	5.67	90.94	59.96	88.63	5.94	2.54
	3	8.92	0.07	85.97	2.08	20.79	1.36	5.12	96.52	50.10	69.56	10.31	2.20
-2	12	10.03	0.29	82.29	-2.28	105.17	4.94	6.61	92.91	60.20	89.61	2.83	2.41
	9	9.88	0.22	83.25	-2.25	75.63	3.43	5.91	92.34	61.25	84.47	4.08	2.62
	6	9.93	0.14	80.43	-2.23	48.83	2.39	5.87	90.96	60.93	78.67	5.86	2.51
	3	9.93	0.07	81.82	-1.79	23.07	1.39	5.46	86.91	48.75	77.49	10.10	2.15

Table 4.3: Heat Pump test results summary at high water supply temperature of 55 °C.

Set Point **T_a (°C)**	**HC (kW)**	ΔT	$\dot{m}_w$ **(kg/s)**	**RH (%)**	T_a **(°C)**	ω **(Hz)**	**P (kW)**	$Pr(-)$	**DLT (°C)**	ε **(%)**	ϑ **(%)**	I_{loss} **(%)**	**COP**
15	18	10.02	0.43	92.61	14.92	114.23	6.84	4.95	97.53	64.69	85.49	2.05	2.63
	15	9.96	0.36	91.23	14.81	97.42	5.39	4.73	95.63	67.65	93.44	2.60	2.78
	12	9.93	0.29	89.82	14.43	68.64	4.18	4.50	92.82	69.84	94.75	3.35	2.87
	9	9.88	0.22	91.10	14.61	50.52	3.06	4.26	91.70	69.58	90.46	4.58	2.94
	6	9.93	0.14	89.92	14.80	32.49	2.11	3.95	95.99	66.71	89.79	6.63	2.83
	3	9.88	0.07	91.78	14.89	15.97	1.38	3.93	112.82	46.32	71.30	10.14	2.18
7	15	9.86	0.36	87.97	7.62	110.90	6.26	5.93	100.93	61.11	82.99	2.24	2.39
	12	10.00	0.29	87.32	7.08	89.69	4.93	5.90	101.95	61.87	80.73	2.84	2.43
	9	10.06	0.22	89.13	6.20	63.22	3.72	5.86	100.25	63.65	89.13	3.76	2.41
	6	9.98	0.14	87.87	6.36	40.00	2.46	5.30	104.41	65.85	84.19	5.68	2.43
	3	9.96	0.07	88.21	6.59	19.85	1.31	5.50	108.01	49.86	72.06	10.72	2.28
2	12	9.98	0.29	86.85	2.21	94.52	5.43	7.05	106.71	57.16	88.96	2.58	2.21
	9	9.91	0.22	87.93	2.02	69.37	4.07	6.88	107.72	63.57	89.96	3.44	2.21
	6	9.97	0.14	85.45	1.88	44.83	2.70	6.39	107.62	61.69	84.91	5.19	2.22
	3	9.95	0.07	85.19	2.05	24.34	1.37	5.99	111.83	48.67	69.45	10.24	2.14
-2	9	9.91	0.22	81.69	-2.23	81.25	7.07	7.88	101.54	54.73	79.50	1.98	1.27
	6	9.79	0.14	81.93	-2.22	54.53	4.87	7.85	102.46	56.03	76.15	2.88	1.23

4.3 Result analysis

4.3.1 Coefficient of performance (COP)

4.3.1.1 Coefficient of performance (COP): function of supply temperature model

The COP values varied mainly according to the heating capacity, heat supply temperature, and ambient temperature conditions. The general relations mapped into a model based on experimental results with heat supply temperature (denoted as Tw), and ambient temperature (Ta), and operating frequency ($\boldsymbol{\omega}$) with the following equations.

$$COP_{model} = 6.78 - 0.15T_w + 0.00127{T_w}^2 + 0.134T_a - 00244T_w * T_a + 0.0024{T_a}^2 \quad (4.1)$$

The relationship of COP with other heating capacity, heat supply temperature and ambient temperature conditions was presented in the following subsections.

4.3.1.2 COP vs. heating capacity

The COP values varies according to the heating capacity at any constant test conditions, as the operating frequency has a linear relationship to the heating capacity. The electrical power consumptions increase linearly with the increase in heating capacity due to higher refrigerant mass flow rates and proportion between the heat production and power causes an increase in COP up to 60 Hz and beyond that resulting into higher power consumption proportion in comparison to heat production where degradation of COP starts. The relationship between COP and heating capacity at varying water/heat supply temperature from the heat pump system at four tested ambient temperature have been depicted in Figure 4. 1.

Figure 4.1 COP values with varying heat supply temperature and ambient temperature of a) 15 ^{o}C, b) 7 oC, c) 2 ^{o}C d) -2 ^{o}C

At any fixed ambient temperature conditions, the COP values against the heating capacity, at heat supply temperature of 30 ^{o}C, 35 ^{o}C, 40 ^{o}C, 45 ^{o}C, 50 ^{o}C, 55 ^{o}C shows a decreasing trend. The COP relationship could be seen for the range of tested heat supply temperature for any fixed ambient temperature conditions from equation 4.1. The COP values are higher at nominal heating capacity of 9KW in comparison to the highest tested heating capacity of 18KW and lowest tested capacity of 3KW due to operating frequency of nominal value for any individual fixed conditions. The heating capacity, falling closer to the nominal value of frequency of 60Hz have higher COP values while keeping other variables of heat supply temperature and ambient temperature conditions constant. The corresponding frequency values against the tested heating capacities values were shown in Figure 4.2. The frequency values required for the specific heating capacity demand provides valuable information to operate the system based on the market signal and availability of the renewable energy system in the grid. The heat supply temperature could also be used to achieve the required heating capacity with optimized COP values. The system performance become poor at higher frequency above the nominal value of 60 Hz and become worst at lower values below 30Hz due to very poor performance of compressor isentropic efficiency.

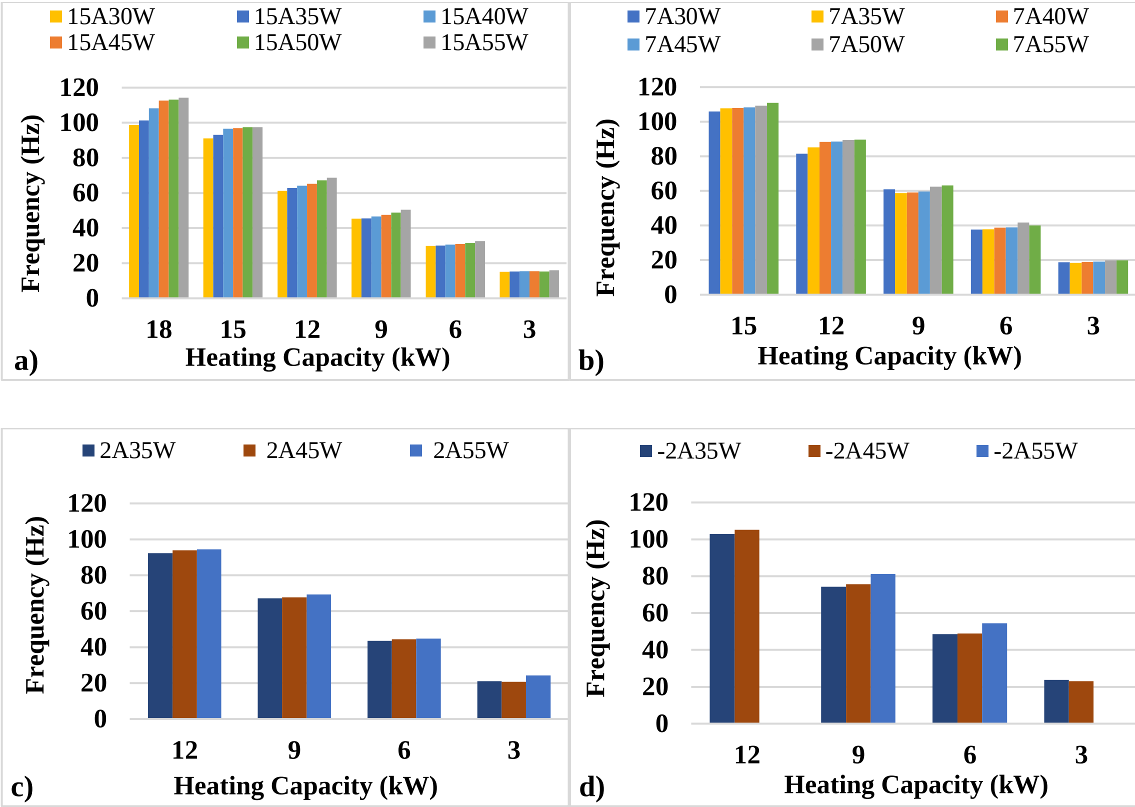


Figure 4.2 Frequency values with varying heat supply temperature and ambient temperature of a) 15 °C, b) 7 °C, c) 2 °C d) -2 °C

The system performance is worst in all tested conditions at lowest frequency of operation, below nominal value compared to the highest frequency of operation (above the nominal value) except at the tests developed at higher ambient conditions of 15 °C. Therefore, it is more beneficial to operate the system at nominal frequency of 60Hz and above frequency values, and to avoid low speed operation below 30Hz at low ambient temperature conditions. At lower ambient conditions with the increase of building heat demand, the system could be operated in more efficient method. The compressor isentropic and volumetric efficiencies, and pressure ratios, heat losses are responsible for the lower performance with upper and lower operating frequency. On other hand, at lower heat capacity the system becomes less efficient because of the poor performance at low speed. Therefore, low speed operation is more beneficial only at higher ambient temperature conditions to avoid extra cycling losses and improve the comfort level by matching load demand to heat supplied, and higher speed operation at low ambient temperature for improvement in thermal comfort. At low ambient temperature where the house load demand increases and to avoid the electric heater requirements it is more beneficial to utilize the high-speed operation. The heat pump operation at higher speed with higher ambient temperature conditions because of the lower house heating demand, becomes unrealistic.

4.3.1.3 COP vs. heat supply temperature

Increasing heat supply temperature has corresponding impact on the individual heating demands at the same ambient test conditions. The low to medium and high heat supply temperature relationship with could be observed from Figure 4.1(a, b, c, d) at respective fixed ambient temperature of 15 °C, 7 °C, 2 °C and -2 °C. The trend for COP is higher at lower heat supply temperature compared to the higher heat supplied temperature due to lower compressor efficiencies and discharge line temperature. The relationship between COP and the heat supply temperature could be observed in more detail with equation 4.1. The COP reduces with the increase of supply temperature for the tested fixed heating capacity range(3-18KW). Based on the experimental results shown here the annual COP for a range of heating loads requirements according to the building age period and property type have been predicted using the developed validated model. The COP get reduced with increasing supply temperature for any constant fixed heating capacity and ambient conditions due to frequency variation and the associated impact on the compressor isentropic and volumetric efficiencies, discharge line temperature increase, pressure ratio and inverter losses. Heat supply temperature reduction for every single 1 °C could improve the heat pump performance by 1-3%. The maximum COP value for the tested ambient conditions of 15 °C was 5.34 at frequency of 61.18Hz with supply temperature of 30 °C for the heating capacity of 12KW. For the same 12kW heating capacity but at higher heat supply temperature of 35 °C, 40 °C, 45 °C, 50 °C, 55 °C the maximum COP for the tested heating capacities were 4.96, 4.36, 3.83, 3.34, and 2.94 with the corresponding increasing demand of operating frequency of 62.87Hz, 64.06Hz, 65.27Hz, 67.27Hz, 68.64Hz respectively. The COP values for heating capacities of 18KW, 15KW, 12KW, 9KW, 6KW, 3KW varies in the range of 3.91-2.63, 4.72-2.78, 5.34-2.87, 5.31-2.94, 5.23-2.83, 4.01-2.18 for a WST variation of 35 °C, 40 °C, 45 °C, 50 °C, 55 °C at ambient conditions of 15 °C (Figure 4.1a). While the frequency variation range for individual heating capacity of 18KW, 15KW, 12KW, 9KW, 6kW, 3KW is from 98.72-114.23Hz, 91.1- 97.42Hz, 61.18-68.64Hz, 45.27-50.52Hz, 30.04-32.49Hz, 15.09-15.97Hz for the increasing WST 35 °C, 40 °C, 45 °C, 50 °C, 55 °C at constant ambient temperature conditions (Figure 4.2a). Interesting to note the frequency variation range with supply temperature from 35 °C-55 °C, with higher values for larger heating capacities and it could be extracted that changing WST has stronger impact in terms of frequency variation for the higher heating capacities than the smaller heating capacity values. The percentage reduction in COP values for each heat capacity is 32.73%, 41.1%, 46.25%, 44.63%, 45.88%, 45.63% by increasing WST values from 30 ^{o}C to 55 ^{o}C. The operation of the compressor at nominal frequency value of 60 Hz shows comparatively high COP values while keeping other conditions constant. For example, at WST of 30 ^{o}C the frequency for the six heating capacities is 98.72Hz, 91.19 Hz, 61.18Hz, 45.27Hz, 29.78Hz, 15.08 Hz with corresponding COP values 3.92, 4.72, 5.34, 5.31, 5.23, 4.01 with the highest COP value when the frequency is 61.18 Hz. The maximum COP value for the tested ambient conditions of 7 ^{o}C as can be seen from Figure 4.1b, obtained was 4.35 at frequency of 61.05 Hz with WST of 30 ^{o}C for the heating capacity of 9kW. For the same ambient air test conditions but with higher heat supply temperature of 35 ^{o}C, 40 ^{o}C, 45 ^{o}C, 50 ^{o}C, 55 ^{o}C the highest COP for the tested heating capacities were 3.84, 3.46, 3.11, 2.75, and 2.43 with the corresponding frequency operation of 58.77Hz, 59.03 Hz, 59.69Hz, 62.48Hz, 63.65 Hz respectively (Figure 4.2b). At ambient temperature conditions of 7 ^{o}C the COP values for each heating capacities of 15kW, 12kW, 9kW, 6kW, 3kW variation range is 3.56-2.43, 3.83-2.43, 4.35-2.41, 4.19-2.43, 3.39-2.28 for the WST variation range of 30 ^{o}C, to 55 ^{o}C respectively (Fig.3b). The corresponding percentage reduction in the COP values were 31.74%, 36.55%, 44.59%, 42 %, 32.74% with increasing WST temperature. While the frequency of operation for the tested heating capacities are in the range of 105.87Hz-110.9 Hz, 81.57Hz-89.69Hz, 61.05Hz- 63.22Hz, 37.65Hz-40Hz, 18.75Hz-19.85 Hz for the WST variation range 30 ^{o}C to 55 ^{o}C (Figure 4.2 b). The frequency of operation for 35 ^{o}C WST at ambient conditions of 7 ^{o}C and heat demands of 15kW, 12kW, 9kW, 6kW, 3kW are 107.69Hz, 85.27Hz, 58.77Hz, 37.78 Hz, 21.73Hz with corresponding COP values of 3.12, 3.35, 3.84, 3.80, 2.81 with the highest value of COP when the frequency was near to the nominal value of 60Hz. The same trend was noted for higher water supply temperature but with lower COP values. Figure 4.1 (c, d) shows the COP and Fig.4.2 (c, d) shows the frequency at 2 ^{o}C & -2 ^{o}C with heating demands of 12KW, 9KW, 6KW, 3KW and with WST of 35 ^{o}C, 45 ^{o}C, 55 ^{o}C. The maximum COP value for the tested ambient conditions of 2 ^{o}C obtained was 3.43 at frequency of 67.19 Hz with WST of 35 ^{o}C for the heating capacity of 9KW. For the same ambient air test conditions but with higher water supply temperature (WST) of 45 ^{o}C, 55 ^{o}C the highest CoP for the tested heating capacities were 2.73 and 2.22 with the corresponding frequency of operation of 67.64Hz, 44.83Hz respectively (Figure 4.2 c). The COP values for 12KW, 9KW,6KW, 3KW for a supply temperature of 35 °C- 55 °C are in the range of 3.04-2.21, 3.43-2.21, 3.29-2.22, 2.79-2.14 respectively at 2 °C (Figure 4.1c). While the corresponding operation range for the frequency are 92.25Hz- 93.52Hz, 67.19Hz-69.37Hz, 21.1Hz-24.34 Hz respectively (Figure 4.2c). The frequency of operation for 12kW, 9kW,6kW, 3kW at 2 °C ambient temperature conditions, and constant WST of 35 °C was 92.24 Hz, 67.18Hz, 43.47 Hz, 21.1 Hz with absolute percentage difference of 53.73%, 11.96%, 27.55%, 64.85% to that of nominal frequency of 60Hz. The highest value of COP is when the absolute percentage difference is the lowest. Similarly for WST of 45 °C the frequencies of operation are 93.84 Hz, 67.64 Hz, 44.39 Hz, and 20.7 Hz with an absolute percentage difference of 56.4%, 12.73%, 26.01%, 65.5% respectively with the highest value of COP when the percentage difference of frequency of operation is lowest to that of nominal frequency of 60 Hz. The tests performance results (COP) with the frequency variation at this lower ambient condition can be seen from Figure 4.1d & Figure 4.2d. The maximum COP value at -2 °C obtained was 3.05 when the WST was 35 °C. The COP values at -2 °C ambient temperature conditions for 12kW, 9kW,6kW, 3kW are 2.81& 2.41, 3.04 & 2.61, 3.05 & 2.50, 2.61& 2.15 for WST of 35 °C, & 45 °C respectively. The frequency variation range for the individual heating demand tests are 102.84-105.17Hz, 74.32 -81.24Hz, 48.64 -54.52Hz, 23.80-23.05Hz.

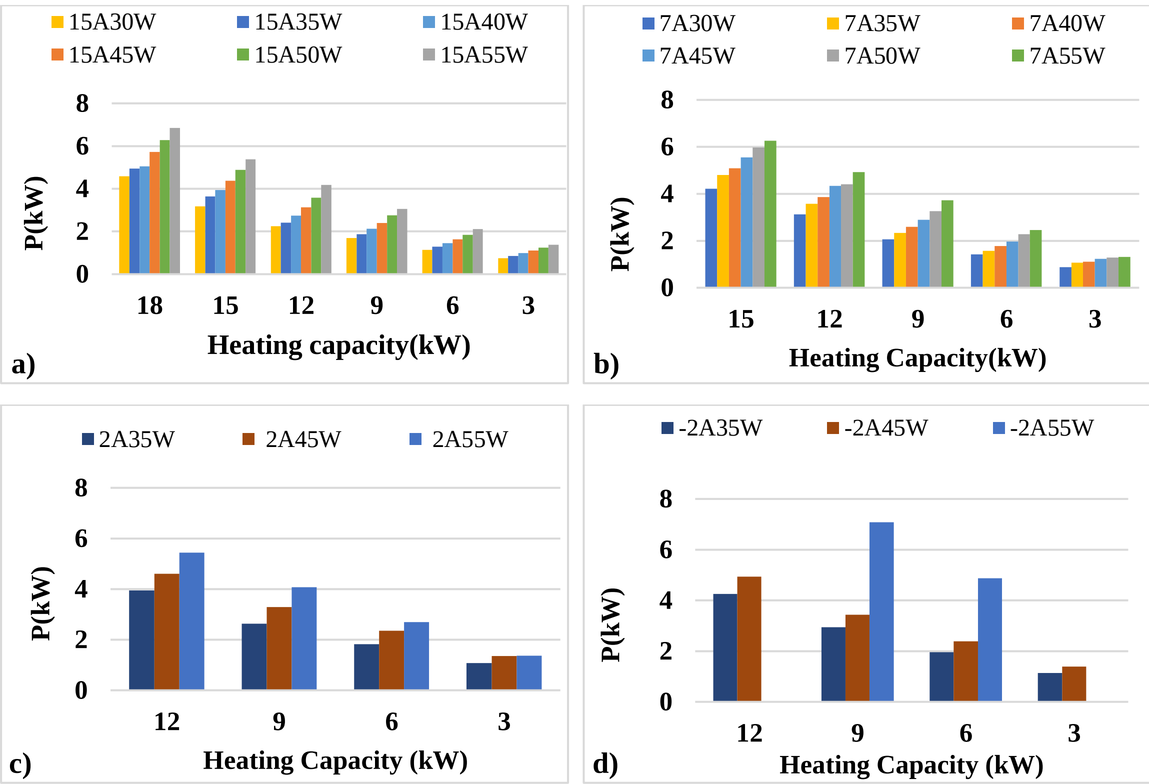


Figure 4.3 Electric power values with varying heat supply temperature and ambient temperature of a) 15 °C, b) 7 °C, c) 2 °C d) -2 °C

The other characteristics properties like power consumptions, pressure ratios, refrigerant mass flow rates, isentropic efficiencies, volumetric efficiencies, inverter losses, and discharge line temperature at different ambient temperature conditions with low to medium and high supply temperature are shown in Figure 4.3 - 4.9. The COP values presented earlier were based on the combined impact of these important system characteristic properties and explained in the following subsections with more details.

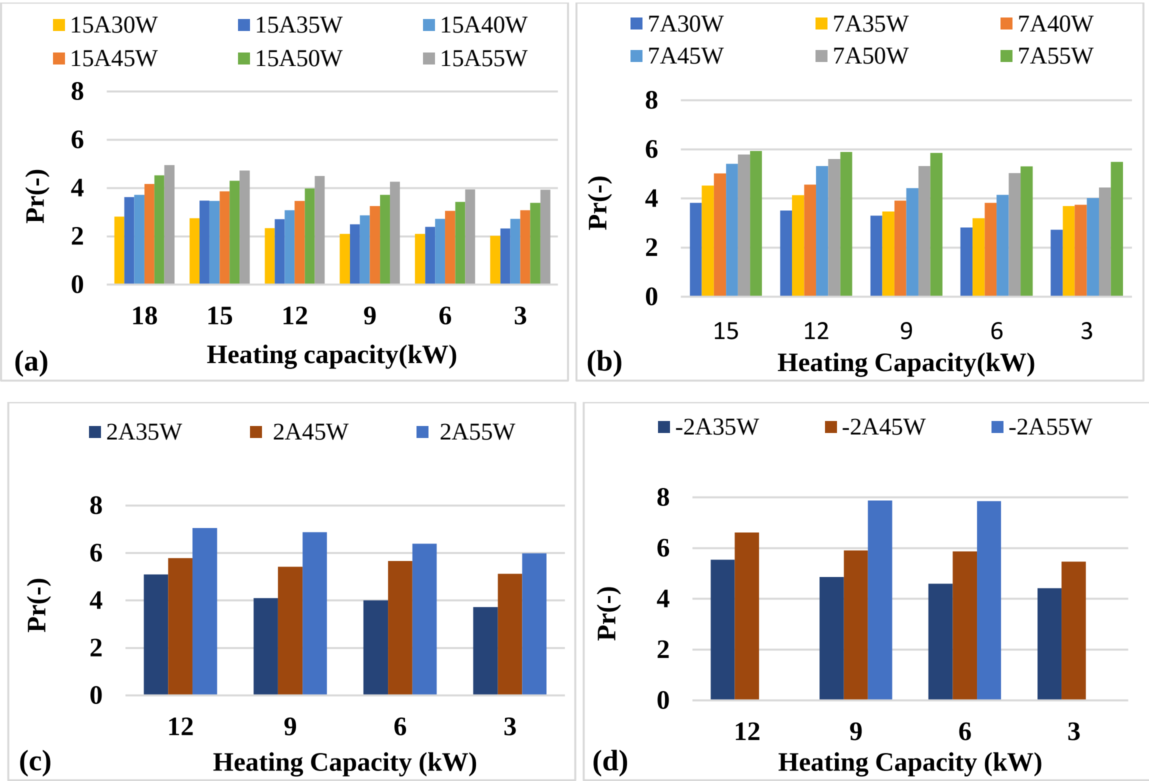
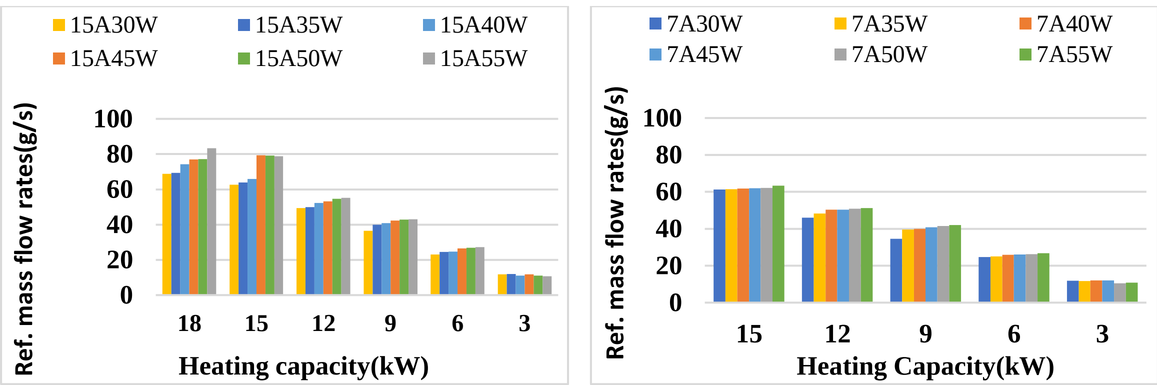


Figure 4.4 Pressure ratio values with varying heat supply temperature and ambient temperature of a) 15 °C, b) 7 °C, c) 2 °C d) -2 °C



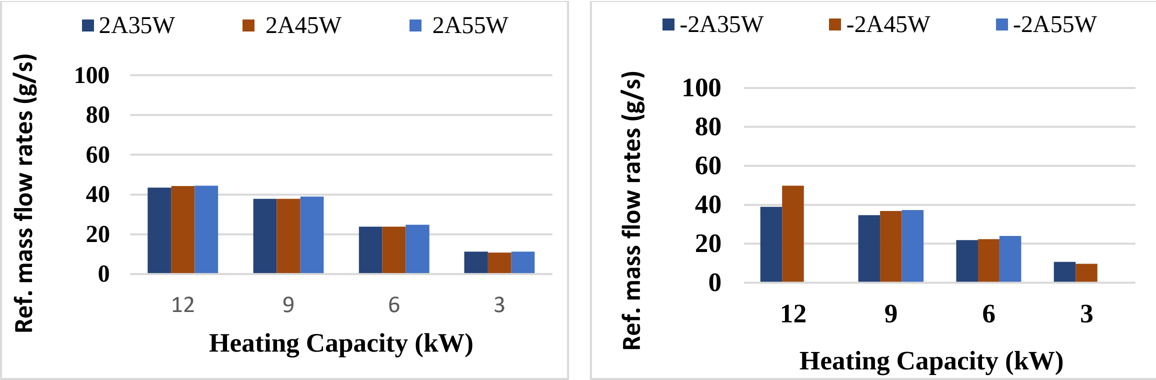


Figure 4.5 Ref. mass flow rates (g/s) values with varying heat supply temperature and a) 15 °C, b) 7 °C, c) 2 °C d) -2 °C

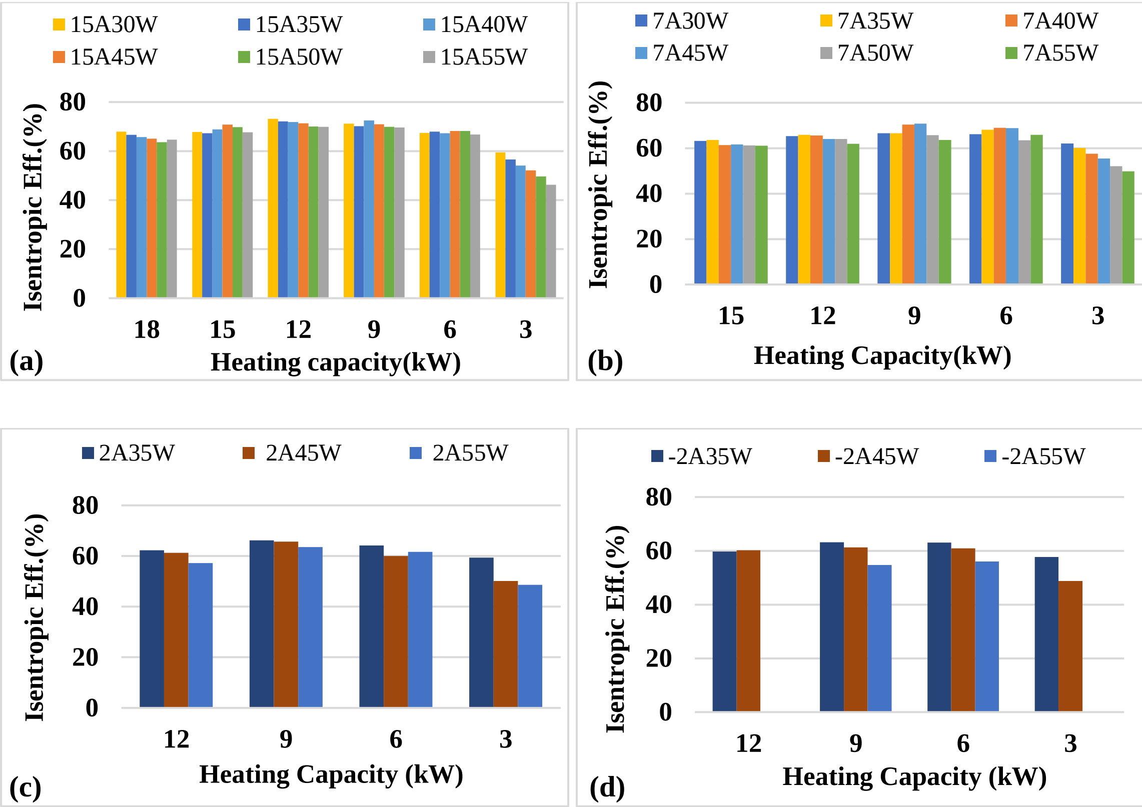
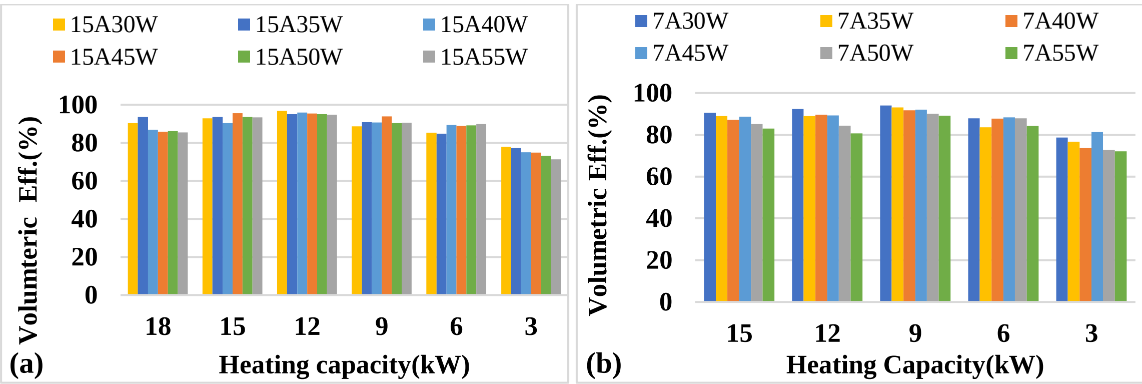


Figure 4.6 . Isentropic efficiency values with varying heat supply temperature and ambient temperature of a) 15 °C, b) 7 °C, c) 2 °C d) -2 °C



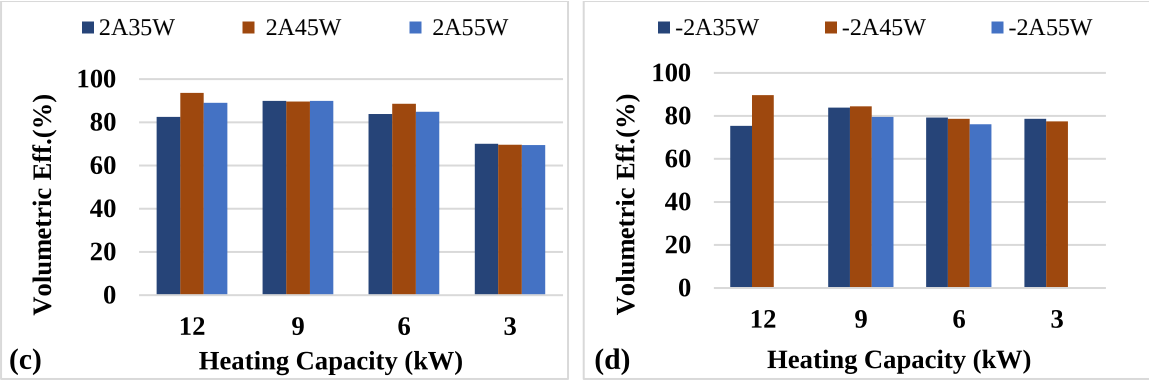


Figure 4.7 Volumetric efficiency values with varying heat supply temperature and ambient temperature of a) 15 °C, b) 7 °C, c) 2 °C d) -2 °C

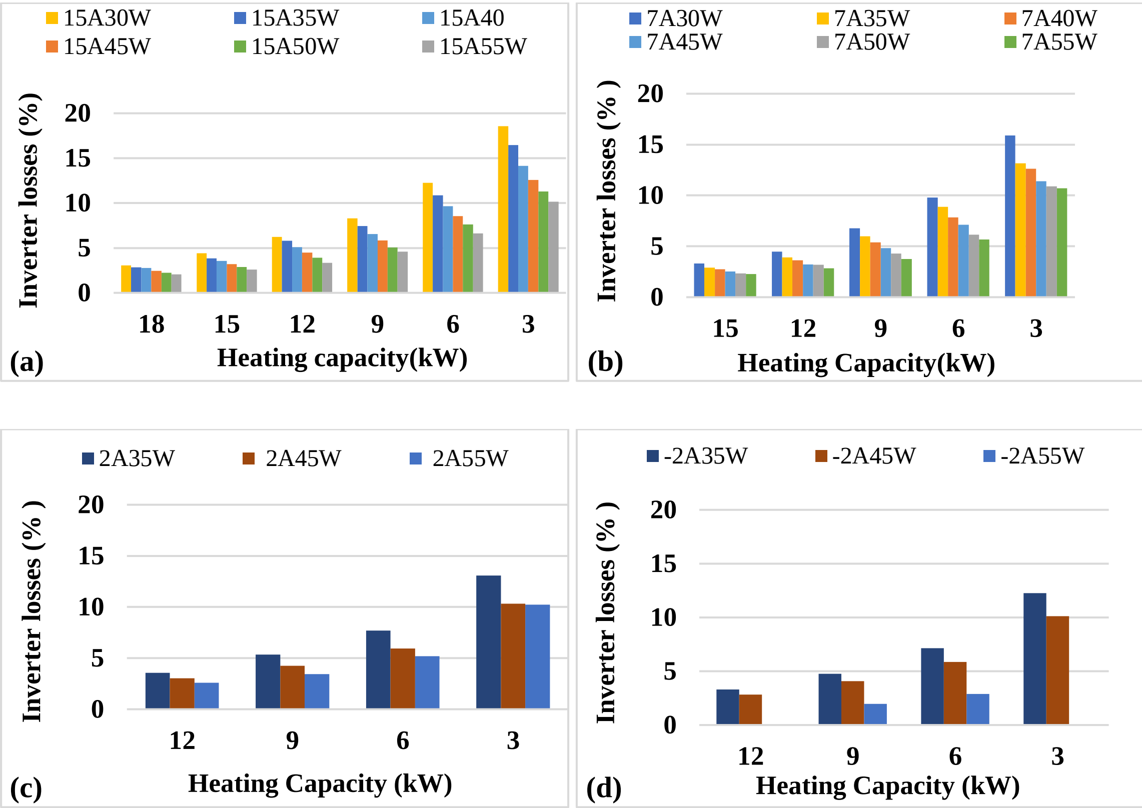
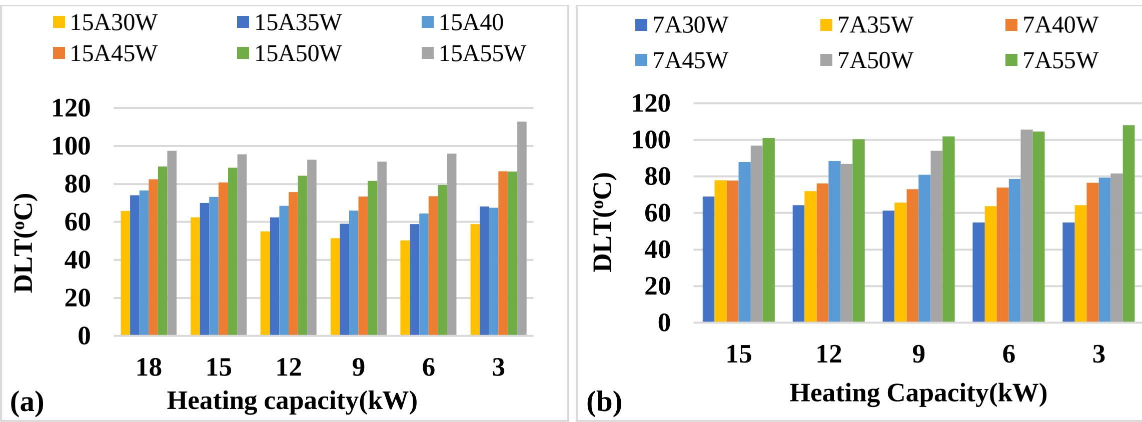


Figure 4.8 Inverter losses values with varying heat supply temperature and ambient temperature of a) 15 °C, b) 7 °C, c) 2 °C d) -2 °C



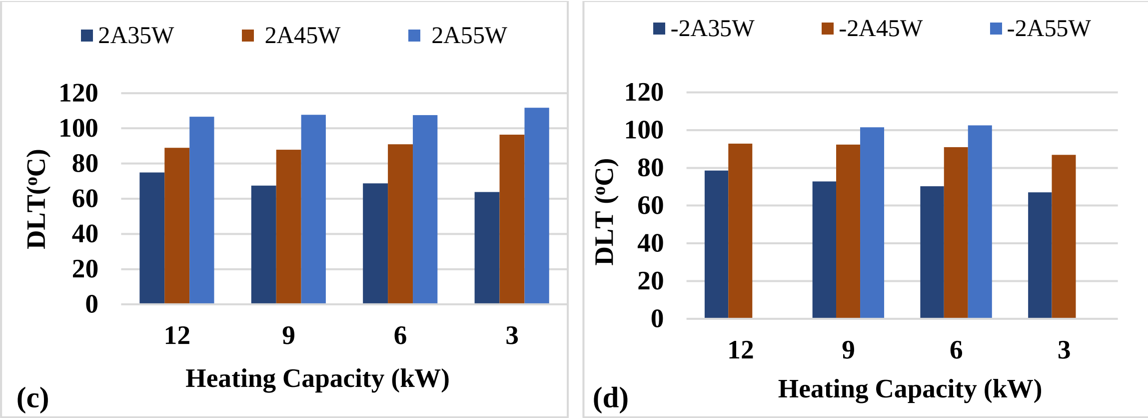
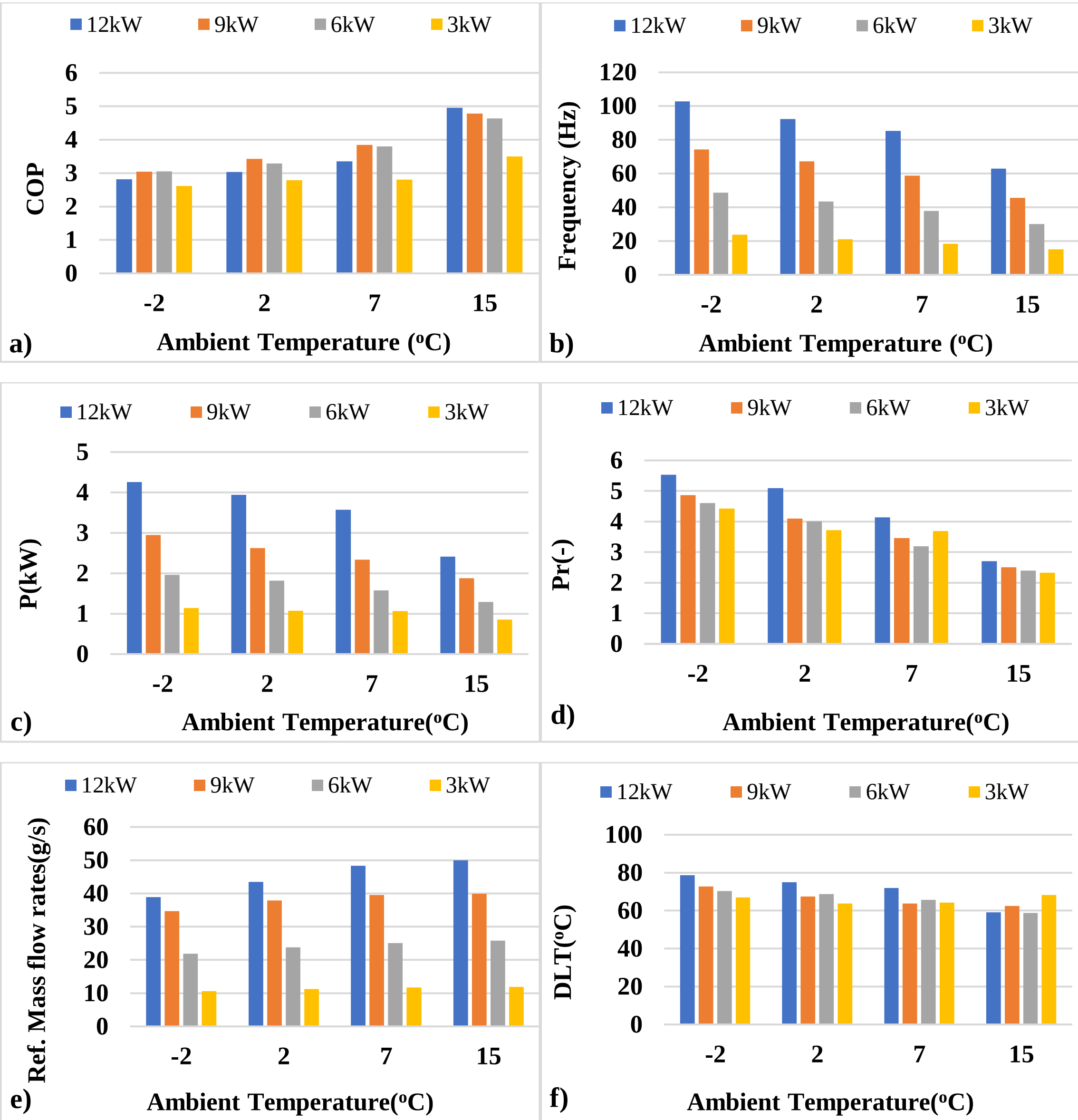


Figure 4.9 DLT values with varying heat supply temperature and ambient temperature of a) 15 ºC, b) 7 ºC, c) 2 ºC d) -2 ºC

4.3.1.4 COP vs. ambient temperature

The impact of three level of heat supply temperature on the system characteristic properties according to the ambient temperature conditions variations could be seen from Figure 4.10 - 4.12. The COP has direct relationship with the ambient temperature, while keeping other variables of supply temperature, and frequency constant. However, the rate of increase in COP value with the increase of ambient temperature varies according to the heat supply temperature value and frequency of operation. At constant frequency with above/below the nominal value lower gradient is observed for increase in COP with ambient temperature. The COP values and the frequency requirements, compressor efficiencies, and other important parameters against the changing ambient temperature conditions of -2 ºC, 2 ºC, 7 ºC, 15 ºC at three heat supply temperature (35 ºC, of 45 ºC, 55 ºC) is presented in this section. The corresponding impact of frequency variation, power consumptions, pressure ratios, refrigerant mass flow rates, discharge line temperature, volumetric and isentropic efficiencies. The COP values trends over the lower supply temperature is higher compared to medium and higher heat supply temperature. The COP values for 35 ºC WST (fixed heat load i.e., 9kW) at ambient conditions of -2 ºC, 2 ºC, 7 ºC, 15 ºC are 3.04, 3.43, 3.84, and 4.78. The percentage change in the COP values from -2 ºC to 15 ºC are 57.23 %. The frequency of operation for meeting 12 KW, 9KW, 6KW, 3KW heating demands for WST of 35 ºC at different ambient conditions are shown in Figure 4.10b. The COP values for a fixed ambient condition depend on the frequency of operation percentage difference with the nominal frequency of 60 Hz. The COP values for -2 ºC and heating capacities of 12KW, 9KW, 6KW, 3KW are 2.81, 3.04, 3.05, and 2.62 and the corresponding frequency of operation are 102.84Hz, 74.32 Hz, 48.63 Hz, 23.8 Hz. Similar trends can be seen at other fixed ambient conditions of 2 °C, 7 °C, 15 °C, but with higher COP values. The frequency of operation at 7 °C for 12KW, 9KW,6KW, and 3KW are 85.27Hz, 58.77 Hz, 37.77 Hz, 21.73 Hz and the corresponding COP values of 3.35, 3.84, 3.80, and 2.81 with the highest value of 3.84 at 58.77 Hz (9KW). Similarly, for constant heat load (i.e., 9KW), COP values at ambient conditions of -2 °C, 2 °C, 7 °C, 15 °C with 45 °C WST are 2.62, 2.72, 3.11, and 3.75. The percentage increase in the COP values for changing ambient temperature from -2 °C to 15 °C are 43.12 % for the fixed load of 9KW and WST of 45 °C. It is worth to note that the percentage increase in COP values from -2 to 15 °C ambient conditions for fixed heat load (i.e., 9KW) is higher at lower WST of 35 °C values than 45 °C. For WST of 55 °C, 45 °C, 35 °C percentage change in COP values for changing ambient conditions from 2 °C to 15 °C are 28.05%, 37.36%. and 39.35% for a constant load of 9KW.



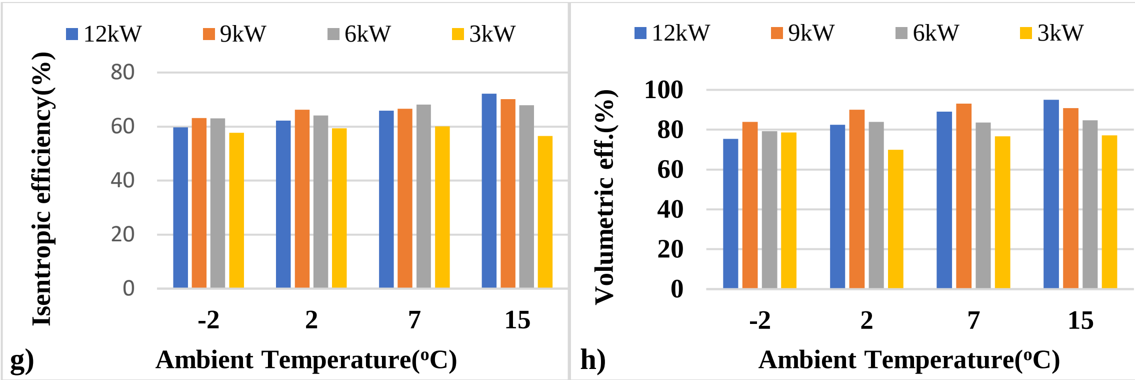
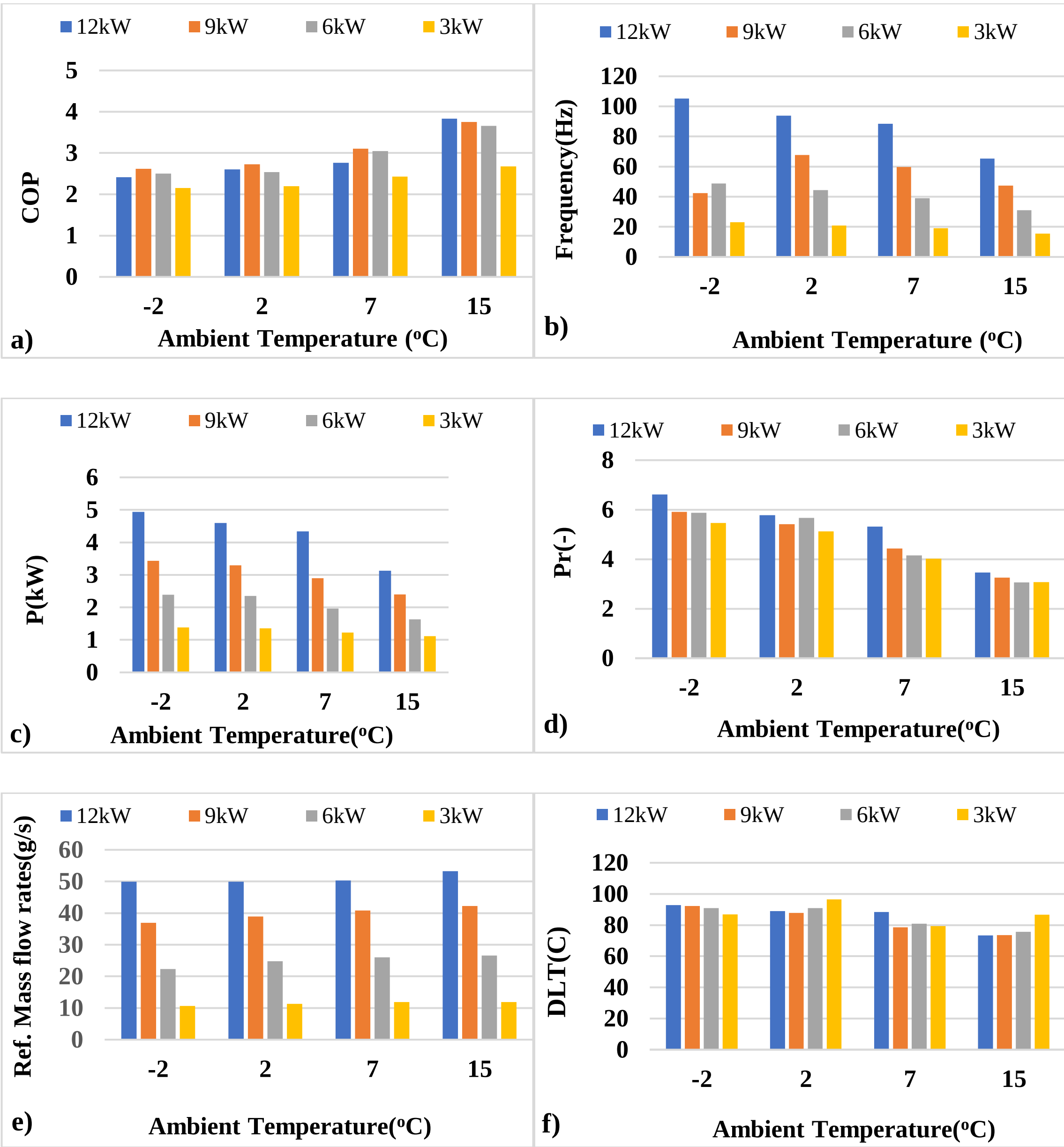


Figure 4.10 HP characteristics properties at low (35 ºC) heat supply temperature a) COP, b) ω (Hz), c) P(KW), d) PR, e) m˙_r (g/s), f) DLT (ºC), g) ε(%), h) θ(%)



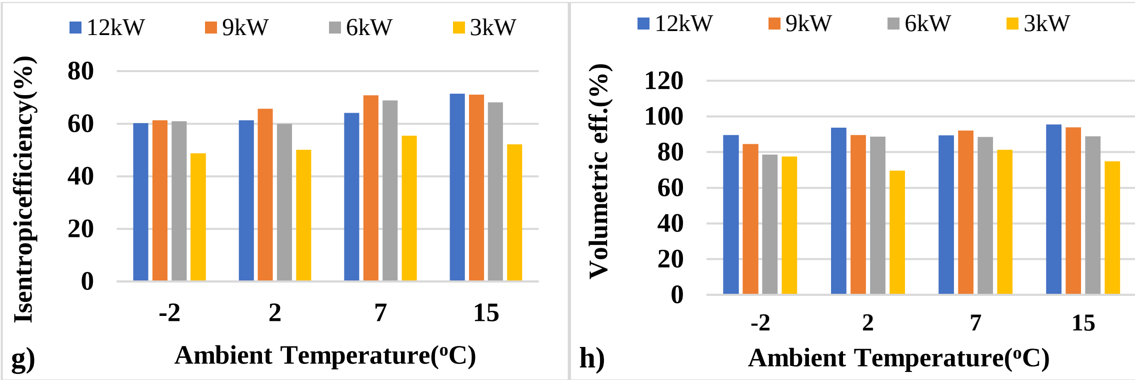
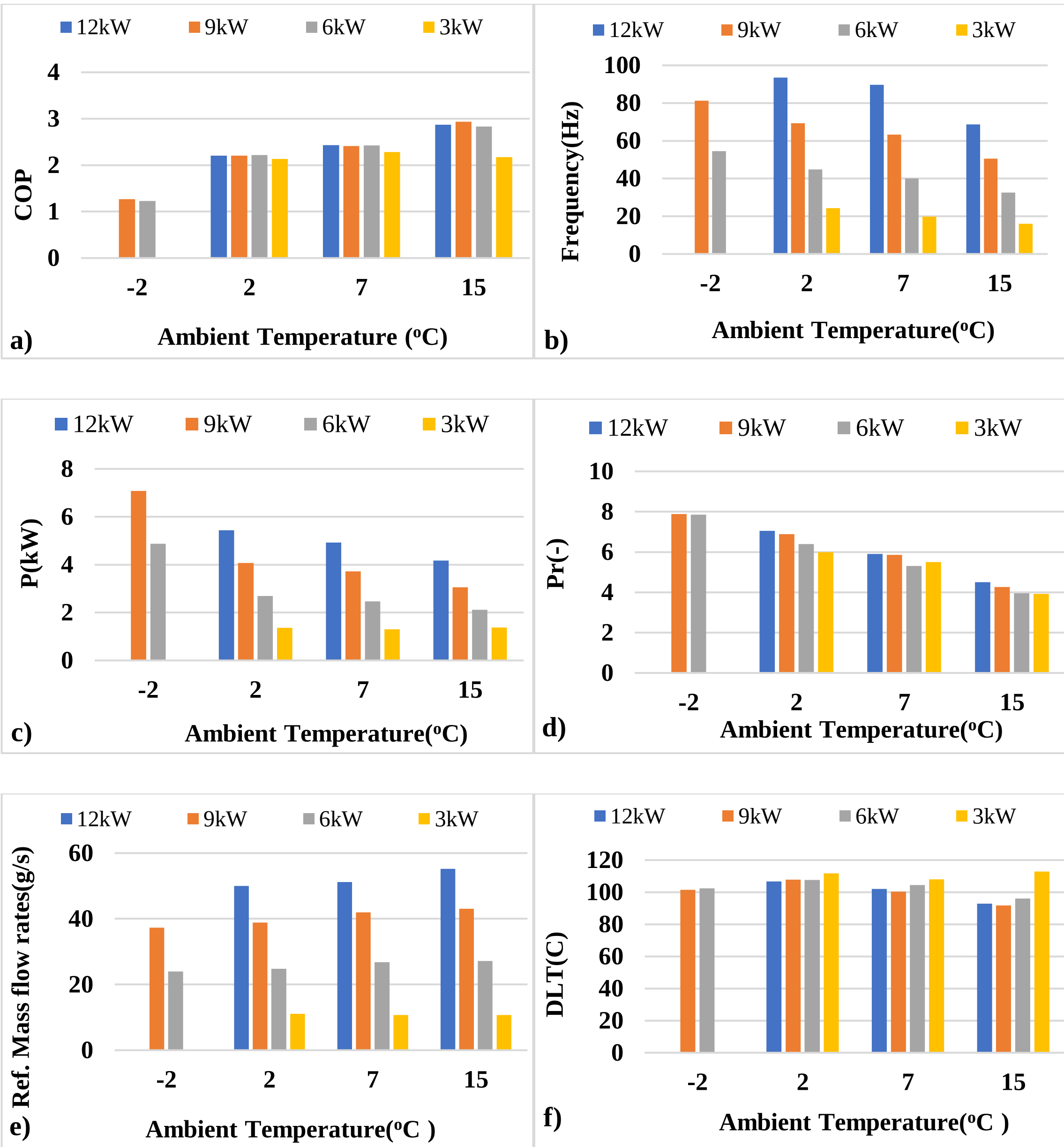


Figure 4.11 . HP characteristics properties at medium (45 ºC) heat supply temperature a) COP, b) ω (Hz), c) P(KW), d) PR, e) $\dot{m}\_r$ (g/s), f) DLT (ºC), g) ε(%), h) θ(%)



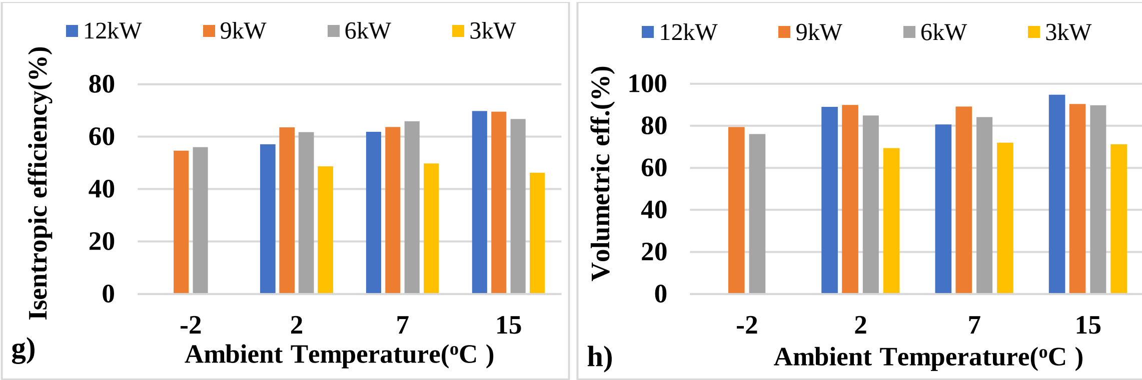


Figure 4.12 HP characteristics properties at high (55 ºC) heat supply temperature a) COP, b) ω (Hz), c) P(KW), d) PR, e) m˙_r (g/s), f) DLT (ºC), g) ε(%), h) θ(%)

4.3.2 Heating Capacity

4.3.2.1 Heating Capacity vs. frequency

The heating capacity has approximately linear relationship with the frequency of operation, and the heat production increases with the increase of the compressor speed, but with a reduced rate at higher frequency at constant heat supply &ambient temperature conditions. Figure 4.13 shows the heating capacity vs. frequency. The fixed heating capacities requires range of operating speed according to the supply and ambient temperature conditions.

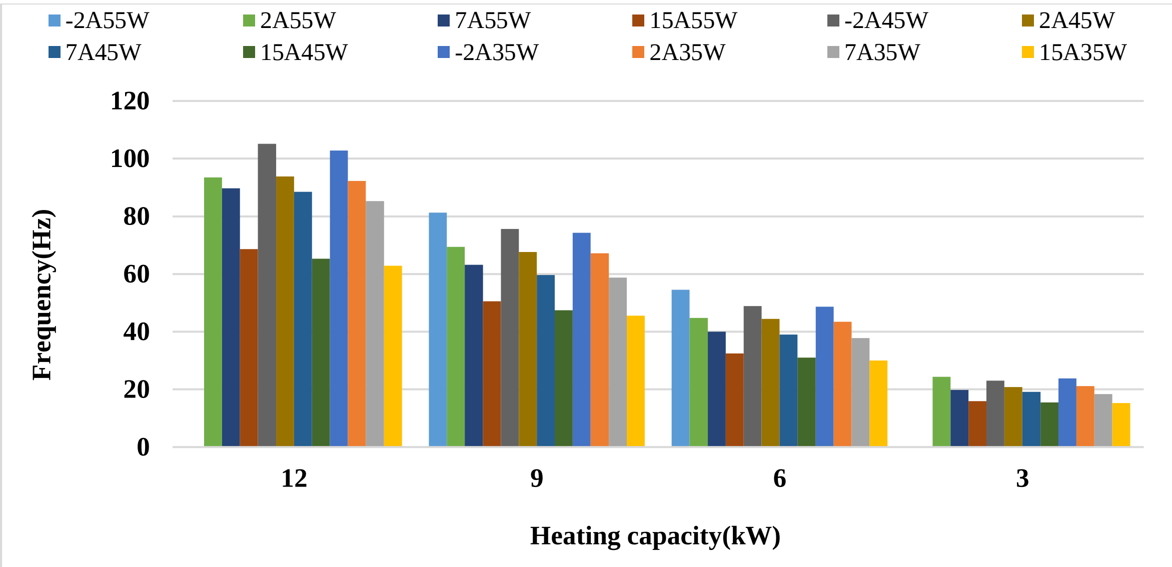


Figure 4.13 Heating Capacity vs frequency (Hz)

4.3.2.2 Heating capacity (Frequency) vs. heat supply temperature

Required frequency has inverse relationship with increasing heat supply temperature, and due to increase in heat supply temperature the operating speed requirement increases to maintain constant heating capacity (Figure 4.13). The heating capacity of 12KW, 9KW, 6KW, and 3KW with constant WST of 35 ^{o}C and ambient temperature of 7 ^{o}C required frequency of operation was 85.27Hz, 58.77Hz, 37.78Hz, and 18.37Hz respectively. The frequency requirements for the same heating capacities are higher with the values of are 88.5Hz, 59.69Hz, 38.95Hz, 19.15Hz at the same ambient temperature conditions but with WST of 45 ^{o}C and further increases when the WST increases to 55 ^{o}C.

4.3.2.3 Heating capacity (Frequency) vs. ambient temperature

The frequency required to maintain the same heating capacity increases with decrease in ambient temperature conditions at constant any constant heat supply temperature (Figure 4.13). The heating capacity get reduced at lower ambient temperature conditions and higher frequency is required to maintain the same heating demand. The heating capacity of 12KW, with WST of 45 ^{o}C requires 65.27Hz, 88.5Hz, 93.84Hz, 105.17Hz at ambient temperature conditions of 15 ^{o}C, 7 ^{o}C, 2 ^{o}C, and -2 ^{o}C respectively. Similar trends have been shown for other heating capacity of 9KW, 6KW, and 3KW but with different percentage difference.

4.3.2.4 Heating capacity (Frequency) vs. electric power consumption(P)

The compressor work increases with the increase of frequency of operation because of the system working on the higher amount of the refrigerant mass flow rate. The power consumption increase as the compressor must work on greater amount refrigerant per volumetric displacement of compressor (Figure 4.3).

4.3.2.5 Heating capacity (Frequency) vs. pressure ratio

The pressure ratio (Pr) increases with increase in frequency, and compressor electric power consumption for the fixed load/source conditions for meeting higher heat load demands (Figure 4.4), resulting in poor compressor performance because of electromechanical losses at higher speeds. The pressure ratio increases because of the compressor work increases due to larger refrigerant mass flow rates at higher speed. The pressure ratio increases because of the increase in enthalpy difference between suction and discharge with increase in water supply temperature for the same ambient temperature conditions. Also, the same is true by reducing the ambient temperature conditions. The increase in thermodynamic work (enthalpy difference) because of higher compression ratio also causes extra power consumption. One of the reasons for lower COP values with increase of WST was the increase in the pressure ratio values, which increases with the electric power consumption of the compressor. The pressure ratio increases with increase in compressor electric power consumption from 2.73 to 3.83 with WST of 30 ^{o}C, 3.13-4.51 to 4.53 with WST of 35 ^{o}C, 3.57

to 5.02 with WST 40 °C, 4.02 to 5.42 with WST of 45 °C, 4.45 to 5.79 with WST of 50 °C, 5.50 to 5.95 with WST of 55 °C. The pressure ratio for the ambient condition ranges increase with higher electric power consumption and the values are 3.72-5.10 for WST of 35 °C, 5 to 5.78 with WST of 45 °C, 4.42 to 5.54 with WST of 55 °C result in poor performance at higher WST for the same heat loads demands.

4. 3.2.5 Heating capacity *(Frequency) vs. refrigerant mass flow rate*

The heating capacity is directly proportional to the refrigerant mass flow rates inside the condenser but with a reduced rate due to increasing heating capacity, given by the following Equation (4.2).

$$\dot{m} = \rho \times \dot{Q} \tag{4.2}$$

Based on the assumptions of evaporator and condenser having same refrigerant mass flow rates. The ambient conditions, and heat supply temperature dictate compressor suction temperature and pressure, which then determines the refrigerant density. The refrigerant mass flow rate inside the evaporator is proportional to the density of refrigerant. The impact of heat supply temperature and the ambient temperature conditions on the refrigerant mass flow rates is illustrated in Figure 4.5. The refrigerant mass flow rate variation range with the compressor speed for the tested ambient conditions of 15 °C was 11.75 g/s -68.90g/s, 11.30g/s -69.40g/s, 11.15-74.30g/s, 11.20g/s -76.90g/s, 11.1g/s-77.1g/s, 10.75g/s-92.40g/s at 35 °C, 40 °C, 45 °C, 50 °C, 55 °C respectively. This could be seen from the Fig. (10-12) that with the increase in ambient temperature the refrigerant mass flow rates increase for the same heating capacity because of the suction temperature and density. Increase in ambient temperature also causes higher pressure to drop across the evaporator and additional restrictions because of higher mass flow rates. The system tries to achieve state of balance after the start-up based on the experienced ambient temperature conditions. The pressure drop inside the evaporator is adjusted according to the ambient conditions, as the heat transmission area, co-efficient of heat transfer are constant for the evaporator. The saturated evaporation temperature with constant temperature difference to that of ambient temperature, is defined by the combination of these properties (i. e. pressure drop adjusted according to the ambient conditions, and the heat transmission area, co-efficient of heat transfer). Then according to the saturated vapour temperature, the evaporating pressure is dictated depending on the properties of refrigerant. At the exit of evaporator, the superheat set point of 10 °C is achieved by expansion valve modulation. The suction density is dictated by the refrigerant temperature at the compressor suction (summation of superheat plus saturated evaporating vapor temperature), and evaporating pressure. With increasing ambient temperature, the suction temperature increases and hence a proportional increase in density results in corresponding refrigerant mass flow rates increase proportionally. The ambient temperature vs. suction temperature and the specific volume (inverse of density) can be seen from the below Figure 4. 14- 4.16 for the respective three heat supply temperature.

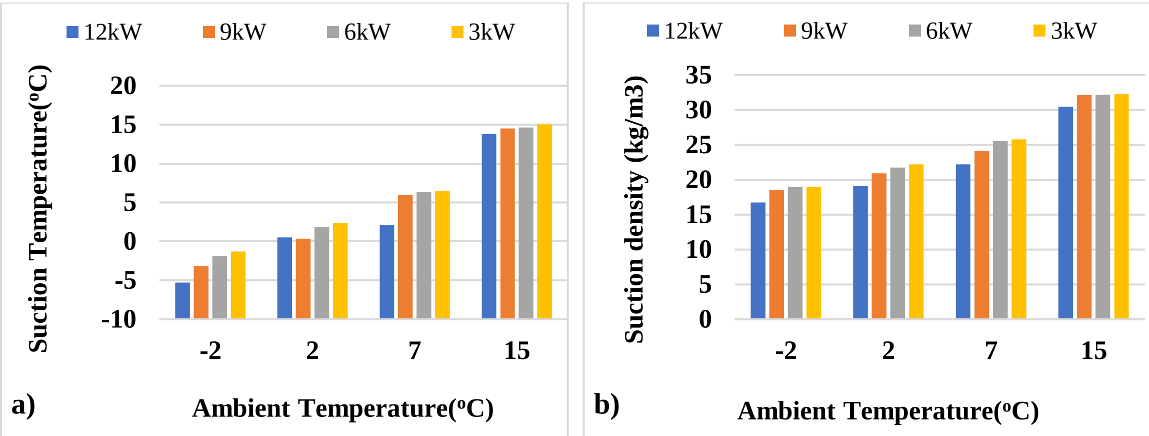


Figure 4.14 Ambient temperature vs. Suction a) temperature, b) density at low heat supply temperature of 35 °C

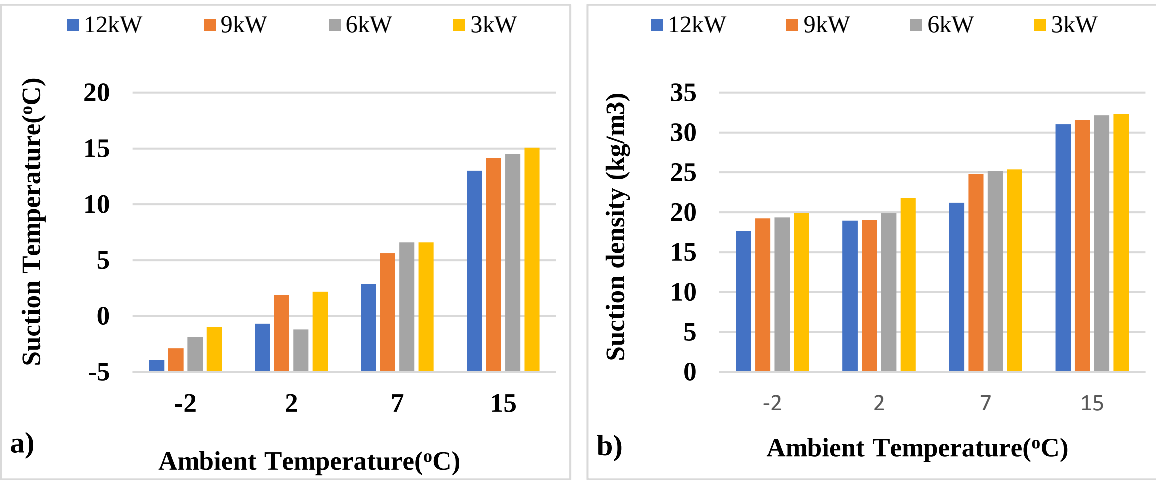
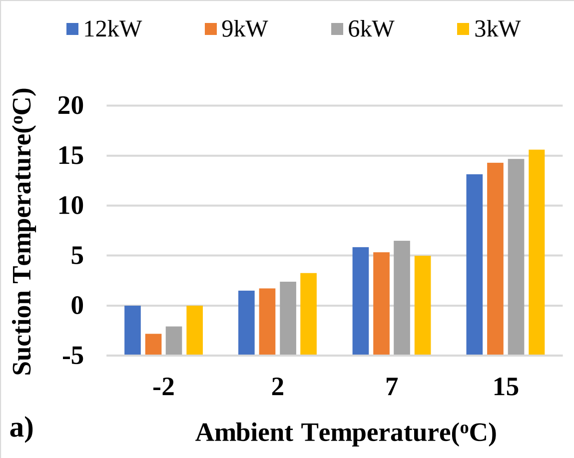


Figure 4.15 Ambient temperature vs. Suction a) temperature, b) density at medium heat supply temperature of 45 °C



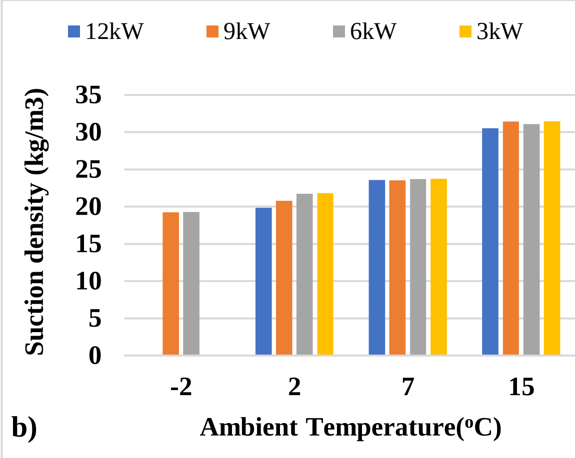


Figure 4.16 Ambient temperature vs. Suction a) temperature, b) density at high heat supply temperature of 55 ºC

As mentioned earlier at higher frequency the heating capacity increases with reducing rate and the same holds for the refrigerant mass flow rates because of the mass flow restrictions at higher speeds due to, i) higher evaporator pressure drops resulting in reduction of suction density, ii) the variation in volumetric efficiency because of the compression process in scroll. The higher pressure drop across evaporator also results in increase of pressure ratio with the increase of frequency. The mass flow rates reduction at higher speeds is due to higher pressure drop inside evaporator which causes reduction in suction density and pressure and subsequently the evaporator mass flow rate was reduced.

4.3.2.6 Heating capacity (Frequency) vs. isentropic efficiency

The isentropic efficiency can be defined as the ratio between the isentropic thermodynamic work to the actual thermodynamic compressor work. A typical trend for the isentropic efficiency was observed for the various tested conditions with the highest value of 73.11% with a pressure ratio of 2.33 at 15 ºC ambient conditions and WST of 30 ºC. The higher enthalpy difference also occurs when the system is operating at frequency above to that of the nominal value of 60Hz resulting in poor isentropic efficiencies due to these higher-pressure ratios. Also, lower isentropic efficiency is observed at frequency below the nominal value, which is attributed to the compressor lubrication issues and other electromechanical losses due to high discharge line temperature. The isentropic work is smaller than the actual thermodynamic compressor power consumption because of these inefficiencies i.e., lubrication issues and other electromagnet losses due to heat-up during compression process. The difference between the actual compressor thermodynamic work performed and measured value of electrical power consumption was attributed to the inefficiencies due to motor losses, inverter losses, friction losses, and heat up due to compression process and motor windings. The enthalpy difference reduces between the discharge and suction ports at higher ambient temperature resulting in higher isentropic efficiency than at low ambient temperature conditions, illustrated in Figure 4.6(a, b, c, d). The maximum isentropic efficiency occurs at nominal value and reduces with the increase of frequency above that value. The difference between the isentropic efficiency at higher frequency to that of nominal value is smaller at low ambient temperature conditions but increases with the increase of ambient temperature of 15 ^{o}C. Similarly, the rate of difference between the efficiency increases with the increase of frequency above the nominal value at higher WST because of increase in discharge line temperature. The worst isentropic efficiency occurs at lowest frequency in all tested conditions because of improper lubrication and leakage issues at low speed. The total isentropic efficiency percentage changes between the minimum and maximum tested heating capacities at ambient temperature conditions of 15 ^{o}C and WST of 35 ^{o}C, 40 ^{o}C, 45 ^{o}C, 50 ^{o}C, 55 ^{o}C are 8.53%, 10.09%, 11.49%, 12.9%, 13.91%, 18.37 % respectively with the highest value when the frequency value is near to 60Hz. The discharge line temperature has shown inverse relationship with isentropic efficiencies. For a constant test conditions discharge line temperature has minimum value when the isentropic efficiency has maximum value. The trend for the DLT value with constant heating capacity and fixed WST, increases with decreasing the ambient temperature conditions resulting at lower compressor isentropic efficiencies. As an example, the DLT for 12kW and with WST of 35 ^{o}C for the four ambient temperature conditions of 15 ^{o}C, 7 ^{o}C, 2 ^{o}C & -2 ^{o}C, were 68.43 ^{o}C, 71.98 ^{o}C, 74.98 ^{o}C, 78.64 with the corresponding isentropic efficiencies of 72.13, 65.86, 62.21, and 59.71, respectively.

4.3.2.7 Heating capacity (Frequency) and volumetric efficiency

The ratio between the actual mass flow rates at suction to the theoretical value can be defined as the volumetric efficiency of the compressor. The variation of the volumetric flow rate and efficiency against the heating capacity can be seen from Figure 4.17-4.19(a, b) respectively. Factor effecting the efficiency is the frequency of operation and the highest volumetric efficiency in all tests is at the point when the compressor operates near to the nominal frequency. The volumetric efficiency also increases according to the ambient temperature. The rate of volumetric efficiency increase at higher frequency reduces because of mass flow rate restrictions. The variation due to compressor frequency on volumetric efficiency and the pressure drop across the evaporator results in a combine effect on the rate of increase of the refrigerant mass flow rate reduction inside the evaporator. This is because, of the volumetric flow rate is directly proportional to the refrigerant mass flow rate and suction density.

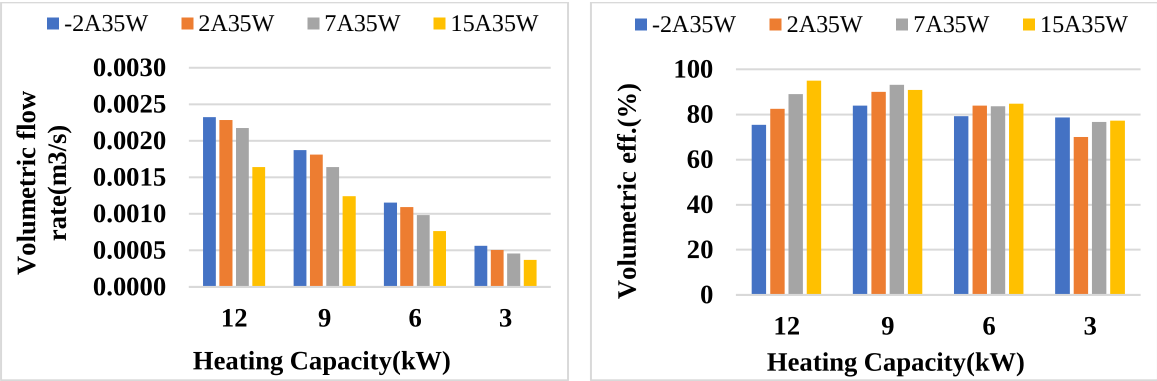


Figure 4.17 Heating capacity vs. Volumetric flow rate, b) Volumetric efficiency at 35 °C

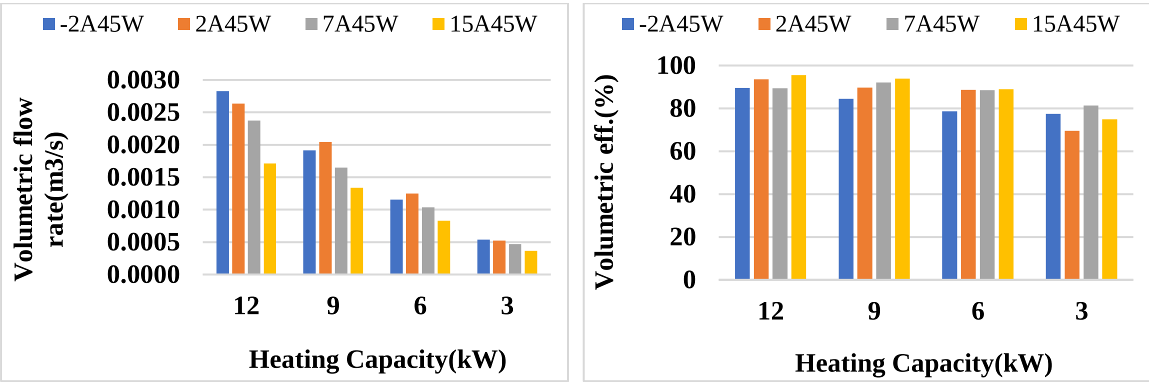


Figure 4.18 Heating capacity vs. Volumetric flow rate, b) Volumetric efficiency at 45 °C

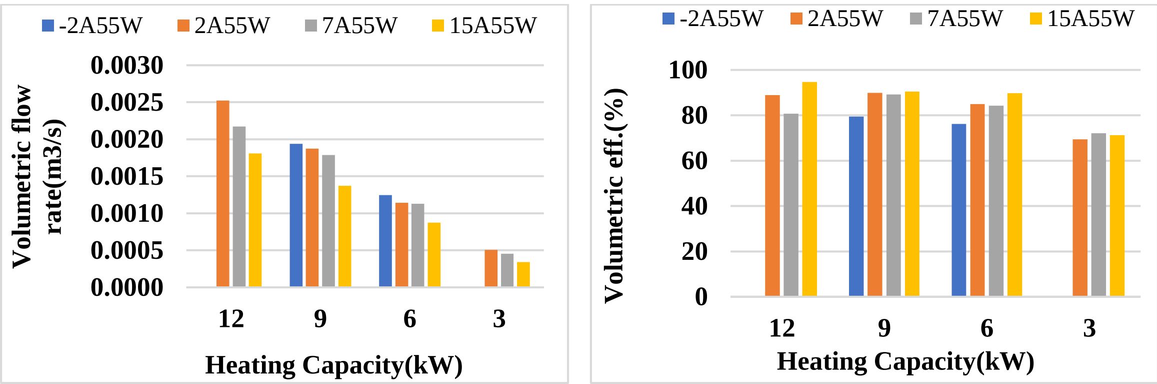


Figure 4.19 Heating capacity vs. Volumetric flow rate, b) Volumetric efficiency at 55 °C

The variation of the volumetric efficiency against the heating capacity can be seen from Figure 4.7 (a, b, c, d) respectively. Factor effecting the efficiency is the frequency of operation, highest volumetric efficiency was experienced in all tests is at the point when the compressor operates near to the nominal frequency. The overall trend of volumetric efficiency also shows an increase according to the ambient temperature. Very rare impact has been observed due to variation in water supply temperature (WST). The rate of increase of volumetric efficiency at higher frequency reduces, because of the mass flow rate restrictions inside the evaporator pressure drop. The variation due to compressor frequency and the pressure drop across the evaporator results in a combine effect on the rate of increase of the refrigerant mass flow rate reduction inside the system, reducing volumetric efficiency rate.

4.3.2.8 Heating capacity (Frequency) vs. inverter losses

After the performance evaluation of the compressor overall performance through the COP values, compressor isentropic and volumetric efficiencies, pressure ratios impact on the system performance, this section presents detailed investigation on the inverter losses *(percentage of total power consumption)* behavior for the ASHP system and the overall contributions in terms of losses. The inverter losses vary with change in compressor frequency for the range of WST and ambient conditions. The inverter losses vary mainly due to the frequency of operation with very small impact due to the source/load side conditions [83, 107]. The proportion of the power consumption by the inverter to that of total system electric power increases with the reduction in frequency because of smaller total power consumption. Additionally, with decrease in ambient temperature conditions and keeping frequency and WST constant results in the higher inverted power consumptions because of the lower efficiency of the inverter. Hence the % loss of inverter increases in proportion to the system total electric power consumption. At WST of 30 ^{o}C, and at frequency of 61Hz the % loss increases from 6.24 to 6.78 due to ambient conditions reduction from 15 ^{o}C to 7 ^{o}C. It was observed that at fixed ambient conditions, WST of 15A30W that with frequency increase (15.08 HZ, 29.78 Hz, 45.27 HZ, 61.18 Hz, 91.19Hz, 98.72Hz) the inverter losses were found to be 212 W, 189W, 266W, 78W, 144W and 183W respectively with the lowest reduction in power at the frequency of 61.18 Hz which is very close to the nominal frequency of 60Hz. However, the percentage inverter losses have inverse relationship with the frequency variation and heating capacities as can be seen from Figure 4.8(a, b, c, d). The higher compressor frequency results in smaller ratio of inverter losses to the total power consumptions. The percentage inverter losses to that of total power consumption reduces with the increase in compressor speed as could be evidenced from the literature review [83]. The percentage inverter losses for the above-mentioned frequency values at 15A30W are

18.56%, 12.26%, 8.31%, 6.24%, 4.41%, 3.05% with the heating capacity values of 3 KW, 6KW, 9KW, 12KW, 15KW, and 18KW respectively. The percentage reduction in the ratio of inverter losses to that of the total power with the increase of frequencies & heating capacities of 3 KW, 6 KW, 9 KW,12 KW,15 KW, and 18KW varies in the range of 15.45% to 2.83% for 15A35 test conditions, 14.16% to 2.77% for the test conditions of 15A40W, 12.58% to 2.45% for the test conditions of 15A45W, 11.29% to 2.23% for the test conditions of 15A50W, 10.14% to 2.05% for the test conditions of 15A55W(Figure 4.8a). The higher the difference between the load and source side temperature results in higher variation range in the inverter losses ratio. The highest inverter losses as a percentage of the total power is 18.56% for 15 ^{o}C ambient conditions occurs at 3KW heating capacity and WST of 30 ^{o}C and subsequent reduction to the value of 10.54% at WST of 55 ^{o}C mainly because of the inverted frequency, with changing WST values and the higher actual power consumption by the HP system at higher WST temperature for the same heating demands. The inverter losses percentage of total compressor power at ambient conditions of 7 ^{o}C, 2 ^{o}C, -2 ^{o}C and corresponding heating capacities could be observed with Figure 4.8 (a, b, c, d). At ambient conditions of 7 ^{o}C, variation range in the inverter percentage losses for 3 kW, 6 KW,9 KW,12 KW,15 KW (Figure 3.8b) is from 15.89% to 3.32% with WST of 30 ^{o}C, 13.17% to 2.92% with WST for 35 ^{o}C, 12.63% to 2.75% for 40 ^{o}C, 11.39 % to 2.52% with 45 ^{o}C, 10.90% to 2.35% with WST of 50 ^{o}C, 10.72 % to 2.27 % with WST of 55 ^{o}C. By comparing the inverter losses at two ambient conditions of 15 ^{o}C and 7 ^{o}C for 3- 15KW range it can be noted that with the reducing the ambient temperature results in further enhancement in losses due to frequency variation for meeting the heat loads demands. The variation range for WST of 30 ^{o}C and ambient conditions of 7 ^{o}C for 3KW to 15KW is 15.89% to 3.32%, with higher values than 12.26% to 3.05% at ambient conditions of 15 ^{o}C & WST of 30 ^{o}C for the same heating capacity range due to the frequency increase for meeting the same heating demands at low ambient temperature.

The losses as a percentage value reduces as the air ambient temperature conditions reduces from 15 ^{o}C to -2 ^{o}C for constant heating capacity and WST. For example, with constant heating capacity of 12KW and WST of 35 ^{o}C the losses values at 15 ^{o}C, 7 ^{o}C, 2 ^{o}C, -2 ^{o}C are 5.80%,3.92%, 3.55%, 3.29% respectively because of changing frequency values of 62.87Hz, 85.27Hz, 92.25Hz, 102.84Hz.

4.3.2.9 Heating capacity (Frequency) vs. discharge line temperature (DLT)

The DLT increase with the increase of the frequency above/below the nominal value with the lowest value at around 60Hz. The compressor discharge temperature has approximately linear inverse relationship with the compressor isentropic efficiencies and increase in DLT value causes degradation of the compressor performance. The compressor discharge temperature increases due to the increase of pressure ratio, which is directly related to the increase of frequency of operation, but only above the nominal value. Below the nominal value of 60Hz, opposite is true and with the decrease in frequency results in higher discharge temperature and lower pressure ratio. The reason for the higher DLT value at above and/below to that of nominal value, is due to compressor heat up and lubrication issues. This also has negative impact on the compressor life, motor, and compression process as well and results in higher power consumption and lower isentropic efficiency. The compressor discharge temperature variation at various heating demands can be seen from the Figure 4.9(a, b, c, d) for four tested ambient temperature conditions and the ranges of WST discussed earlier. At constant source/load side conditions the discharge temperature increases while moving away from the nominal frequency. The minimum compressor discharge temperature is maintained while operating the system near to the nominal frequency and keeping other tested conditions fixed and the same trend was noted for the compressor isentropic efficiency as well. Increasing WST for keeping ambient air temperature conditions and load constant also resulting in higher DLT measured resulting in lower isentropic efficiencies. For example, at 7A30W for 15KW heating demand the DLT measured was 69.05 ^{o}C with isentropic efficiency of 63.24 and becomes 100.93 with isentropic efficiency of 61.11 when the WST changes to 55 ^{o}C for the same heat load. At higher speed of operation for all tested conditions the discharge temperature increase with lowest value near 60Hz frequency. The trend for the discharge line temperature is corresponding to the volumetric and isentropic efficiency of the compressor. Higher mechanical losses associated with the leakage, and lubrication issues and hence gas heating at suction at upper and lower operation to that nominal frequency. The discharge line temperature has inverse relationship with the ambient temperature, increases with temperature reduction. The reason for this was because more compression work was required due to higher pressure and temperature lift.

4.3.3 Electric power(P) consumptions

The electric power consumption relies on three i.e., frequency, ambient temperature, and water supply temperature (WST). The electric power consumption has inverse relationship with ambient temperature at constant heat loads, and WST. The power consumption, frequency of operation and ambient temperature relationship can be seen for three different tested water supply temperature can be seen from Figure 4.3 (a, b, c, d) and represented by equation 4.3 below.

$$P = 0.385 + 0.00013 * \omega^2 + 0.000066 * \omega * T_w + 0.00001{T_w}^2 + 0.00036 * \omega * T_a + 0.000288 * T_w * T_a - 6.33 * 10^{-6} * \omega * T_w * T_a - 0.00094 * {T_a}^2 \quad (4.3)$$

4.3.3.1 Electric power vs. supply temperature, ambient temperature

Increasing ambient temperature results in higher heating capacity for constant frequency and hence the power consumption requirements also increase. However, the rate of increase of power consumption increases with the increase of frequency above the nominal value, at all ambient temperature conditions and water supply temperature. At ambient temperature conditions of 15 °C with WST of 30 °C and frequency of operation 15.08Hz, 29.78Hz, 45.27Hz, 61.18Hz, 91.19Hz, 98.72Hz with the power consumption value of 0.75KW, 1.14KW, 1.69KW, 2.24KW, 3.17KW, 4.59KW with the highest increase in power consumption at maximum frequency of operation. The same trend is observed at higher supply temperature of 35 °C, 40 °C, 45 °C, 50 °C, 55 °C at the fixed ambient temperature conditions. However, at lower ambient temperature conditions the rate of power consumption reduces. The compressor work increases with the increase of refrigerant mass flow rate as the compressor must work on greater amount refrigerant per its volumetric displacement. The compressor pressure ratio increases because of the increase in enthalpy difference between suction and discharge with increase in supply temperature for the fixed ambient temperature conditions. Also, the same is true by reducing the ambient temperature conditions. The increase in thermodynamic work (enthalpy difference) because of higher compression ratio also requires more power consumption. The higher enthalpy difference also occurs when the system is operating at frequency above to that of the nominal value of 60Hz resulting in poor isentropic efficiencies due to these higher-pressure ratios. Also, lower isentropic efficiency is observed at frequency below the nominal value, which is attributed to the compressor lubrication issues and other electromechanical losses due to high discharge line temperature. The isentropic work is smaller than the actual thermodynamic compressor power consumption because of these inefficiencies due to lubrication issues and other electromagnet losses, i.e., heat-up during compression process. The enthalpy difference reduces between the discharge and suction ports at higher ambient temperature resulting in higher isentropic efficiency than at low ambient temperature conditions. The maximum isentropic efficiency occurs at nominal value and reduces with the increase of frequency above that value. The difference between the isentropic efficiency at higher frequency to that of nominal value is smaller at low ambient temperature conditions but increases with the increase of ambient temperature of 15 ^{o}C. Similarly, differential rate between the efficiency increases with the increase of frequency above the nominal value at higher WST because of increase in discharge line temperature (Figure 3.12). The worst isentropic efficiency occurs at lowest frequency at all tested conditions because of improper lubrication and leakage issues at low speed. The isentropic efficiency, defined as the ratio between the isentropic thermodynamic work to the actual thermodynamic compressor work. The actual compressor power measured was used to calculate the final electrical efficiency. The difference between the actual thermodynamic work and measure value of electrical power consumption was because of the inefficiencies due to motor winding losses, inverter losses (harmonics, heat, power consumption by inverter), friction losses, heat up due to compression process.

4.3.3.2 The electric power consumption vs. frequency

The earlier pointed out correlation between the power consumption against water supply temperature, ambient temperature conditions are further dependent at individual heating capacity and frequency of operation. The higher refrigerant mass flow rate requires higher compression work, and hence more power is consumed at higher frequency and heating capacity. For the constant heat capacity, the frequency requirements increase with the increase of WST and hence higher refrigerant mass flow results into more compression work. At higher frequency, the isentropic efficiency reduction causes higher power consumption.

4.4 Water supply/return temperature difference(ΔT) impact on the HP performance

4.4.1 COP vs. Frequency

The variation in COP, frequency against the heating capacity because of delta T value for fixed heat supply temperature of 40 ^{o}C, 45 ^{o}C, 50 ^{o}C, and 55 is shown in Figure 4.20, and 4.21 respectively. At 40 ^{o}C WST the highest COP occurs when the delta T value is 20 at heating capacity of 12KW with a value of 4.60 and the worst COP was found at 3KW test with the COP value of 2.93 when the delta T was 5. The same pattern has been shown for higher WST of 45 °C, 50 °C, and 55 °C but with different COP values. The reason for poor performance with lower delta T value with constant heat load was because of the higher DLT, pressure ratio and comparatively lower isentropic efficiency, However, at lower heating capacity of 9KW, 6KW, and 3KW the variation in performance becomes less obvious because of the smaller difference between in frequency requirements.

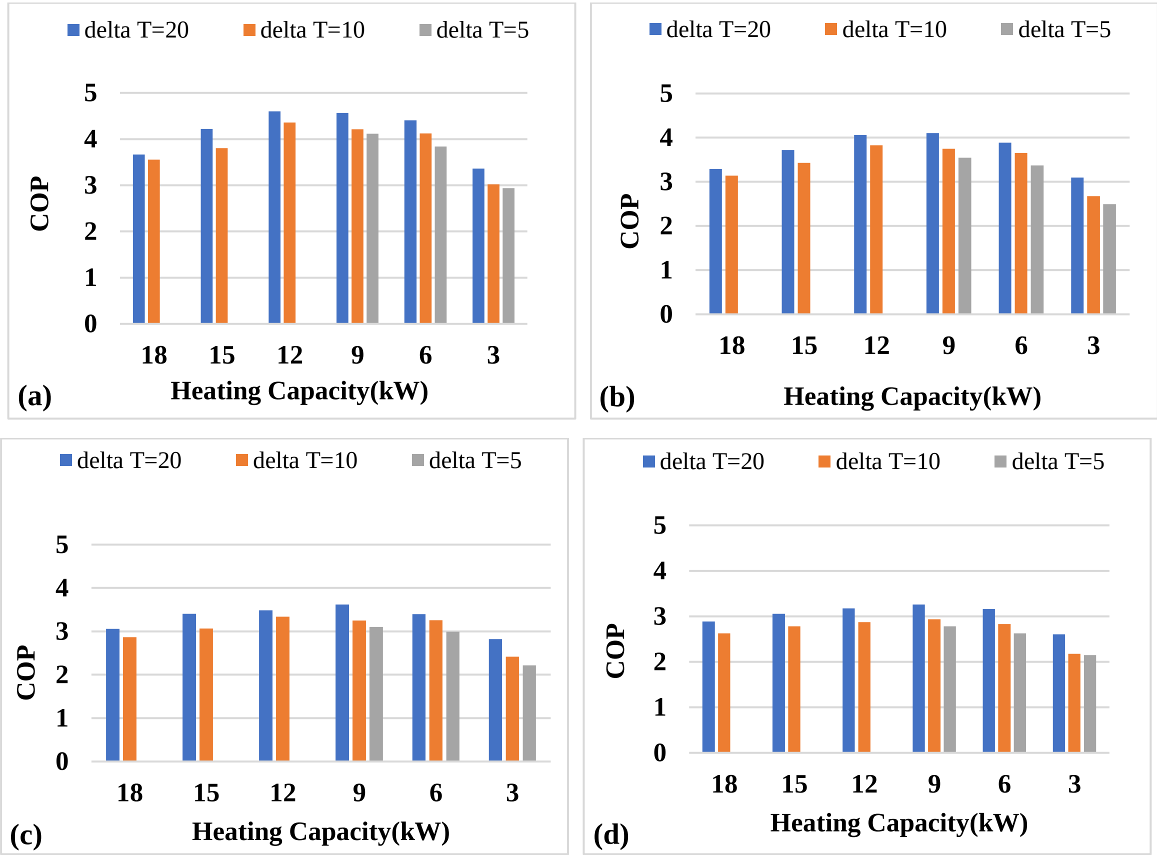
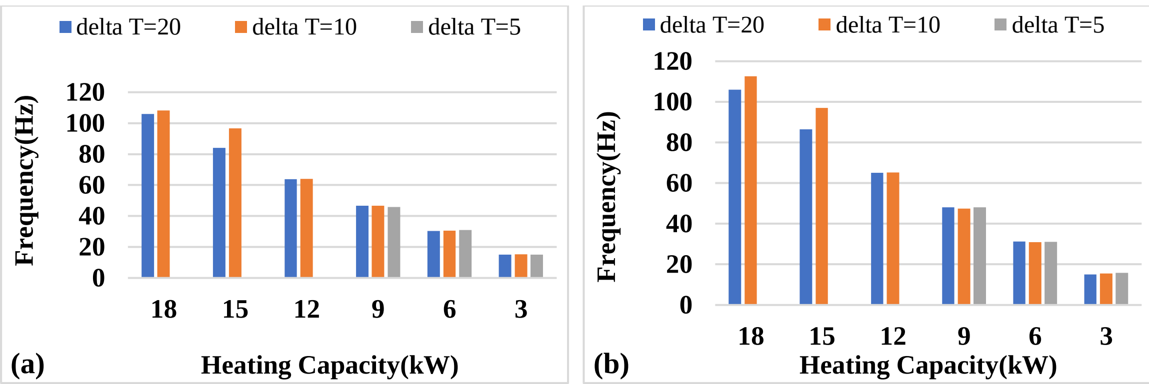


Figure 4.20 . Heating capacity vs. COP with delta (ΔT) of 5, 10, 20 °C at heat supply temperature of a) 40 °C, b) 45 °C, c) 50 °C, d) 55 °C



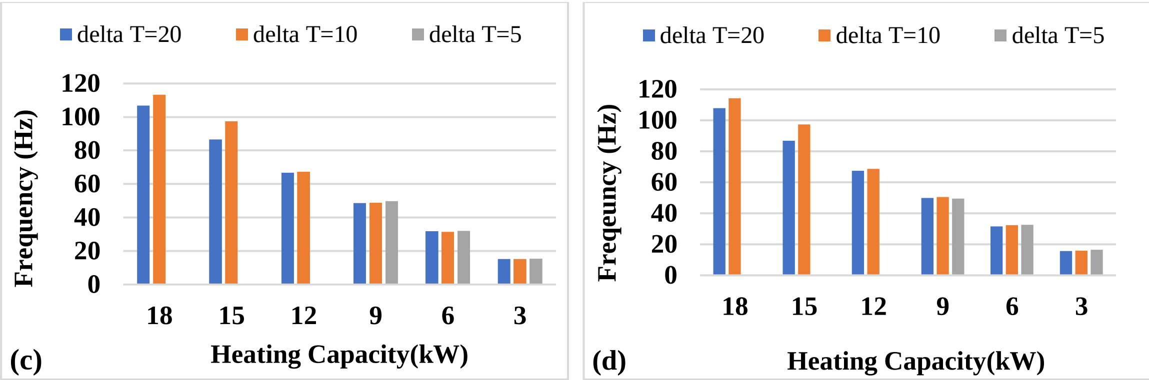


Figure 4.21 Heating capacity vs. frequency with delta (ΔT) of 5, 10, 20 °C at heat supply temperature of a) 40 °C, b) 45 °C, c) 50 °C, d) 55 °C pressure ratio (Pr) impact due to delta T

The pressure ratio varies from a minimum value of 2.01 to maximum 4.95 with the corresponding electric power consumption value of 0.75KW to 6.84KW for delta T=10(Figure 4.22). For WST of 40 °C the pressure ratio is 2.40, 2.72, and 2.76 when the delta T is changing from 20, 10 and 5 for 3KW test.

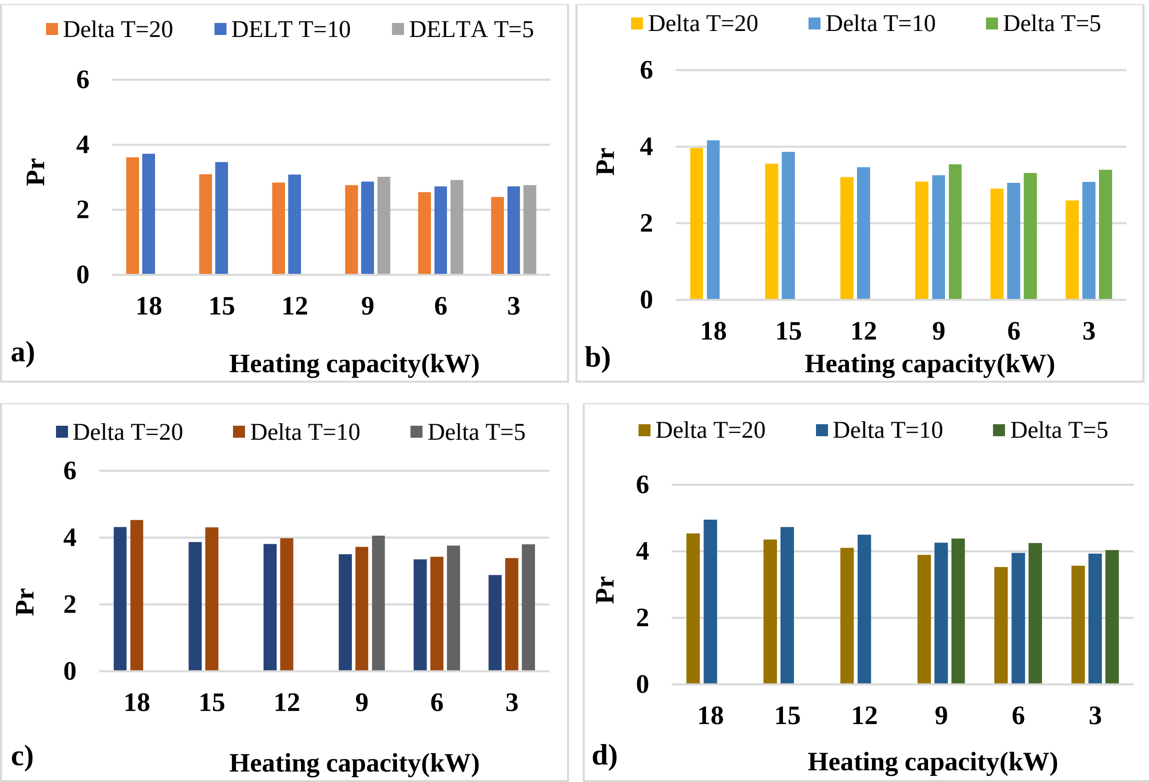


Figure 4.22 . Heating capacity vs. Pressure ratio with delta (ΔT) of 5, 10, 20 °C at heat supply temperature of a) 40 °C, b) 45 °C, c) 50 °C, d) 55 °C

4.4.2 Impact of delta T at isentropic efficiencies and DLT

The isentropic efficiencies & DLT for four WST of 40 °C, 45 °C, 50 °C, and 55 °C with changing delta T have been shown in Figure 4.23 & 4.24 respectively. Both parameters significantly impact the COP values due to changing delta T. The isentropic efficiency has inverse relationship with increasing DLT. Figure 4.25 & 4.26 shows the COP, and frequency values for delta T of 20 and 5 respectively, at the same ambient condition of with a range of WST temperature. Because of this water pump flow rates limitations, the heat load generation of 18KW, 15KW, 12KW for delta T of 5 was not possible and the tests were only conducted at these low heating loads of 9KW, 6KW and 3KW. For delta T of 20, &5 the COP value reduces with increasing WST temperature but with different rates. For water supply temperature of 40 °C and entering water temperature of 20 °C the COP values were 3.66, 4.22, 4.60, 4.57, 4.41, 3.36 for the heating capacities with the same pattern for higher WST but with smaller values. The maximum COP values for WST of 40 °C was 4.60 when the heating capacity was 12KW, while at higher WST of 45 °C, 50 °C , 55 °C maximum COP occurs at 9kW with values of 4.10, 3.62, 3.28, due to this frequency of operation. For lower delta T of the tests were performed for WST of 25 °C, 30 °C, 35 °C,40 °C, 45 °C, 50 °C, and 55 °C and presented. The COP values variation for each heating demands of 18KW, 15KW, 12KW, 9KW, 6KW and 3KW with delta T of 20 and WST of 40 °C to 55 °C are in the range 3.66-2.88, 4.22-3.05, 4.60-3.28, 4.57-3.26, 4.41-3.16, 3.36-2.61. For delta T of 5 with heating capacity 9KW, 6KW and 3KW and with WST variation from 25 °C to 55 °C the variation range is 5.57-2.78, 5.42-2.62, 4.05-2.15 respectively.

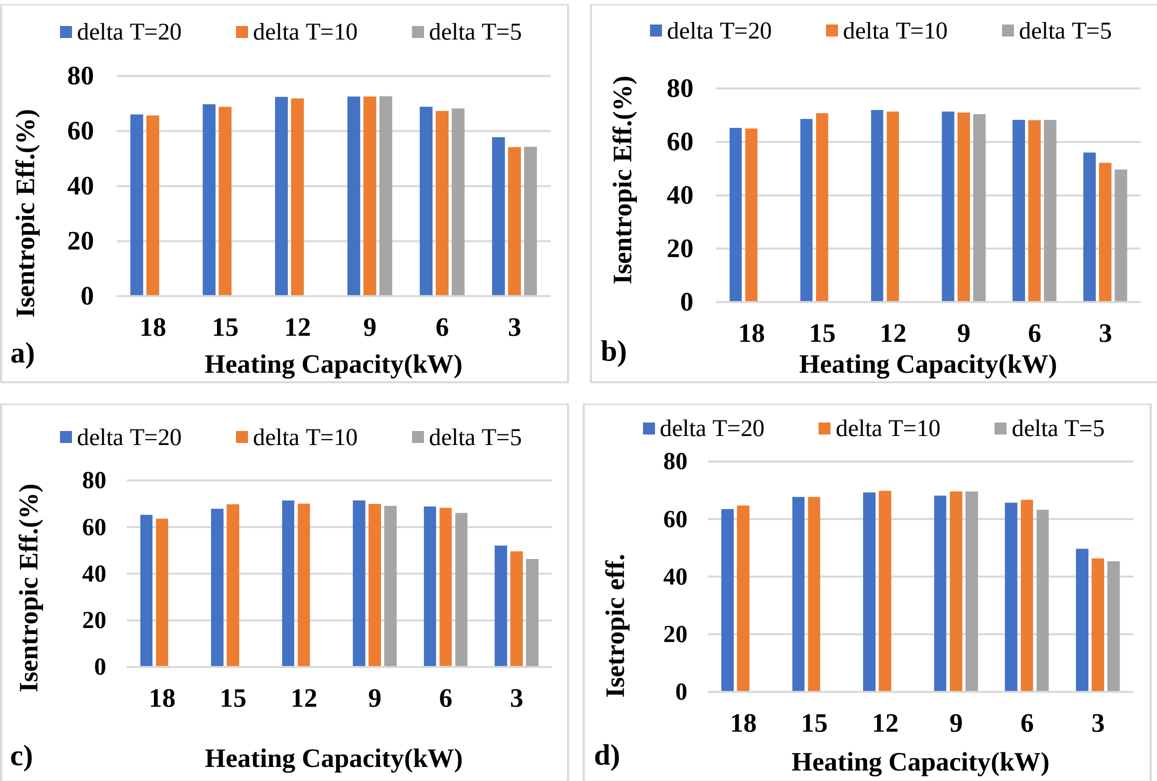


Figure 4.23 Heating capacity vs. isentropic efficiency with delta (ΔT) of 5, 10, 20 °C at heat supply temperature of a) 40 °C, b) 45 °C, c) 50 °C, d) 55 °C

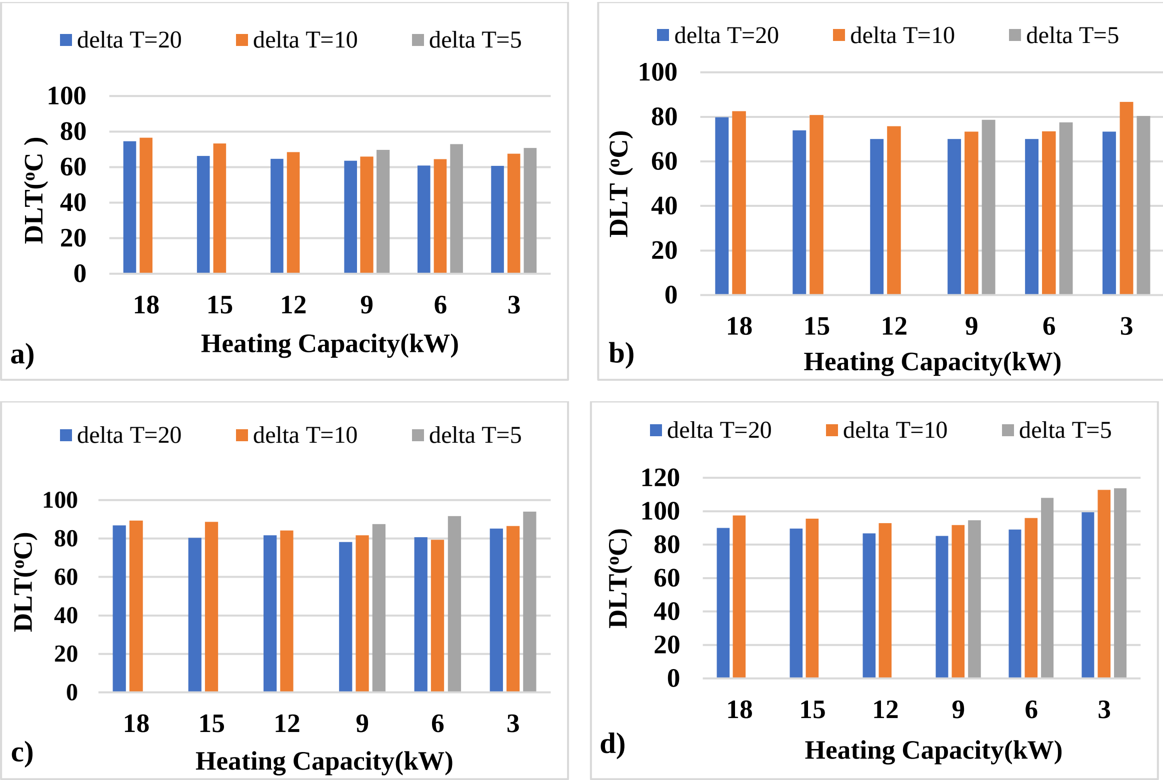


Figure 4.24 Heating capacity vs. DLT with delta (ΔT) of 5, 10, 20 ºC at heat supply temperature of a) 40 ºC, b) 45 ºC, c) 50 ºC, d) 55 ºC

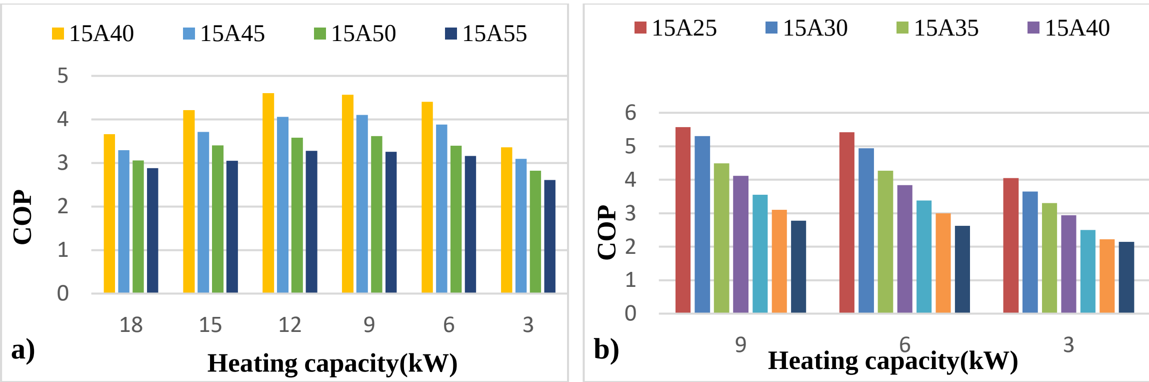


Figure 4.25 Heating Capacity vs. COP for a) (ΔT) =20 b) (ΔT) =5

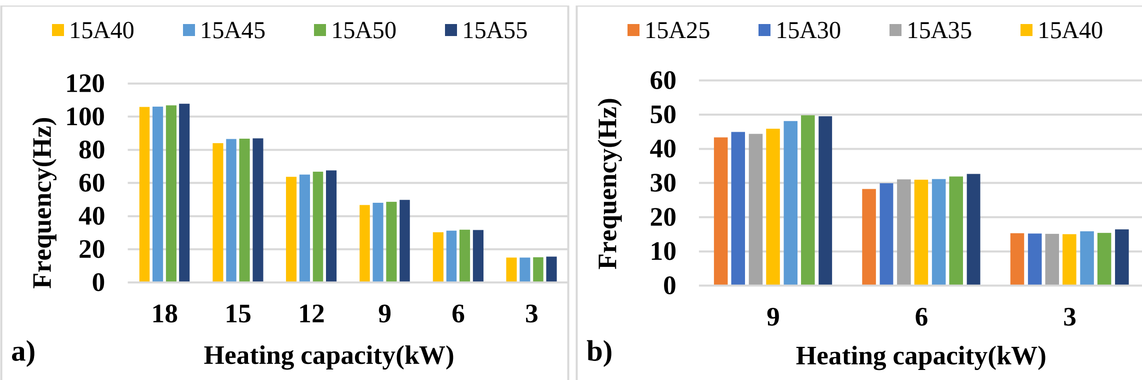


Figure 4.26 Heating Capacity vs. frequency for a) (ΔT) =20 b) (ΔT) =5

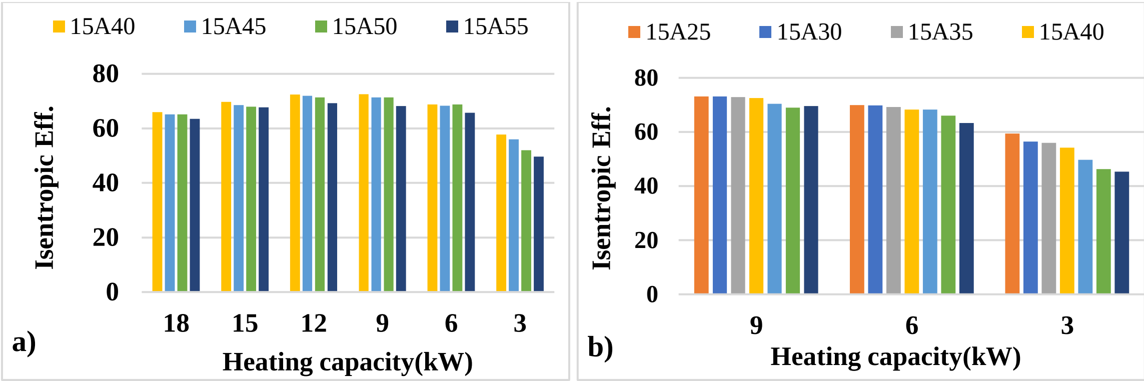


Figure 4.27 Heating Capacity vs. Isentropic Eff for a) (ΔT)=20, b) (ΔT) =5

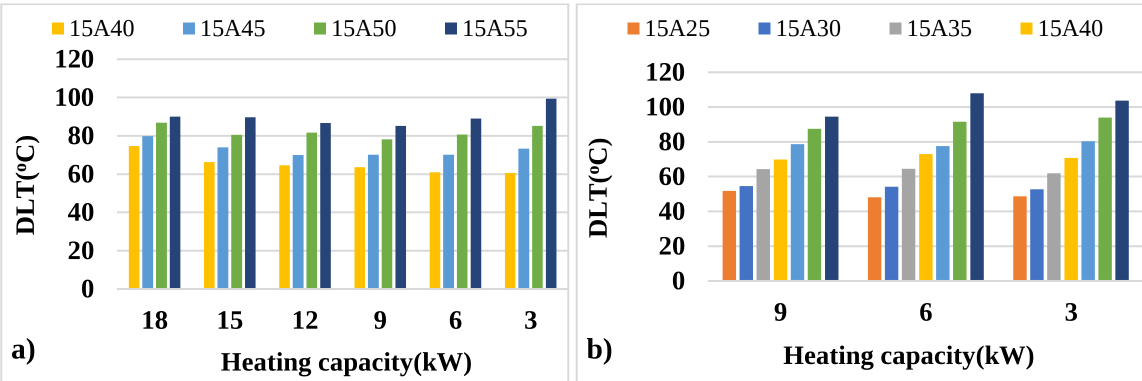


Figure 4.28 Heating Capacity vs. DLT for a) (ΔT) =20 b) (ΔT) =5

The isentropic efficiencies, and DLT at the changing delta T value are shown in Fig 4. 27 & 4. 28 respectively. The DLT for delta T =20 (Fig 28 a, b) at six heat heating capacity varies from 74.61 ºC-90.06 ºC, 66.27 ºC -89.67 ºC, 64.69-86.73 ºC, 63.56 ºC-85.18 ºC, 60.94 ºC-89.10, 60.62-99.43 when the WST changes from 40 ºC to 55 ºC, while reverse pattern was noticed for isentropic efficiencies with variation of 69.95-63.50, 69.75-67.73, 72.45-69.21, 72.55-68.14, 68.80 -65.67, 57.78-49.66. The DLT for delta T of 5 with 9KW, 6KW, 3KW with WST variation from 25 ºC-55 ºC are in the range of 51.82 ºC-94.55 ºC, 48.05 ºC-107.93 ºC, 48.72-103.80 ºC with the isentropic variation from 73.09-69.58, 69.92-63.30, 59.42-45.36 with reverse trend.

4.5 Error Analysis

The error associated with individual measuring devices mentioned in the methodology was utilized to find the total error for the heating capacity, refrigerant mass flow rate, electric power consumption and the COP. Every experimental work has an associated uncertainty to a certain extent. Independent variables measurement contributes to the final experimental results error. The error analysis in this study was performed from the methodology developed by (ASHRAE Guideline 2-2005, 1986) using the following equations.

$$Q_{Error} = Q * \sqrt{\left(\frac{\dot{m}_{w_{error}}}{\dot{m}_w}\right)^2 + \left(\frac{\Delta T_{Error}}{\Delta T}\right)^2} \tag{4.4}$$

$$\dot{m}_{ref\_error} = \dot{m}_{ref} * \sqrt{\left(\frac{\dot{m}_{w_{Error}}}{\dot{m}_w}\right)^2 + \left(\frac{\Delta T_{Error}}{\Delta T}\right)^2 + \left(\frac{\Delta h_{C\_error}}{\Delta h}\right)^2} \tag{4.5}$$

$$CoP_{error} = CoP * \sqrt{\left(\frac{Q_{HP_{Error}}}{Q_{HP}}\right)^2 + \left(\frac{P_{Error}}{P}\right)^2} \tag{4.6}$$

The percentage error for every quantity varies according to the measured values and has been considered in the analysis. The percentage error in the heating capacity varies from minimum value of ±4.61% to a maximum value of ±21.12% when the heat capacity varies from at 18KW to 3KW, respectively. The refrigerant mass flow rate error varies between ±4.39% at 0.083 kg/s to ±17.53% at 0.011 kg/s. The electric power was measured with ±1% accuracy. The error for COP varies between ±0.26 % at COP of 3.92 to ±6.98 % at a COP of 2.18.

4.6 Chapter summary

In this chapter the experimental results of the tested ASHP system have been presented to form the basis for the upcoming chapters regarding the domestic heat load applications and HP retrofit assesment different housing stock. The challenges with the variable speed operarations were highlighted. The variable speed compressor could provide the range of heating capcity due to speed variation at constant sourc/load side conditions but with its own implications. These needs to be considered in combinations with the associated benfitis of VSM. Some of the key points based on the tesitng results could be summarized as follows;

a) The performance strongly depends on the supply temperature with the COP value of 3.84 (7 ^{o}C ambient air temperature) when the heat supply was 35 ^{o}C and becomes 2.41 when the supply temperature become 55 ^{o}C at nominal heating capacity of 9kW.
b) The three variables mainly impact the system COP during testing conditions are the heating capacity, ambient temperature conditions, and heat supply temperature.
c) The heating capacity has linear relationship with operating frequency at fixed load/source side conditions and impact the compressor efficiencies.
d) The COP value is maximum when operating at nominal value of 60Hz at all tested conditions irrespective of the ambient and heat supply temperature. The COP degrades with the increase and decrease to that nominal frequency because of

compressor efficiencies, inverter losses, discharge line temperature (DLT), and pressure ratios.

e) For a constant WST of 35 ^{o}C, frequency modulation for 12KW, 9KW, 6KW and 3KW heating capacities were in the range of 102.8-62.86 Hz, 74.31-45.6Hz, 48.63-30 Hz, 23.8-15.21 Hz at ambient temperature conditions of -2 ^{o}C, 2 ^{o}C, 7 ^{o}C & 15 ^{o}C respectively.

f) The delta T value have a significant impact on the HP performance and compressor efficiencies, so it needs to be carefully considered while using the heat pump system for domestic heat load applications.

g) The recommended point of operations for the heat pump compressor is 60Hz at all ambient temperature conditions and heat supply temperature in variable speed mode to avoid the negative impact on the compressor efficiencies and DLT while using the heat pump technology for demand side management. However, the realistic system operations depends on the house heat demand and over/under production could results in cycling losses/electric heater requirements. Therefore, the thorough investigations with different property types with different heat load demands in UK climatic conditions needs considerations for the annual COP calculations.

Chapter 5: Domestic heat load applications for UK housing stock

5.1 Introduction

The testing results and performance establishment at different load conditions form the basis of this study, with the simple numerical model and domestic heat load applications. The issue with ASHP system is its performance degradation at lower ambient temperature conditions and more compressor work is required, putting extra pressure on network. The HP design for meeting the highest peak load demand requires larger components size, with increased cost, and negative impacts the overall annual system performance due to higher cycling losses during off-peak period. The peak load demand occurs only for the limited period of the year and the idle operation of the system for the remaining period causes losses and fluctuations. In seeking a system which can modulate its capacity according to the load requirements and operate over a wide range, a variable speed compressor based ASHP system is a viable option. The variable speed compressor based ASHP system fulfill the heating demand at lowest ambient temperature conditions by increasing the speed without the needs for oversizing. The potential performance improvement with the control mode, and heat supply temperature for the old aged Irish housing stock is considered in this study. Annual performance improvement in the range of 1-25% with VSM compared to FSM was reported depending on HP type (ground/air source) compressor used, operating speed range, and design load [31]. The reason for performance improvement was better part load condenser, and evaporator efficiency, smaller number of cycles, back up electric heater requirements, lower losses due to defrosting, supply temperature [31]. The heat supply temperature with intermittent operations during the on-cycle needs to be higher compared to the continuous system operation leading toward high energy consumptions due to higher condensation and lower evaporation temperature [63]. Limited number of articles were found investigating the performance improvement and energy savings with control mode and considering heat supply temperature simultaneously but for other HP types [31]. The GSHP seasonal performance improvement were investigated with hydronic heating distribution system aiming at increased system performance due to capacity control approaches and heat supply temperature [31,71]. The reduction in heat supply temperature from high (55^{o}C) to low (35 ^{o}C) result in improvement seasonal performance factor by 30-35% [31]. Three types hydronic heating distribution systems, i.e., underfloor, fan assisted hydronic coil unit, and traditional wet radiator were considered in analysis for comparative study between VSM & FSM. The thermal inertia of various hydronic heating distribution system influences the cyclic properties of HP, the response time at start-up and consequently its overall performance [71]. The negative impact due to cycling get reduced with VSM of control because of load matching with heat supplied [31]. However, there were certain issues during steady state HP operation with varying compressor speed compared to nominal speed of operation i.e., compressor isentropic, volumetric efficiency, pressure ratio, and discharge line temperature, and inverter losses [82]. These challenges with the utilization of the variable speed compressor-based heat pump system for domestic heat load applications was rarely investigated. The isentropic efficiency was highest at 65% at nominal speed with pressure ratio of 2.2, while the volumetric efficiency linearly degraded from 98% to 83% with pressure ratio variation range between 1.5 to 5.6 due to varying speed (35-75 HZ) [37]. The low to medium and high heat supply temperature on variable speed compressor based ASHP analysis was conducted at different steady state test conditions aiming at UK housing stock retrofit applications [22]. The COP value improved in the range of 30-40 % due to heat supply temperature reduction from 55 ^{o}C to 35 ^{o}C for the individual tested conditions. ASHP system with nominal heating capacity of 9.8KW was developed aiming at retrofit applications for old age 1900s Mid-Terraced house in Belfast climatic conditions [54]. The performance was evaluated at very high heat supply temperature of 60 ^{o}C for the range of experienced ambient conditions under the laboratory conditions [54]. The limited frequency modulation range between 37.5 Hz and 75 Hz, resulting in poor load match to the heat supplied during VSM of control with annual COP of 2.15[54]. Based on the above literature review on the ASHP experimental development, simulating constant heat loads under the controlled laboratory conditions to look into the inside happening with the speed variations, and the annual performance evaluation at low to medium and high heat supply temperature in both modes for capacity control was missing to the best of author,s knowledge in the context of UK housing stock. The earlier comparitive studies with capacity control were performed either under single steady state conditions for optimal frequency, individual component analysis (i.e compressor isentropic and volumetric efficiencies, inverter losses), single heat supply temperature. Therefore, the research questions investigated in this study could be summarized as, i) Numerical modeling of the HP performance based on the testing results and validations, ii) The annual performance improvement and energy saving with VSM in comparison to FSM with low to medium and high supply temperature with different property types iii) part load operation and the importance of balance point in two modes of control.

5.2 Data collection for building energy analysis

This section is dedicated to further details regarding the tools and materials used in this chapter. First of all, the domestic heat load demands in each bin for the five different property types considered in the analysis have been explained, followed by the number of hours occurring in each bin in Belfast climatic conditions using Meteonorm weather file data [101]. The load demand in each bin and the bin distribution are key factors and indicate the HP performance with the importance of design outdoor temperature (DOT) value.

5.2.1 Domestic heat load demands

The tested HP system was designed and developed with an aim to meet domestic household heat load demand commonly required inside the UK residential building for combined space heating (SH) and domestic hot water (DHW) demand. The old age period (1900-1949) property type including Detached (142 m^2), Semi-detached (133 m^2), End terraced (124 m^2), Mid terraced (94 m^2), and Flats (75 m^2) were considered in the analysis. The heat load demand including both SH and DHW demand for five property types were adapted from the average experimental results [105, 108] for different property type, is depicted in Figure 5.1.

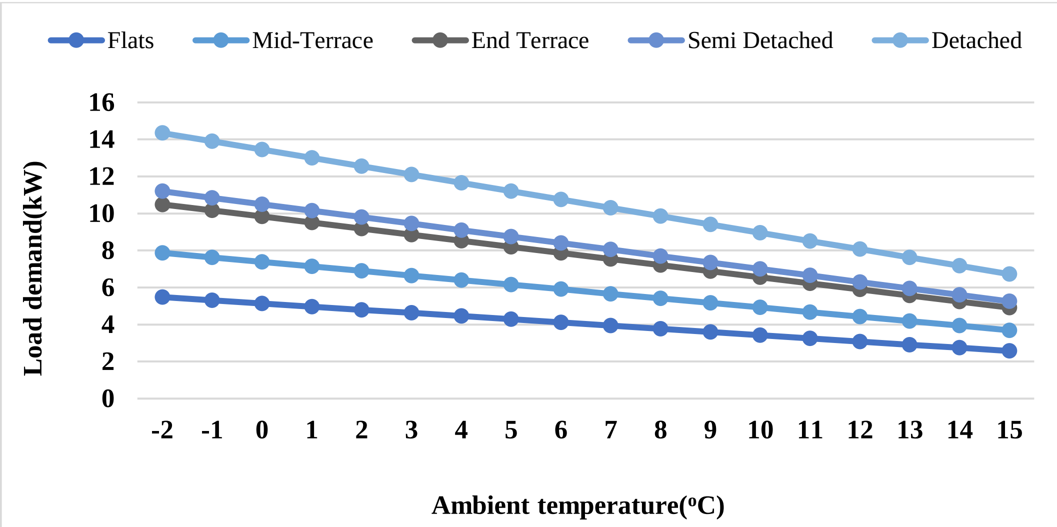


Figure 5.1 Average heat load demand for different property types [105,108]

5.2.2 Bin distribution for Belfast climatic conditions

The hourly bin distribution depicted in Figure 5.2, for Belfast climatic conditions were retrieved from TRNSYS 17 Meteonorm weather file data [101]. The largest number of hours during the whole year period was measured at 7 ^{o}C.

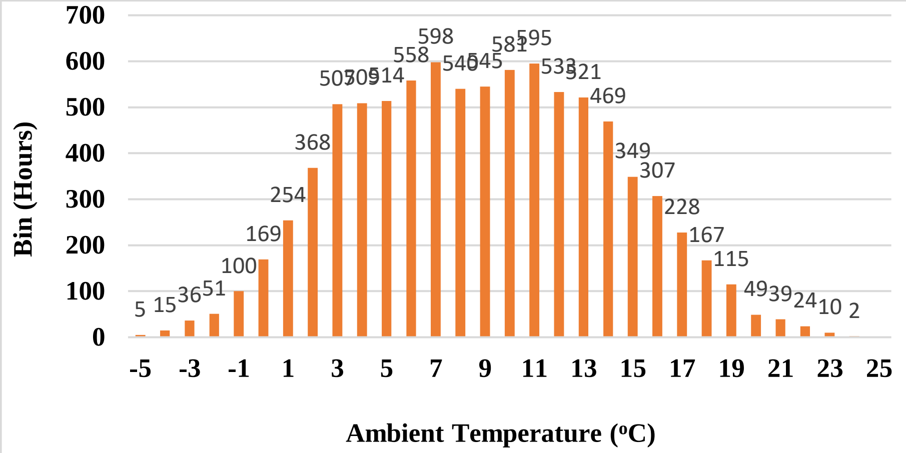


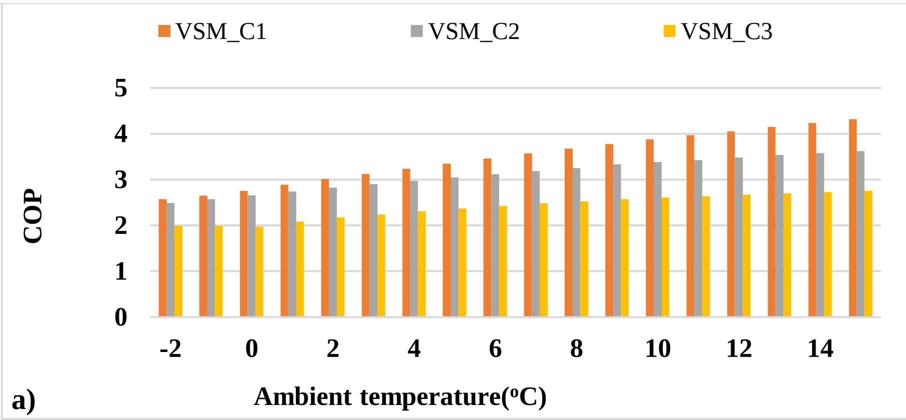
Figure 5.2 . Annual bin distribution for Belfast climatic conditions

5.3 Result analysis

The HP performance from the tested results using the bin approach for the annual COP evaluation into different property types were used. The system annual performance simulated against the heat load demand in two control mode (VSM & FSM) and at three heat supply temperature considered here as C1, C2, and C3 have been presented in the following sub-sections for different property types. The COP values varies according to the part load operation, heat supply temperature and control mode. The role of transient losses become very crucial while comparing the control mode for the capacity control for different heat load demand in the evaluation of annual performance.

5.3.1 Part load performance in different property types

The HP system performance were predicted using the model function in MATLAB software, a proprietary multi-paradigm programming language and numeric computing environment developed by MathWorks [72]. This allows plotting of functions and data, implementation of algorithms, and interfacing with programs written in other languages. The part load performance according to the different property types with two control modes of operation (VSM & FSM). The HP meets heat load demand across the experienced ambient temperature conditions in VSM of control for all five property types by adjusting heating capacities via compressor speed. Keeping balance between the house heat loss and HP heat capacity means maintaining the thermostat temperature at set point, fully controlled system resulting in higher thermal comfort. The thermal heat load demands for each property type were shown earlier influences the HP performance in each bin. The part load performance with VSM in each bin for five property types over the experienced ambient temperature conditions, at three heat supply temperature is shown in Figure 5.3-5.7. The COP values with Detached type, representing the highest heat load demand (Figure. 5.3) at high ambient temperature conditions are higher compared to other property types because of the part load factor and the continuous compressor operation near to the nominal speed values. In case of Flat type, representing the lowest heat demand the COP values at higher ambient temperature degrades due to lower speed operation and additional cycling losses. At lowest ambient temperature conditions of -2 ^{o}C, the Detached type building causes lower COP values due to inefficient compressor operation at full speed, while other property types of Semi-detached, end terraced, Mid terraced and Flat type shows COP improvement due to near compressor operation at nominal speed. Transient losses associated with the system are coming mainly due to cycling losses and defrost. During VSM the overall system efficiency is improved by continuous operation due to match between heat load demand and heat supplied over full range of experienced ambient temperature conditions for the five property types .



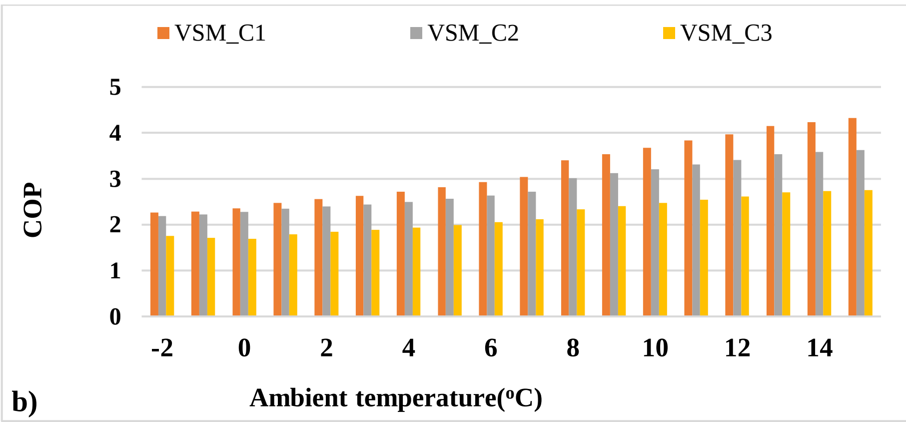


Figure 5.3 Detached type in VSM, a) without, b) with considering defrost

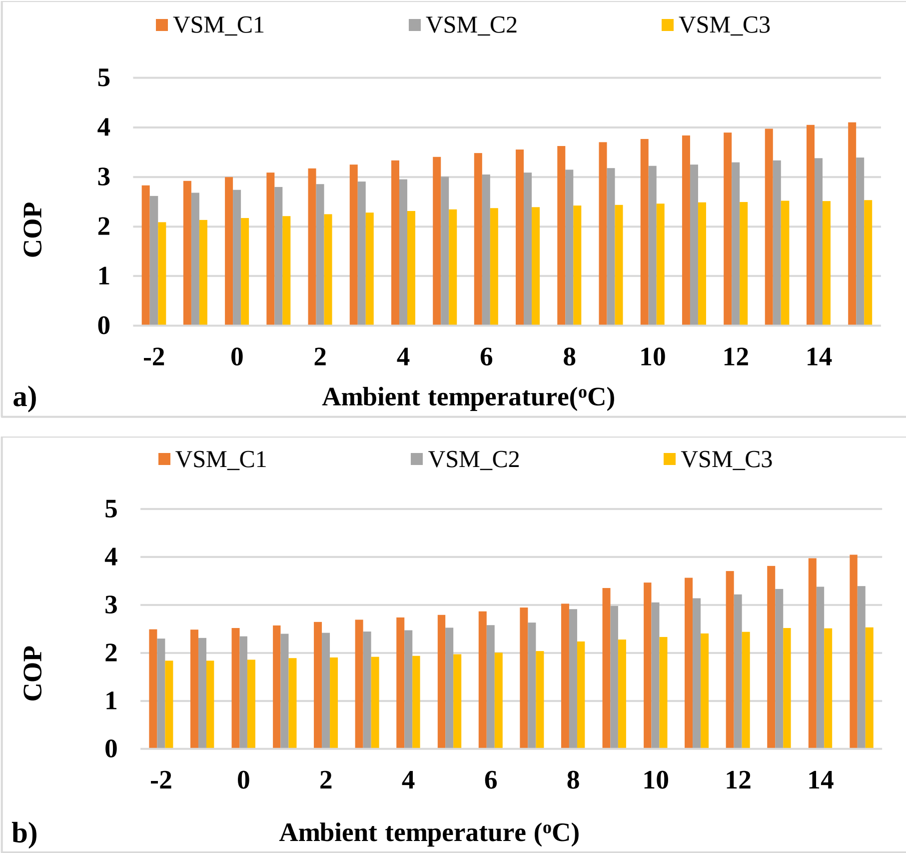


Figure 5.4 Semi-detached type, a) without, b) with considering defrost

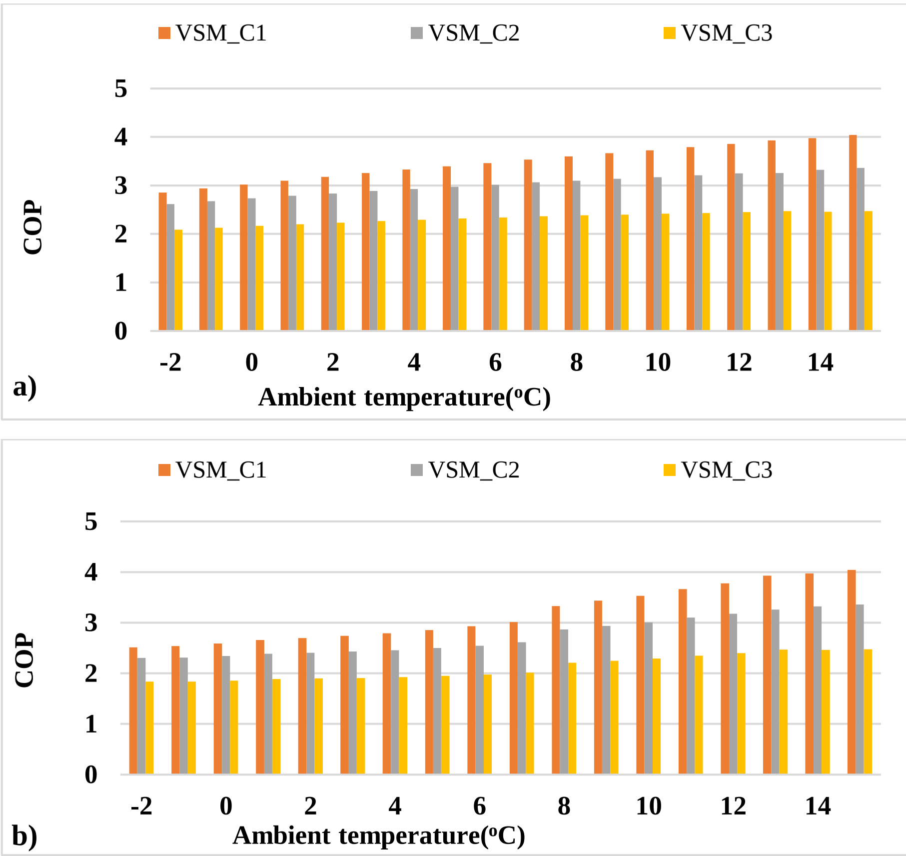


Figure 5.5 End Terraced, a) without, b) with considering defrost

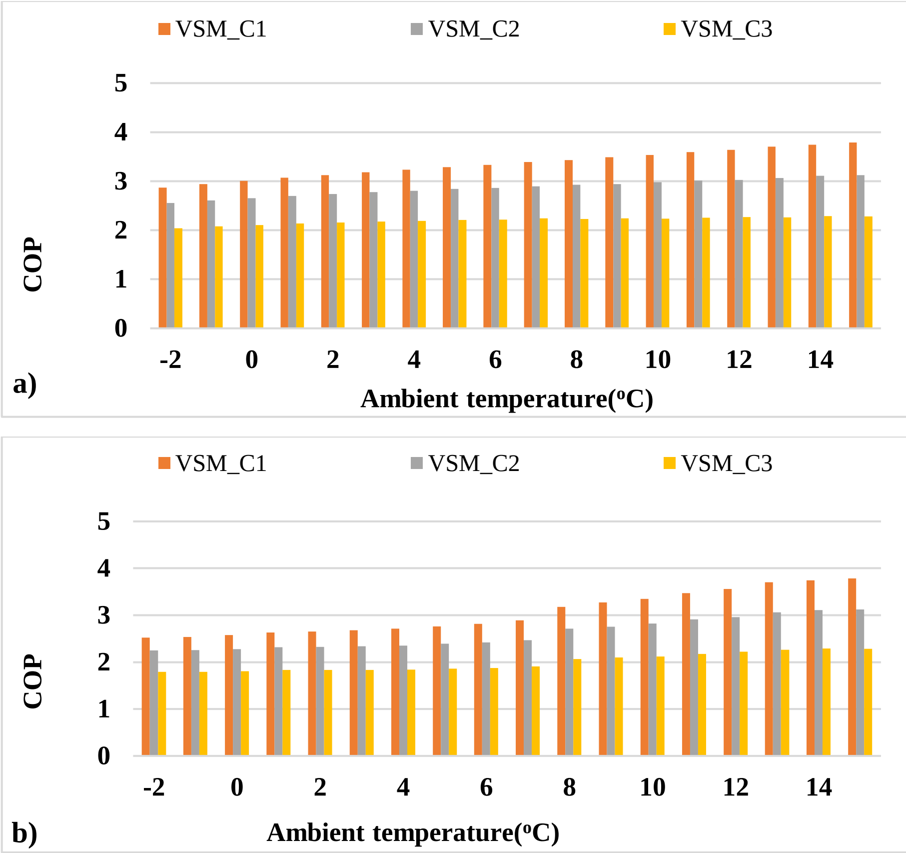


Figure 5.6 Mid terraced, a) without, b) with considering defrost

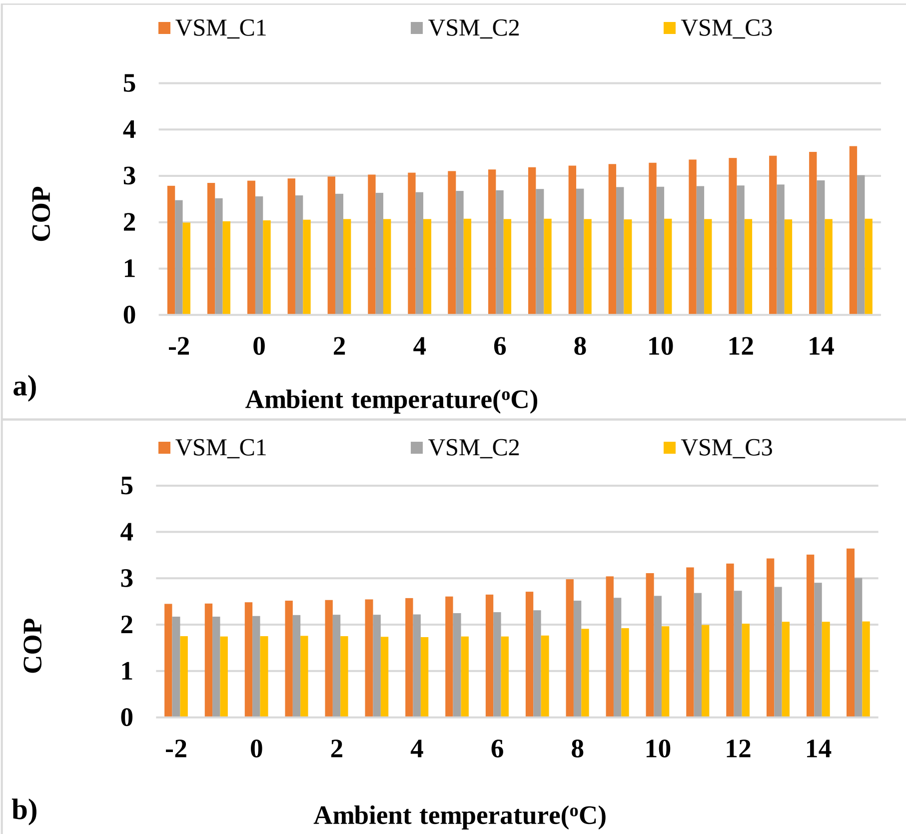


Figure 5.7 Flat type, a) without, b) with considering defrost

The main issue is frosting effect occurring at surface of evaporator at low ambient temperature and high relative humidity due to air flow rate reduction that causes low evaporating temperature and pressure, resulting in poor performance (COP) a higher electrical energy consumption and increased compressor pressure ratio. The COP values before/after frosting effect considerations based on testing results using interpolation and extrapolations techniques were presented in Figures 5.3- 5.7. Figure 3.8a shows COP values for the three cases based on the level of heat supply (C1, C2, C3) without considering the transient losses during FSM. The COP values were maximum in each bin due to nominal speed of operation. However, the maximum COP in each bin only would not help in favour of fixed speed operation for the system as it causes over and under heat production in case of respective operation from above and below to that of balance point. The overproduction causes overshoot the room temperature resulting in cycling losses and lower production causes the backup electrical heater requirements and both these impact network stability at large scale installations of such systems. The system operation in FSM at maximum COP was possible at nominal value, but without matching the heat load values and required ON/OFF cycles for the capacity control resulting in cycling losses.

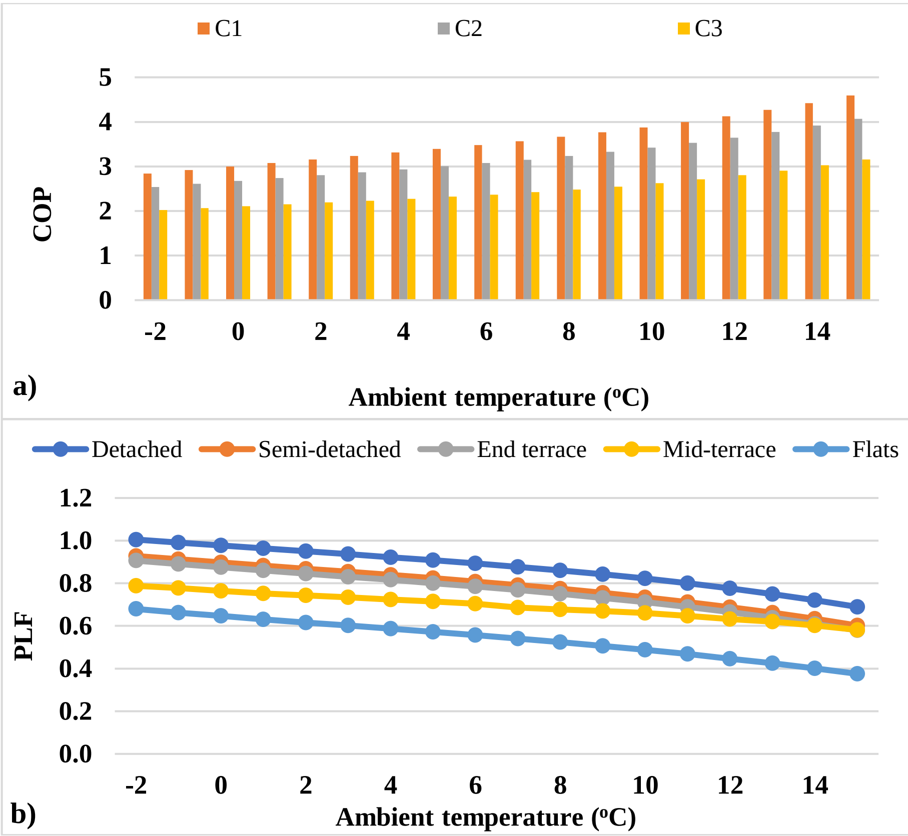
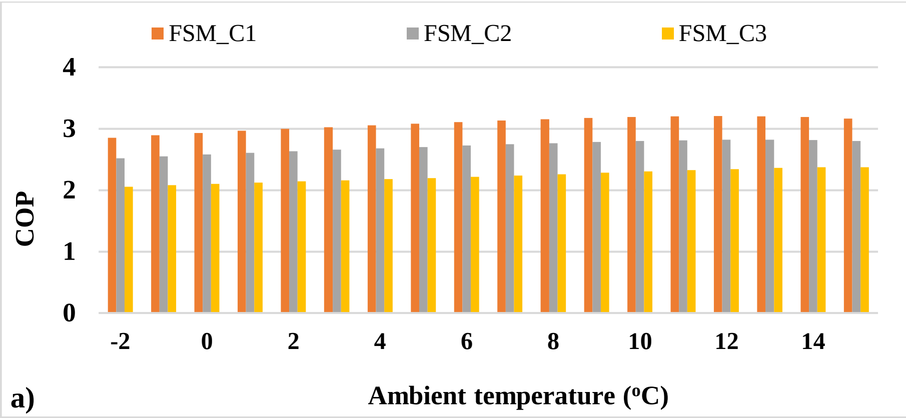
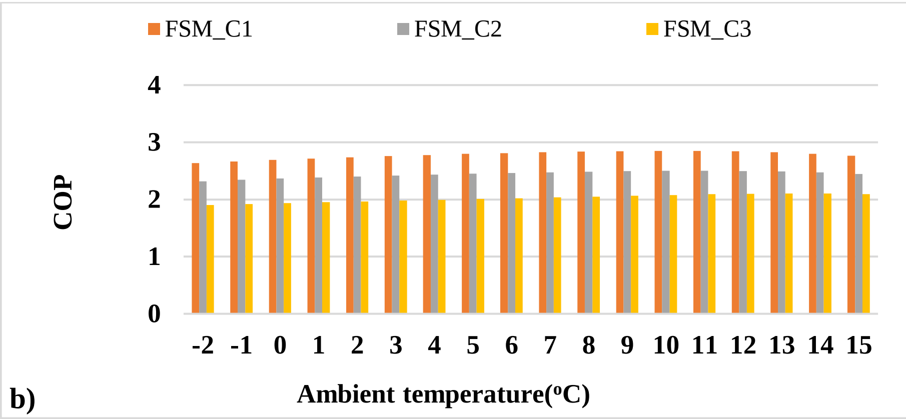


Figure 5.8 FSM operation a) COP values without considering cycling losses b) Correction factor for all five-property type according to thermal load demand

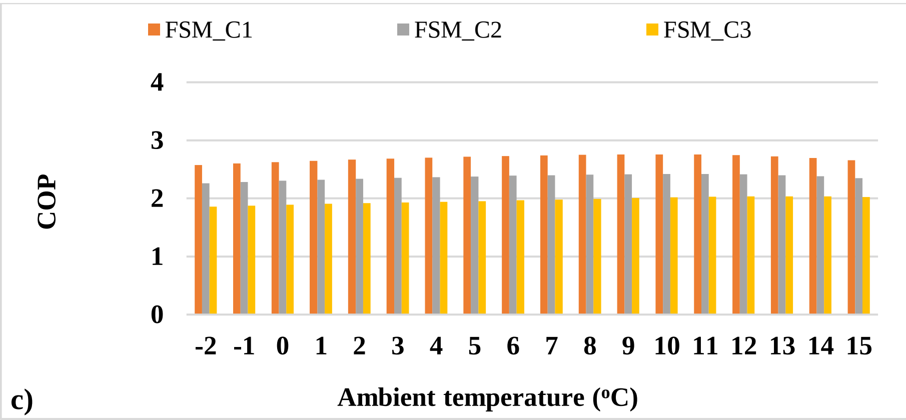
The intermittent operation has negative impact on compressor durability and thermostat overshoot/and undershoot issues. The PLF for the five property types considering the transient losses is shown in Figure 5.8b. The PLF depends upon the thermal load demand and HP part load operation in each bin for the individual property type. Figure 5.9(a-e) depicts the COP values for five property types at part load operation considering the cycling losses with FSM at three level of heat supply temperature (C1, C2, C3).



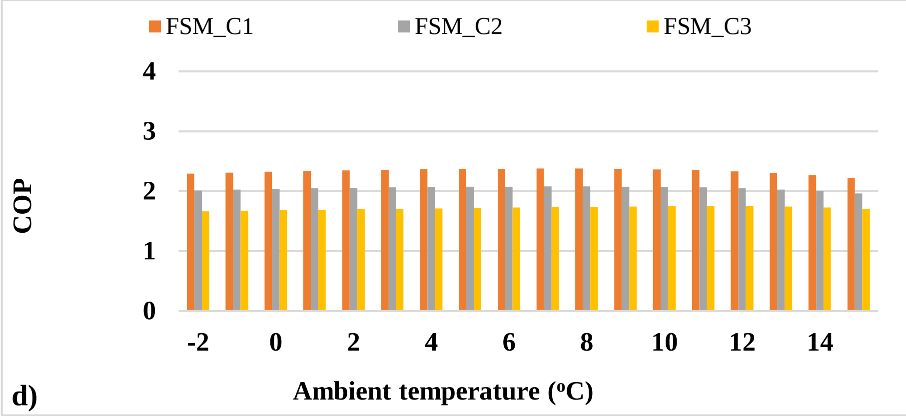
a)



b)



c)



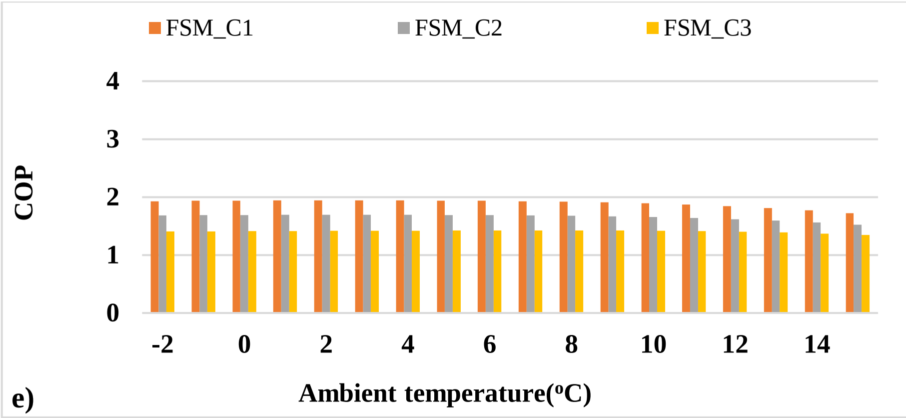


Figure 5.9 COP values after part load factor (PLF) with a) Detached type, b) Semi-detached type, c) End terrace type, d) Mid-terrace, e) Flat type

5.3.2 Electric power consumption and heating production difference (FSM vs. VSM)

The comfort level, network stability is higher in case of VSM because of good match between load demand and supplied energy. However, the thermal comfort, and network stability could not be the only criteria for choosing the VSM operation over FSM of control as the poor performance may results in higher energy consumption and carbon emissions. The operating mode selection needs to be prioritised in terms of performance as well and the VSM could be given preference when the cycling losses with FSM at maximum COP are large enough degrading the system overall performance. The VSM could meet the load demand for all property types without auxiliary heater requirements while in FSM the HP either produces extra heat and consume additional power or short of the heat and with requirements of the auxiliary heater. Difference in electric power consumption, and heating capacity for VSM against FSM for the respective five-property was shown in Figure 5.10(a, b)-5.14(a,b). The positive values indicates when the HP in FSM consumes additional power or produce extra heat while the negative values indicate when the HP could not meet the required heat load demand in FSM and auxiliary heater is required. Individual property type has its own balance point and moves towards left (lower ambient temperature values) as the load demand reduces from the Detached property type to the Flat type. The HP additional heating capacity production in FSM causes the fluctuation in temperature control inside the built environment on customer side. The network instability on grid side is also caused due to intermittent operation and higher power requirements resulting in grid frequency fluctuation. The additional heat produced causes reduction in ON-cycle duration and increase OFF-cycle. The thermostat temperature overshoot because of increased heating capacity at higher ambient temperature is one of the main issues specifically with the underfloor heating distribution system (35 ^{o}C) having thermal lag due to high thermal inertia. On the other hand, thermostat temperature undershoots become a problem at very low ambient temperature when the HP heating capacity reduces with low heat supply temperature. The balance point has impact on the overall annual COP for different property types. In case of Detached type, the HP with FSM of control the heat production was lower than the required heat load below the balance point of 9 ^{o}C. The back-up electric heating becomes essential for meeting the heat load in the range of -2 to 9 ^{o}C resulting in lower overall annual COP. In case of Semi-detached type building the back-up, electric requirements start with a balance point 6 ^{o}C and subsequently reduces to 5 ^{o}C, 0 ^{o}C in case of End terraced, and Mid terraced type. Flats type additional heat is produced over at all experienced ambient temperature conditions.

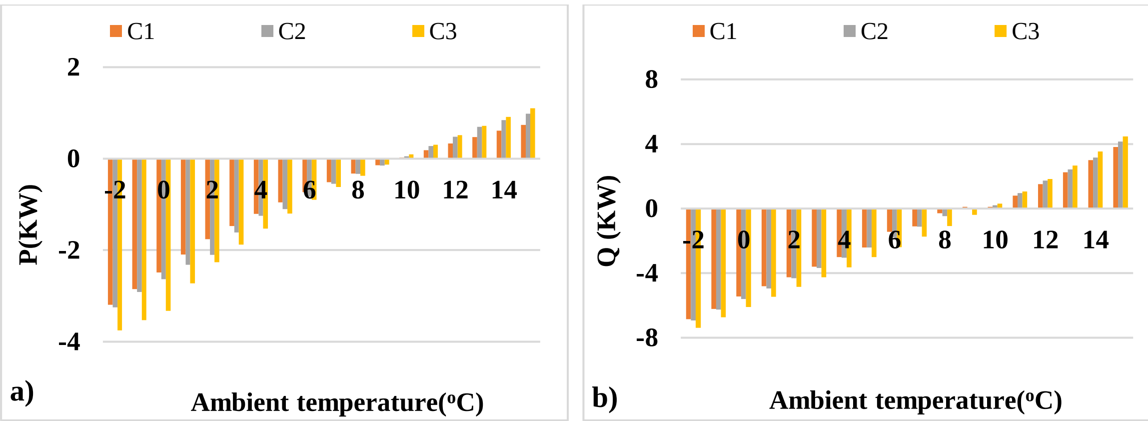


Figure 5.10 HP difference (VSM vs. FSM) for Detached type in a) Power (P) consumptions, b) heat production

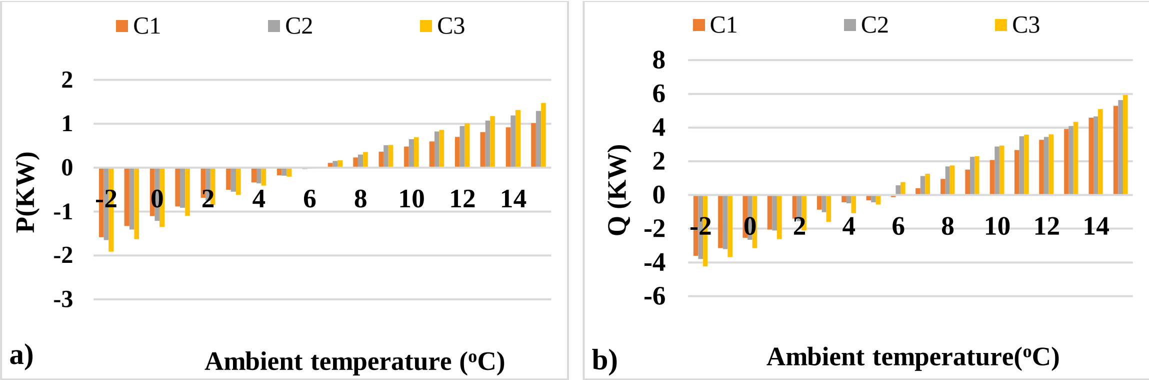


Figure 5.11 . HP difference (VSM vs. FSM) for Semi-detached type in a) Power (P) consumptions, b) heat production

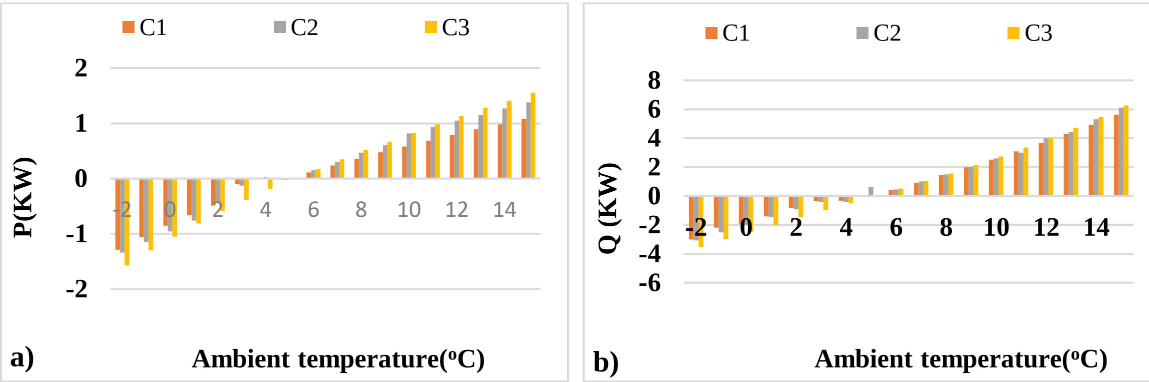


Figure 5.12 HP operating mode difference (VSM vs. FSM) for End terraced type in a) Power (P) consumptions, b) heat production

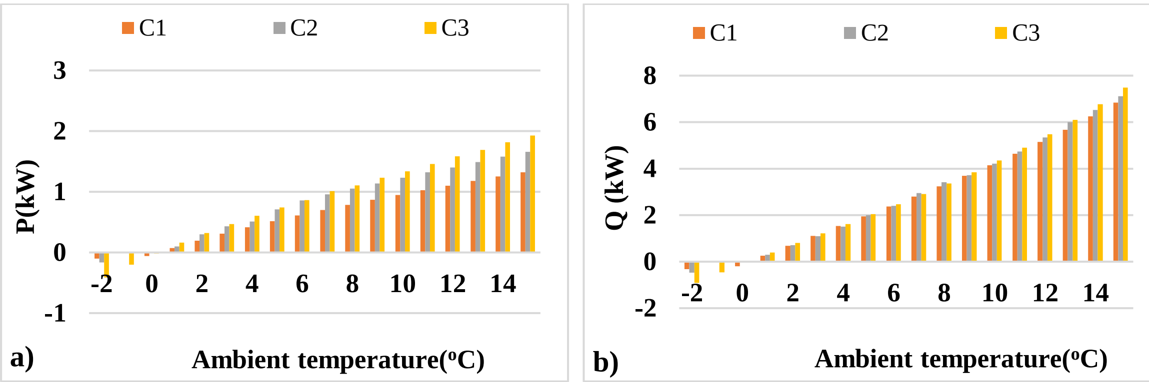


Figure 5.13 HP operating mode difference (VSM vs. FSM) for Mid terraced type in a) Power (P) consumptions, b) heat production

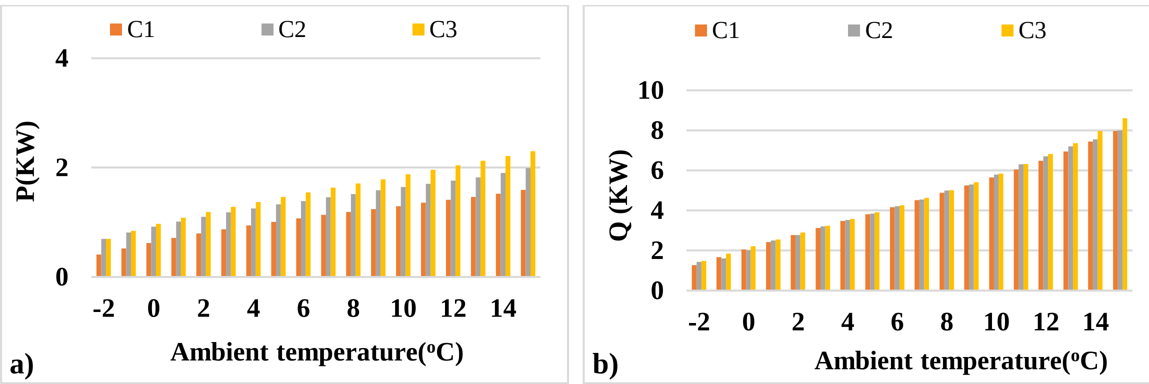


Figure 5.14 HP operating mode difference (VSM vs. FSM) for Flat type in a) Power (P) consumptions, b) heat production

The annual heat load demand for all five property types in each bin was shown in Figure 5.15. The Flat type representing the lowest heat demand in all bins with the Detached type building represent the highest load demand. The 7 °C occurs for the largest hours during the complete year resulting in the highest heat load demand for the Belfast climatic conditions. The property type with balance points closer to the 7 °C in Belfast climatic conditions would have even better performance even during FSM. The corresponding required electrical energy consumption for the HP (FSM & VSM) for the five property types is shown in Figure 5.16-5.20. As discussed above in the case of the FSM where the HP is unable to meet the required heat load demand at lower ambient temperature the back-up electric heater was utilized.

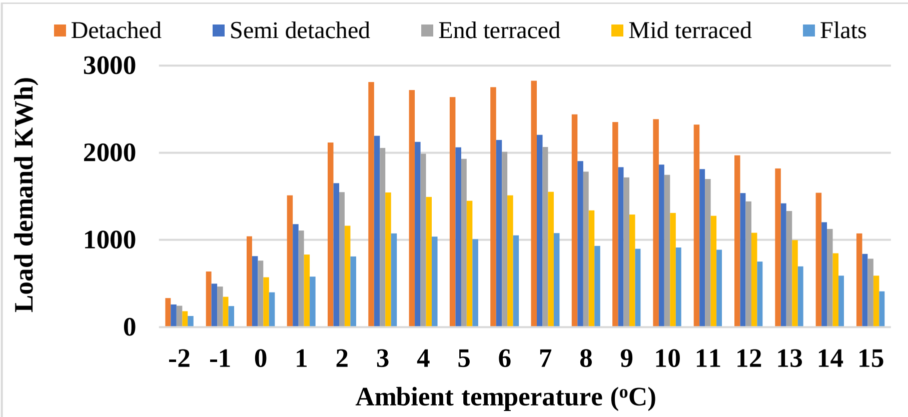


Figure 5.15 Annual heat load demand (KWh) for all property types

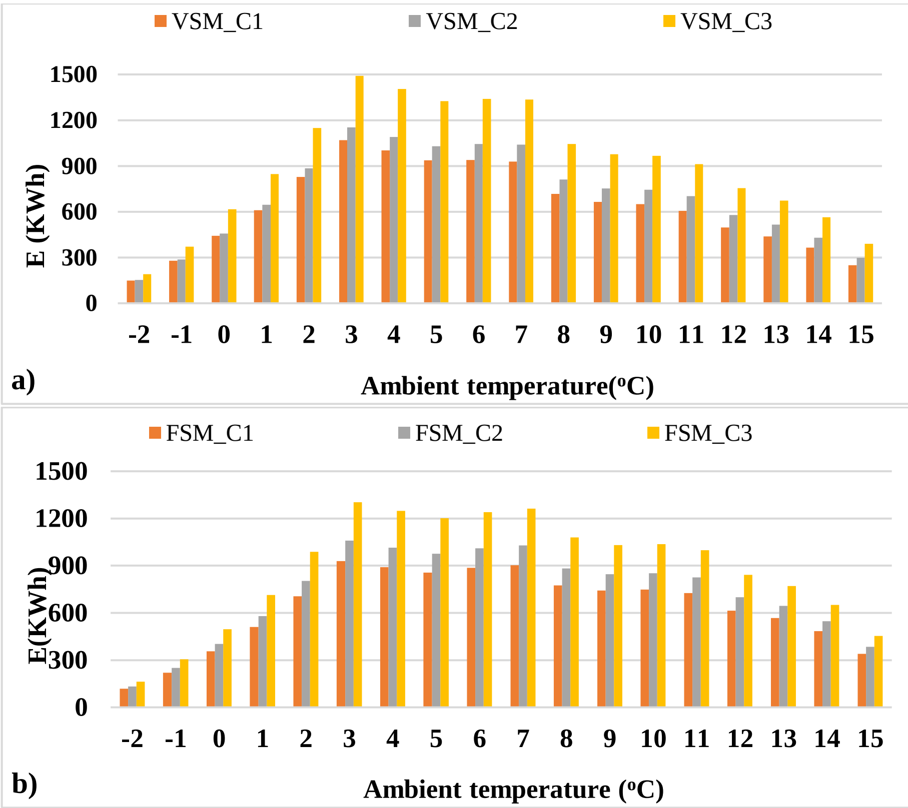


Figure 5.16 HP annual electrical energy consumption (E) for Detached type in a) VSM, b) FSM

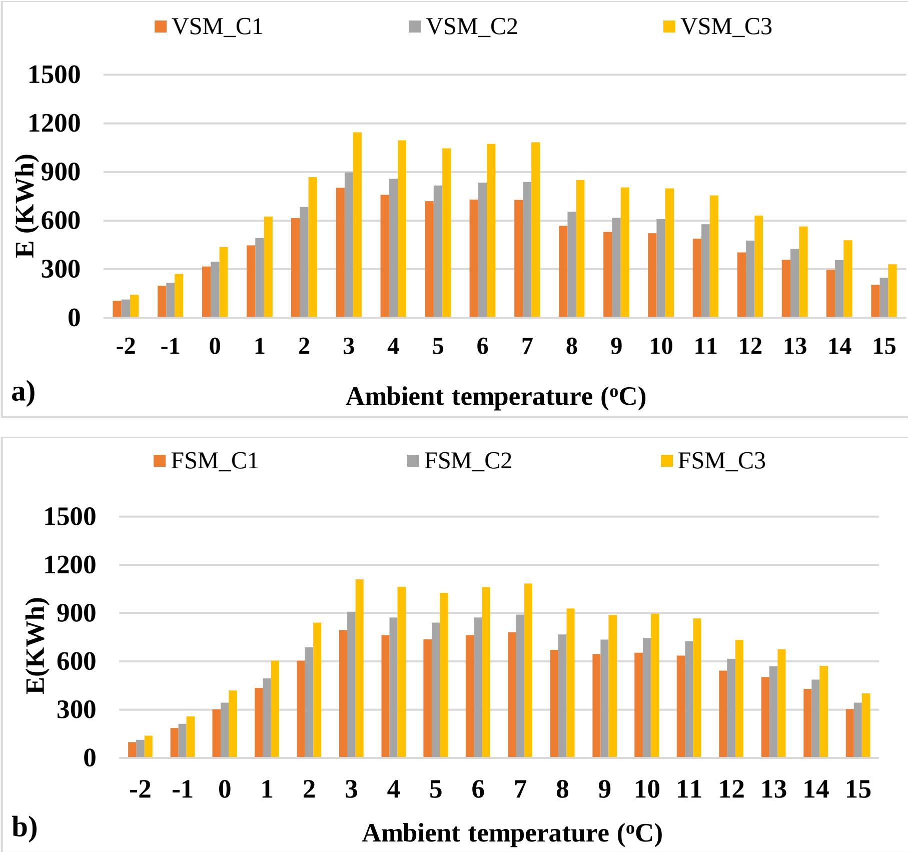


Figure 5.17 Annual energy consumption (E) for Semi-detached type in a) VSM, b) FSM

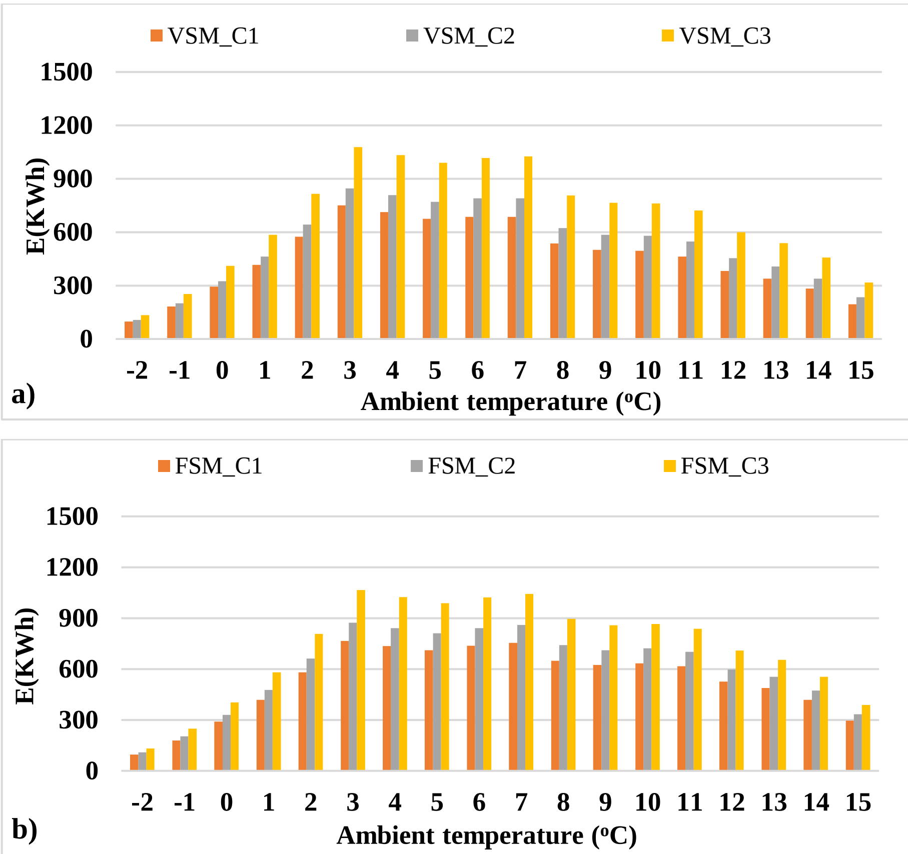


Figure 5.18 HP annual energy consumption (E) for End-terraced type in a) VSM, b) FSM

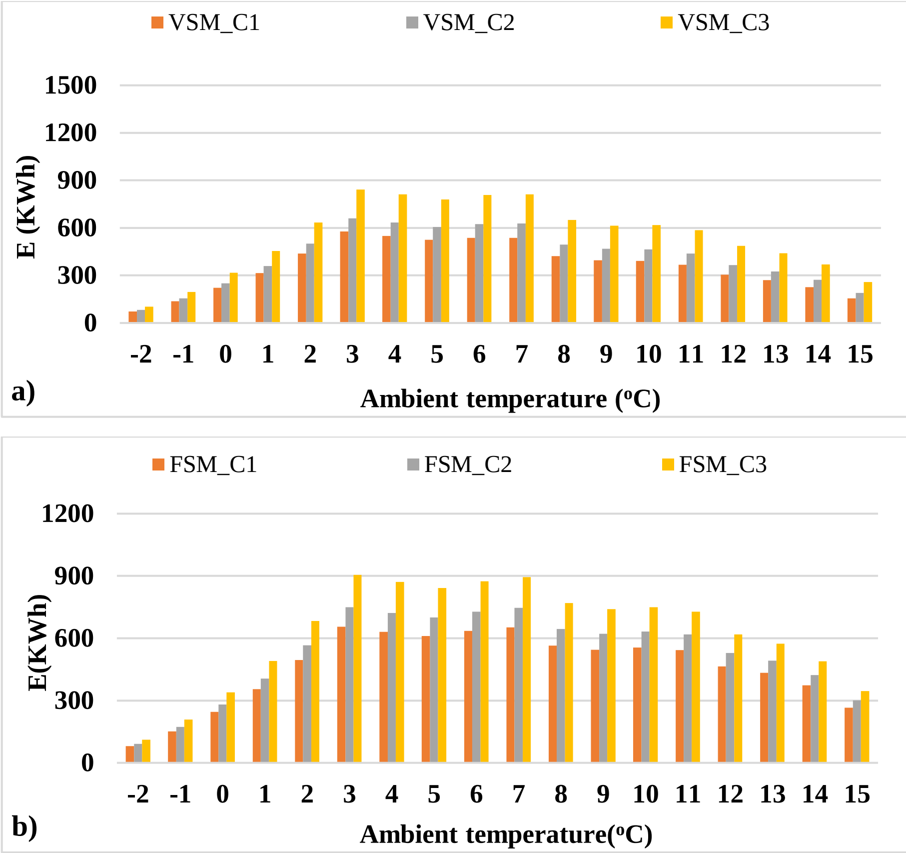


Figure 5.19 HP annual energy consumption (E) for Mid-terraced type in a) VSM, b) FSM

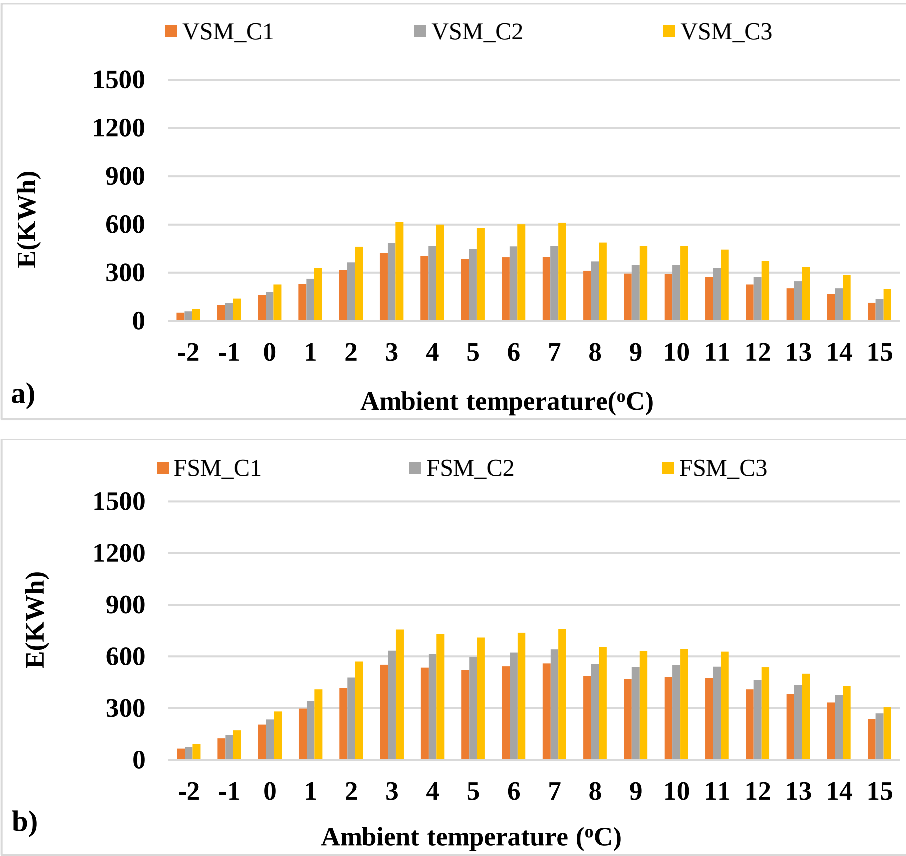


Figure 5.20 HP annual electrical energy consumption (E) for Flat type in a) VSM, b) FSM

5.4 Annual performances

The summary results for the overall annual COP, HP useful annual heat output, electrical energy demand in two modes of operation, three cases (C1, C2, C3) considered, back-up electric heater contributions for the five property types are presented in Table 5.1. The annual COP depends on the annual building load demand, considered case, back-up electric heater (EH) requirements, and operating mode of control.

Table 5.1: HP annual performance in different property types

Property type	**Heat output (KWh)**	**COP (VSM)**			**COP (FSM)**			**Electric demand (KWh)_VSM**			**Electric demand (KWh)_FSM**			
		C1	C2	C3	C1	C2	C3	C1	C2	C3	C1	C2	C3	EH
Flats	14564	2.89	2.46	1.87	1.87	1.64	1.40	5042	5912	7774	7774	8878	10405	0
Mid terrace	20914	3.05	2.61	2.00	2.30	2.03	1.71	6853	8008	10457	9090	10290	12229	52
End terrace	27846	3.16	2.75	2.12	2.43	2.26	1.90	8801	10144	13157	11448	12345	14693	892
Semi detached	29740	3.18	2.77	2.14	2.41	2.27	1.92	9356	10726	13884	12353	13120	15516	1516
Detached	38085	3.16	2.74	2.13	2.10	2.15	1.91	12067	13429	17433	18109	17676	19925	5333

The annual COP improvement, and energy savings due to operating mode of control (VSM vs. FSM), and heat supply temperature (C1 vs.C3) with different percentage for all property type depending on the part load factor was shown in Figure 5.21 & 5.22, respectively. The control mode of operations have significant impact on the COP improvement for the HP system but depends on the property type according to the heat load demand and operations over balance point (Figure 5.21). The highest improvement with COP values and energy savings at all three level of heat supply (C1, C2, C3) was achieved with Flat type, and this was due to the idle operations of the HP system at lower heat load demand and cycling losses during FSM. Similarly, the COP improves, and energy saving is increased with lower heat supply in comparison to higher heat supply temperature for both control mode but with different percentage values. The COP improvement and energy savings due to heat supply temperature during VSM is higher for all property types in comparison to the FSM. It could be observed that the VSM operation at lower heat supply temperature is more beneficial in comparison to high heat supply temperature. The annual COP during FSM is maximum with a value of 2.27 for the Semi-detached type due to balance point match and the maximum number of hours operation for Belfast Climatic conditions at 7 ^{o}C. The backup electric heater requirements are highest for the Detached type building with a demand of 5333 KWh during FSM.

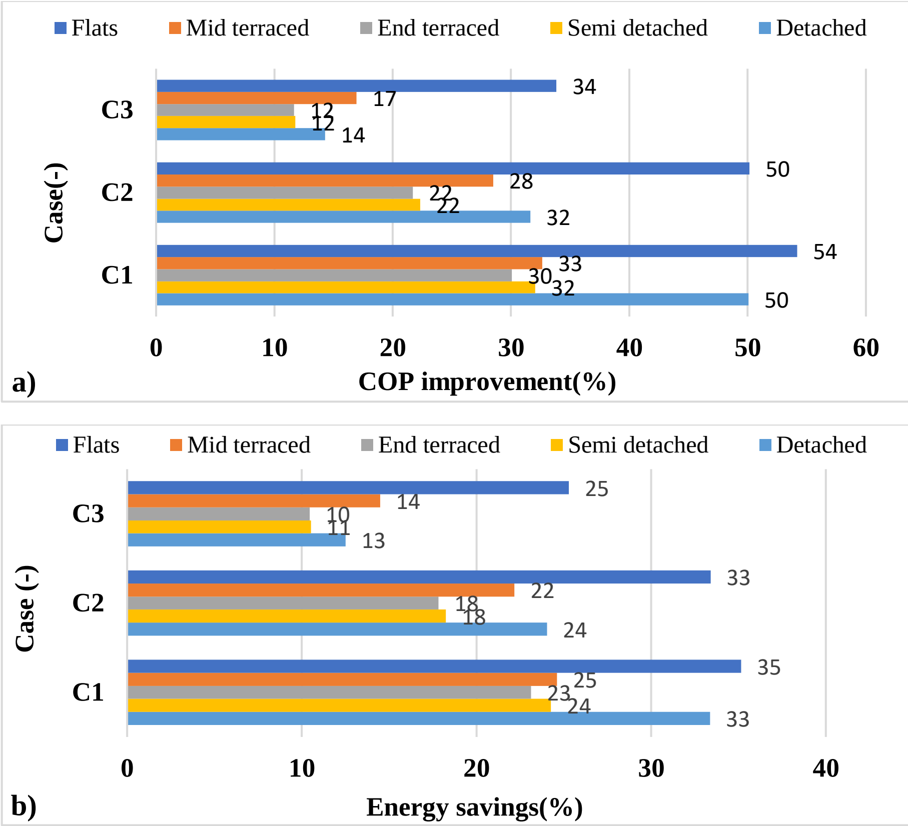


Figure 5.21 Improvement with operating mode (VSM vs. FSM) in annual a) COP's improvement (%), b) energy savings (%)

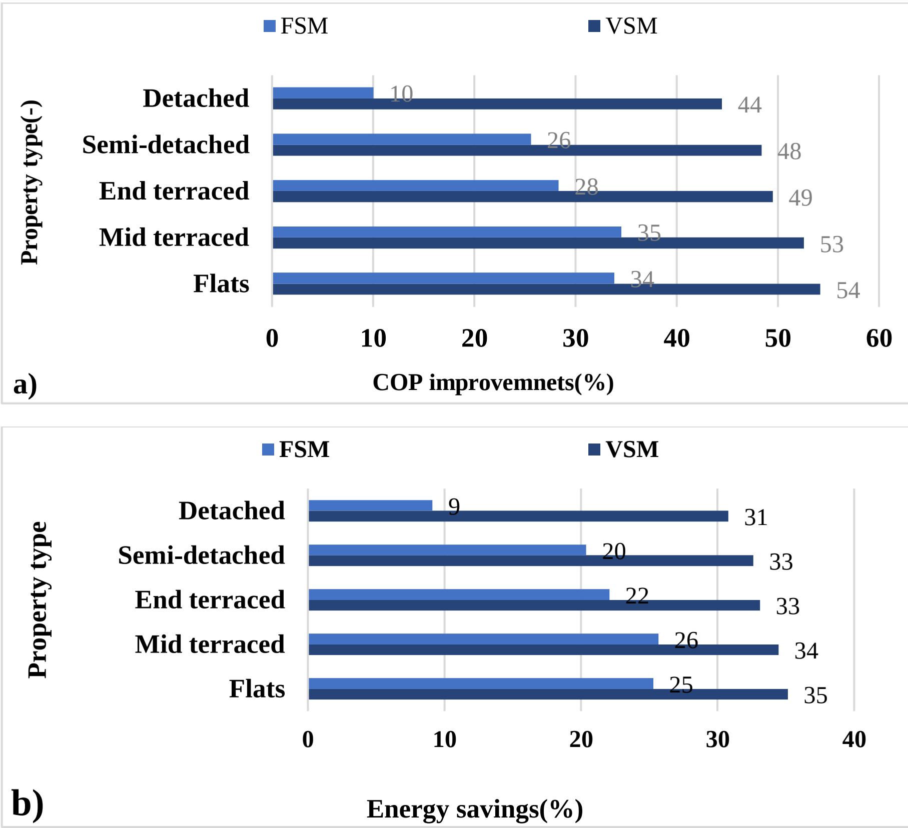


Figure 5.22 Improvement with heat supply temperature (C1 vs. C3) in a) COP, b) Energy savings (%)

5.4.1 Age period annual COPs

Table 5.2 summarizes the HP annual useful heat output, COPs, electric energy consumptions for the five property types, in two control modes, three cases, with four age period for Belfast region, UK. The heat demand depends on the property type, age period and increases for the older age period due to poor insulation properties, and lower thermal inertia. Among the five property types considered Flats representing the lowest annual heat demand ranges from 14564KWh to 6670KWh, with Detached type representing highest annual heat demand in the range of 38085KWh to 25807KWh with the corresponding age period with the maximum demand for the old age period (1900-1949). The HP annual COP for different property types depends on the HP operating capacity and part load ratio, and mode of control. The COP relation with the heat demand variation due to age factors for all five property types was not linear but depends on the HP heating capacity match to the load demand in each bin and part load factor. The HP proves to be least efficient in Flat's type for all age period compared to other type and this was because of higher losses and lower part load operations. Among the Flat type of old age period (1900-1949), representing the highest COP. The PLF with different load demand for all five property types according to the four age period of 1900-1949, 1950-1975, 1976-1990, and 1991-2007 was depicted in Figure 5.23 (a,b,c,d).

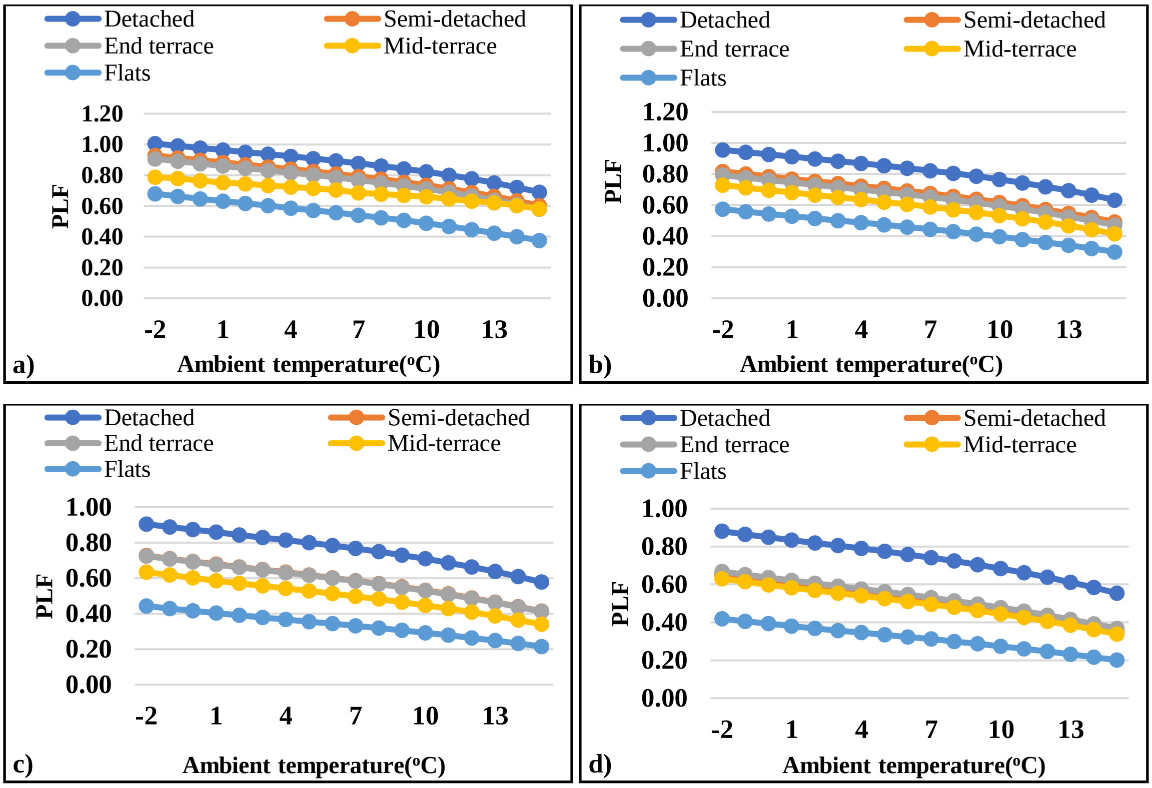


Figure 5.23 Part load factor (PLF) with age periods of a) 1900-1949, b) 1950-1975, 1976-1990, 1991-2007

Table 5.2: HP performance in different property types, age period.

Property type	Age period	Annual heat output	COPS VSM C1	C2	C3	FSM C1	C2	C3	Annual Electric demand (KWh) VSM C1	C2	C3	FSM C1	C2	C3
	1900-1949	14564	2.89	2.46	1.87	1.87	1.64	1.40	5042	5912	7774	7774	8878	10405
	1950-1975	10806	2.79	2.38	1.80	1.53	1.34	1.16	3878	4536	6009	7050	8061	9350
	1976-1990	7224	2.73	2.34	1.73	1.14	0.99	0.87	2642	3088	4164	6359	7281	8343
Flats	1991-2007	6670	2.74	2.34	1.74	1.07	0.93	0.81	2433	2846	3843	6252	7161	8187
	1900-1949	20914	3.05	2.61	2.00	2.30	2.03	1.71	6853	8008	10457	9090	10290	12229
	1950-1975	16749	2.95	2.51	1.91	2.04	1.79	1.52	5684	6663	8747	8200	9354	11021
Mid	1976-1990	12787	2.84	2.43	1.79	1.72	1.51	1.29	4510	5270	7125	7432	8492	9906
terrace	1991-2007	12671	2.83	2.43	1.80	1.71	1.50	1.28	4470	5221	7056	7409	8466	9873
	1900-1949	27846	3.16	2.75	2.12	2.43	2.26	1.90	8801	10144	13157	11448	12345	14693
	1950-1975	20069	3.03	2.59	1.98	2.26	1.99	1.68	6620	7742	10126	8896	10094	11977
End	1976-1990	16516	2.94	2.53	1.88	2.03	1.78	1.51	5618	6531	8779	8154	9303	10955
terrace	1991-2007	14127	2.87	2.43	1.80	1.84	1.61	1.37	4923	5821	7867	7690	8783	10283
	1900-1949	29740	3.18	2.77	2.14	2.41	2.27	1.92	9356	10726	13884	12353	13120	15516
	1950-1975	21409	3.06	2.66	2.00	2.32	2.06	1.73	6993	8061	10727	9215	10404	12380
Semi	1976-1990	16632	2.95	2.54	1.89	2.03	1.78	1.51	5644	6560	8815	8177	9328	10988
detached	1991-2007	13370	2.85	2.44	1.81	1.77	1.55	1.33	4690	5476	7399	7544	8618	10070
	1900-1949	38085	3.16	2.84	2.18	2.10	2.15	1.91	12067	13429	17433	18109	17676	19925
	1950-1975	32449	3.19	2.83	2.18	2.31	2.25	1.93	10179	11467	14910	14043	14402	16795
	1976-1990	27701	3.16	2.78	2.12	2.42	2.26	1.89	8761	9975	13097	11432	12268	14630
Detached	1991-2007	25807	3.14	2.75	2.08	2.43	2.22	1.86	8219	9392	12379	10613	11605	13876

The overall annual COP trend in VSM shows increase with the increase of building heat demand for four property types (Flats, Mid terrace, End terrace, Semi-detached), and get reduced for Detached types due to the operating point for the HP. The age factor effects the COP values for the specific case studied, and control mode.

5.4.2 Energy consumptions

The annual electricity consumption depends on property type, age period, COP value, control mode. The old age period consumes more electrical energy due to poor insulation and higher heat demands for the same property type. Among all the property type and age period considered, the detached building consumes with age period of (1900-1949) consume the highest energy with poor thermal inertia, while the Flats with the age period (1991-2007 onwards) consume the least energy due to lower heat demands because of lower heat loss.

5.4.3 Performance improvement with operating mode of operation (VSM vs. FSM)

The change in mode of control, heat supply temperature impacts the system COPs, and energy consumptions all property types, and ages period but with different percentage amount. The COP values during FSM are smaller than VSM with comparative results and improvement displayed in Figure 5.24. For all age period the highest percentage (%) improvement in COP is possible for the Flats (C1) and the reason for this is higher cycling losses and lower part load fact value shown earlier (Figure 5.23) for each building type, and all age period during FSM followed by C2& C3.

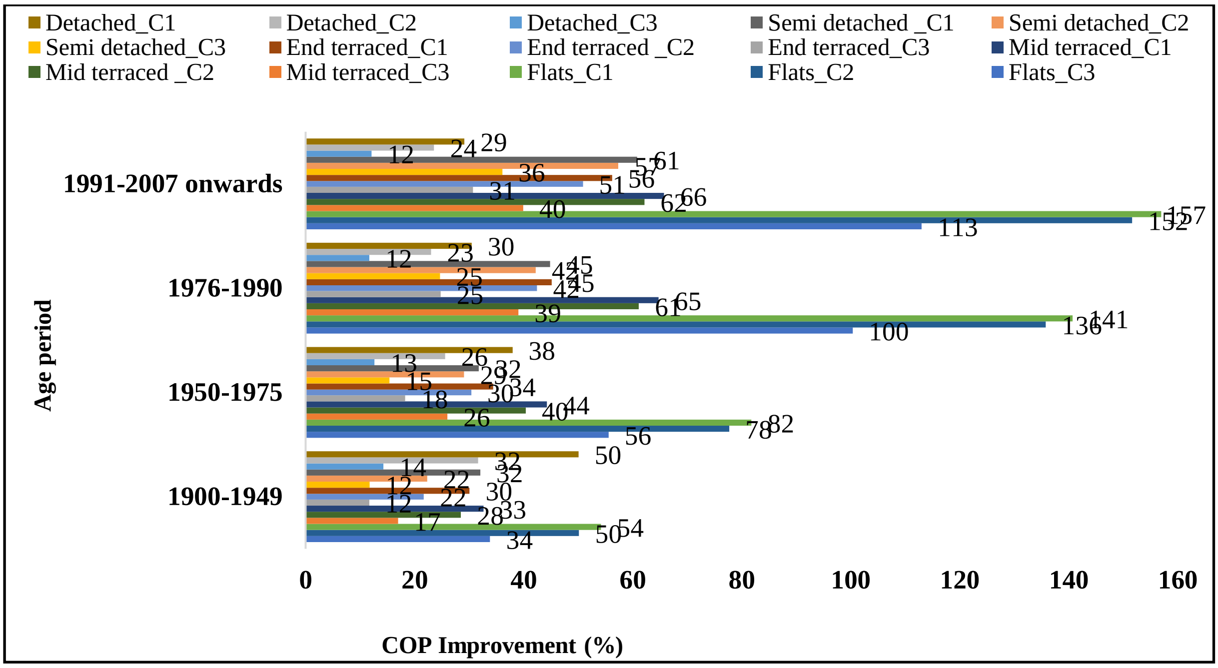


Figure 5.24 HP annual a) COP improvement (%) with operating mode (VSM vs. FSM) for all property type and age period

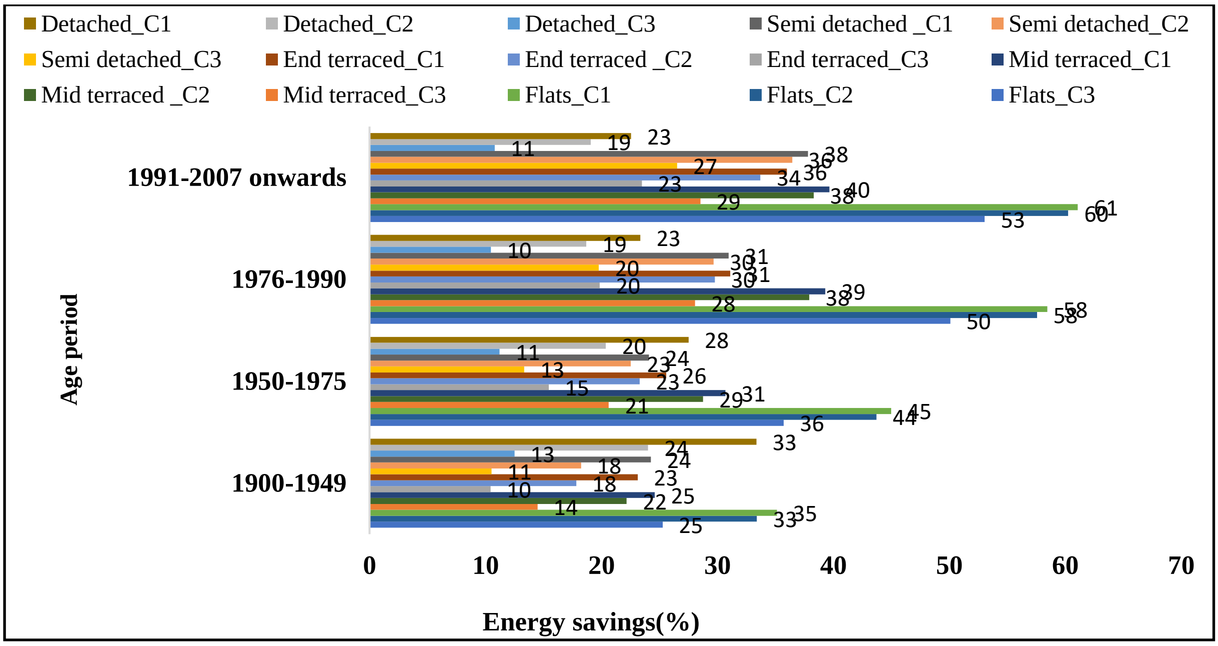


Figure 5.25 . HP annual energy savings (%) due to operating mode of control (VSM vs. FSM) for all property type and age period

The lowest improvement (%) in COP occurs in detached type (C3) for all age period except the first age period (1900-1949), where the lowest improvement occurs for the end terraced (C3) with value of 11.67%. The minimum COP improvement occurs in case of Detached building with 10-12% (C3) was possible for the detached building. The lowest improvement for VSM over FSM is because of reduced cycling losses. The annual energy savings (%) is obtained with VSM in comparison to FSM is shown in Figure 5.25. Among the considered cases, building type, and age period the highest energy savings of up to 61% was possible for the Flats(C1) with age period (1991-2007 onwards).

5.5 Other climatic conditions (VSM of operations considered only)

This section is dedicated to investigating the climatic conditions impact on the HP performance for the three cases, cost, and carbon emission analysis for five building property types, and three age period only in VSM. The FSM of the developed HP have not been considered analysed in this section as the FSM control method have has either no effect or very little impact due to climatic conditions evidenced from the previous section results for 1900s Mid terrace house. The performance at climatic conditions of Valentia, Dublin, Belfast, and Aviemore for each property type is depicted in respective tables (5.3-5.7). Although the Belfast climatic conditions have been investigated and analysed for the HP in two control modes (VSM& FSM) in previous section for all property types and age period. Therefore, in this section for Belfast climatic conditions, only the HP performance result have been shown for comparison purposes. The retrofit assessment has been presented only in VSM of control for other three climatic conditions in next chapter.

5.5.1 Annual Performance analysis (COP)

The influence of the climactic conditions on the building heat demand and the corresponding impact on the HP performance for the four considered location have been shown in Table 5.3-5.7. Valentia with mildest climatic conditions requiring annual heat demand for all property type and corresponding age period is comparatively lower than the other climatic conditions for the same kind property types, and age period. Among the five property types considered, the Flats type building representing the lowest heat demand while the Detached type building representing highest annual heat demand for the corresponding age period and climatic conditions. The old age period (1900-1949), Flats type having the highest annual heat demand with a value of 13304 KWh in Valentia and increases to 15944KWh in case of Aviemore, with the respective values of 6093KWh, and 7303KWh for the modern Flats type building with age period duration of 1991-2007 onwards. In Valentia, the Detached type building has the highest annual heat demand of 34789KWh for the old age period (1900-1949) with least value of 23574 KWh for the modern building (1991-2007 onwards), and this become 41694KWh &28253KWh in Aviemore. These annual heat demand variations due to climatic conditions has also influence on the annual COP and electrical energy consumption of the HP. This has been discussed in the following sections. In Valentia, the annual COP for the developed HP have highest value for all building types and corresponding age period and degrades with the HDDS increase for colder climatic conditions due to bin distribution difference. Among the five property types considered the Detached building shows higher annual COP value trend, with the HP (C1) annual COP of 3.51 for old age period (1900-1949) in Valentia and become 2.9 for Aviemore with a percentage reduction of 17.3%. The HP performance for Dublin and Belfast lies in between the milder and colder climatic conditions, with respective annual COP values of 3.25 and 3.16. The Flats property type represent the lowest COP values among all five considered property types. The age period impact on the annual COP for all property types have been discussed in section **3.7** and hence will not be discussed here to avoid repetition. In Valentia, the Flats type building for old age period (1900-1949) have annual COP value of 3.08 and become 2.76 for Aviemore with a percentage reduction of 27.3%. The annual COP values for Dublin & Belfast is 2.94 & 2.89 for the same age period. The other property types including Mid terrace, End terrace, and Semi-detached have shown similar impact on annual COP values due to climatic conditions. The Mid terrace house old age period (1900-1949) annual COP values for the four climatic conditions of Valentia, Dublin, Belfast, and Aviemore are 3.27, 3.11, 3.05, 2.9, respectively.

5.5.2 Annual electrical energy consumption

The annual electrical energy consumption is strongly influenced by the climatic conditions, age period, property type and cases considered, and control modes. As mentioned earlier the focus of this section is climatic conditions impact on the system performance and energy consumption, as the other factors have been thoroughly investigated earlier. For all property types, age period the corresponding cases considered the climatic conditions for Valentia is less demanding for energy in comparison to other climatic conditions of Dublin, Belfast, and Aviemore. For example, Semi-detached type (1900-1949), the annual electrical energy consumption for the HP (C1) is 7875KWh in Valentia and increases to 8854KWh, 9356KWh, and 10933KWh for Dublin, Belfast and Aviemore, respectively. For detached type property (1900-1949), the energy consumption increases due to heat demand to 9917 KWh, 11341 KWh, 12067KWh, 14370 KWh for the respective four climatic conditions.

Table 5.3: HP performance in Flat type property at all age period and four locations

	Annual heat output (KWh)		COPS			Annual Electric demand (KWh)		
Location	**Age**		C1	C2	C3	C1	C2	C3
	1900-1949	13304	3.08	2.59	1.93	4325	5133	6876
	1950-1975	9871	2.97	2.50	1.84	3323	3942	5350
	1976-1990	6599	2.93	2.47	1.78	2250	2671	3717
Valentia	1991-2007 onwards	6093	2.93	2.47	1.78	2076	2465	3431
	1900-1949	14106	2.94	2.50	1.89	4795	5640	7459
	1950-1975	10467	2.84	2.42	1.81	3688	4331	5775
	1976-1990	6997	2.79	2.37	1.75	2510	2946	4009
Dublin	1991-2007 onwards	6461	2.79	2.38	1.75	2313	2716	3700
	1900-1949	14564	2.89	2.46	1.87	5042	5912	7774
	1950-1975	10806	2.79	2.38	1.80	3878	4536	6009
	1976-1990	7224	2.73	2.34	1.73	2642	3088	4164
Belfast	1991-2007 onwards	6670	2.74	2.34	1.74	2433	2846	3843
	1900-1949	15944	2.76	2.38	1.84	5781	6708	8688
	1950-1975	11831	2.66	2.31	1.78	4440	5128	6658
	1976-1990	7908	2.61	2.27	1.73	3025	3484	4561
Aviemore	1991-2007 onwards	7303	2.62	2.27	1.73	2785	3212	4218

Table 5.4: HP performance in Mid-terraced type at all age period and four locations

	Annual heat output (KWh)		COPS			Annual Electric demand		
Location	Age		C1	C2	C3	C1	C2	C3
	1900-1949	19104	3.27	2.76	2.09	5850	6918	9158
	1950-1975	15299	3.14	2.65	1.98	4871	5777	7710
	1976-1990	11681	3.02	2.55	1.84	3870	4582	6360
Valentia	1991-2007 onwards	11574	3.02	2.55	1.84	3832	4536	6292
	1900-1949	20257	3.11	2.65	2.02	6511	7634	10012
	1950-1975	16222	3.00	2.55	1.93	5404	6356	8385
	1976-1990	12385	2.89	2.46	1.81	4290	5030	6850
Dublin	1991-2007 onwards	12272	2.89	2.46	1.81	4250	4983	6783
	1900-1949	20914	3.05	2.61	2.00	6853	8008	10457
	1950-1975	16749	2.95	2.51	1.91	5684	6663	8747
	1976-1990	12787	2.84	2.43	1.79	4510	5270	7125
Belfast	1991-2007 onwards	12671	2.83	2.43	1.80	4470	5221	7056
	1900-1949	22896	2.90	2.51	1.94	7901	9134	11794
	1950-1975	18336	2.81	2.42	1.87	6527	7574	9809
	1976-1990	13999	2.71	2.35	1.77	5165	5966	7891
Aviemore	1991-2007 onwards	13872	2.71	2.35	1.77	5120	5913	7818

Table 5.5: HP performance in End terraced type at all age period and four climatic conditions

	Annual heat output (KWh)		**COPS**			**Annual Electric demand (KWh)**		
Location	Age		C1	C2	C3	C1	C2	C3
	1900-1949	25437	3.42	2.92	2.23	7435	8697	11404
	1950-1975	18333	3.24	2.74	2.06	5656	6694	8880
	1976-1990	15087	3.14	2.66	1.94	4812	5662	7781
Valentia	1991-2007 onwards	12905	3.05	2.55	1.84	4230	5057	7016
	1900-1949	26971	3.23	2.79	2.15	8338	9653	12564
	1950-1975	19439	3.09	2.63	2.00	6290	7382	9696
	1976-1990	15997	3.00	2.57	1.90	5341	6230	8425
Dublin	1991-2007 onwards	13683	2.92	2.46	1.81	4683	5556	7563
	1900-1949	27846	3.16	2.75	2.12	8801	10144	13157
	1950-1975	20069	3.03	2.59	1.98	6620	7742	10126
	1976-1990	16516	2.94	2.53	1.88	5618	6531	8779
Belfast	1991-2007 onwards	14127	2.87	2.43	1.80	4923	5821	7867
	1900-1949	30486	2.97	2.62	2.04	10248	11644	14960
	1950-1975	21971	2.88	2.49	1.93	7624	8823	11405
	1976-1990	18081	2.80	2.44	1.85	6449	7421	9799
Aviemore	1991-2007 onwards	15466	2.74	2.35	1.77	5638	6593	8717

Table 5.6: HP performance in Semi-detached type at all age period and four locations

	Annual heat output (KWh)		**COPS**			**Annual Electric demand (KWh)**		
Location	Age		C1	C2	C3	C1	C2	C3
	1900-1949	27166	3.45	2.96	2.27	7875	9174	11989
	1950-1975	19557	3.28	2.81	2.08	5966	6955	9424
	1976-1990	15193	3.14	2.67	1.94	4837	5689	7815
Valentia	1991-2007 onwards	12213	3.03	2.57	1.85	4024	4760	6597
	1900-1949	28805	3.25	2.82	2.18	8854	10201	13244
	1950-1975	20736	3.12	2.70	2.02	6643	7683	10278
	1976-1990	16109	3.00	2.57	1.90	5367	6258	8461
Dublin	1991-2007 onwards	12950	2.90	2.48	1.82	4461	5227	7112
	1900-1949	29740	3.18	2.77	2.14	9356	10726	13884
	1950-1975	21409	3.06	2.66	2.00	6993	8061	10727
	1976-1990	16632	2.95	2.54	1.89	5644	6560	8815
Belfast	1991-2007 onwards	13370	2.85	2.44	1.81	4690	5476	7399
	1900-1949	32558	2.98	2.64	2.06	10933	12341	15834
	1950-1975	23438	2.91	2.55	1.94	8065	9207	12069
	1976-1990	18208	2.81	2.44	1.85	6482	7456	9843
Aviemore	1991-2007 onwards	14637	2.72	2.36	1.78	5372	6203	8206

Table 5.7: HP performance in Detached type at all age period and four climatic conditions

Annual heat output (KWh)			COPS			Annual Electric demand (KWh)		
Location	Age		C1	C2	C3	C1	C2	C3
	1900-1949	34789	3.51	3.08	2.37	9917	11289	14674
	1950-1975	29641	3.48	3.05	2.32	8506	9704	12783
	1976-1990	25304	3.42	2.97	2.23	7404	8527	11361
Valentia	1991-2007 onwards	23574	3.38	2.93	2.19	6970	8055	10783
	1900-1949	36888	3.25	2.90	2.24	11341	12706	16504
	1950-1975	31429	3.27	2.89	2.21	9613	10872	14195
	1976-1990	26830	3.23	2.83	2.14	8301	9485	12510
Dublin	1991-2007 onwards	24996	3.21	2.80	2.11	7794	8937	11835
	1900-1949	38085	3.16	2.84	2.18	12067	13429	17433
	1950-1975	32449	3.19	2.83	2.18	10179	11467	14910
	1976-1990	27701	3.16	2.78	2.12	8761	9975	13097
Belfast	1991-2007 onwards	25807	3.14	2.75	2.08	8219	9392	12379
	1900-1949	41694	2.90	2.66	2.06	14370	15685	20283
	1950-1975	35524	2.97	2.67	2.08	11971	13314	17106
	1976-1990	30326	2.97	2.64	2.04	10199	11482	14884
Aviemore	1991-2007 onwards	28253	2.96	2.62	2.01	9537	10781	14024

5.6 Chapter summary

The heat pump (HP) technology was found potential candidate for domestic heat load applications for the range of housing stock in UK. Five different old age period (1900-1949) residential property types were considered in the analysis for Belfast climatic conditions. The system meets the required heat load demand for all property types with improved performance by matching the heat supplied to that of heat load demand. In this study the ASHP annual COP in Belfast climatic conditions have been assessed for five different property types based on the testing and modelling results analysis. The impact of heat supply temperature and control mode of operation have significant impact on the system performance. The FSM operation for the maximum COP during steady state conditions was not valuable in terms of overall annual COP due to transient losses. The following conclusions could be extracted from the above study.

a) The VSM of operation is more beneficial in terms of annual COP, electric power consumption compared to FSM.
b) During FSM operations largest annual COP occurs for property type when the balance point and bin temperature (for largest number of hours) falls closely to each other.
c) The COP improvement and the corresponding energy savings during VSM due to heat supply temperature level was higher than FSM for all age periods, property types.

Chapter 6: HP Retrofit Assessment

6.1 Introduction

In this study the HP performance and retrofit assessment for the 1900s Mid-terraced test house is conducted first followed by other property types investigated in chapter 5. The HP retrofit instead of other heating technologies including very high heat supply temperature, electric heating option, oil/gas boilers at range of percentage efficiencies were considered. The performance analysis of the other five property types including Flats, Mid terraced, End terraced, Semi-detached, and Detached with age period of (1900-1949, 1950-1975, 1976-1990, and 1991-2007 onwards) were investigated.

6.2 Methodology

6.2.1 1900s Mid terraced test house

First, the 1900s mid terrace typical 3-bedroom test house in Belfast, Northern Ireland, with floor area of 105 m^2 have been considered for the retrofit assessment [10, 21]. This has been followed by other property types, age periods. The combined heat load demand calculated experimentally includes both domestic hot water (DHW) & space heating (SH) [10] could be seen in Figure 6.1. The HP in two control modes, i.e., VSM & FSM at three heat supply temperature level of 35 °C, 45 °C, and 55 °C, were evaluated for the performance improvement, energy consumption, cost and carbon emission savings potential once retrofitted instead of other heating technologies.

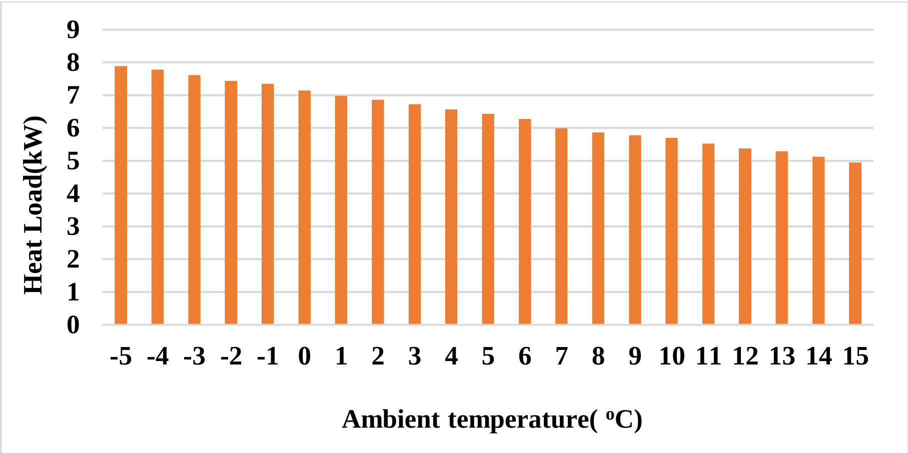


Figure 6.1 1900s Mid terraced test house heat load demand including both SH and DHW

6.2.2 HP performance for the test house

The HP performance and the COP value in each bin for the 1900s mid terraced building type in Northern Ireland (Belfast) for the two control approaches (VSM & FSM) and three case studies have been shown in the following subsections. The COP values varies according to the three case studies (C1, C2, C3), heat demand and ambient temperature conditions in each bin. The HP performance was strongly dependent on the part load factor, ambient temeprature and heat supply temperature. Defrost strategy was found complicated and different approaches have been used in the literature including as a COP reduction parameter [49]. In this study defrost was considered as a COP reduction parameter using real experimental tests. The HP operates according to load demand, with continous full capacity operation(in case of higher load demand than heat production with back-up electric heater requiremtns), and at part load in case when heating capcity is higher than the required load demand. The capacity is controlled at lower heat load demand using VSM & FSM of control. The HP COP values in each bin for with/without considering the transient cycling losses in two control modes are presented in the following;

a) Variable speed mode (VSM)

In VSM of control, the HP operates continuously at partial loads to follow the building load demand. The COP values in each bin according to the ambient temperature considering the defrost and without defrost value have been shown in Figure 6.2.

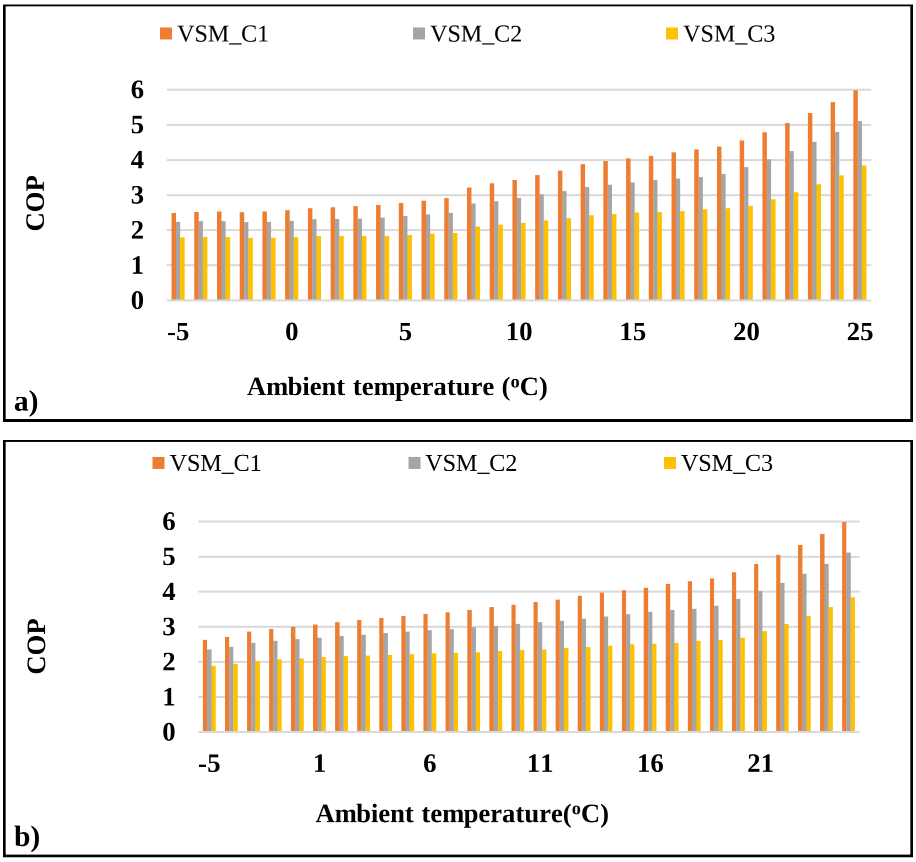


Figure 6.2 HP COP values in each bin in VSM, a) defrost considered, b) No defrost considered

The COP reduction due to defrost at range of ambient temperature conditions was established experimentally and the extrapolation and interpolation technique were used to extend the degradation caused at all ambient temperature conditions range. Beyond the certain point (12 ^{o}C) the defrost value become negligible.

b) Fixed speed mode(FSM)

The HP operation in fixed speed mode (FSM) with the highest COP at nominal frequency with 50% heating capacity have been shown in Figure 6.3 a. The steady state testing results were based on the testing results. However, after considering the cycling losses and correction factor results in COP reductions as could be seen in Figure. 6.3 b. The COP correction factor according to the building thermal load is shown in Figure 6.4. The part load fact (PLF) was calculated with the approach mentioned in the test standard BSEN14511 for the variable speed heat pump systems and explained in the methodology section. The ideal would be to have the real experimental results for these losses calculation and field trials required, but in the absence of the real data the standard recommendations have been followed [35]. The losses increase with the decrease in house load demand and causes the HP operation at very lower COP values when the compressor operation is at very low speed.

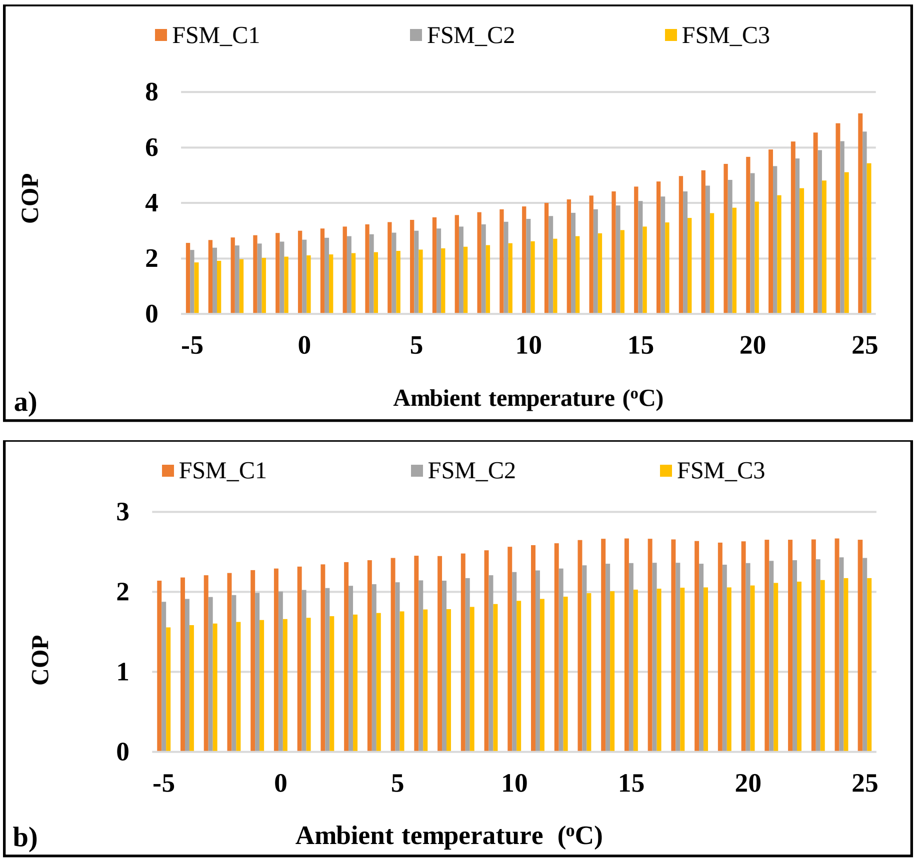


Figure 6.3 HP COP values in FSM a) steady state, b) after cycling losses

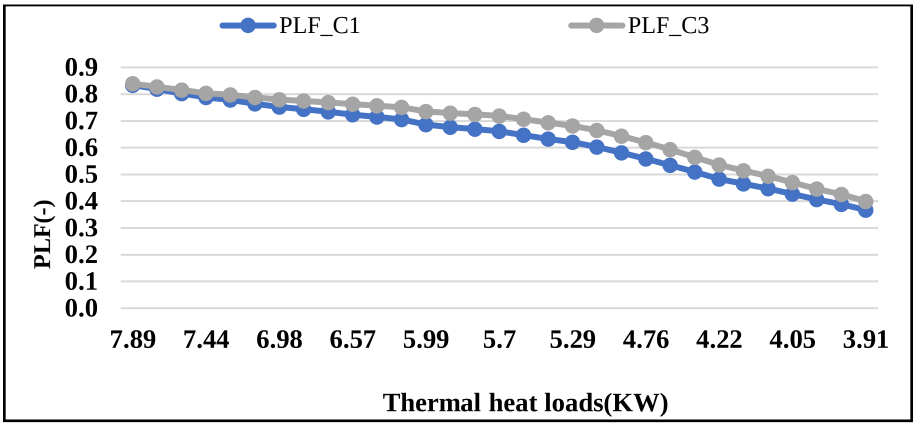


Figure 6.4 Part load factor (PLF) for the 1900s Mid terraced test house load demand

The HP (VSM) could vary compressor speed to meet the maximum heat demands at maximum percentage of the experienced ambient temperature conditions. In contrast operating the system in FSM produces heating capacity, either less or more than the house heat demand. In case of lower heat production, the back-up heater is required, while over production results in continuous ON/OFF cycles. The power consumption & heat production differences using FSM compared to matched load in VSM was experimentally measured and presented in Figure 6.5 & Figure 6.6 respectively. The negative values show the point where HP could not meet the house heat demand, and positive values indicate extra heat output during FSM.

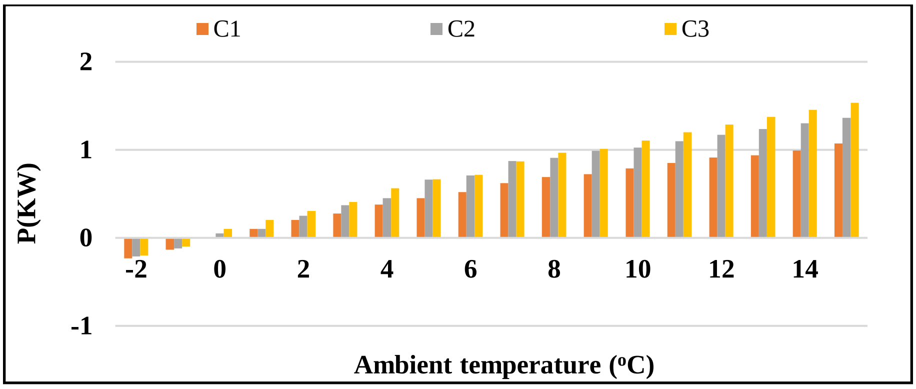


Figure 6.5 Power consumption difference in operating control (FSM vs. VSM) differences

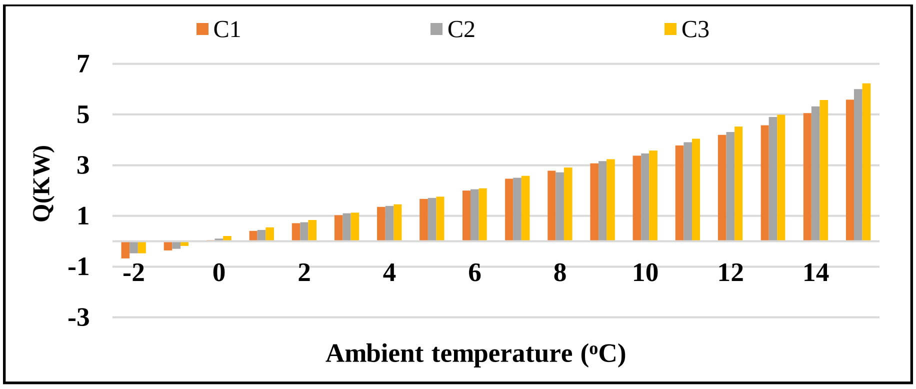


Figure 6.6 Heat production difference in operating control (FSM vs. VSM)

The test house heat load demand for Belfast climatic conditions in each bin with total annual heat demand of 23429KWh is depicted in Figure 6.7. The 7 °C bin records the highest heat demand due to higher number of hours annually occurring.

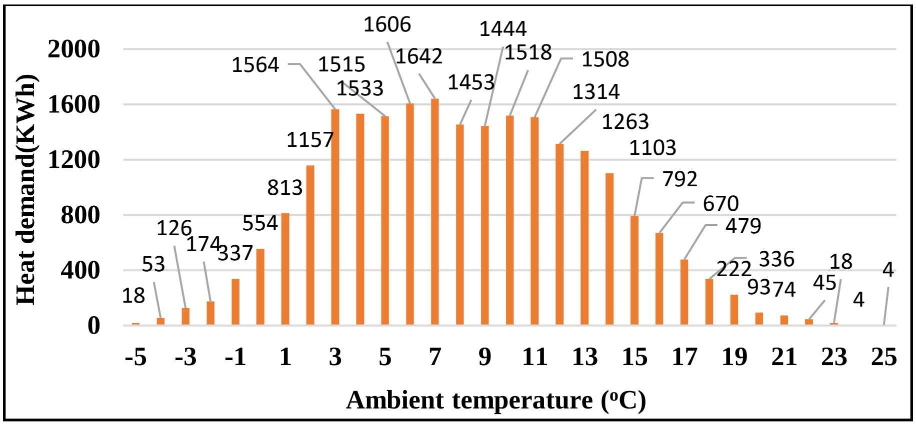


Figure 6.7 Heat energy demand (KWh) in each bin for Belfast climatic conditions

The total annual electrical energy consumption for test house strongly depends on the frost effect at all heat supply temperature. The total electrical energy demand for three cases (C1, C2, C3), considered is shown in Figure 6.8 (a, b).

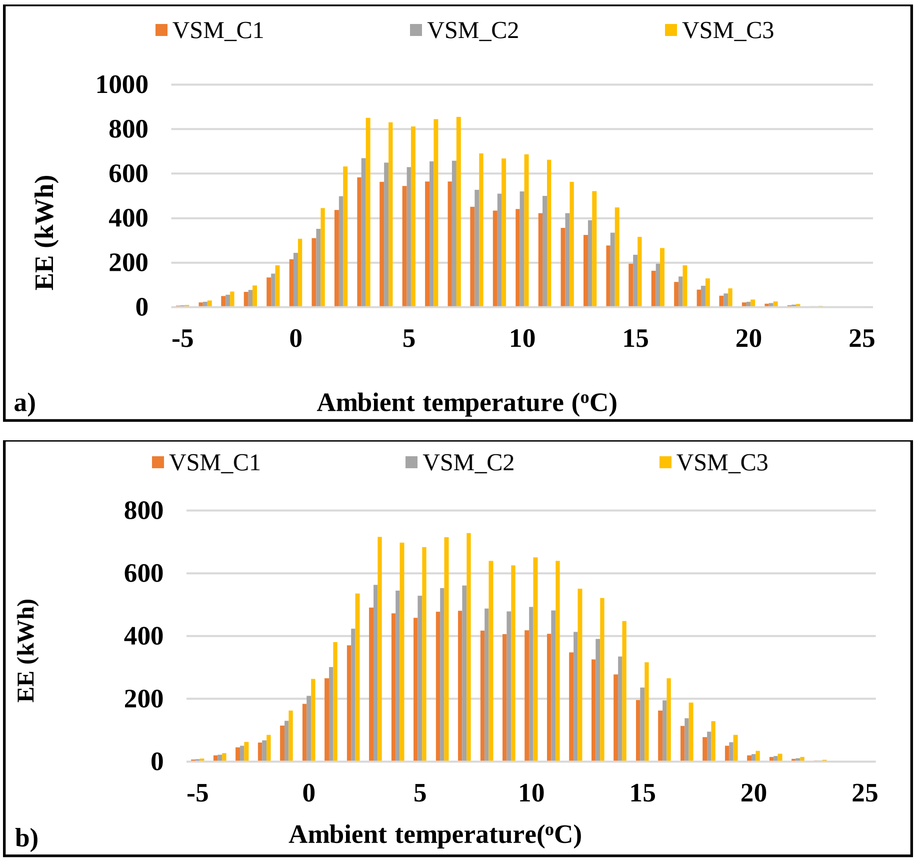


Figure 6.8 Electrical energy (KWh) consumptions in VSM a) with b) without defrost

The respective values for C1, C2, C3, without defrost considerations were 6697KWh, 7828KWh, and 10210 KWh, which increases subsequently to 7418KWh, 8661KWh, and 11280KWh when defrost is considered. This causes the annual COP reduction by approximately by 10.5 %. In the literature the approximate reduction for seasonal performance evaluation of 13% [36] was reported, and the difference was being attributed with climatic conditions and even higher for the specific colder conditions. During ON/OFF control the cycling losses is dependent on percentage capacity of the unit operation [109]. The COP correction factor at different heating capacity was shown in Figure 6.4. The results for electrical energy consumption using FSM operation causes significant variations with/without cycling losses can be noticed from Figure 6.9 (a, b).

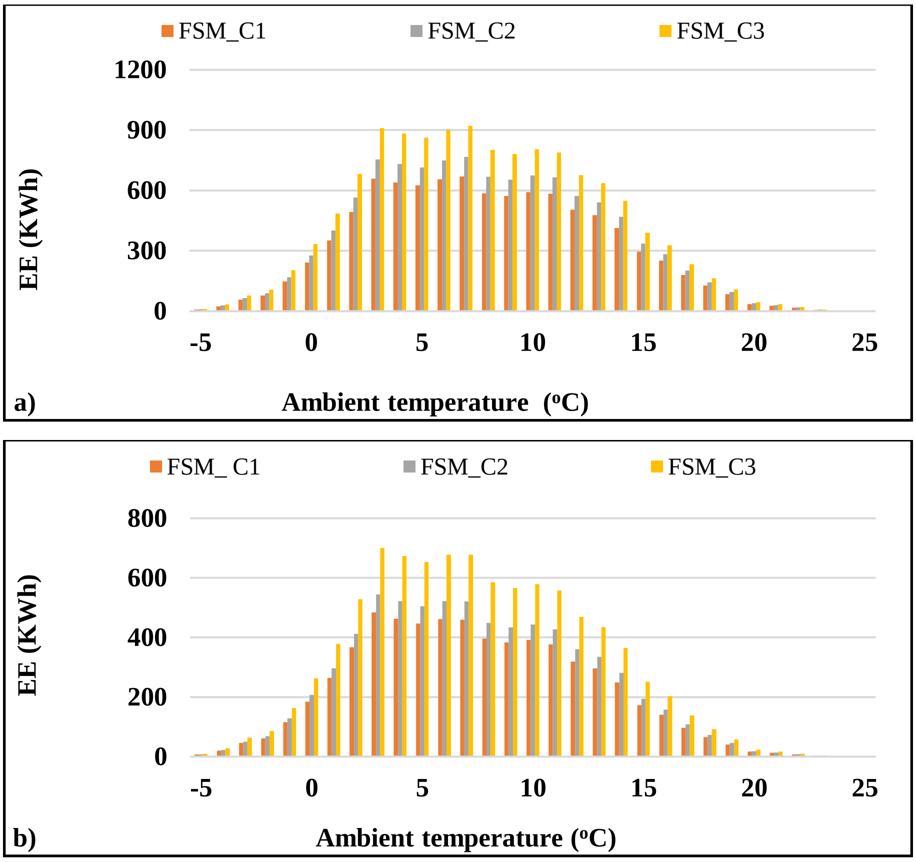


Figure 6.9 Electrical energy consumption in FSM a) with, b) without cycling losses

6.2.3 Other heating technologies

The developed HP technology was compared as a retrofit option instead of other heating technologies including oil boilers (OB), gas boilers (GB), electric heating (EH), and air-sourced very high temperature heat pump (VHTHP) at three (3) different percentage (%) efficiency as described in Table 6. 1.The OB/ GB have been compared at percentage efficiency of 90%, 80%, and 70%, representing range of newly advanced condensing boilers to earlier heavy weight less efficient boilers to investigate different age period installations and corresponding energy savings and carbon emissions reduction. The very high temperature heat pump (VHTHP) system was based on actual installed experimental results for the test house with COP of 2.15 at constant heat supply temperature of 75 ^{o}C and become 2.32, with applying of weather compensation control strategy for the cascaded HP units [42]. The third COP value of 2.12 utilized were from the earlier experimental development at lab scale units were considered, [54]. The cost and carbon emission factor with comparison of differenet fuels types consumption have been performed using the corresponding fuel price shown in **Table 7**. The electricity prices was 0.175 (£/KWh), oil price of 0.068(£/KWh), and gas price of 0.047(£/KWh) respectively [110] were utilized. The greenhouse gas emission factor were used from GHGs emission report [12], with oil & gas having values of 0.243 & 0.203 respectivley.

Table 6.1: Other investigated heating technologies with percentage efficiencies/COP, cost, and carbon emission factor

Heating Technology types	**Efficiency/COP (-)**			**Fuel Price (£/KWh)**	**Carbon emission factor (-)**
	1	**2**	**3**		
OB	90%	80%	70%	0.068	0.243
GB	90%	80%	70%	0.047	0.203
EH	100%	95%	90%	0.175	0.29
VHTHP	2.32	2.15	2.12	0.175	0.29

6.3 Results analysis

6.3.1 HP Annual, seasonal, and monthly performance

The annual, seasonal (S1, S2, S3, S4) house heat demand, electrical energy consumptions, COP values are summarized in Table 6.2 for three considered cases, during the two mode of control. The COP values depends on the heat supply temperature, mean hourly ambient temperature of the climatic conditions, and control mode of operation. The HP annual useful heat output was found to be 23429 KWh with electric power consumption variation according to case considered, and control modes. The COP value trends shows higher values in VSM in comparison to FSM in all cases, with C1 shows superior performance over C2& C3 for the same annual heat load demand. The seasonal COP values ranges between 3.85 to 2.77 in VSM control, while the range becomes 2.62-2.34 during FSM of control for the C1. Other case studies (C2, &C3) have also shown the trend with the highest COP value in case of S3 and lowest values in case of S1.

Table 6.2: HP performance for the 1900s Mid-terraced test house results for Belfast

	Type of control (-)	FSM					VSM				
	Seasons/Annual	S1	S2	S3	S4	Annual	S1	S2	S3	S4	Annual
No.	Mean ambient temperature (ºC)	4.05	7.68	14.08	9.34	8.79	4.05	7.68	14.08	9.34	8.79
	Annual useful heat output(kWh)	7478	6120	4121	5765	23429	7478	6120	4121	5765	23429
1	Input power(kWh)	3195	2487	1572	2292	9466	2699	2019	1070	1772	7419
	Annual COPs	2.34	2.46	2.62	2.52	2.48	2.77	3.03	3.85	3.25	3.16
2	Input power(kWh)	3595	2834	1783	2609	10737	3102	2349	1279	2078	8661
	Annual COPs	2.08	2.16	2.31	2.21	2.18	2.41	2.61	3.22	2.77	2.71
3	Input power(kWh)	4347	3397	2081	3111	12813	3956	3045	1717	2719	11280
	Annual COPs	1.72	1.80	1.98	1.85	1.83	1.89	2.01	2.40	2.12	2.08

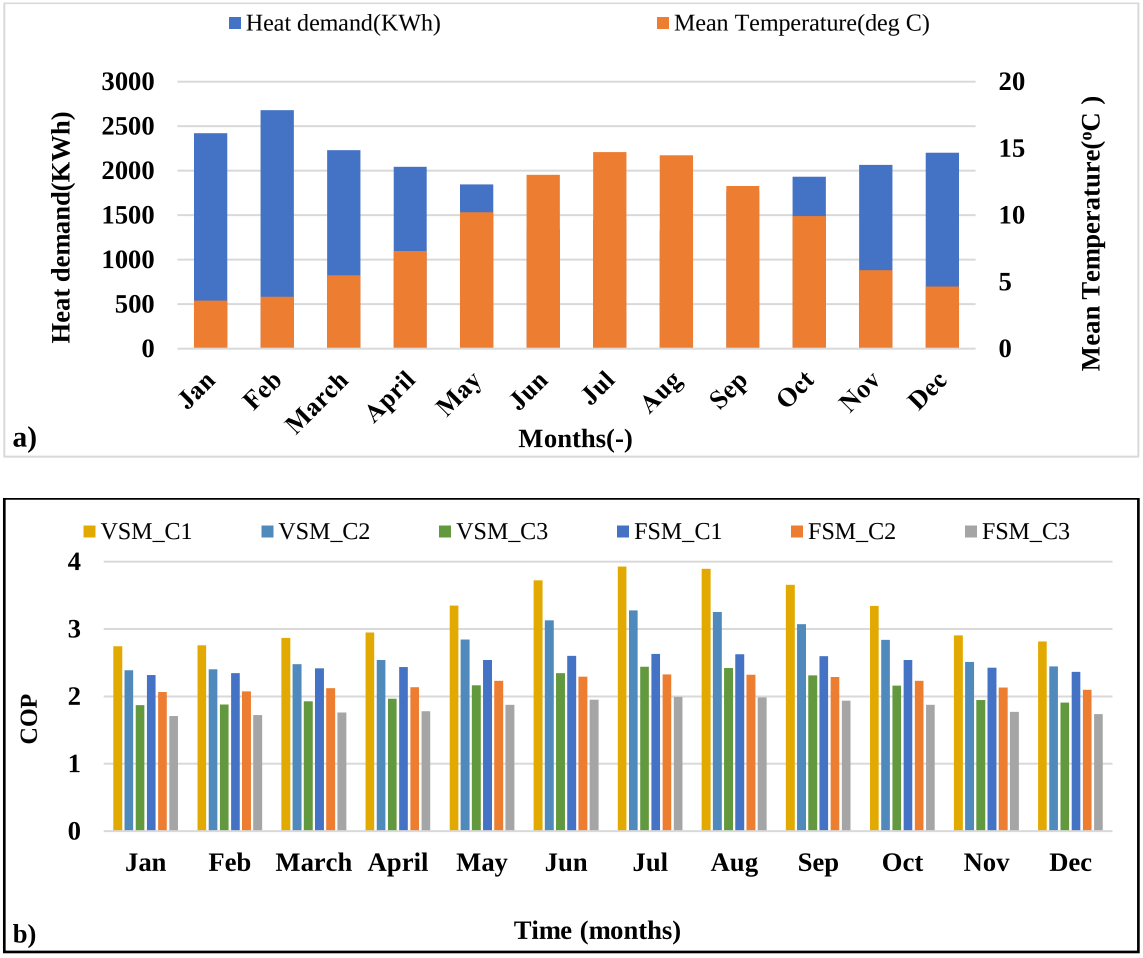
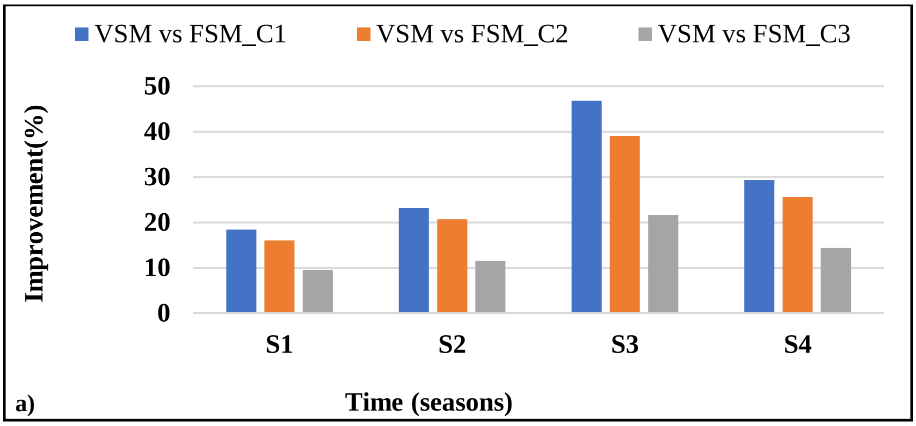


Figure 6.10 Monthly a) load demand (KWh), b) HP COP in Belfast climatic conditions

Monthly COP values showing higher COP (Figure 6.10) at higher mean ambient temperature month in comparison to the months of lower mean hourly ambient temperature in both control mode and for all cases. The monthly values range from 3.93 to 2.74 with the highest value in July and smallest value in month of January in VSM(C1). The trend is maintained with other cases as well and for FSM control but with different percentages variations.



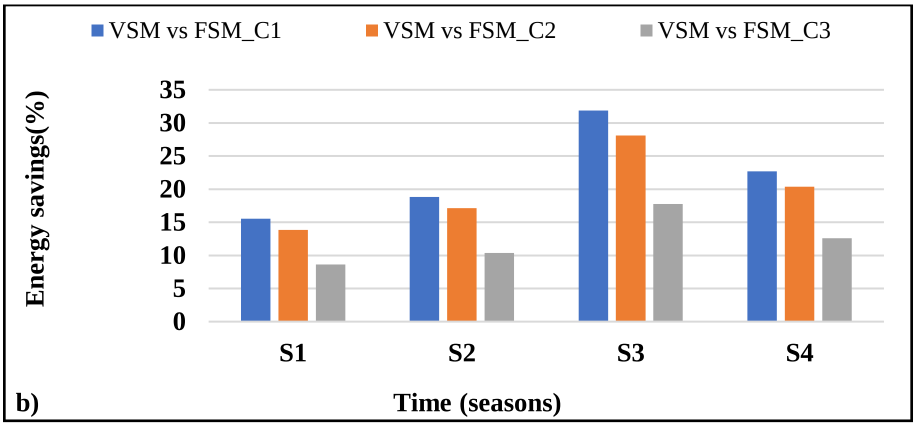


Figure 6.11 Control mode (VSM vs. FSM) improvement (%) in a) COP, b) Energy savings

The seasonal percentage (%) improvement in COP, energy consumption due to chang in control mode (VSM vs. FSM) with C1,C2,and C3 is illustrated in Figure 6.11 (a,b). The potential improvement depends on the climatic conditions during the period with highest during summer season S3 and lowest value during the winter season S1. The highest COP improvent during summer season with VSM compared to FSM was because of the additional cycling losses during FSM and the idle operaion of the system for most of the time. The highest energy savings with a value of 31.89% occurs for S3 with the lowest improvement in S1 with the value of 15.54 % for C1 with the same trend for other C2 & C3.

6.3.2 HP annual COPs with other climatic conditions

The annual performance depends on the climatic conditions with the highest COP value in milder climatic conditions of Valentia for lower supply temperature(C1) with a value of 2.55 during FSM and increases to 3.40 during VSM. The reason for this was higher mean weighted average hourly ambient temperature of 10.65 ^{o}C. The COP value in other climatic conditions was lower than Valentia because of higher HDDs, and poor performance at lower mean weighted average hourly ambient temperature. Among the considered climatic conditions the lowest annual COP occurs in Aviemore, with value of 2.38 & 2.98 during respective FSM & VSM (C1) of control. The COP values in C1 in both control modes are either higher or equal to 2.5, hence could be considered as eligible for renewable heat incentives scheme proposed in Northern Ireland (NI) [10]. The only exception was Aviemore during FSM in case of C1. The annual COPs changes significantly with the change in control mode of operation plus the heat supply temperature, and climatic conditions. The annual performance results were summarized in Table 6.3.

Table 6.3: HP annual performance results in four (4) different climatic conditions

Control mode (-)		**FSM**				**VSM**			
Case	Location	Valentia	Dublin	Belfast	Aviemore	Valentia	Dublin	Belfast	Aviemore
No. (-)	T_{wma}	10.65	9.48	8.82	6.79	10.65	9.48	8.82	6.79
	Heat output(kWh)	22335	23025	23429	24636	22335	23025	23429	24636
1	Input power(kWh)	8769	9209	9466	10338	6576	7136	7419	8263
	Mean Annual COPs	2.55	2.50	2.48	2.38	3.40	3.23	3.16	2.98
2	Input power(kWh)	9980	10464	10737	11530	7750	8355	8661	9558
	Mean Annual COPs	2.24	2.20	2.18	2.14	2.88	2.76	2.71	2.58
3	Input power(kWh)	11848	12466	12813	13886	10205	10919	11280	12332

The COP improvement due to change in control mode (VSM vs. FSM) & heat supply temperature (C1 vs. C2 & C1 vs. C3) in four climatic conditions have been shown in Figure 6.12 (a, b). The highest COP improvement (%) due to change in control mode occurs in case of C1 in milder climatic conditions of Valentia with 33.4% increase and become 25% in Aviemore. It could be concluded that the benefit associated with VSM against FSM is more in milder climatic conditions at low supply temperature(C1) with similar results by other researchers [28,43]. The highest COP improvement during VSM for (C1 vs. C3) was found 55% in Valentia.

6.3.3 HP energy consumptions in other climatic conditions

The energy savings in percentage (%) due to control mode and heat supply temperature in four locations with different climatic conditions have been shown in Figure 6.13 (a, b). The energy savings get reduced for the colder climatic condition of Aviemore due to change in control mode and heat supply temperature also highlighted by other researchers [23,68, 69]. In case of FSM energy savings percentage change due to varying climatic conditions is very minute and with smaller range of around 35% to 26% respectively (C1 vs. C3).

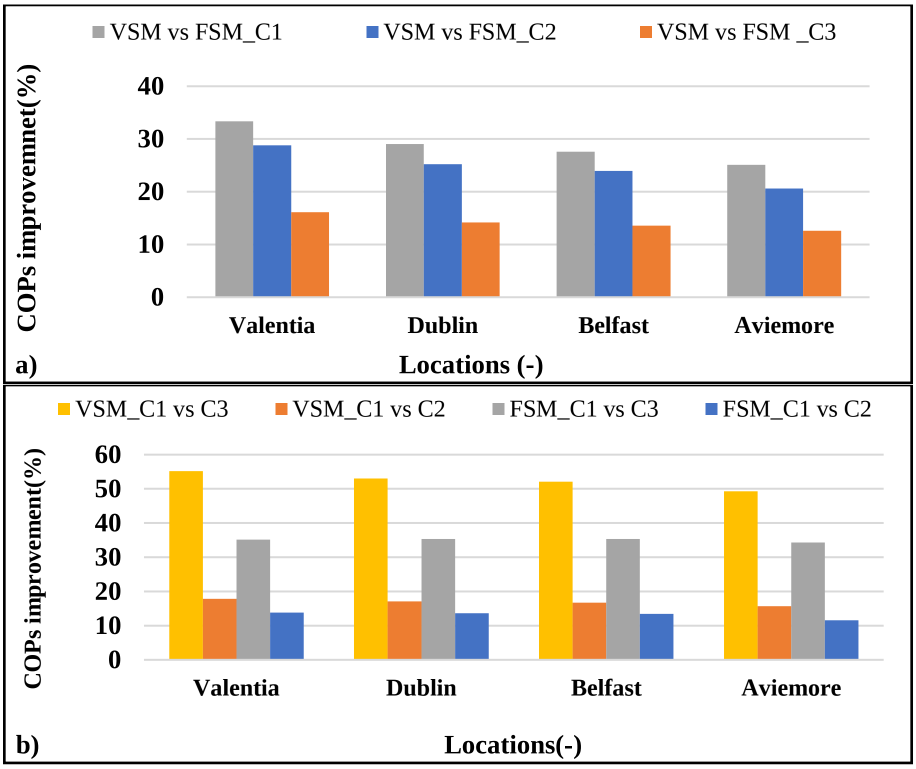


Figure 6.12 Annual COP improvement (%) due to a) control mode (VSM vs. FSM), b) heat supply temperature (C1 vs. C3)

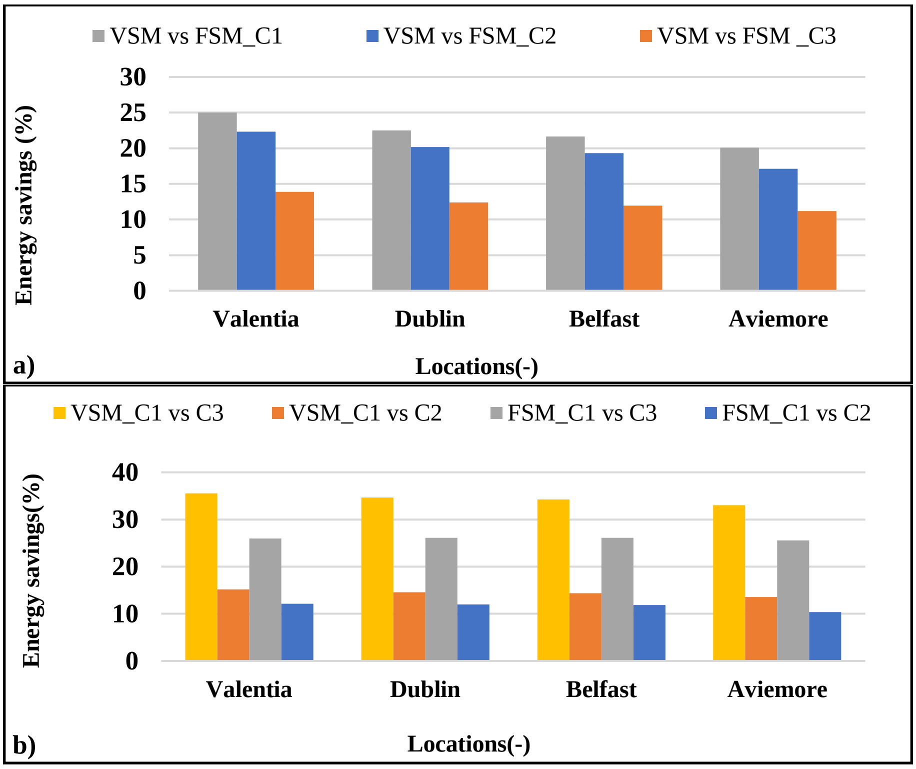


Figure 6.13 . Annual energy savings (%) due to a) control mode, b) heat supply temperature

6.4 HP retrofit assessment

The HP retrofit in two modes for capacity control at three supplied water temperature levels (C1, C2,C3) with four climatic conditions have been assesed instead of OB/GB, EH and with the existing installed air sourced VHTHP inside test house [10]. The energy price for different fuel type, and carbon emission factor were utilized mentioned in Table 6.1. The results for annual running cost, carbon emissions with considered heating technologies and percentage (%) efficiencies is presented in Table 6.4. The VHTHP carbon emissions and electric energy consumptions in comparison to the developed system depends on the supply temperature. The climatic conditions influence the HP performance as a retrofit option. The HP annual running cost, and carbon emissions changes according to climatic conditions mainly because of two reasons, i.e., changing heat load demand, and HP performance. In case of boilers (oil/gas), and electric heating option the annual running cost varies only because of changing house heat load demand. The air sourced- VHTHP performance real experimental data was only available for the Belfast climatic conditions. Therefore, while comparing cost and carbon savings with the other heating technologies in the following subsections, VHTHP was presented only for Belfast climatic conditions. The control mode of operation, and heat supply temperature (C1, C2, C3) significantly affect the annual running cost and carbon emissions. In the following two subsections the HP annual running cost & carbon emissions savings in percentage have been compared to other heating technologies, when the HP retrofitted into the reference building. The developed HP in both control mode performance in terms of carbon emissions and primary energy consumptions is higher than the fossil fuel-based oil/gas boilers and electric heating option. However, the developed HP in terms of carbon emissions was found only more valuable at lower heat supply temperature in contrast to VHTHP.

6.4.1 Annual running cost savings

As a rule of thumb, HP to be competitive with the gas/oil boilers in relation to the annual running cost the criteria mentioned in Eq. (6.1), needs to be fulfilled.

$$Annual\ COP_{hp} \geq \frac{electricity\ price\ per\ KWh}{gas/oil\ boiler\ price\ per\ KWh} \qquad (6.1)$$

The percentage (%) cost savings when the OB, GB, very high heat supply temperature ASHP , and EH is retrofitted in the reference test house with the developed HP in the four considered climatic conditions were presented in Figure 6.14-6.17(a, b).

Table 6.4: HP retrofit assessment for the 1900s Mid-terraced test house instead of other investigated heating technologies

Developed HP		**Belfast**			**Valentia**			**Dublin**			**Aviemore**	
Case/efficiency No: (-)	**1**	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**
						1) Variable speed mode (VSM)						
Annual heat output (KWh)	23429	23429	23429	22335	22335	22335	23025	23025	23025	24636	24636	24636
Annual Input power (KWh)	7419	8661	11280	6576	7750	10205	7136	8355	10919	8263	9558	12332
Annual COP	3.16	2.71	2.08	3.40	2.88	2.19	3.23	2.76	2.11	2.98	2.58	2.00
Annual running cost (£)	1298	1516	1974	1151	1356	1786	1249	1462	1911	1446	1673	2158
Annual CO_2 emissions(kg)	2151	2512	3271	1907	2248	2959	2069	2423	3167	2396	2772	3576
						2) Fixed speed mode (FSM)						
Annual Input power (KWh)	9466	10737	12813	8769	9980	11848	9209	10464	12466	10338	11530	13886
Annual COP	2.48	2.18	1.83	2.55	2.24	1.89	2.50	2.20	1.85	2.38	2.14	1.77
Annual running cost (£)	1657	1879	2242	1535	1747	2073	1612	1831	2182	1809	2018	2430
Annual CO_2 emissions(kg)	2745	3114	3716	2543	2894	3436	2671	3035	3615	2998	3344	4027
						3) Oil boilers						
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	26033	29287	33470	24816	27918	31907	25583	28781	32893	27373	30795	35194
Annual running cost (£)	1770	1991	2276	1688	1898	2170	1740	1957	2237	1861	2094	2393
Annual CO_2 emissions(kg)	6326	7117	8133	6030	6784	7753	6217	6994	7993	6652	7483	8552
						4) Gas boilers						
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	26033	29287	33470	24816	27918	31907	25583	28781	32893	27373	30795	35194
Annual running cost (£)	1224	1376	1573	1166	1312	1500	1202	1353	1546	1287	1447	1654
CO_2 emission(kg)	5285	5945	6794	5038	5667	6477	5193	5843	6677	5557	6251	7144
						5) Electric heater						
Efficiency (%)	100%	95%	90%	100%	95%	90%	100%	95%	90%	100%	95%	90%
Annual Input power (KWh)	23429	24662	26033	22335	23510	24816	23025	24237	25583	24636	25932	27373
Annual running cost (£)	4100	4316	4556	3909	4114	4343	4029	4241	4477	4311	4538	4790
Annual CO_2 emissions(kg)	6794	7152	7549	6477	6818	7197	6677	7029	7419	7144	7520	7938
						6) Very high temperature HP						
Supply Temperature(°C)	55-75	60	75	65-75	60	75	65-75	60	75	65-75	60	75
Annual Input power (KWh)	10099	10897	11052	9627	10388	10535	9925	10709	10861	10619	11459	11621
Annual COP	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12
Annual running cost (£)	1767	1907	1934	1685	1818	1844	1737	1874	1901	1858	2005	2034
Annual CO_2 emissions (kg)	2929	3160	3205	2792	3013	3055	2878	3106	3150	3079	3323	3370

The positive values indicate the running cost for the developed HP was smaller than other heating technologies, while negative values means that the developed HP is not cost effective. In Valentia, during VSM (C1), could save cost in comparison to OB/GB, EH option at all percentage efficiencies. Similar trends of cost savings have been shown for other climatic conditions with the only exception of GB at 90% efficiencies. In Dublin, Belfast, and Aviemore, VSM(C1) instead of GB at 90% shows negative respective values of -4%, -6%, and -12%, means no money savings. It could be anticipated that the cost savings for the HP reduces with the colder climatic conditions. The other two cases considered (C2, C3) could also shows the same trends. The VHTHP compared for cost savings is only shown for Belfast (Figure 6.15). The HP in comparison to VHTHP have shown savings (C1 &C2) for both control mode (VSM& FSM) with negative values when it comes to C3, and the reason is superior performance of VHTHP at higher supply temperature. During FSM money was saved for C1, when compared to OB at all efficiency, with some negative values at 90% & 80% efficiency GB and increasing trend in negative cost savings as we move from milder to severe climate conditions. The C3 have shown negative cost savings also for OB at 90%, 80% in addition to GB (90%, 80%,70%) for FSM. To summarize discussions, it could be stated that HP have more saving potential for running cost at C1 during VSM.

6.4.2 Carbon emission analysis

The carbon emission savings at C1, C2, C3 in two control modes (VSM & FSM) compared to other heating options is shown in Figure 6.18-6.21**.** The carbon emission savings depends on control method, climatic conditions, case considered (C1, C2, C3) and percentage efficiency for other heating technologies. In case of FSM the carbon emission savings is lower in comparison to VSM for all cases considered. The HP have shown positive values for carbon emissions savings compared to all other heating technologies even at highest efficiency with the only exception of VHTHP. The developed HP when retrofitted with GB at the highest percentage efficiency of 90% could cut the carbon emission in VSM(C1) ranges from 62% to 57% according to the climatic conditions from milder to severe. The respective values ranges become 71% to 66% when the GB is operated at 70% efficiency. The HP in VSM (C1) when compared with the most efficient (90%) modern oil boilers could cut the carbon emissions in the range of 68 -64% with the increase in savings between 75% - 72% for old heavy weight OB (70%) efficiency.

6.5 Payback period analysis

The payback period analysis for VSM, and low temperature heating distribution installations is summarized in Table 6.5. The methodology was explained in more details in the chapter 2 for the economic analysis. The annual running cost savings with the HP operation depends on the control mode of operations, and installed heating distributions systems. The cost of control devices is reducing with advancement in technology [28]. The HP at low heat supply temperature of 35 ^{o}C needs additional installation cost of £6000 [75] and the payback period was found as 10.2 years in FSM, and reduces to 9 years in case of VSM, shown in the table.

The additional cost associated with VSM in contrast to FSM is calculated as £1000 [28] with payback period depending on the case considered. The cost analysis in the present calculation using the simple payback period approach by Eq. (6.2).

$$Pay\ back\ period = \frac{Additional\ cost\ due\ to\ control\ devices/heating\ distribution\ system(£)}{Annual\ cost\ savings(£)} \quad (6.2)$$

Table 6.5: Payback period due to additional control devices (VSM vs. FSM) & heating distribution cost (C1 vs.C3)

Control mode (VSM vs. FSM)/case considered (C1 vs. C3)	**Annual cost savings(£)**	**Payback period (-)**
VSM vs. FSM at 35 °C heat supply temperature (C1)	358	2.8
VSM vs. FSM at 55 °C heat supply temperature (C3)	268	3.7
Heating distribution installation cost (C1 vs. C3) in VSM	675	8.9
Heating distribution installation cost (C1 vs. C3) in FSM	585	10.2

(Installation cost for underfloor heating option = 6000 pounds), (additional control devices capital cost is 1000 pounds for VSM in comparison to FSM)

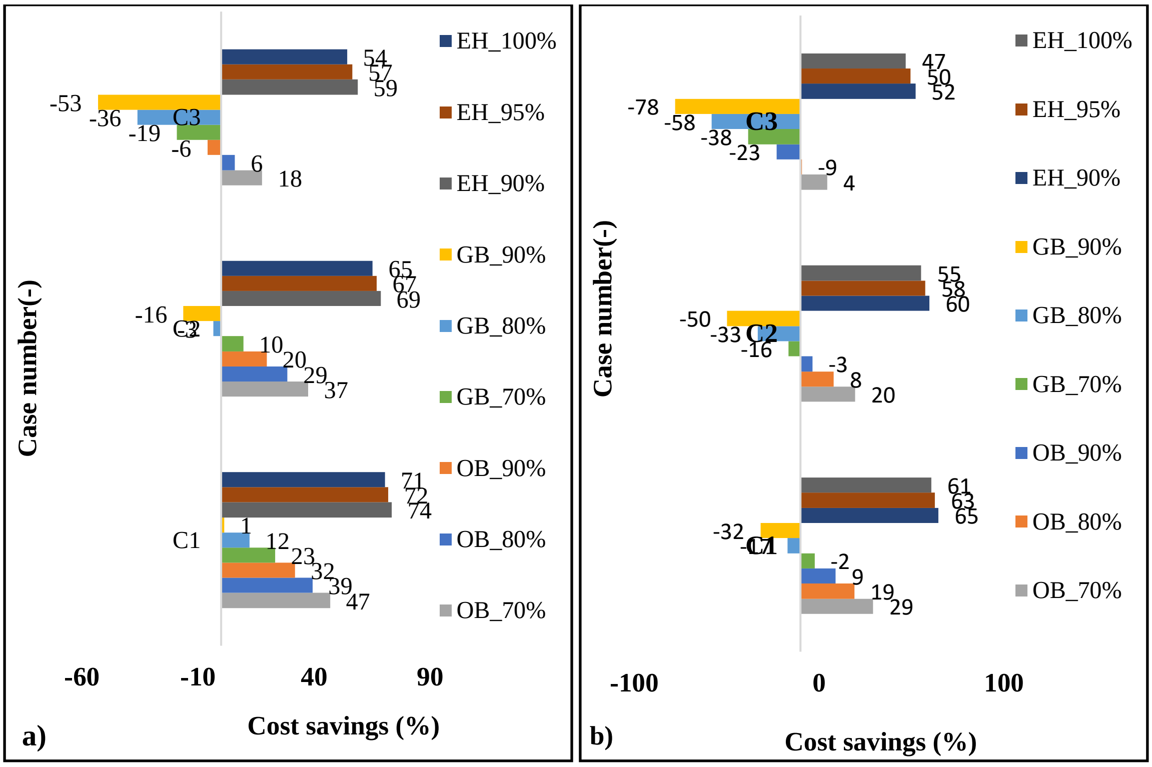


Figure 6.14 Valentia climatic conditions, a) VSM, b) FSM

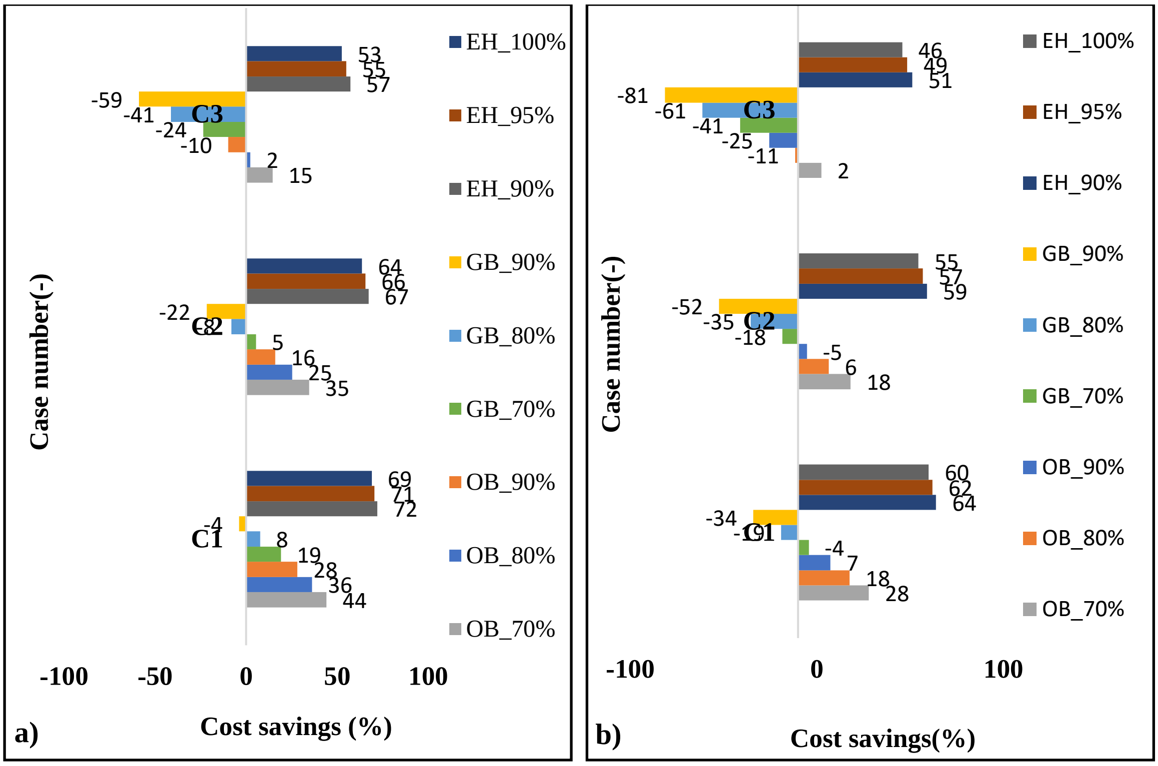


Figure 6.15 Dublin climatic conditions, a) VSM, b) FSM

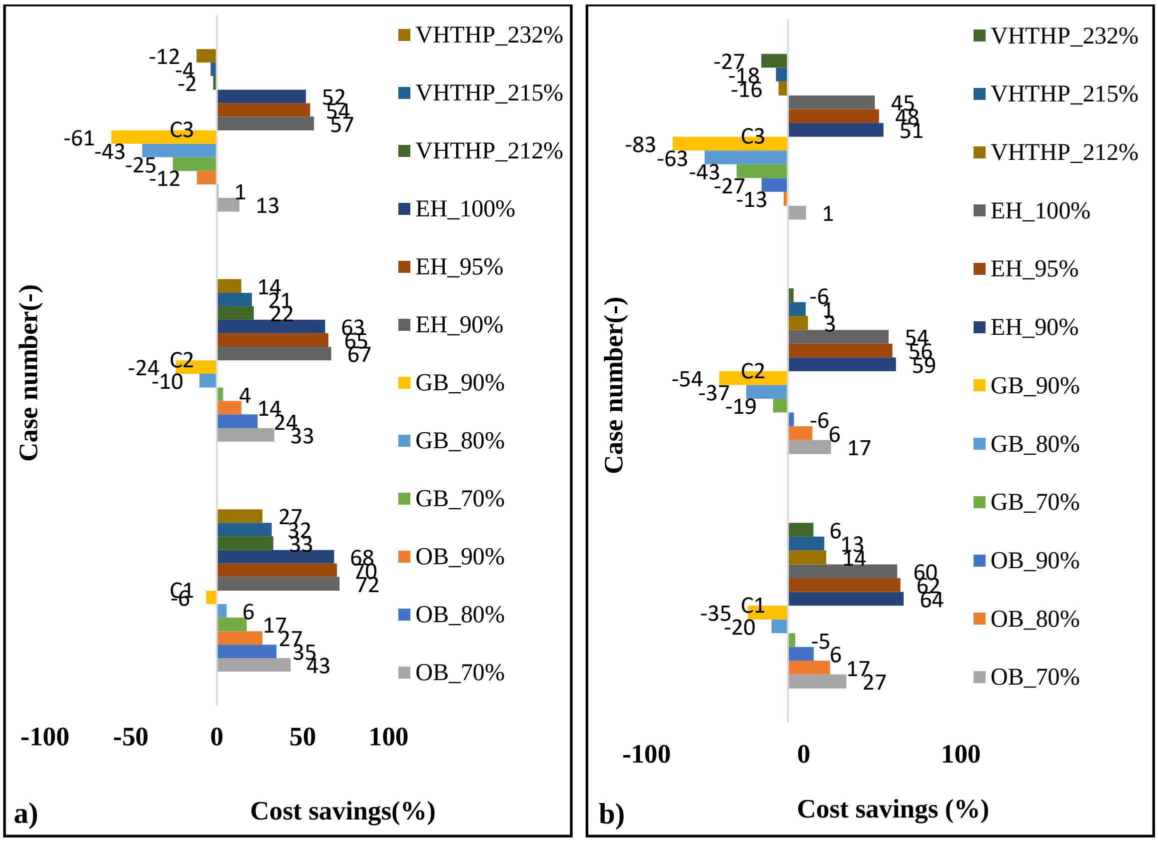


Figure 6.16 Belfast climatic conditions, a) VSM, b) FSM

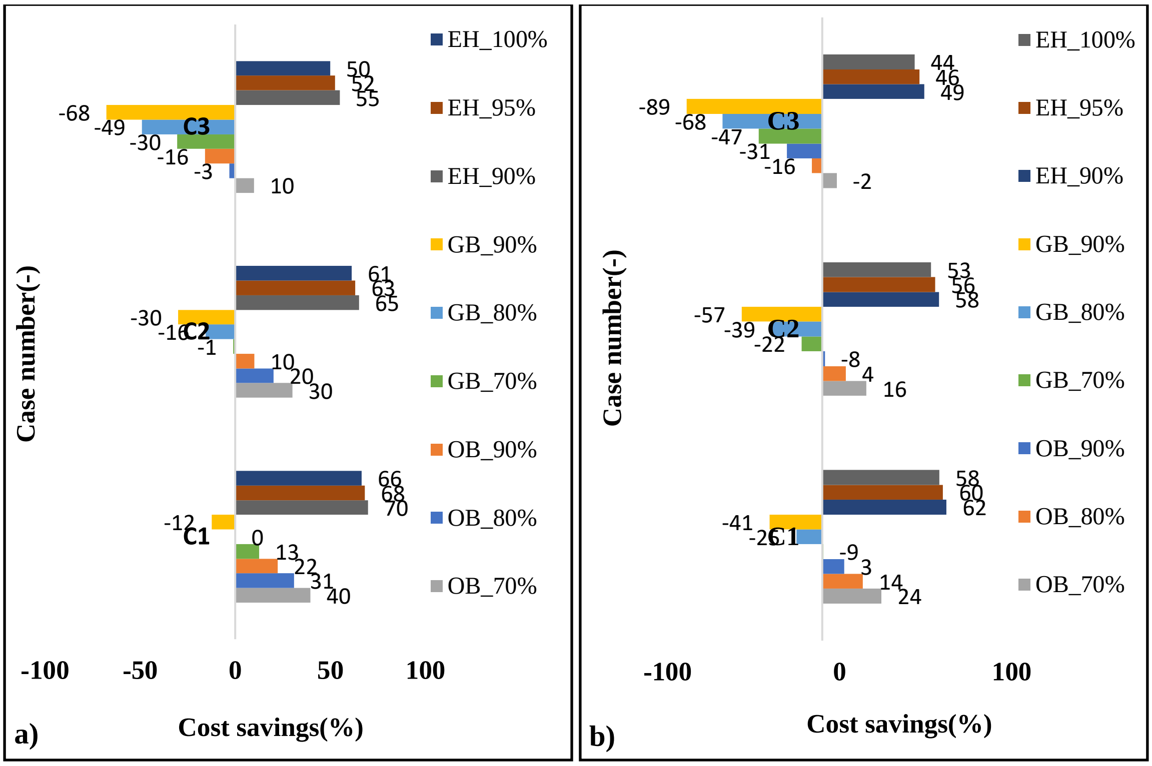


Figure 6.17 Aviemore climatic conditions, a) VSM, b) FSM

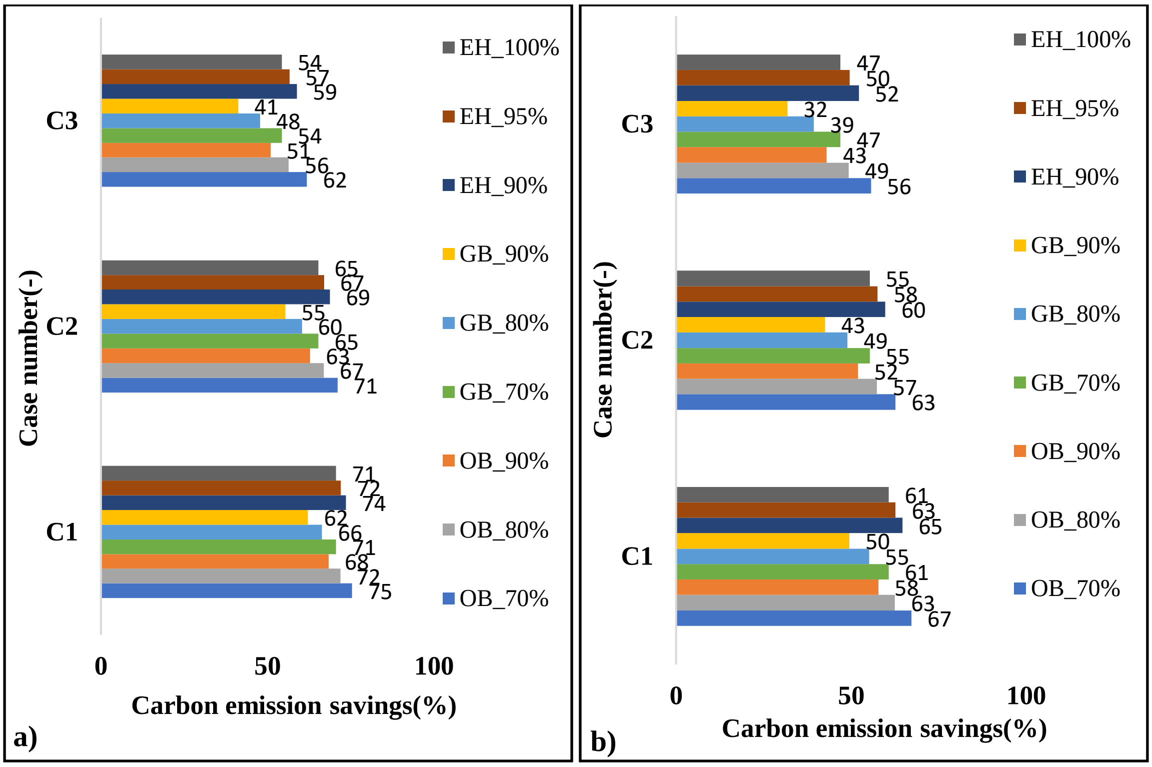


Figure 6.18 Valentia climatic conditions, a) VSM, b) FSM

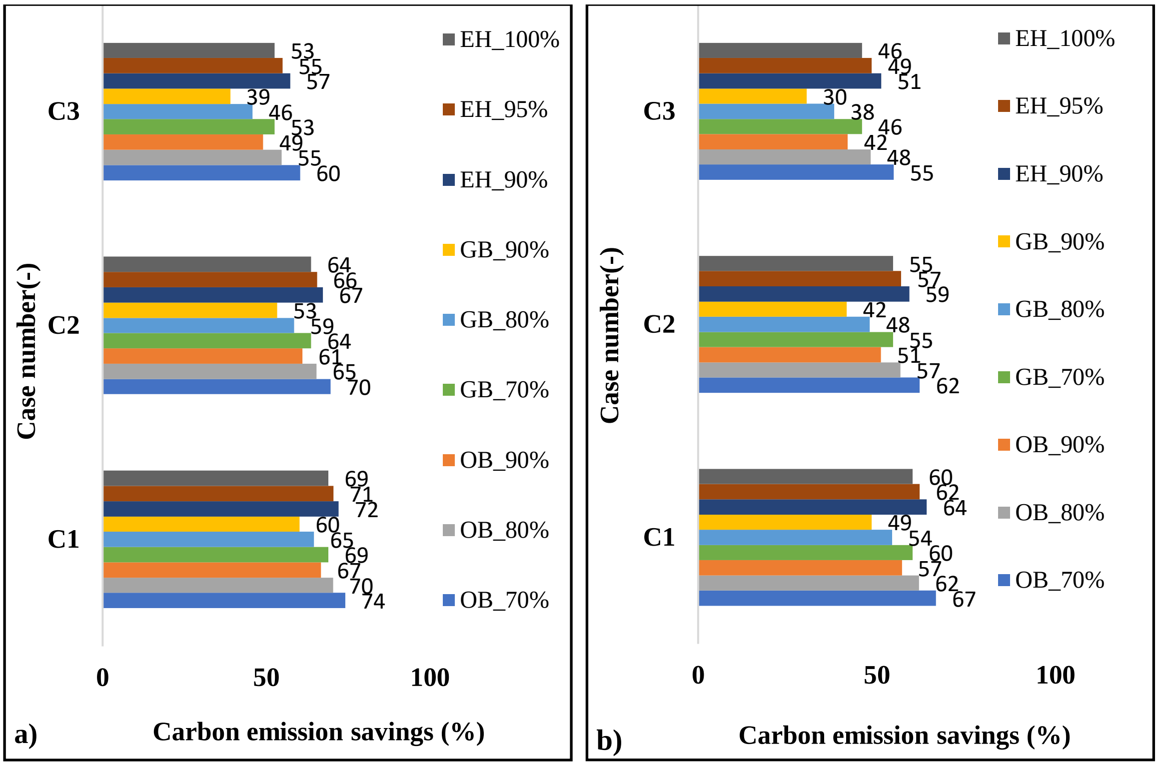


Figure 6.19 Dublin climatic conditions, a) VSM, b) FSM

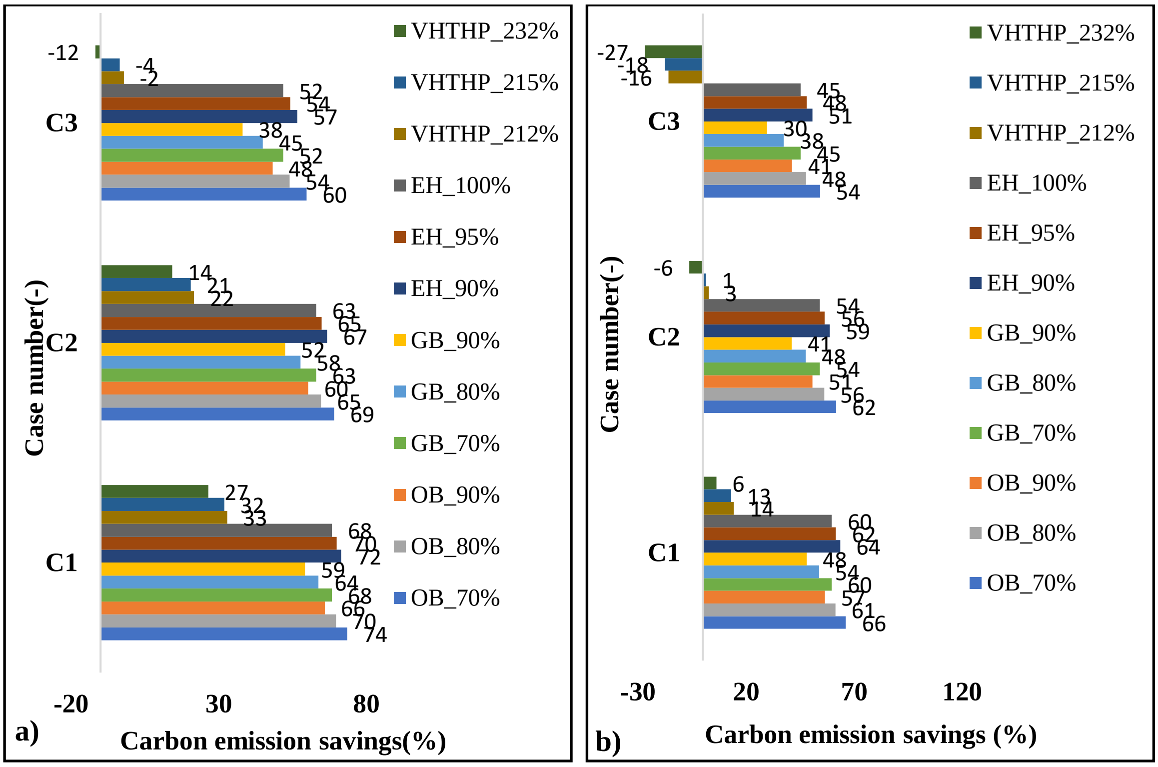


Figure 6.20 Belfast climatic conditions, a) VSM, b) FSM

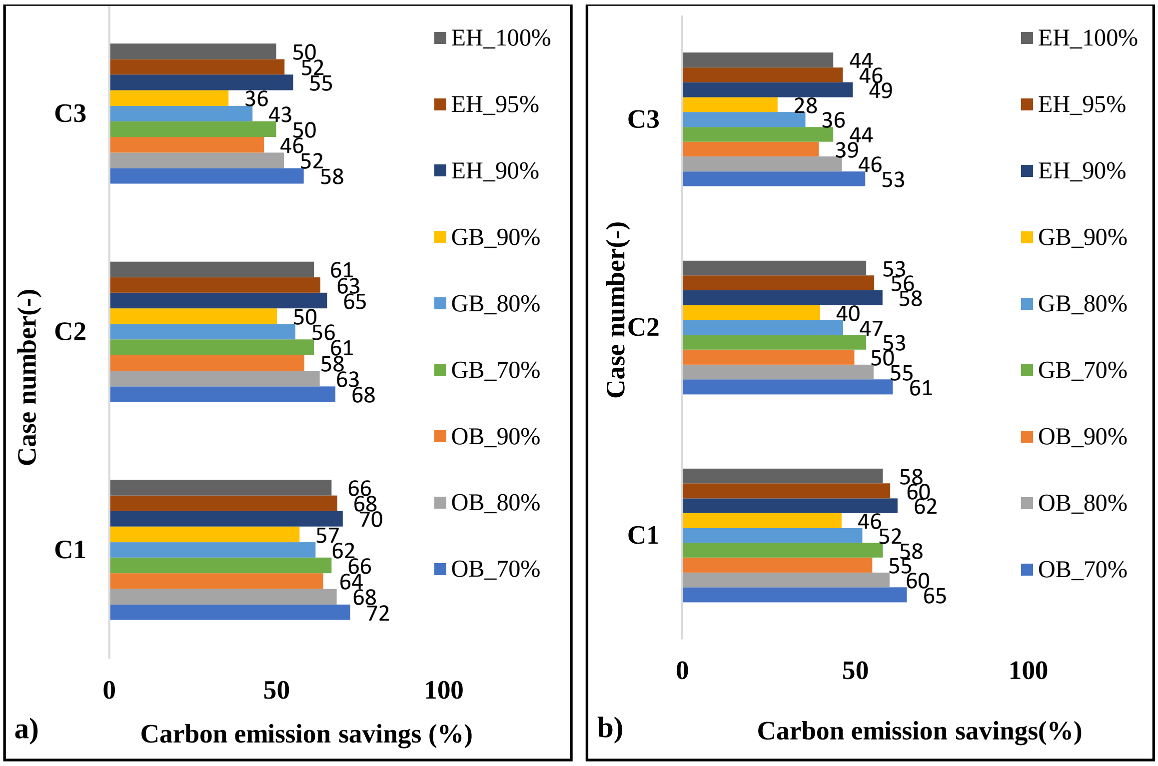


Figure 6.21 Aviemore climatic conditions, a) VSM, b) FSM

6.6 Retrofit Assessment with other property types & ages period in Belfast region

The HP performance and retrofit assessment and annual running cost, carbon emissions, energy input, heat output in addition to all other heat technologies considered for five property types with four age period have been presented in this section. The other heating technologies including oil/gas boilers, EH and very high temperature ASHP comparative results can be observed from Tables 6.6- 6.10, for Flats, Mid terraced, End terraced, Semi-detached, and Detached type respectively. The Flats annual running cost varies in the range of £882 to £426 for VSM(C1) and become £1360 to £1094 for FSM(C1) according to the age period (Table 6.6). Similarly, for VSM (C3) the range is in between £1361 to £498, while for FSM(C3) range increases to £1821 to £1433. The annual running cost for other heating technologies could be observed.

6.6.1 HP running cost and carbon emission savings analysis with different property types

The annual running cost and carbon emissions savings with the developed HP while meeting the same heat demand for different property types, age period, in comparison to other heating technologies operating at different percentage efficiencies will be discussed in this section. The savings depends on the property type, operating mode, three cases (C1, C2, and C3) for the HP, age factor, operating efficiency for other heating technologies and climatic conditions. The climatic conditions of Belfast have been considered, discussed, and presented in this section. The other three climatic conditions of Valentia, Dublin and Aviemore with all property types and age period have been moved to Appendix B. The first age period (1900-1949) mentioned here as old age period show positive values with cost savings for developed HP VSM (C1) in comparison to other heating technologies except the GB at 90% efficiency where the cost savings become negative with the values of -10%, -6%, -5%, and -6% for the four building types (Mid terrace, End terrace, Semi-detached, Detached). The Flats type savings become negative at 80% GB efficiency in addition to the 90% with the value of -3% and -16% respectively. The modern buildings with age period (1991-2007) the retrofitting of the developed HP become less advantageous with negative cost savings at 90%, & 80% GB with values of -22% & -9%, -18% & -5%, -17%& -4%, -18%, -4% for the Flats, Mid terrace, End terrace, Semi-detached, respectively. In case of Detached type, GB at 90% efficiency only is beneficial than the HP with cost saving value of -7%. The carbon emissions savings for the developed HP in VSM(C1) is positive for all property types and age period. Similarly, the VSM with other two cases can be analysed for cost and carbon emission savings for all property types and respective age period.

Table 6.6: HP retrofit assessment for Flat property type and four age period in Belfast

Developed HP	**1900-1950**			**1951-1975**			**1975-1990**			**1991-2007 onwards**		
1) Variable speed mode (VSM)												
Case number (-)	1	2	3	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	5042	5912	7774	3878	4536	6009	2642	3088	4164	2433	2846	3843
Annual running cost (£)	882	1035	1361	679	794	1052	462	540	729	426	498	673
Annual C02 emissions(kg)	1462	1715	2255	1125	1315	1743	766	896	1207	706	825	1114
2) Fixed speed mode (FSM)												
Annual Input power (KWh)	7774	8878	10405	7050	8061	9350	6359	7281	8343	6252	7161	8187
Annual running cost (£)	1360	1554	1821	1234	1411	1636	1113	1274	1460	1094	1253	1433
Annual C02 emissions(kg)	2255	2575	3018	2044	2338	2711	1844	2112	2419	1813	2077	2374
3) Oil Boilers												
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	16182	18205	20806	12007	13508	15438	8026	9030	10320	7411	8338	9529
Annual running cost (£)	1100	1238	1415	816	919	1050	546	614	702	504	567	648
Annual C02 emissions (kg)	3932	4424	5056	2918	3282	3751	1950	2194	2508	1801	2026	2316
4) Gas Boilers												
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	16182	18205	20806	12007	13508	15438	8026	9030	10320	7411	8338	9529
Annual running cost (£)	761	856	978	564	635	726	377	424	485	348	392	448
C02 emission(kg)	3285	3696	4224	2437	2742	3134	1629	1833	2095	1505	1693	1934
5) Electric Heater												
Efficiency (%)	100%	95%	90%	100%	95%	90%	100%	95%	90%	100%	95%	90%
Annual Input power (KWh)	14564	15331	16182	10806	11375	12007	7224	7604	8026	6670	7021	7411
Annual running cost (£)	2549	2683	2832	1891	1991	2101	1264	1331	1405	1167	1229	1297
Annual C02 emissions(kg)	4224	4446	4693	3134	3299	3482	2095	2205	2328	1934	2036	2149
6) Very High Temperature VSHP												
Water Supply Temperature(oC)	65-75	60	75	65-75	60	75	65-75	60	75	65-75	60	75
Annual Input power (KWh)	6278	6774	6870	4658	5026	5097	3114	3360	3407	2875	3102	3146
Annual COP	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12
Annual running cost (£)	1099	1185	1202	815	880	892	545	588	596	503	543	551
Annual C02 emissions(kg)	1821	1964	1992	1351	1458	1478	903	974	988	834	900	912

The cost and carbon emission savings strongly depends on the performance result of the developed HP discussed in previous section. The cost and carbon emission savings have been discussed for VSM vs. FSM in the respective sections dedicated to individual property type.

Table 6.7: HP retrofit assessment for Mid-terraced type and four age period in Belfast

Developed HP	**1900-1949**			**1950-1975**			**1976-1990**			**1991-2007 onwards**		
1) Variable speed mode (VSM)												
Case number (-)	1	2	3	**1**	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**
Annual Input power (KWh)	6853	8008	10457	5684	6663	8747	4510	5270	7125	4470	5221	7056
Annual running cost (£)	1199	1401	1830	995	1166	1531	789	922	1247	782	914	1235
Annual C02 emissions(kg)	1987	2322	3033	1648	1932	2537	1308	1528	2066	1296	1514	2046
2) Fix speed mode (FSM)												
Annual Input power (KWh)	9090	10290	12229	8200	9354	11021	7432	8492	9906	7409	8466	9873
Annual running cost (£)	1591	1801	2140	1435	1637	1929	1301	1486	1734	1297	1482	1728
Annual C02 emissions(kg)	2636	2984	3546	2378	2713	3196	2155	2463	2873	2149	2455	2863
3) Oil Boilers												
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	23238	26142	29877	18610	20936	23927	14208	15984	18267	14079	15838	18101
Annual running cost (£)	1580	1778	2032	1265	1424	1627	966	1087	1242	957	1077	1231
Annual C02 emissions(kg)	5647	6353	7260	4522	5087	5814	3453	3884	4439	3421	3849	4399
4) Gas Boilers												
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	23238	26142	29877	18610	20936	23927	14208	15984	18267	14079	15838	18101
Annual running cost (£)	1092	1229	1404	875	984	1125	668	751	859	662	744	851
C02 emission(kg)	4717	5307	6065	3778	4250	4857	2884	3245	3708	2858	3215	3674
5) Electric Heater												
Efficiency (%)	100%	95%	90%	100%	95%	90%	100%	95%	90%	100%	95%	90%
Annual Input power (KWh)	20914	22015	23238	16749	17630	18610	12787	13460	14208	12671	13338	14079
Annual running cost (£)	3660	3853	4067	2931	3085	3257	2238	2356	2486	2217	2334	2464
Annual C02 emissions(kg)	6065	6384	6739	4857	5113	5397	3708	3903	4120	3674	3868	4083
6) Very High Temperature VSHP												
Water Supply Temperature(°C)	65-75	60	75	65-75	60	75	65-75	60	75	65-75	60	75
Annual Input power (KWh)	9015	9727	9865	7219	7790	7900	5512	5948	6032	5462	5893	5977
Annual COP	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12
Annual running cost (£)	1578	1702	1726	1263	1363	1383	965	1041	1056	956	1031	1046
Annual C02 emissions(kg)	2614	2821	2861	2094	2259	2291	1598	1725	1749	1584	1709	1733

Table 6.8: HP retrofit assessment for End-terraced type and four age period in Belfast

Developed HP	**1900-1949**			**1950-1975**			**1976-1990**			**1991-2007 onwards**		
1) Variable speed mode (VSM)												
Case number (-)	1	2	3	**1**	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**
Annual Input power (KWh)	8801	10144	13157	6620	7742	10126	5618	6531	8779	4923	5821	7867
Annual running cost (£)	1540	1775	2303	1158	1355	1772	983	1143	1536	861	1019	1377
Annual C02 emissions(kg)	2552	2942	3816	1920	2245	2936	1629	1894	2546	1428	1688	2281
2) Fix speed mode (FSM)												
Annual Input power (KWh)	11448	12345	14693	8896	10094	11977	8154	9303	10955	7690	8783	10283
Annual running cost (£)	2003	2160	2571	1557	1767	2096	1427	1628	1917	1346	1537	1799
Annual C02 emissions(kg)	3320	3580	4261	2580	2927	3473	2365	2698	3177	2230	2547	2982
3) Oil Boilers												
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	30940	34808	39781	22299	25086	28670	18351	20644	23594	15697	17659	20182
Annual running cost (£)	2104	2367	2705	1516	1706	1950	1248	1404	1604	1067	1201	1372
Annual C02 emissions(kg)	7519	8458	9667	5419	6096	6967	4459	5017	5733	3814	4291	4904
4) Gas Boilers												
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	30940	34808	39781	22299	25086	28670	18351	20644	23594	15697	17659	20182
Annual running cost (£)	1454	1636	1870	1048	1179	1348	862	970	1109	738	830	949
C02 emission(kg)	6281	7066	8075	4527	5093	5820	3725	4191	4790	3186	3585	4097
5) Electric Heater												
Efficiency (%)	100%	95%	90%	100%	95%	90%	100%	95%	90%	100%	95%	90%
Annual Input power (KWh)	27846	29312	30940	20069	21125	22299	16516	17385	18351	14127	14871	15697
Annual running cost (£)	4873	5130	5415	3512	3697	3902	2890	3042	3211	2472	2602	2747
Annual C02 emissions(kg)	8075	8500	8973	5820	6126	6467	4790	5042	5322	4097	4312	4552
6) Very High Temperature VSHP												
Water Supply Temperature(°C)	65-75	60	75	65-75	60	75	65-75	60	75	65-75	60	75
Annual Input power (KWh)	12003	12952	13135	8651	9335	9467	7119	7682	7790	6089	6571	6664
Annual COP	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12
Annual running cost (£)	2100	2267	2299	1514	1634	1657	1246	1344	1363	1066	1150	1166
Annual C02 emissions(kg)	3481	3756	3809	2509	2707	2745	2064	2228	2259	1766	1906	1932

Table 6.9: HP retrofit assessment for Semi-detached type and four age period in Belfast

Developed HP	**1900-1949**			**1950-1975**			**1976-1990**			**1991-2007 onwards**		
1) Variable speed mode (VSM)												
Case number (-)	1	2	3	**1**	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**
Annual Input power (KWh)	9356	10726	13884	6993	8061	10727	5644	6560	8815	4690	5476	7399
Annual running cost (£)	1637	1877	2430	1224	1411	1877	988	1148	1543	821	958	1295
Annual C02 emissions(kg)	2713	3111	4026	2028	2338	3111	1637	1902	2556	1360	1588	2146
2) Fix speed mode (FSM)												
Annual Input power (KWh)	12353	13120	15516	9215	10404	12380	8177	9328	10988	7544	8618	10070
Annual running cost (£)	2162	2296	2715	1613	1821	2166	1431	1632	1923	1320	1508	1762
Annual C02 emissions(kg)	3582	3805	4500	2672	3017	3590	2371	2705	3186	2188	2499	2920
2) Oil Boilers												
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	33044	37175	42485	23788	26761	30584	18480	20790	23760	14855	16712	19100
Annual running cost (£)	2247	2528	2889	1618	1820	2080	1257	1414	1616	1010	1136	1299

Annual C02 emissions(kg)	8030	9033	10324	5780	6503	7432	4491	5052	5774	3610	4061	4641
3) Gas Boilers												
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	33044	37175	42485	23788	26761	30584	18480	20790	23760	14855	16712	19100
Annual running cost (£)	1553	1747	1997	1118	1258	1437	869	977	1117	698	785	898
C02 emission(kg)	6708	7546	8625	4829	5433	6209	3751	4220	4823	3016	3393	3877
4) Electric Heater												
Efficiency (%)	100%	95%	90%	100%	95%	90%	100%	95%	90%	100%	95%	90%
Annual Input power (KWh)	29740	31305	33044	21409	22536	23788	16632	17507	18480	13370	14073	14855
Annual running cost (£)	5204	5478	5293	3747	3944	4163	2911	3064	3234	2340	2463	2600
Annual C02 emissions(kg)	8625	9078	9583	6209	6535	6898	4823	5077	5359	3877	4081	4308
5) Very High Temperature VSHP												
Water Supply Temperature(^{o}C)	65-75	60	75	65-75	60	75	65-75	60	75	65-75	60	75
Annual Input power (KWh)	12819	13832	14028	9228	9958	10099	7169	7736	7845	5763	6218	6306
Annual COP	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12
Annual running cost(£)	2243	2421	2455	1615	1743	1767	1255	1354	1373	1008	1088	1104
Annual C02 emissions(kg)	3717	4011	4068	2676	2888	2929	2079	2243	2275	1671	1803	1829

Table 6.10: HP retrofit assessment for Detached property type and four age period in Belfast

Developed HP	**1900-1949**			**1950-1975**			**1976-1990**			**1991-2007 onwards**		
1) Variable speed mode (VSM)												
Case number (-)	1	2	3	**1**	**2**	**3**	**1**	**2**	**3**	**1**	**2**	**3**
Annual Input power (KWh)	12067	13429	17433	10179	11467	14910	8761	9975	13097	8219	9392	12379
Annual running cost (£)	2112	2350	3051	1781	2007	2609	1533	1746	2292	1438	1644	2166
Annual C02 emissions(kg)	3499	3894	5056	2952	3325	4324	2541	2893	3798	2383	2724	3590
2) Fix speed mode (FSM)												
Annual Input power (KWh)	18109	17676	19925	14043	14402	16795	11432	12268	14630	10613	11605	13876
Annual running cost (£)	3169	3093	3487	2458	2520	2939	2001	2147	2560	1857	2031	2428
Annual C02 emissions(kg)	5252	5126	5778	4073	4176	4871	3315	3558	4243	3078	3366	4024
3) Oil Boilers												
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	42317	47606	54407	36054	40561	46355	30779	34626	39572	28675	32259	36868
Annual running cost (£)	2878	3237	3700	2452	2758	3152	2093	2355	2691	1950	2194	2507
Annual C02 emissions(kg)	10283	11568	13221	8761	9856	11264	7479	8414	9616	6968	7839	8959
4) Gas Boilers												
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	42317	47606	54407	36054	40561	46355	30779	34626	39572	28675	32259	36868
Annual running cost (£)	1989	2237	2557	1695	1906	2179	1447	1627	1860	1348	1516	1733
C02 emission(kg)	8590	9664	11045	7319	8234	9410	6248	7029	8033	5821	6549	7484
5) Electric Heater												
Efficiency (%)	100%	95%	90%	100%	95%	90%	100%	95%	90%	100%	95%	90%
Annual Input power (KWh)	38085	40089	42317	32449	34156	36054	27701	29159	30779	25807	27166	28675
Annual running cost (£)	6665	7016	7405	5679	5977	6309	4848	5103	5386	4516	4754	5018
Annual C02 emissions(kg)	11045	11626	12272	9410	9905	10456	8033	8456	8926	7484	7878	8316
6) Very High Temperature VSHP												
Water Supply Temperature(oC)	65-75	60	75	65-75	60	75	65-75	60	75	65-75	60	75
Annual Input power (KWh)	16416	17714	17965	13986	15092	15306	11940	12884	13066	11124	12003	12173
Annual COP	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12	2.32	2.15	2.12
Annual running cost (£)	2873	3100	3144	2448	2641	2679	2089	2255	2287	1947	2101	2130
Annual C02 emissions(kg)	4761	5137	5210	4056	4377	4439	3463	3736	3789	3226	3481	3530

6.6.1.1 HP cost savings with FLATS type in two control modes

The following Figure 6.22-6.25(a, b) showing the cost savings for the four-age period in two control modes when the oil/gas boiler, EH and high temperature HP is retrofitted with the develop HP. The positive values shows that the annual running cost is less than the other heating technologies and negative values indicated that running cost is higher. The VSM(C1) for the developed HP have cost savings in comparison to all other heating technologies except GB AT 90%, and 80% efficiency. The cost savings also varies with the age period. For other two cases (C2, C3) VSM have shown some more negative values. The high heat supply temperature HP performance was found superior as the high heat supply temperature is more beneficial than the developed HP at C3 almost in all cases. The four-age period shows increasing trend of negative cost saving in VSM(C3) with the values of (-24 %, -29%, -34%, and -34%) in comparison to VHTHP at COP value of 2.32. The FSM operation of the system is only beneficial in terms of cost when compared to the EH option and become expensive in comparison to all other heating technologies even at lowest efficiency.

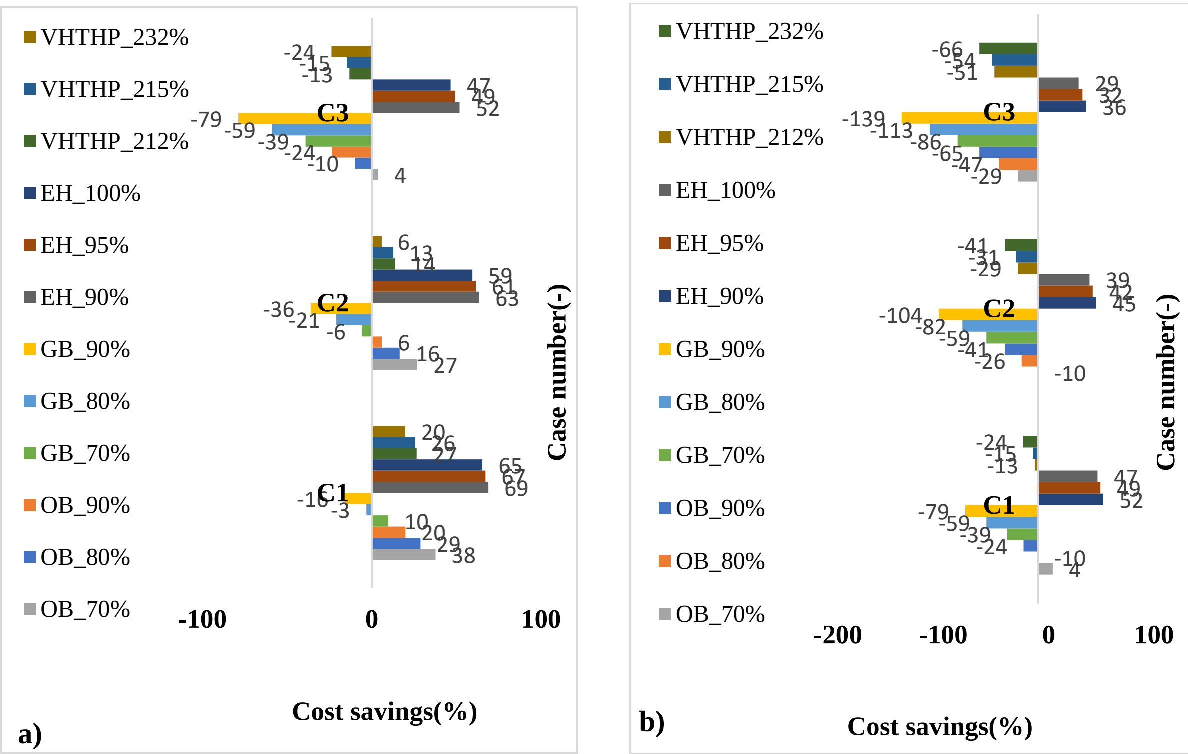


Figure 6.22 HP cost savings in flat type with age period of 1900-1949, a) VSM, b) FSM

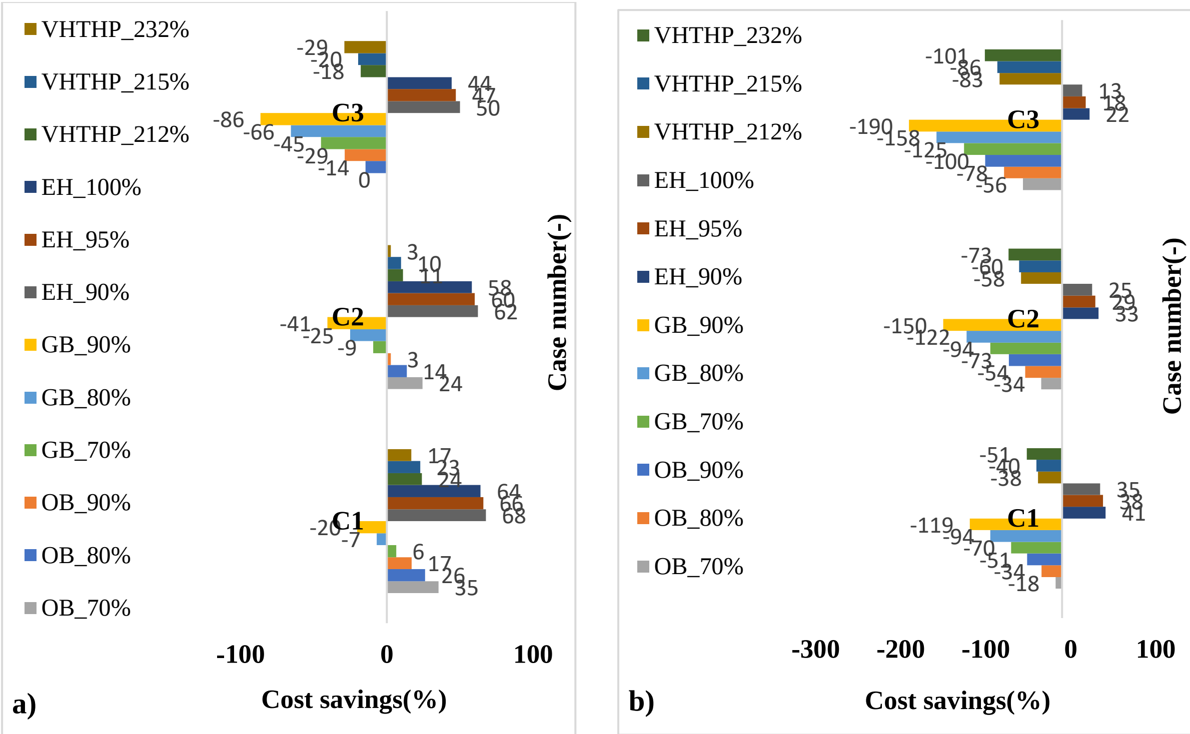


Figure 6.23 HP cost savings in flat type with age period of 1950-1975, a) VSM, b) FSM

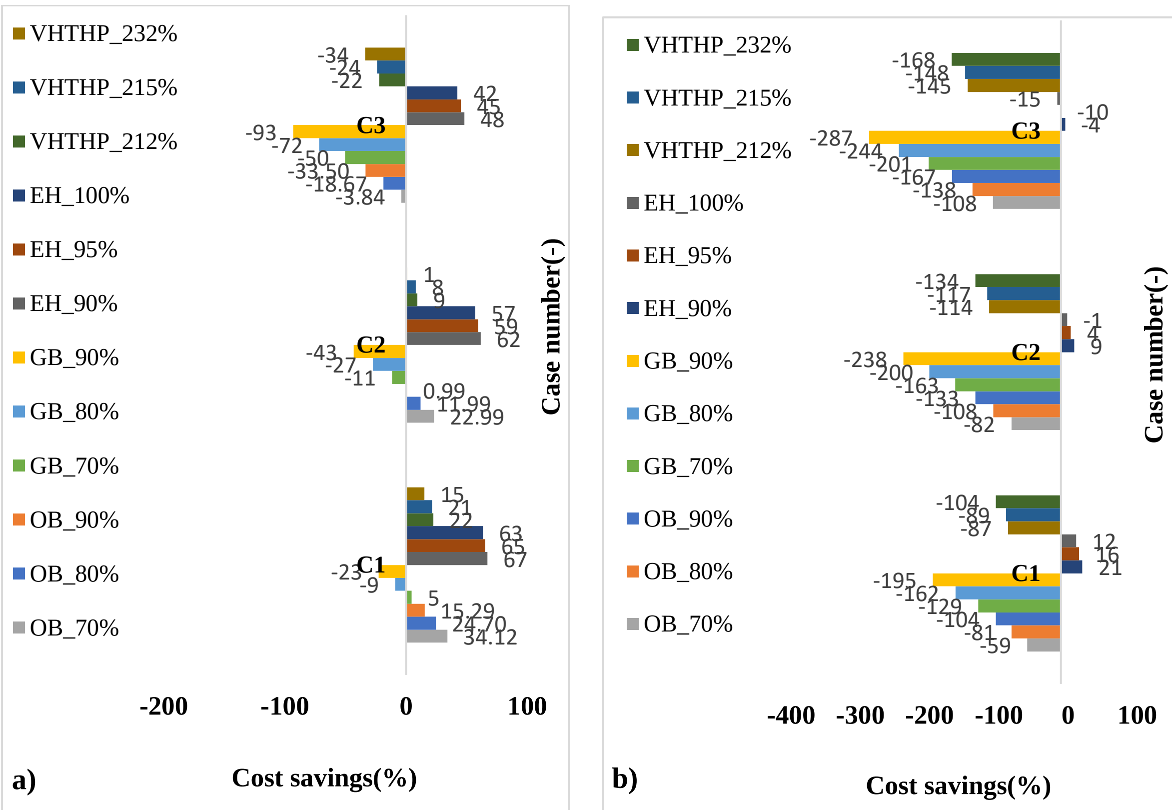


Figure 6.24 HP cost savings in flat type with age period of 1976-1990, a) VSM, b) FSM

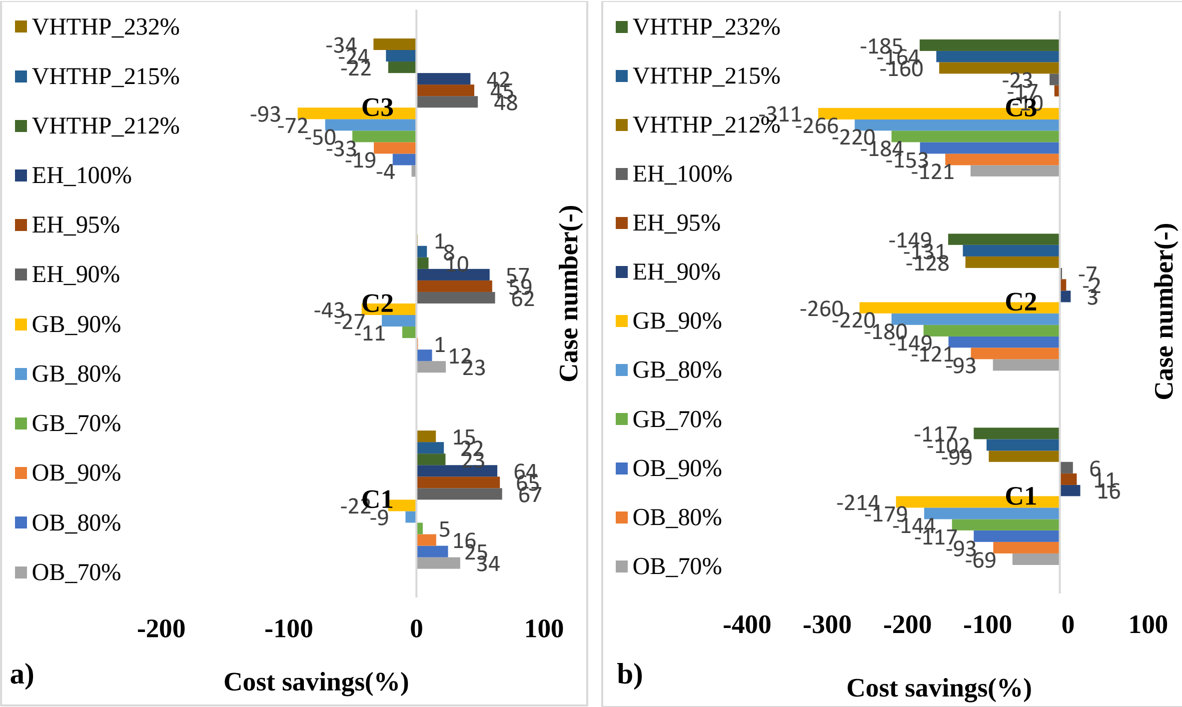


Figure 6.25 HP cost savings in flat type with age period of 1991-2007, a) VSM, b) FSM

6.6.1.2 HP carbon emission savings with FLATS type

The annual carbon emissions for meeting the heat demand for Flats with different age period, and the heating technology type was presented in Table 6.6, while the carbon emission savings for the developed HP have been shown in Figure 6.26-29, when employed instead of either fossil fuel-based boilers (oil& gas) or with the EH, and high temperature ASHP. The carbon emission savings while operating in VSM is possible for the retrofitted oil/gas boilers, EH, high temperature ASHP at all percentage efficiency with the developed HP (C1, &C2), but in case of C3, where only the high temperature ASHP have less carbon emissions and savings become negative. But the developed HP have superseded the other heating technologies in carbon emissions and savings is obtained in all cases, age period. The carbon emission savings for the HP in comparison to the GB at highest efficiency of 90% varies in the ranges from 31%, 48%, and 55 % to the 26%, 45%, and 53% for the three respective cases considered according to the age period (1900-1949) & (1990-2007 onwards). The savings increases with the GB percentage efficiency become low (80%, and 70%). Similarly, the OB at 90% in comparison to the developed HP emit 43%, 56%, and 63% for the three respective cases and age period of 1900-1949 and become 55%, 66%, and 71% when the OB efficiency become 70%. In FSM, the results are different and the high temperature ASHP even outperform better in comparison to the developed HP with less carbon emissions at all cases (C1, C2, C3), age period. The HP in comparison to OB/GB carbon emission saving is positive for the first age period in all cases (C1, C2, C3) and become negative with the age period, which means that the OB/GB retrofitting is more beneficial for the old age period (1900-1949).

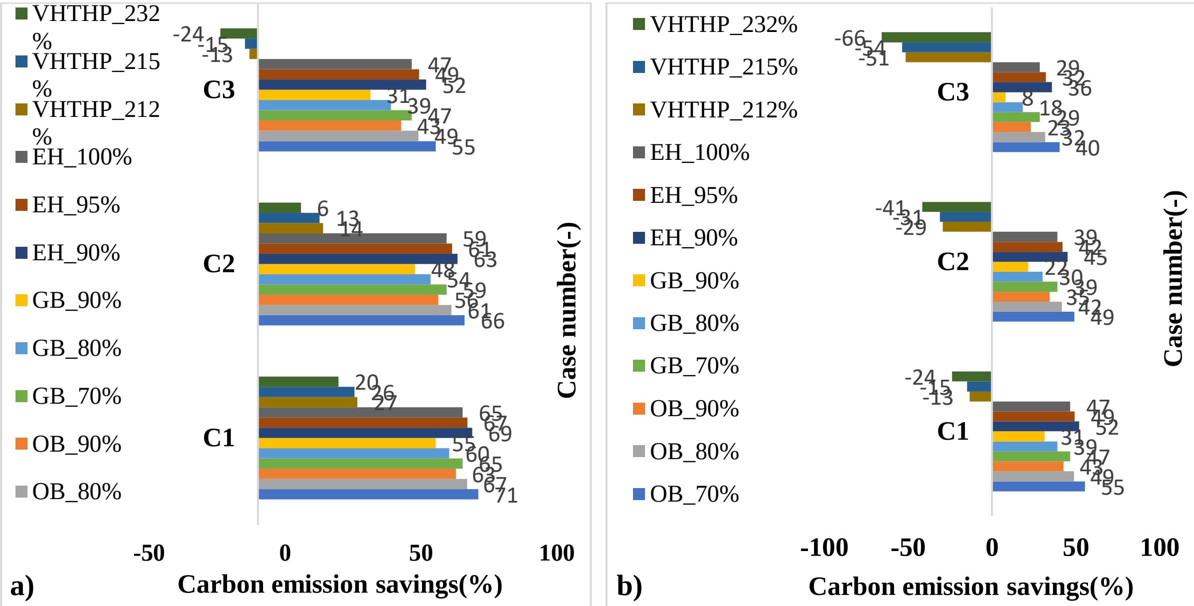


Figure 6.26 HP carbon savings in flat type with age period of 1900-1949, a) VSM, b) FSM

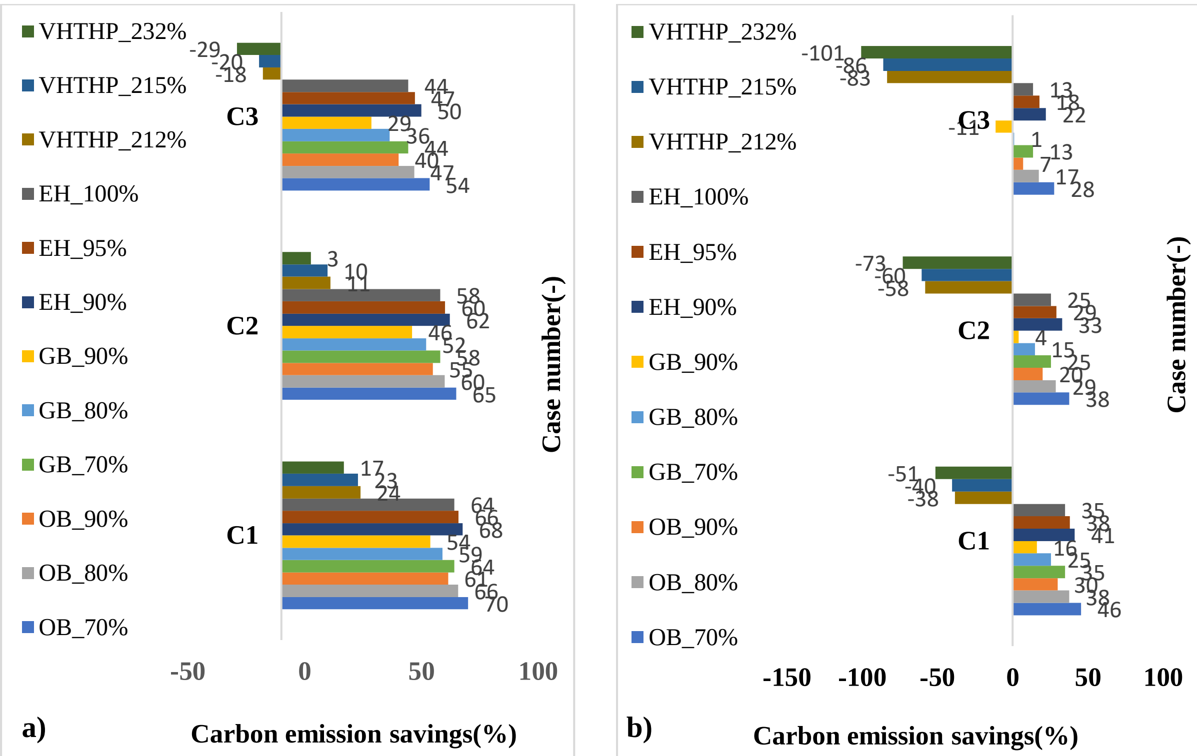


Figure 6.27 HP carbon savings in flat type with age period of 1950-1975, a) VSM, b) FSM

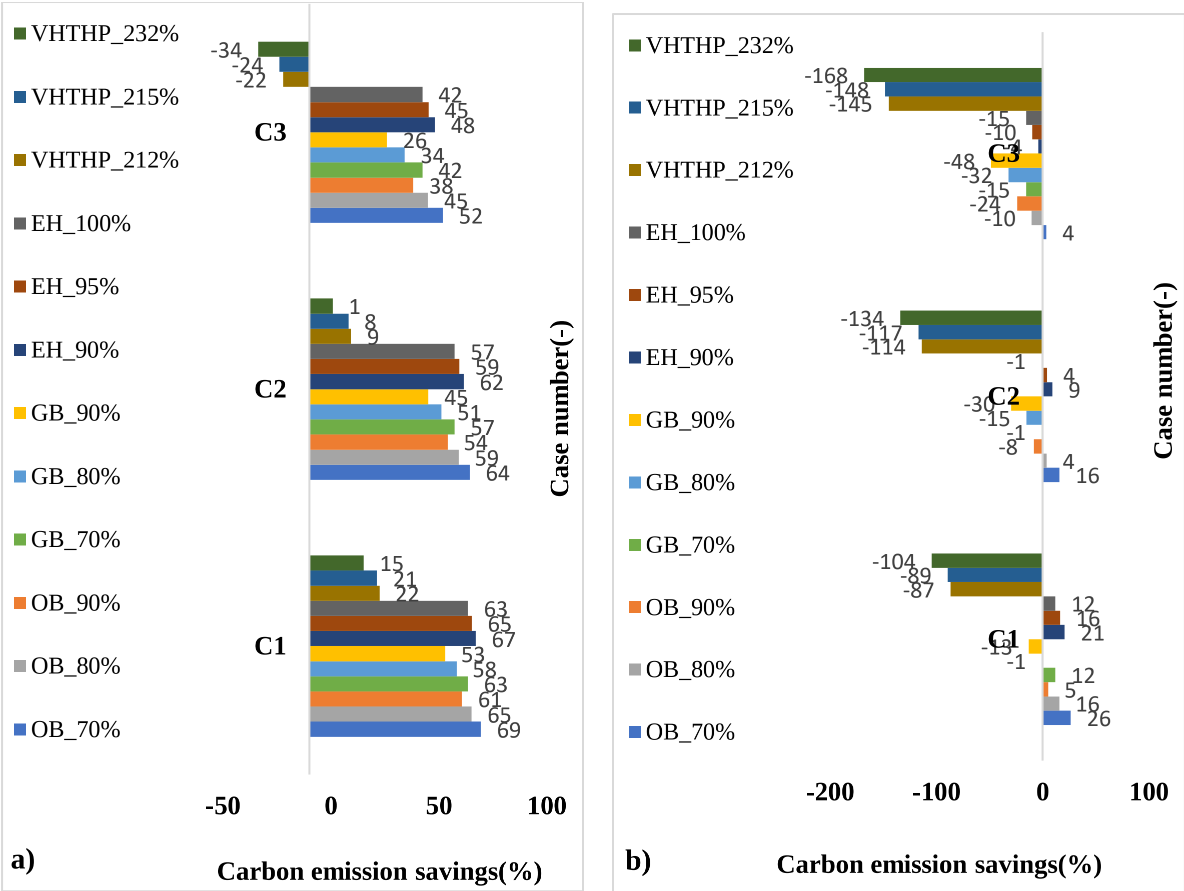


Figure 6.28 HP carbon savings in flat type with age period of 1976-1990, a) VSM, b) FSM

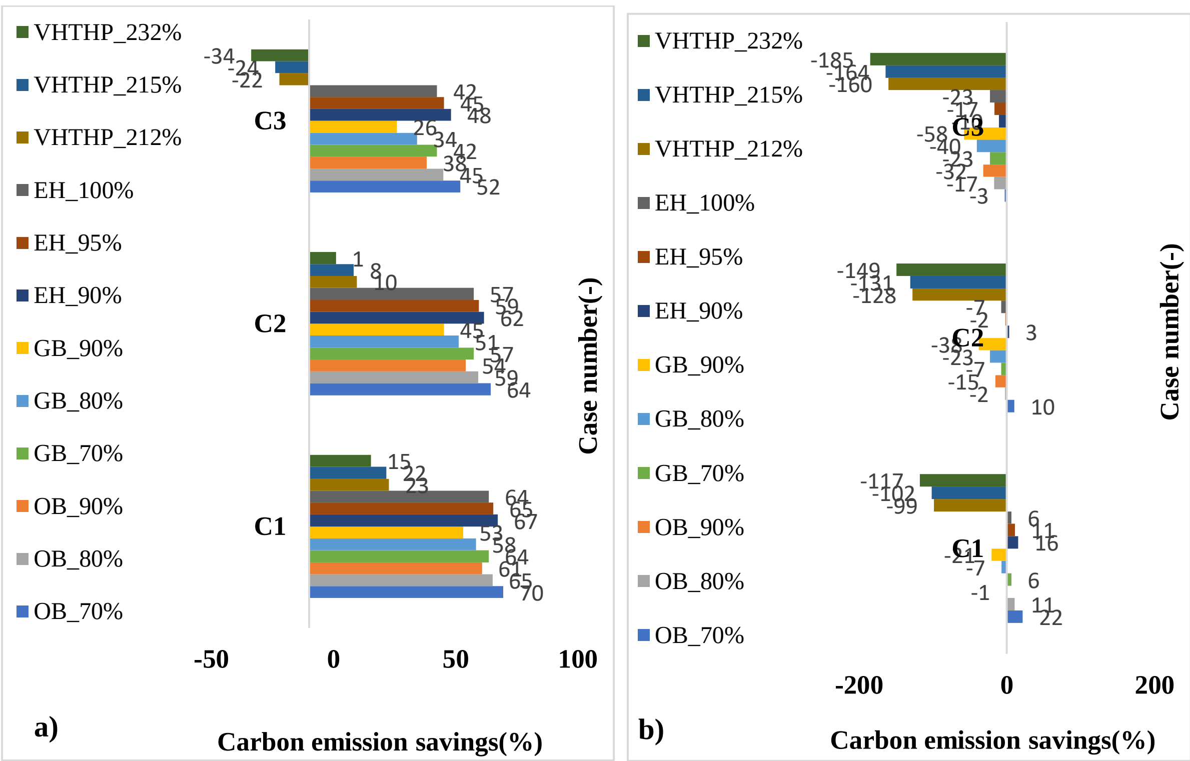


Figure 6.29 HP carbon savings in flat type with age period of 1991-2007, a) VSM, b) FSM

6.6.1.3 HP cost savings with Mid terraced type in two control modes (FSM vs. VSM)

The cost saving with the Mid terraced property type and four age period with two control mode have been shown in Figure 6.30(a,b)-6.33(a,b). The overall trend for cost savings reduces with the four-age period in both control mode and the HP system saves least money compared to the other heating technologies with the modern building having age period of 1991-2007. For example, if we investigate the cost savings with the VSM for the HP against the EH option (at 100% efficiency) for the age period of 1900-1949 building is 67%,62%, and 50% for the case studied C1, C2, and C3 which subsequently reduces to 65%, 59%, and 44% for the age period of 1991-2007. The reasons for this is the load demand match to the heat supplied for larger percentage by the HP at nominal operating point/and closer operation in case of old age period property types. Additionally, the HP system is not beneficial in terms of cost savings when compared to 90% efficiency advanced gas boilers at all three cases, both control mode, and all four-age period. The reason for this is the falling gas prices and better performance efficiency of the gas boiler. However, the difference become less when compared at lower heat supply temperature(C1) due to better HP performance.

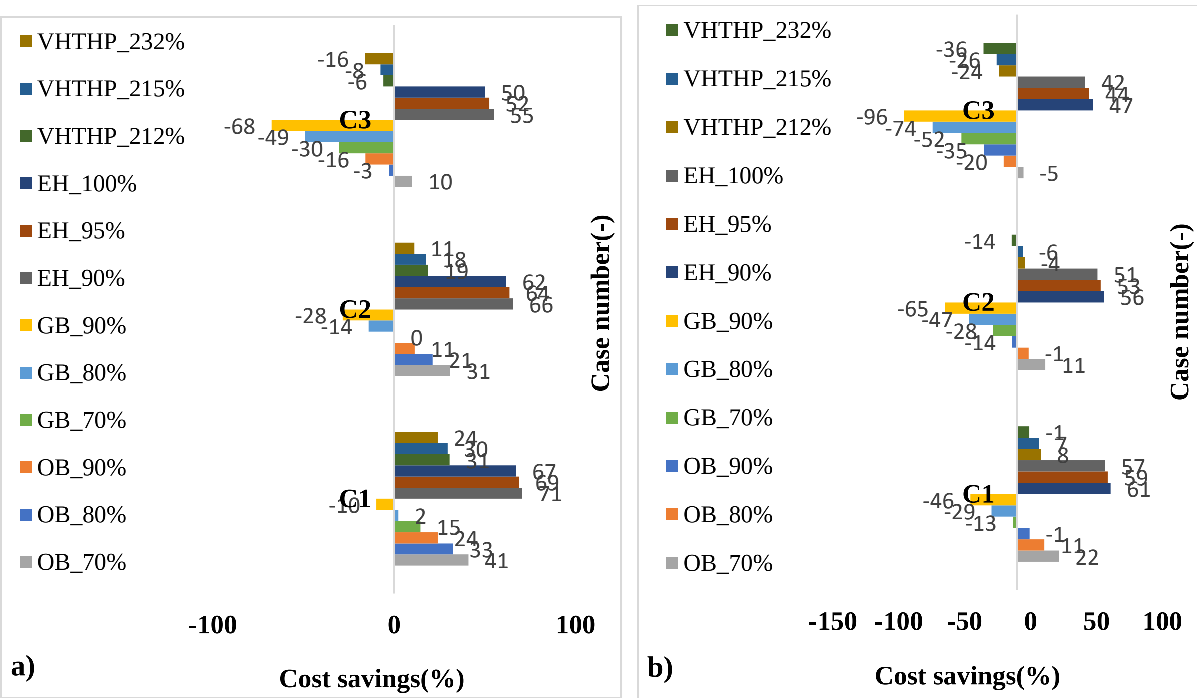


Figure 6.30 HP cost savings in Mid-terrace with age period of 1900-1949, a) VSM, b) FSM

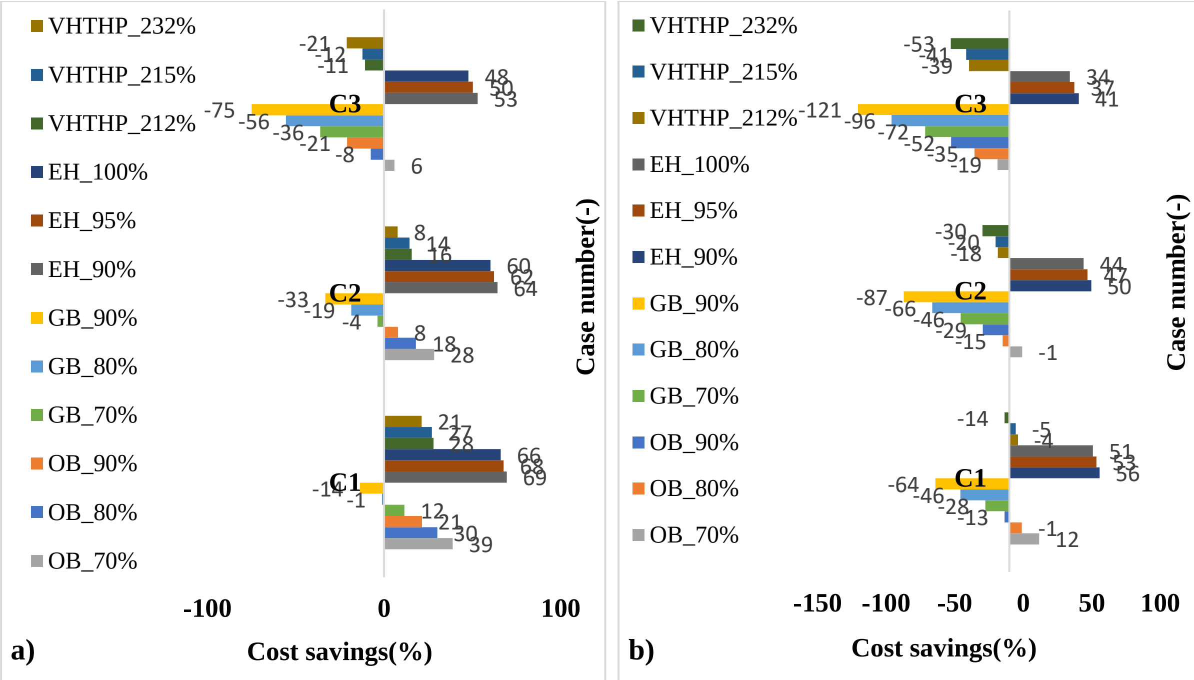


Figure 6.31 HP cost savings in Mid-terrace with age period of 1950-1975, a) VSM, b) FSM

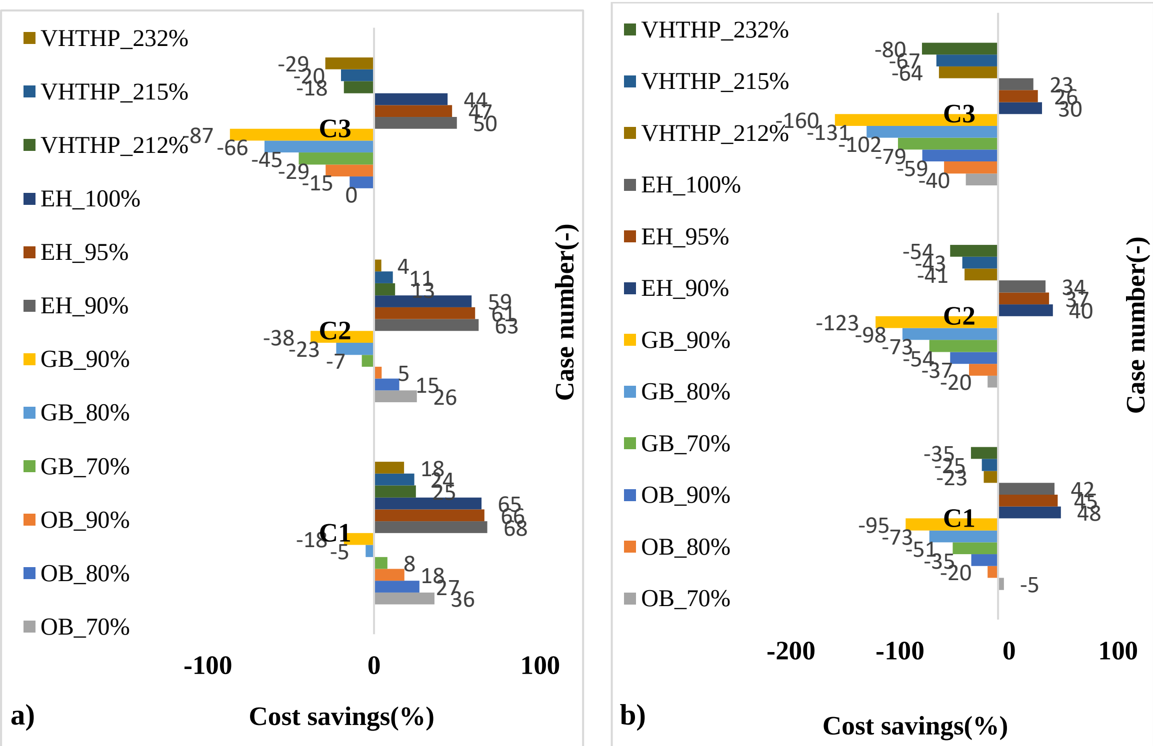


Figure 6.32 HP cost savings in Mid-terrace with age period of 1950-1975, a) VSM, b) FSM

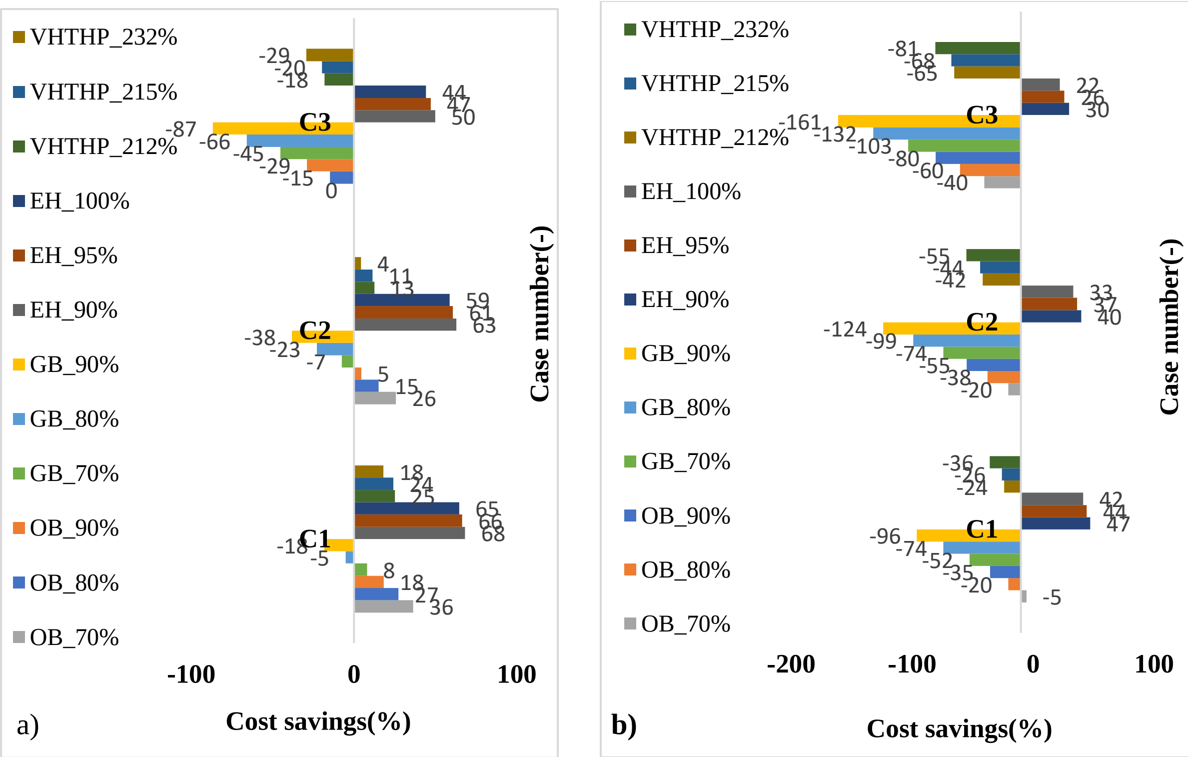


Figure 6.33 HP cost savings in Mid-terrace with age period of 1991-2007, a) VSM, b) FSM

6.6.1.4 HP carbon emission savings with Mid terraced in two control modes (FSM vs. VSM)

The carbon emission savings with the Mid terraced property type and four age period with two control mode have been shown in Figure 6.34-6.37. The HP(VSM) shows carbon emission savings in comparison to all other heating technologies at all percentage efficiencies with the only exception of very high heat supply temperature ASHP at supply temperature of 55 °C with increased cost of 16%. However, in case of FSM operation the HP is also not economical in terms of carbon emission savings at medium heat supply temperature(C2) in comparison to VHTHP. In comparison to all other technologies at all percentage the HP is favourable option and save annual carbon emissions with different percentage efficiencies.

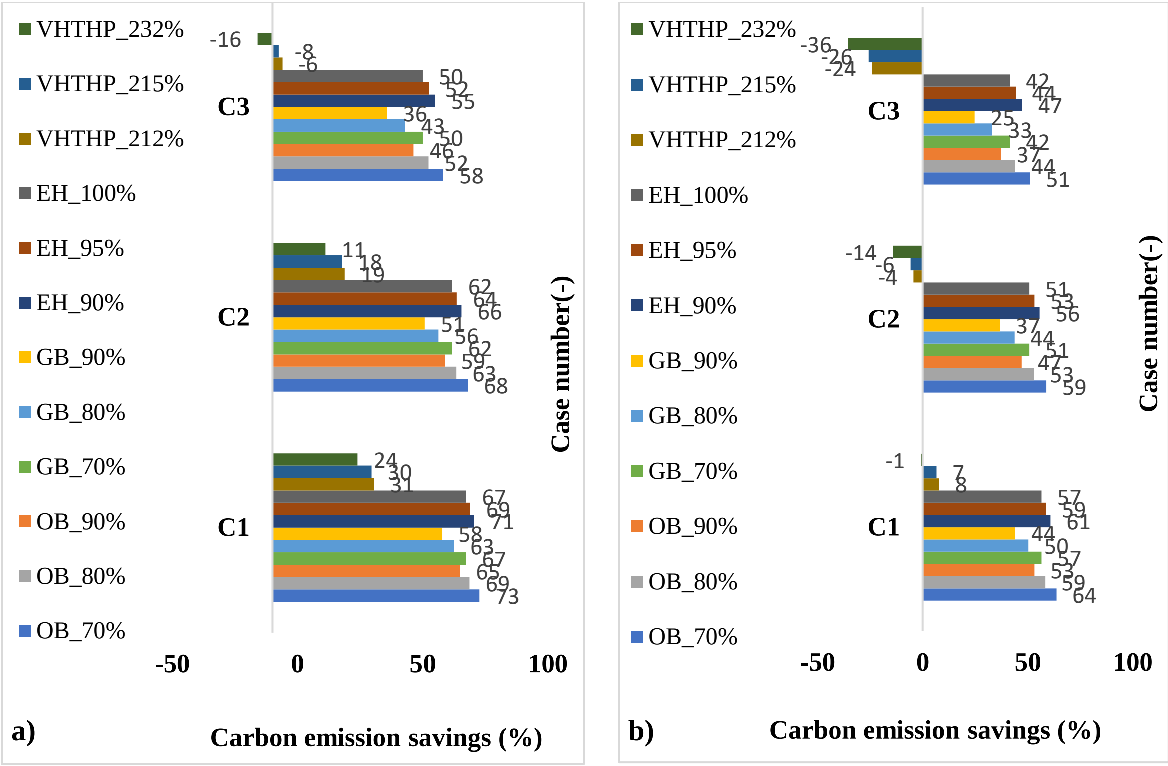


Figure 6.34 HP carbon emission savings in Mid-terrace with age period of 1900-1949, a) VSM, b) FSM

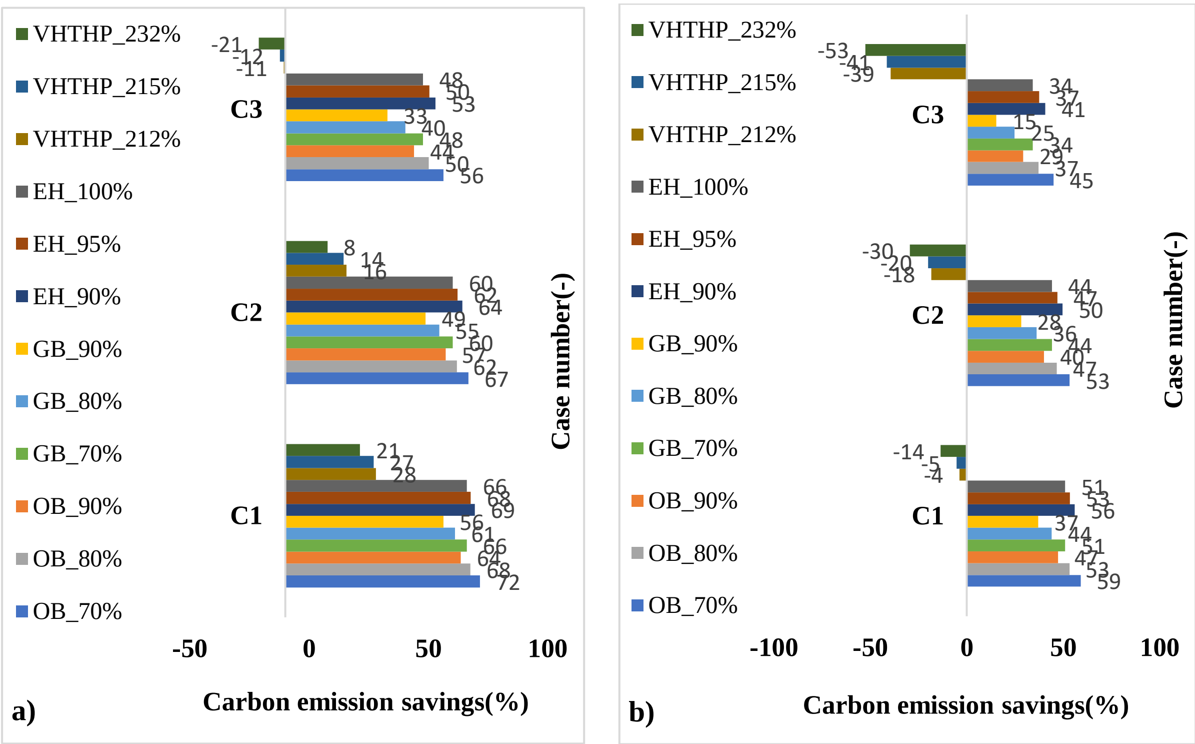


Figure 6.35 HP carbon emission savings in Mid-terrace with age period of 1950-1975, a) VSM, b) FSM

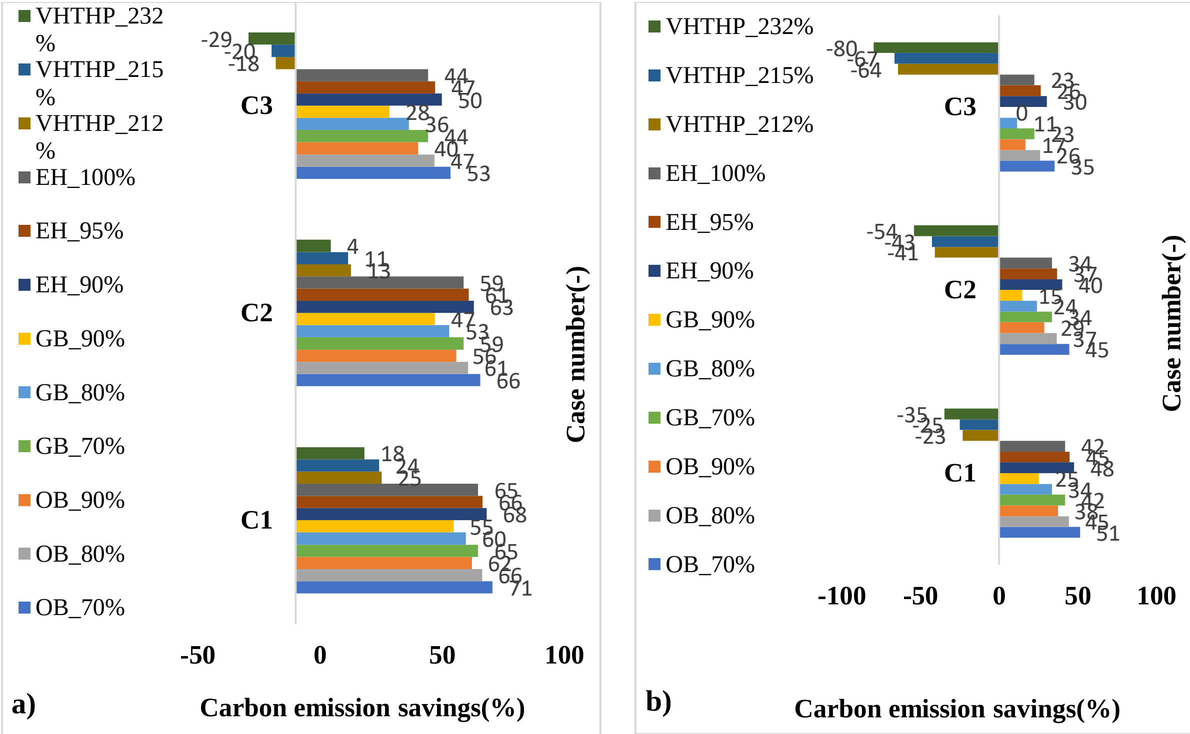


Figure 6.36 HP carbon emission savings in Mid-terrace with age period of 1976-1990, a) VSM, b) FSM

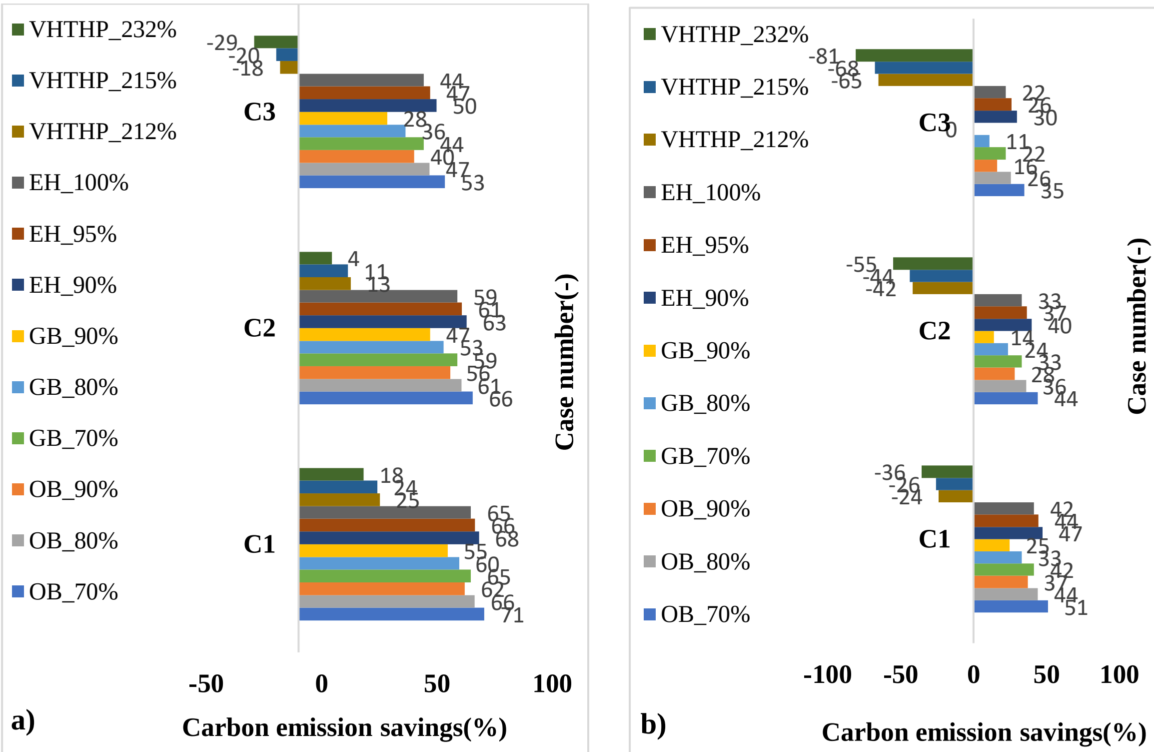


Figure 6.37 HP carbon emission savings in Mid-terrace with age period of 1991-2007, a) VSM, b) FSM

6.6.1.5 HP cost savings with End terraced in two control modes (FSM vs. VSM)

The cost savings with End terraced property type at four age period in two control mode is shown in Figure 6.38-6.41. The annual running cost for all the technologies depends on the age period of the property due to varying annual load demand and the part load operations of the HP system. The overall performance for the HP system and cost savings for all age period is better for the End-terraced type than the Mid-terraced property type for the corresponding four age period. This is evidenced from the results of the cost savings for the HP against the EH(100% efficiency) for the age period of 1900-1949 is 68%, 64%, and 53%, respectively for C1,C2, and C3. The respective savings were 67%,62%, and 50% for C1,C2, and C3 for the Mid-terraced property for the same age period and operating EH efficiency. The reason for this is the better part load performance of the HP system in case of End terraced property type compared to the Mid-terraced property type and this highlights the importance of suitable design of the HP system for individual heat load applications.

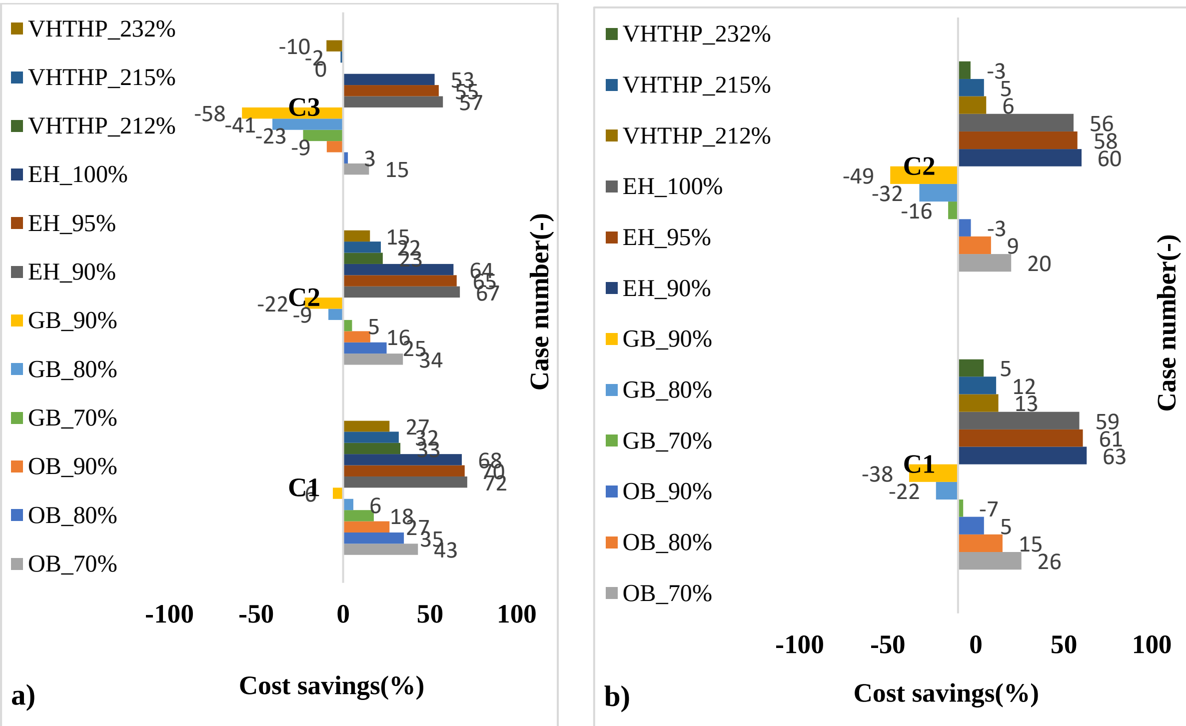


Figure 6.38 HP cost savings in End-terrace with age period of 1900-1949, a) VSM, b) FSM

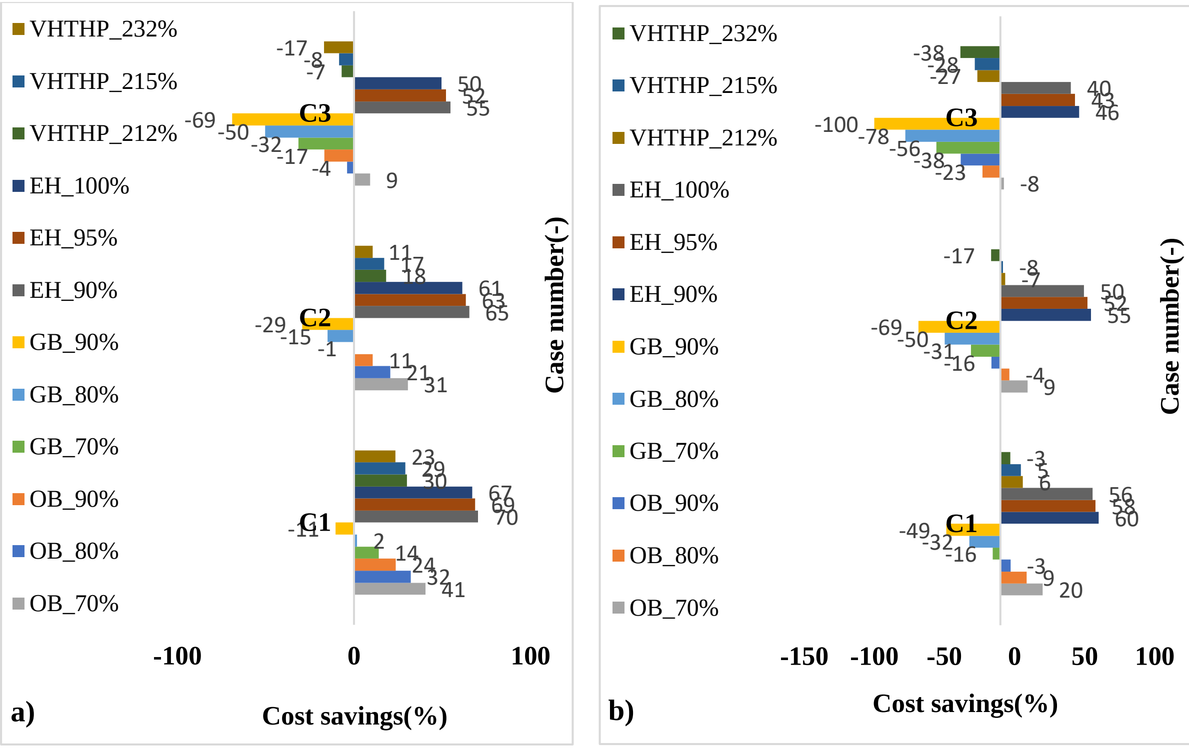


Figure 6.39 HP cost savings in End-terrace with age period of 1950-1975, a) VSM, b) FSM

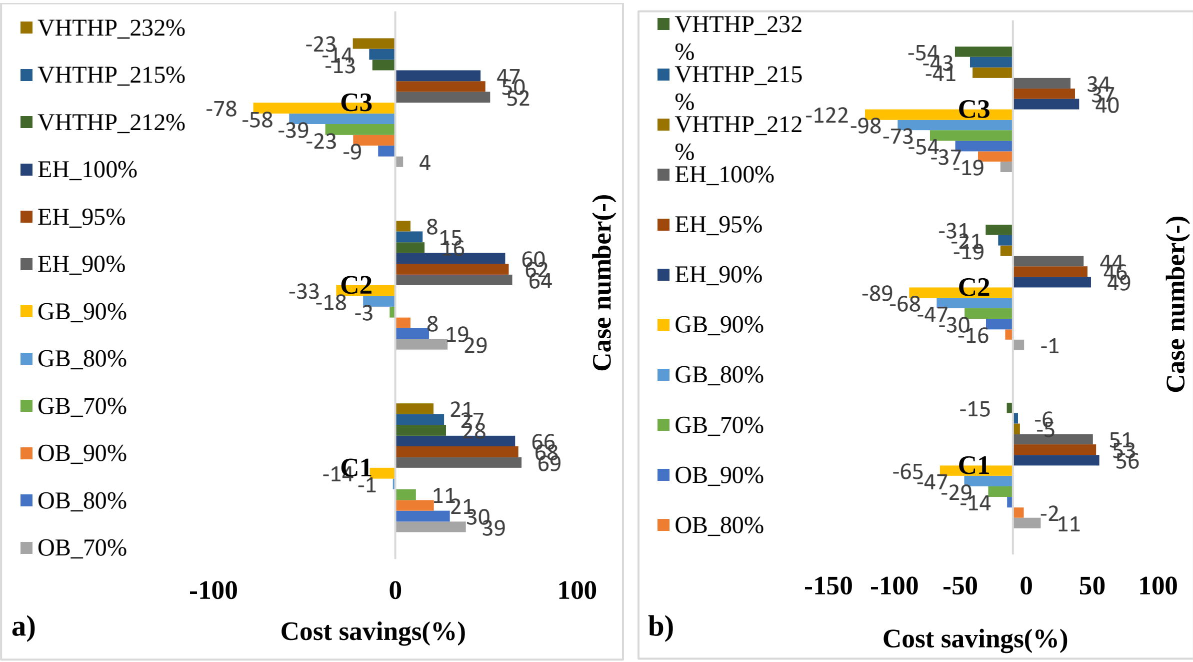


Figure 6.40 HP cost savings in End-terrace with age period of 1976-1990, a) VSM, b) FSM

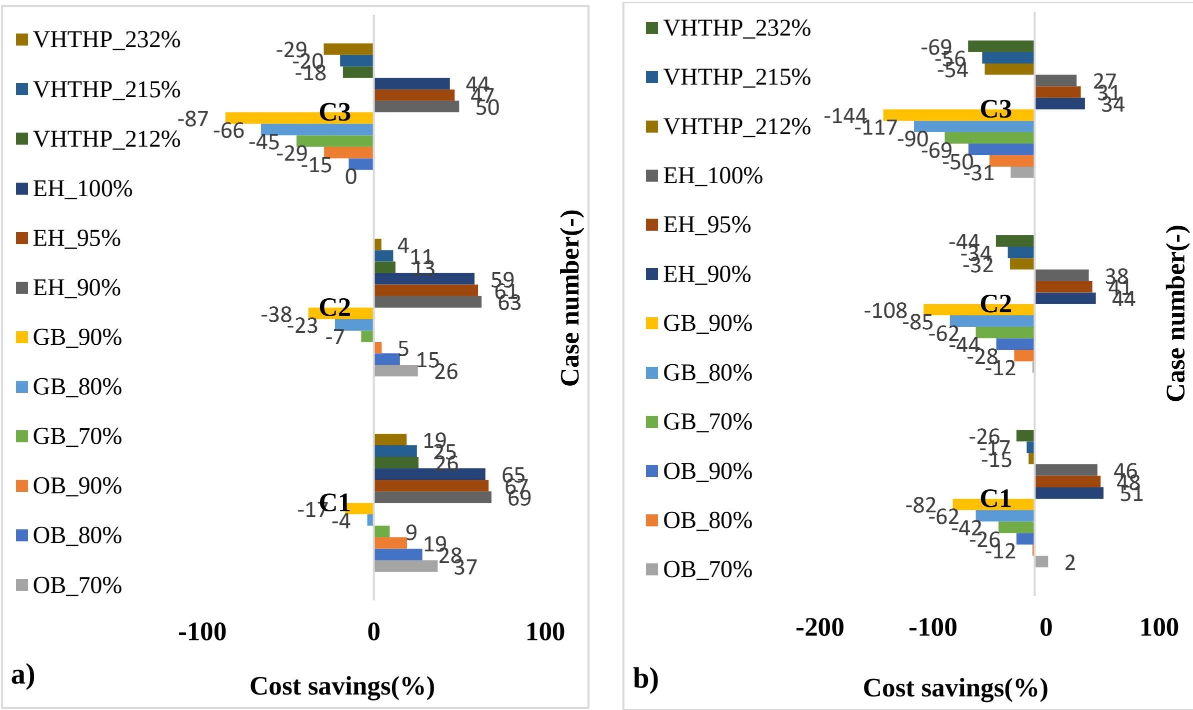


Figure 6.41 HP cost savings in End-terrace with age period of 1991-2007, a) VSM, b) FSM

6.6.1.6 HP carbon emission savings with End terraced in two control modes (FSM vs. VSM)

The HP carbon emission savings with End terraced property type at four age period in two control mode is shown in Figure 6.42-6.45. The annual carbon emission savings for the HP system in comparison to all technologies varies according to the age period of the property. Similar trends of additional carbon emission savings for the End terraced property type in comparison to the Mid-terraced property type have been shown due to the better HP performance. The HP in VSM have shown carbon emission savings for all cases at all age period against all other heat technologies with the only exception for the very high heat supply temperature ASHP at C3 and this is due to the suitability of the cascaded ASHP system at high heat supply temperature(C3) in comparison to our developed ASHP system. However the developed system in VSM outperforms even the very high heat supply temperature in terms of carbon emissions at low and medium heat supply temperature (C1, C2).

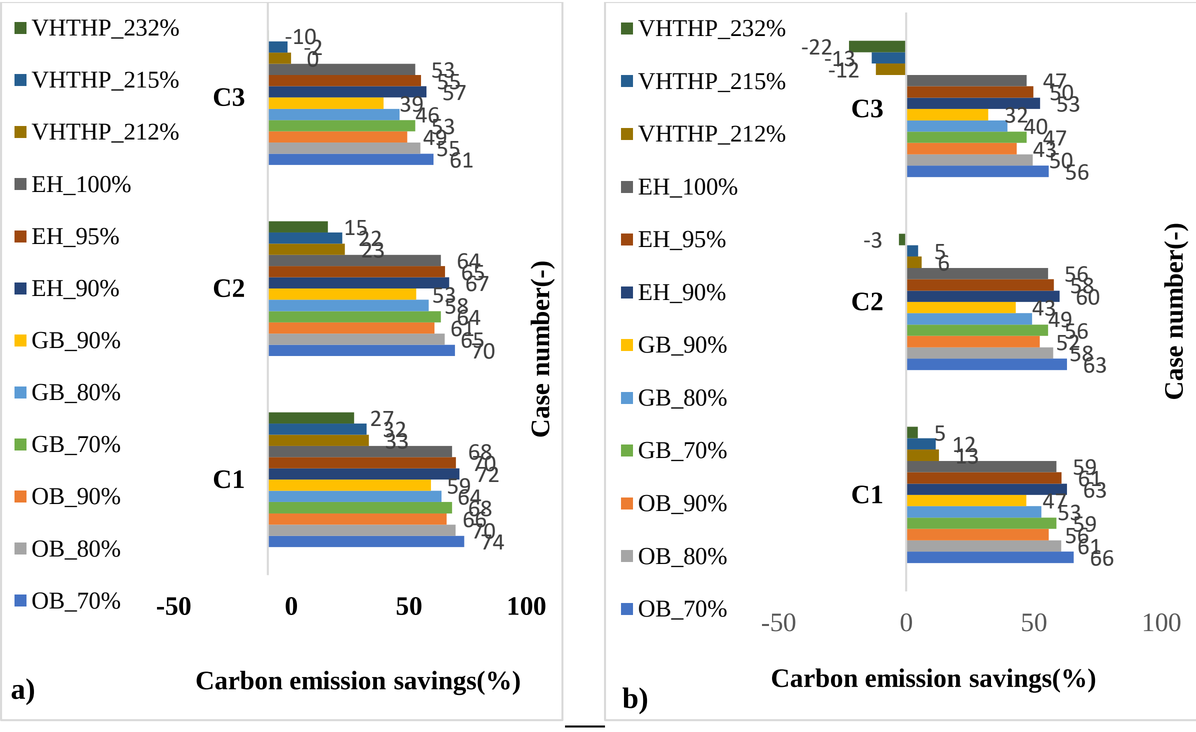


Figure 6.42 Carbon emission savings in End-terrace with age period of 1990-1949, a) VSM, b) FSM

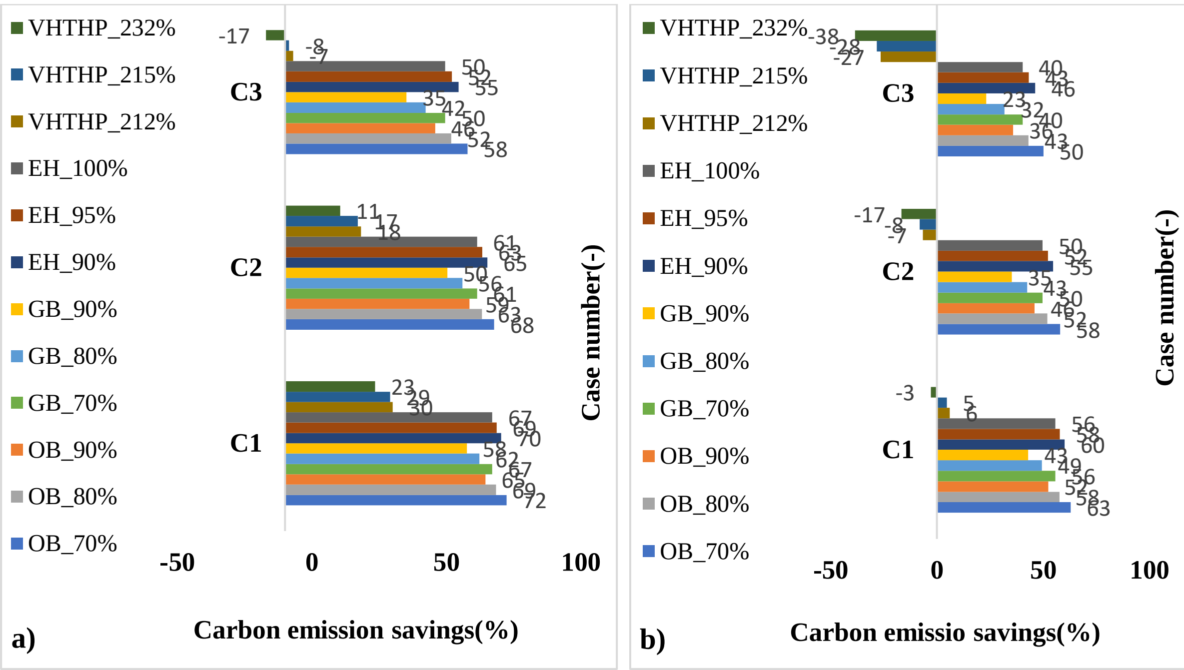


Figure 6.43 HP carbon emission savings in End-terrace with age period of 1950-1975, a) VSM, b) FSM

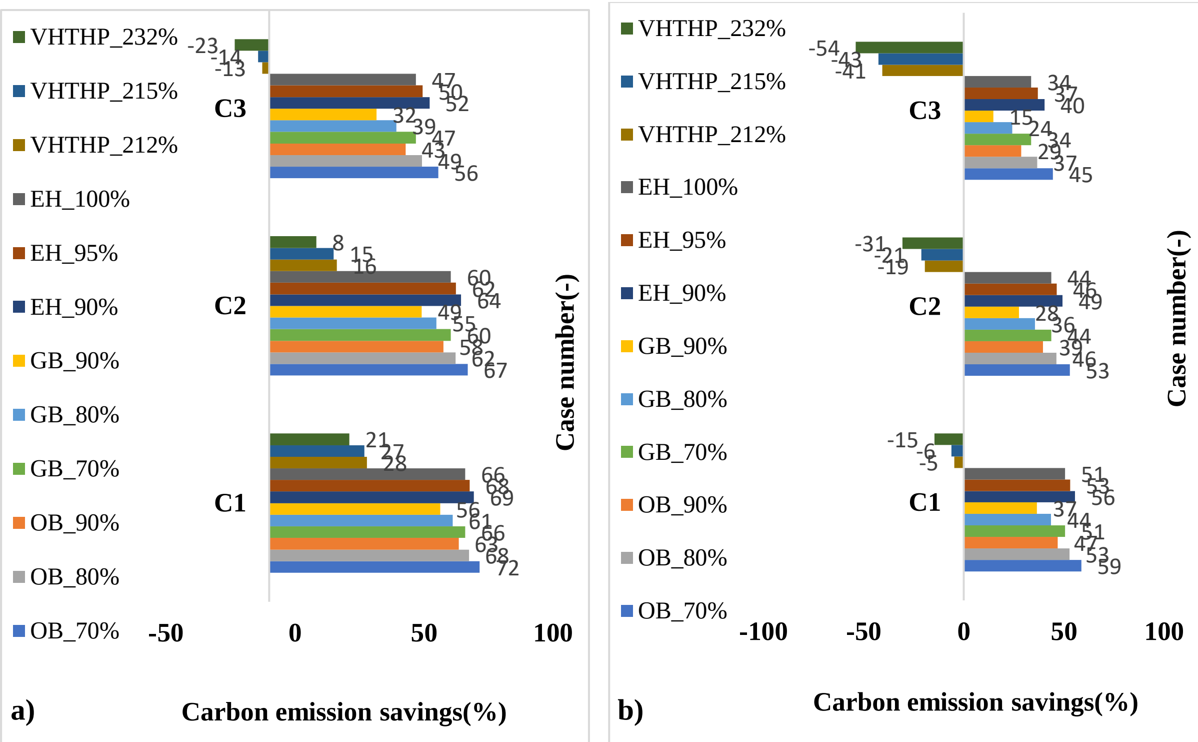


Figure 6.44 HP carbon emission savings in End-terrace with age period of 1976-1990, a) VSM, b) FSM

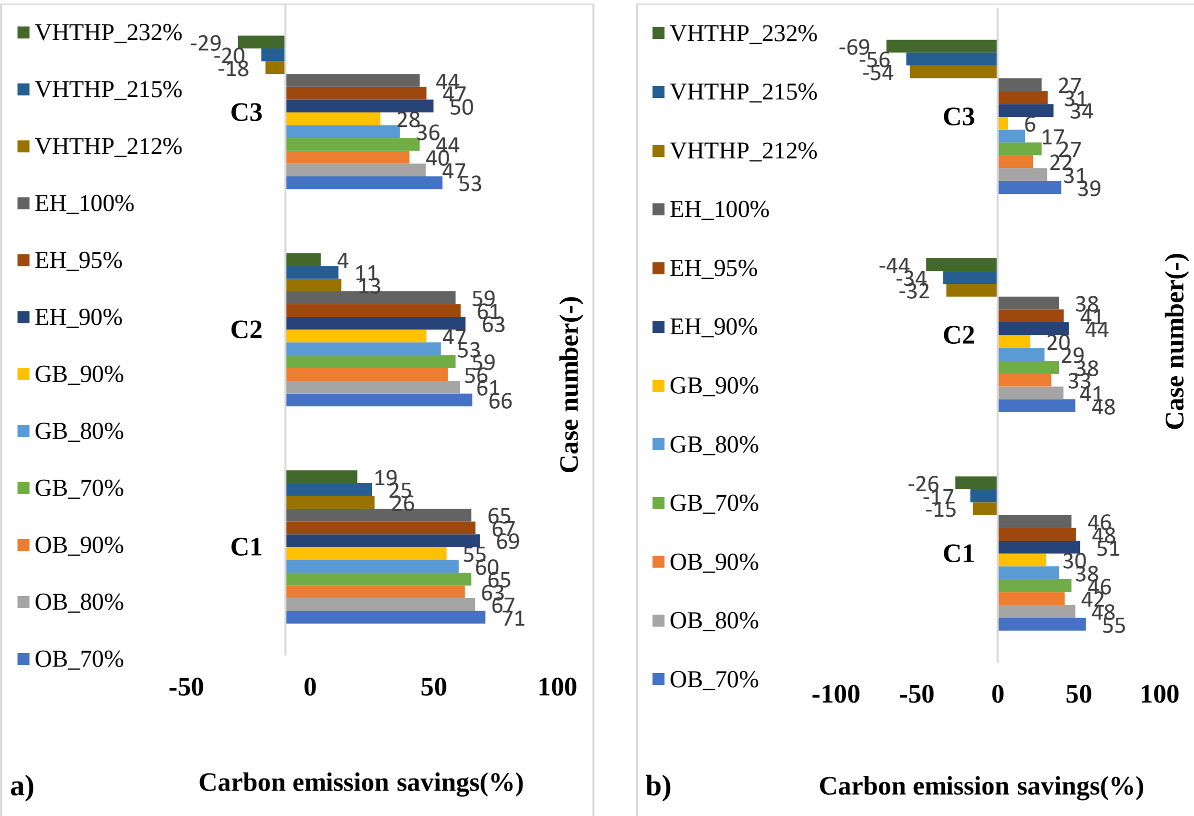


Figure 6.45 HP carbon emission savings in End-terrace with age period of 1991-2007, a) VSM, b) FSMHP cost savings with in two control modes (FSM vs. VSM)

6.6.1.7 HP cost savings with Semi-detached in two control modes (FSM vs. VSM)

The cost savings with Semi-detached property type at four age period in two control mode is shown in Figure 6.46-6.49. The annual running cost for all the technologies strongly depends on the age period of the property and the HP performance at part load operations. As the thermal load characteristics of the Semi-detached property types are very close to the End-terraced property types and hence the HP have shown similar cost savings in contrast to other heating technologies. Just to highlight the cost savings with End-terraced type(1900-1949) for the HP system(VSM) in C1, C2, and C3 against the EH(100% efficiency) was 68%, 64%, and 53%, while in case of Semi-detached property type this becomes 69%, 64%, and 53%. Other age periods have shown the same pattern.

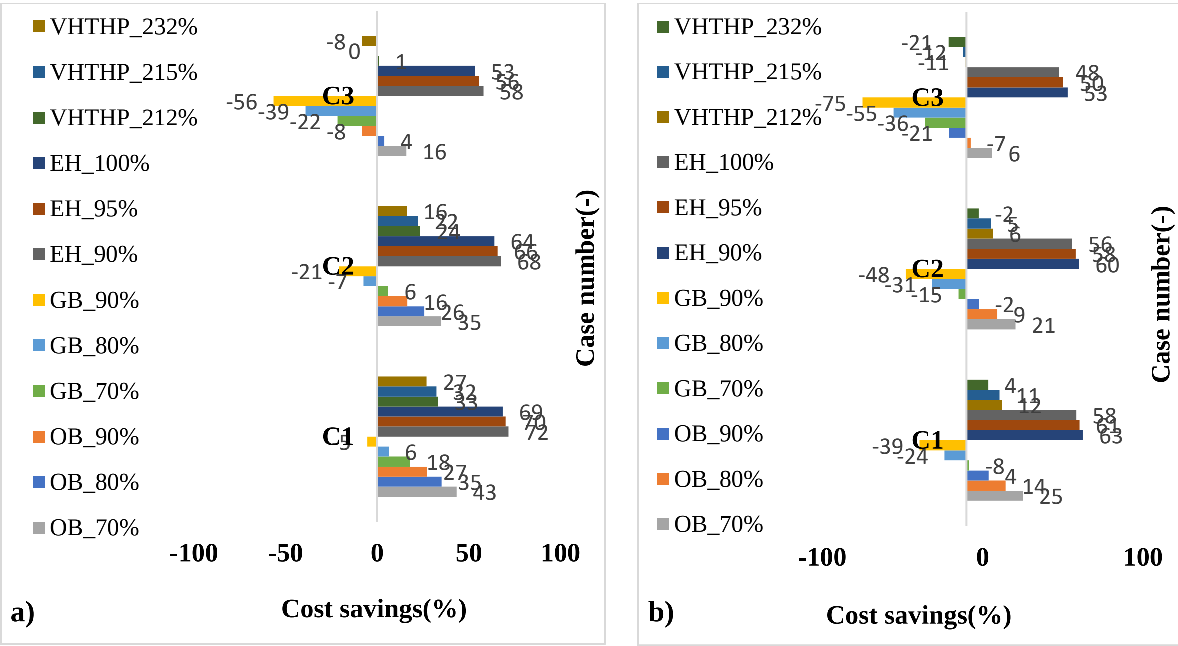


Figure 6.46 HP cost savings in Semi-detached with age period of 1900-1949, a) VSM, b) FSM

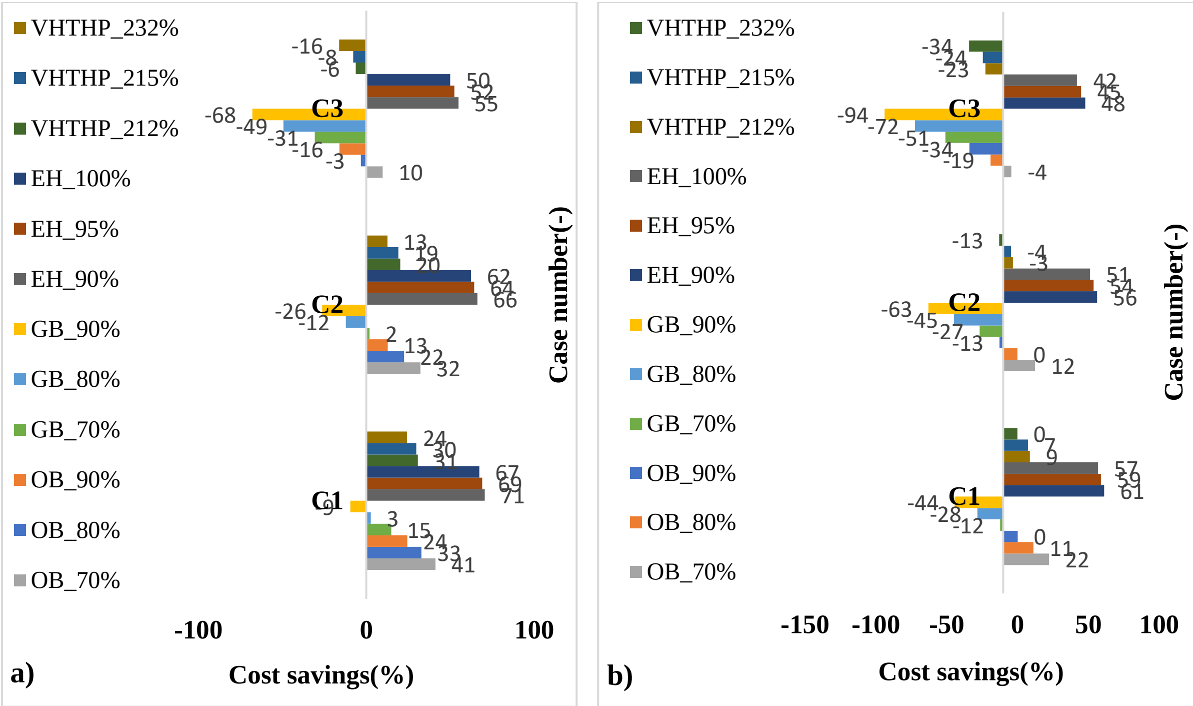


Figure 6.47 HP cost savings in Semi-detached with age period of 1950-1975, a) VSM, b) FSM

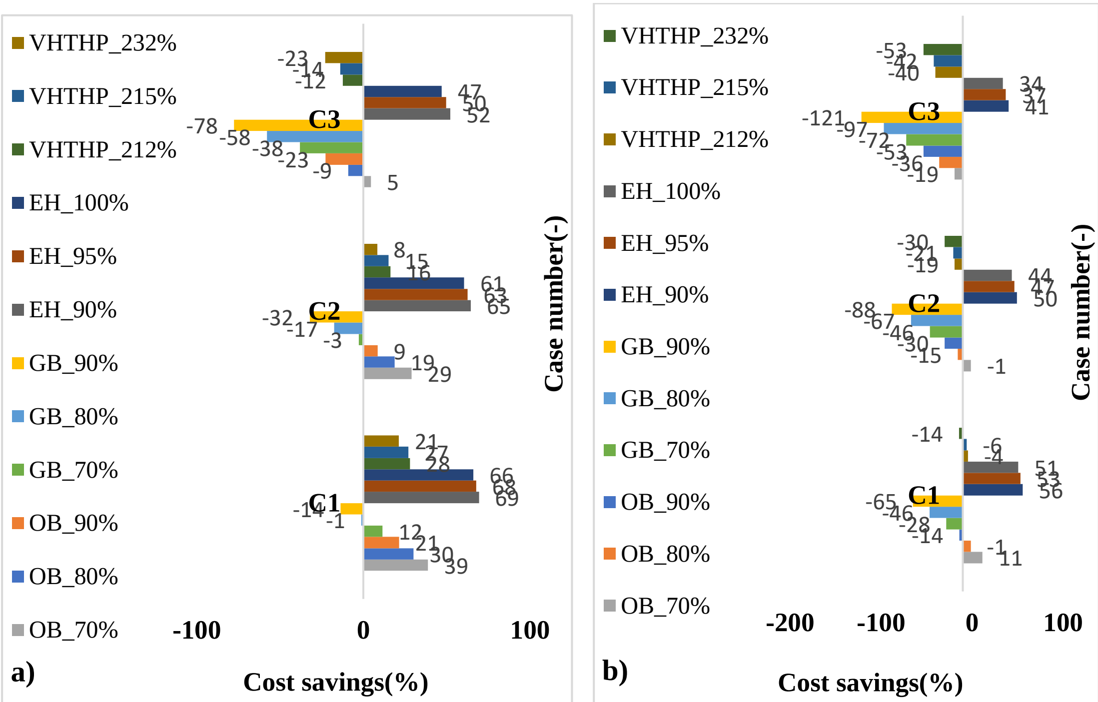


Figure 6.48 HP cost savings in Semi-detached with age period of 1976-1900, a) VSM, b) FSM

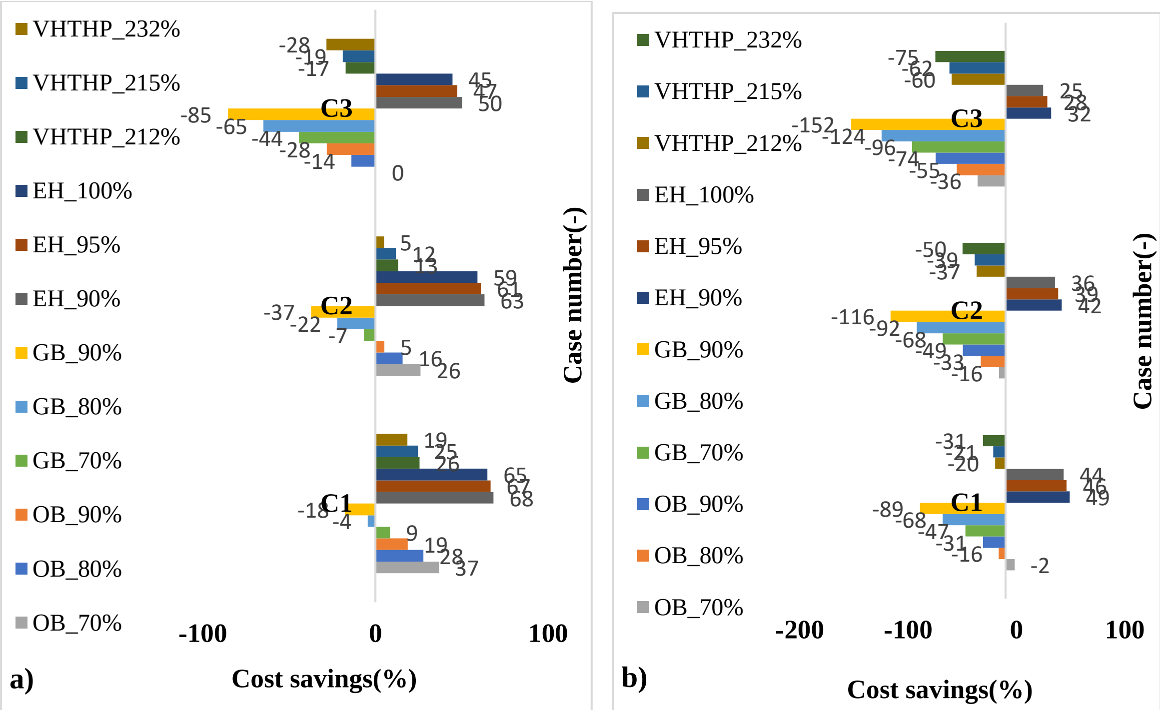


Figure 6.49 HP cost savings in Semi-detached with age period of 1991-2007, a) VSM, b) FSM

6.6.1.8 HP carbon emission savings with Semi-detached in two control modes (FSM vs. VSM)

The carbon emission savings with Semi-detached property type at four age period in two control mode is shown in Figure 6.50-6.53. The HP(VSM) have shown carbon emission savings in almost all cases compared to other heating technologies except the very high heat supply temperature ASHP at C3 with the reasons explained earlier explicitly. However, for the developed HP(FSM) have also shown some negative values for C2, and C1 against the very high heat supply temperature ASHP and this is due to the poor performance of the developed HP system in FSM. The percentage savings with the HP technologies are appropriate and verify the potential of the decarbonizations of the domestic heat sector by 2050. However, this needs additional initial capital cost and the installations of the low temperature heat supply distribution systems.

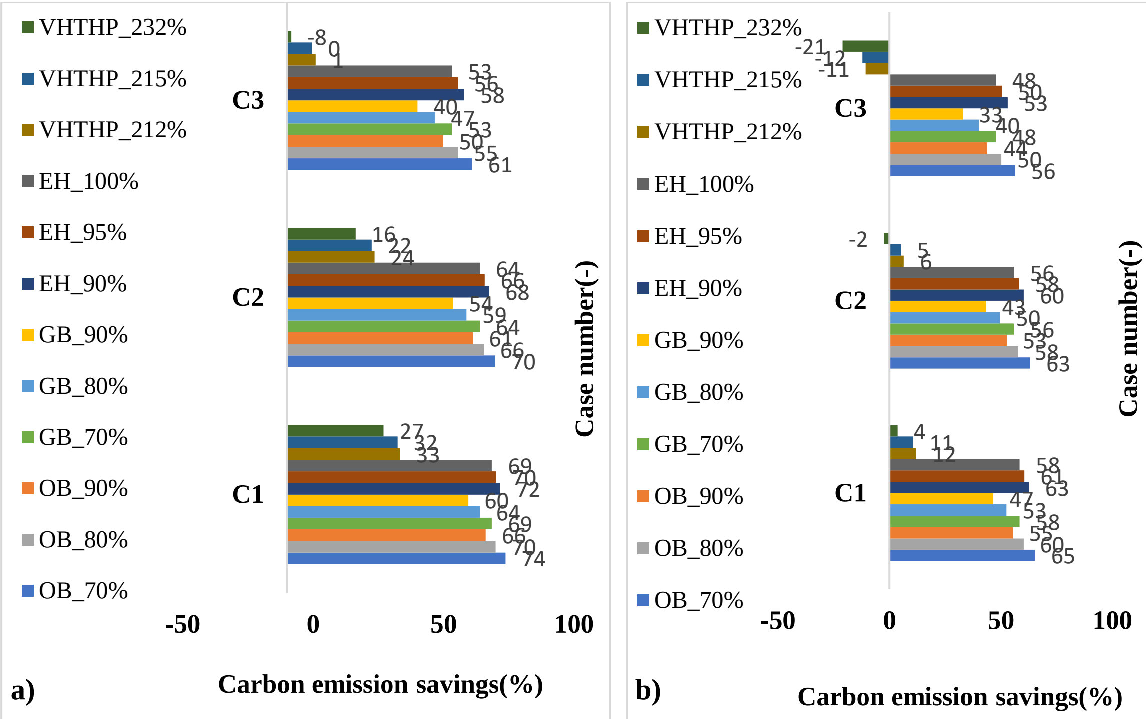


Figure 6.50 HP carbon emission savings in Semi-detached with age period of 1900-1949, a) VSM, b) FSM

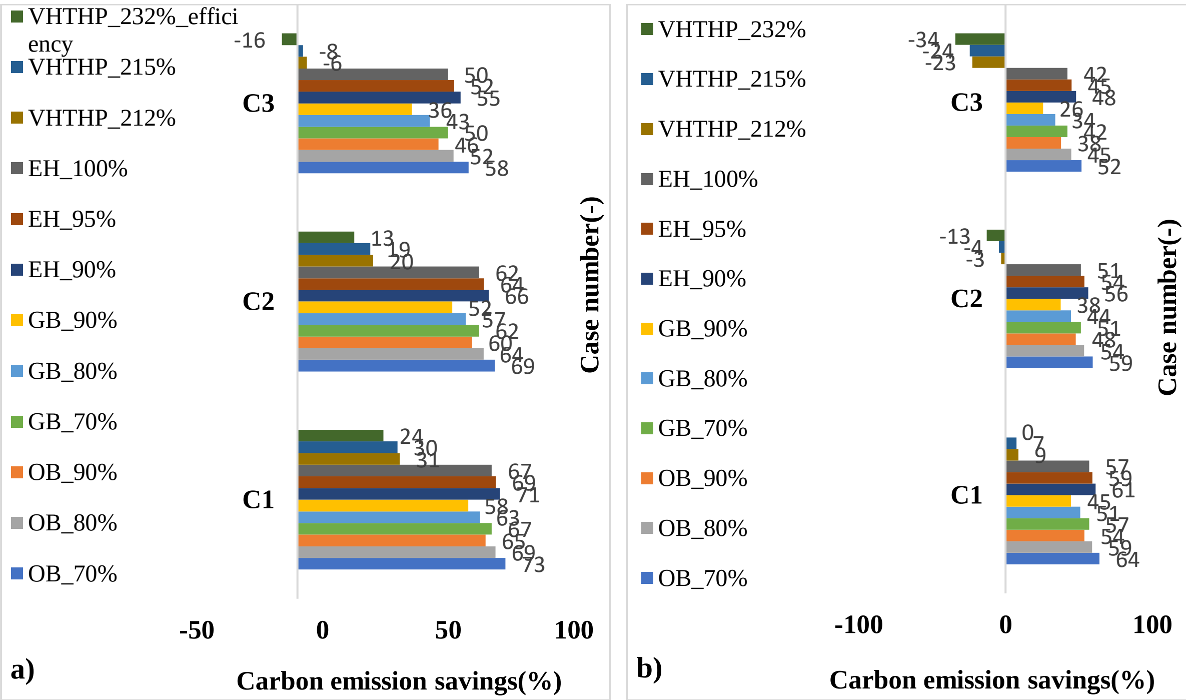


Figure 6.51 HP carbon emission savings in Semi-detached with age period of 1950-1975, a) VSM, b) FSM

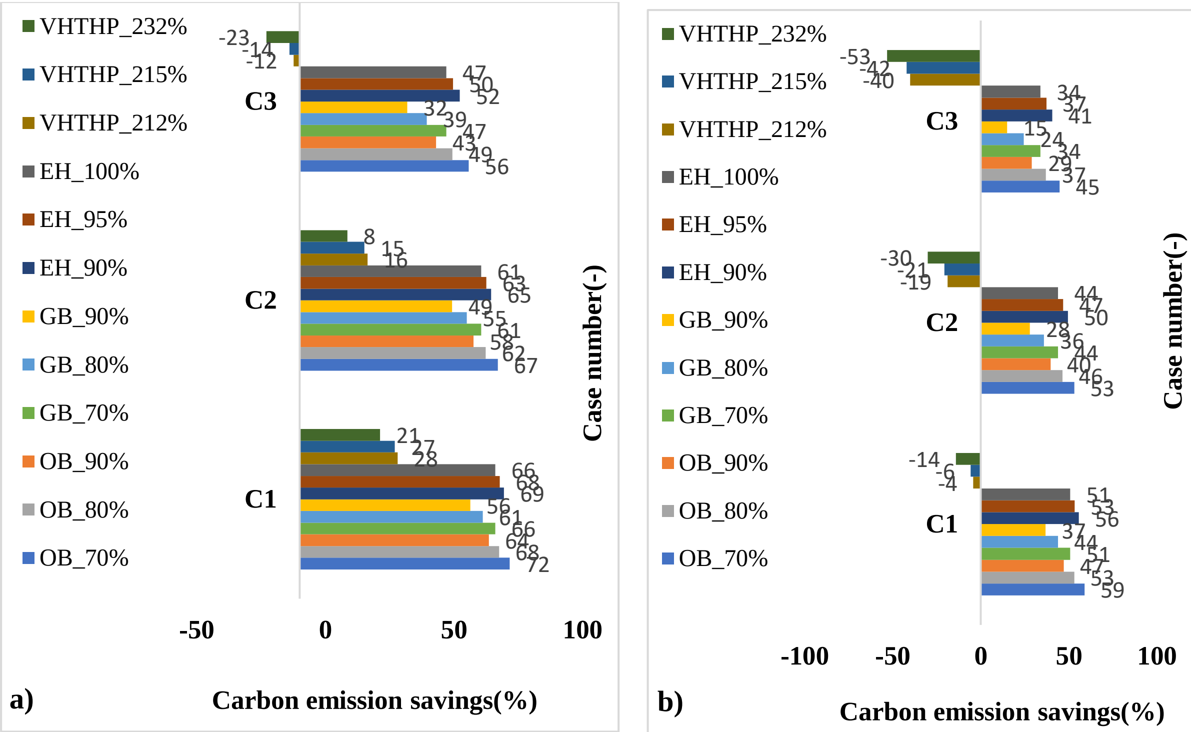


Figure 6.52 HP carbon emission savings in Semi-detached with age period of 1976-1990, a) VSM, b) FSM

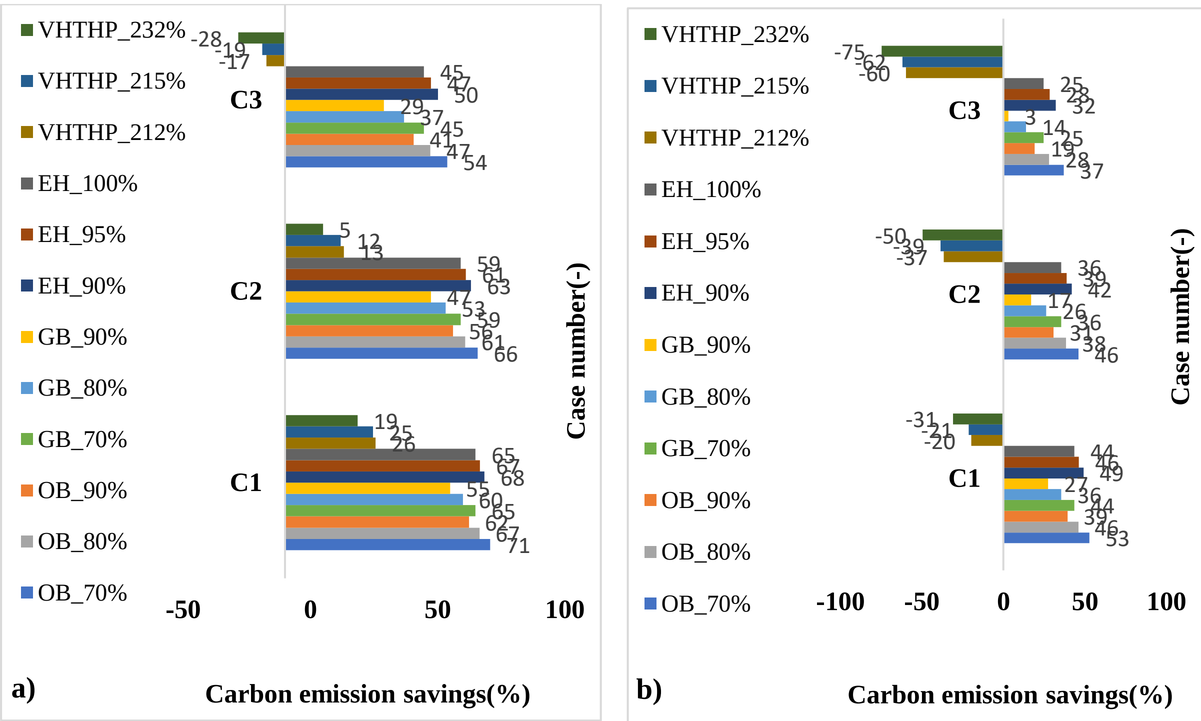


Figure 6.53 HP carbon emission savings in Semi-detached with age period of 1991-2007, a) VSM, b) FSM

6.6.1.9 HP cost savings with Detached in two control modes (FSM vs. VSM)

The cost savings with Detached property type at four age period in two control mode is shown in Figure 6.54-6.57. The annual running cost for all the technologies depends on the age period of the property. The HP(VSM) outperforms the oil boilers at all percentage efficiencies and age period with the only exception at C3, and the OB operating at 90% efficiency where the savings become negative (-6%, -6%, -10%, -11%) at the respective four age period. The reason is the poor performance and COP value for the heat pump at high heat supply temperature. However in contrast the 90%, and 80% OB operations have outperformed the HP in FSM at all cases (C1,C2, and C3).

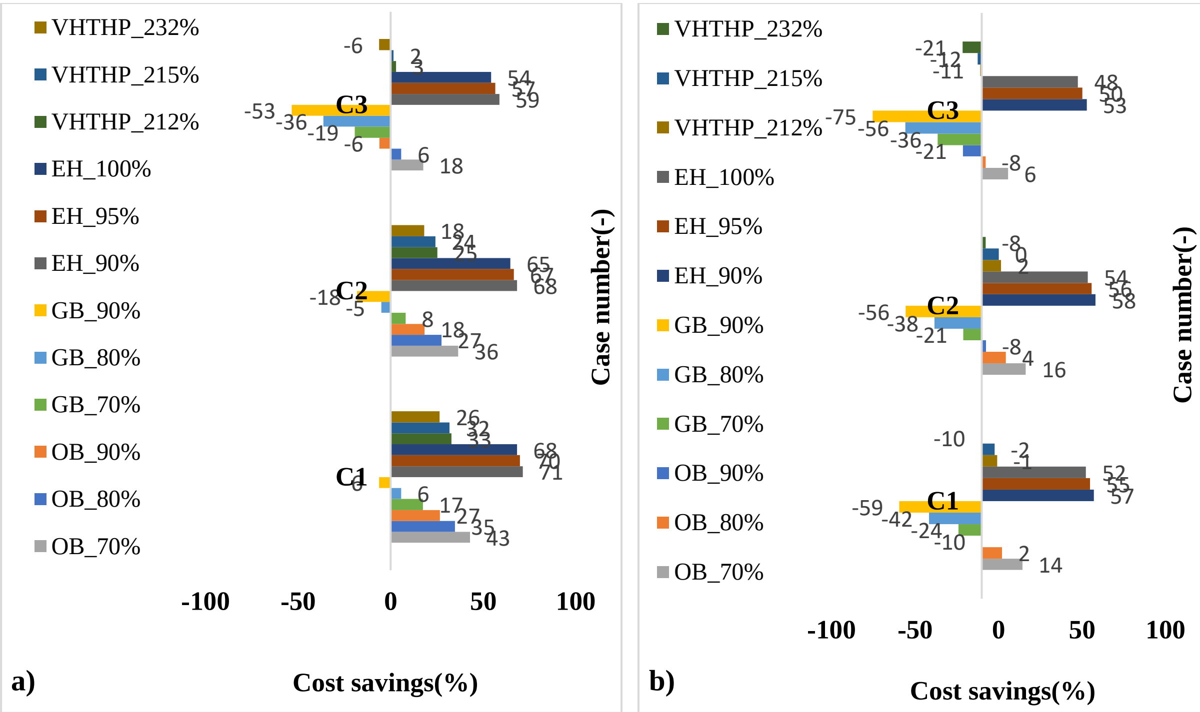


Figure 6.54 HP cost savings in Detached type with age period of 1900-1949, a) VSM, b) FSM

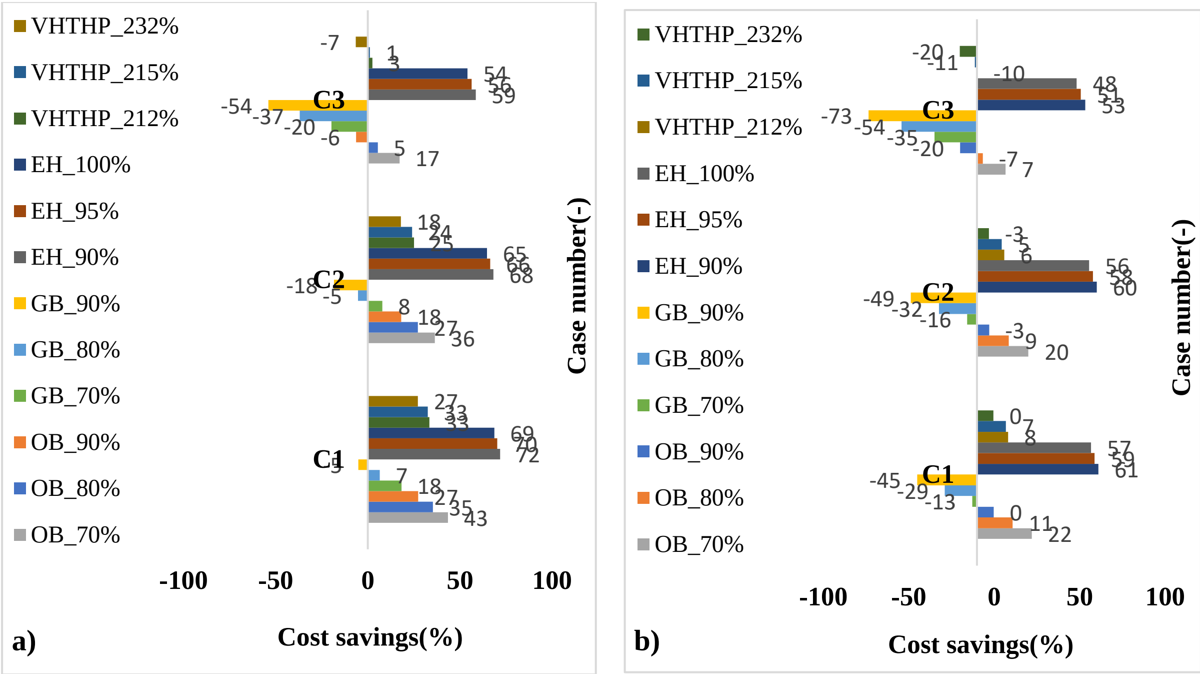


Figure 6.55.HP cost savings in Detached type with age period of 1950-1975, a) VSM, b) FSM

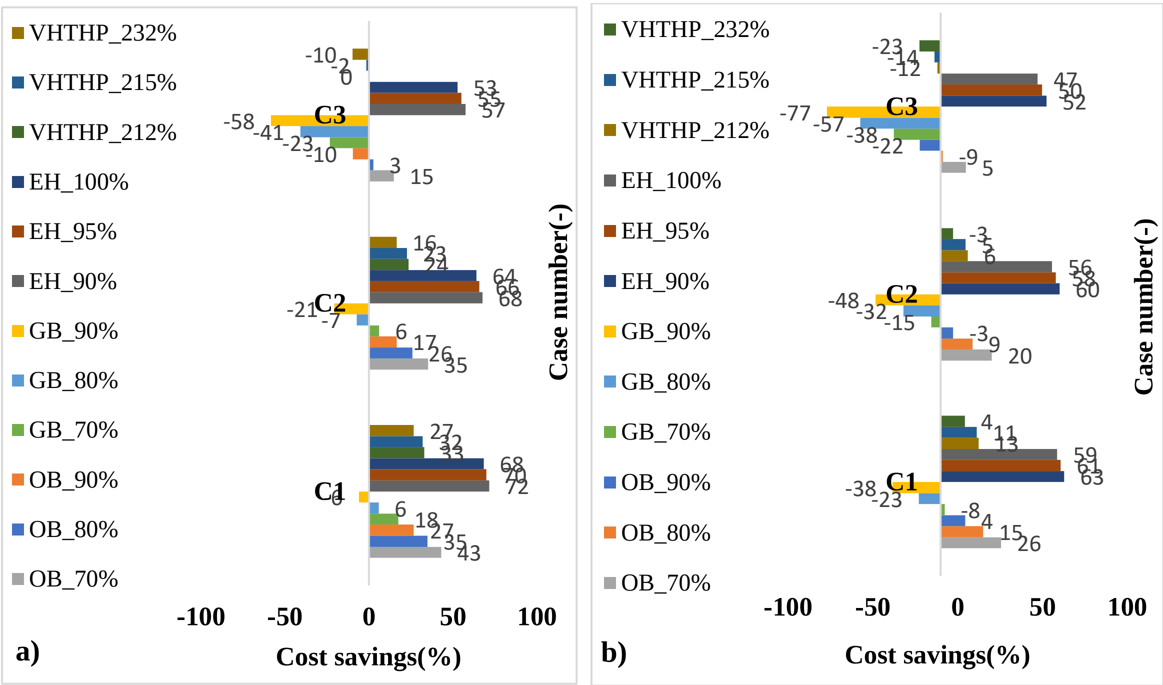


Figure 6.56 HP cost savings in Detached type with age period of 1976-1990, a) VSM, b) FSM

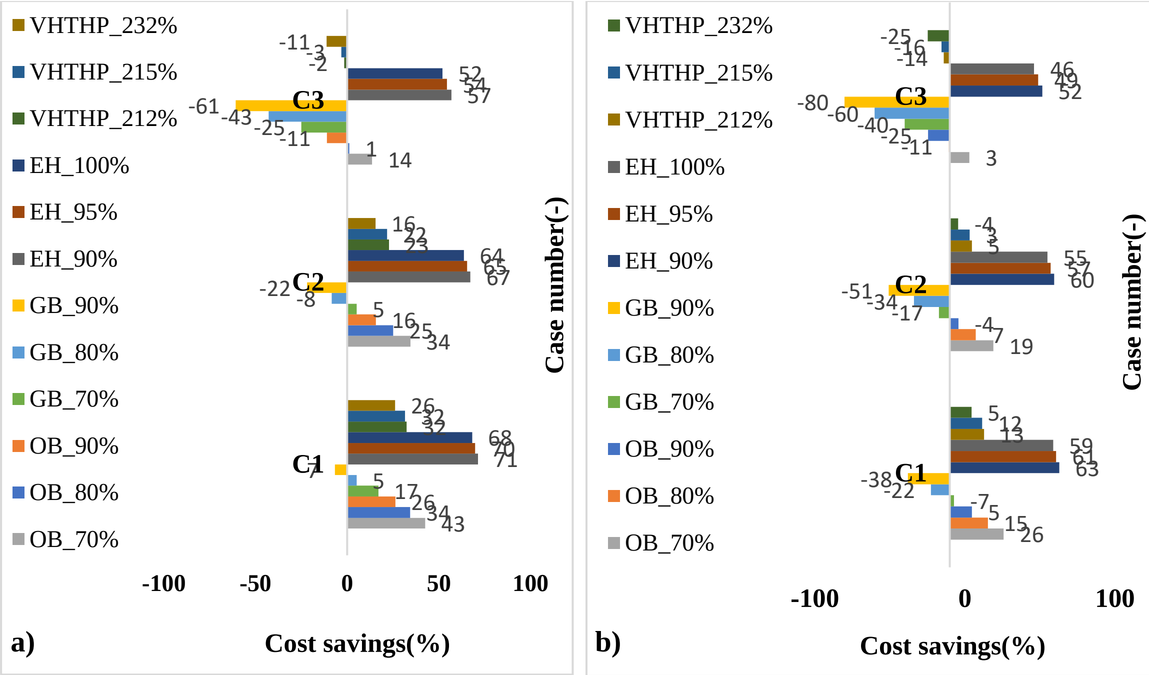


Figure 6.57 HP cost savings in Detached type with age period of 1991-2007, a) VSM, b) FSM

6.6.1.10 HP carbon emission savings with Detached in two control modes (FSM vs. VSM)

The carbon emission savings with Detached property type at four age period in two control mode is illustrated in Figure 6.58-6.61. The HP at all cases, both control modes, all four age periods considered was valuable in contrast to fossil fuel based (OB/GB) and EH option in terms of carbon emissions savings. However, a few exceptions to the developed HP system against the vary high heat supply temperature ASHP occurs at C3 in VSM and at C2, and C1 as well in FSM, where the savings become negative and are responsible for additional emissions.

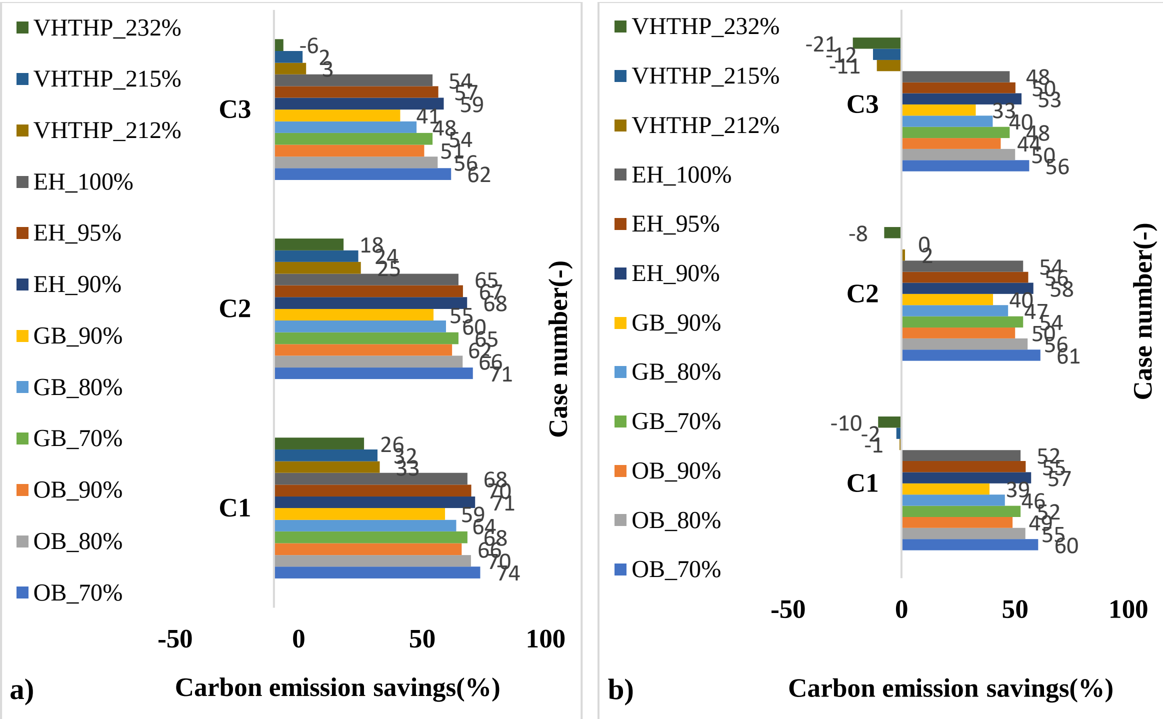


Figure 6.58 HP carbon emission savings in Detached type with age period of 1900-1949, a) VSM, b) FSM

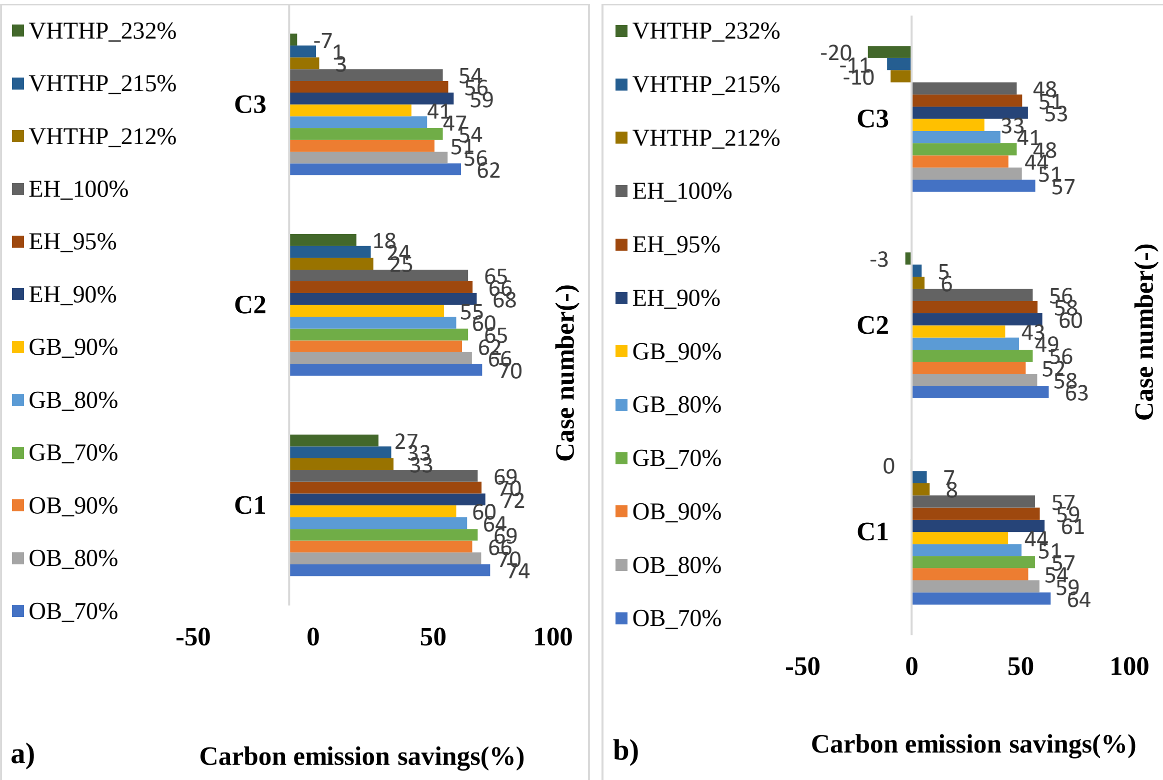


Figure 6.59 . HP carbon emission savings in Detached type with age period of 1950-1975, a) VSM, b) FSM

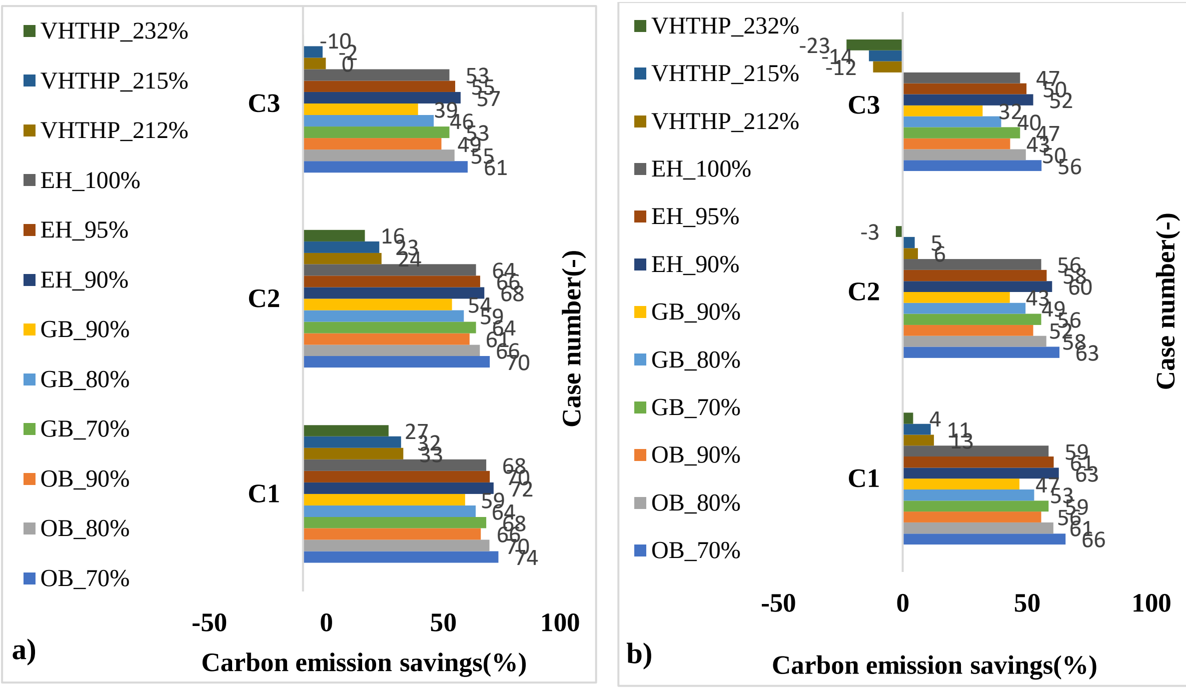


Figure 6.60 HP carbon emission savings in Detached type with age period of 1976-1990, a) VSM, b) FSM

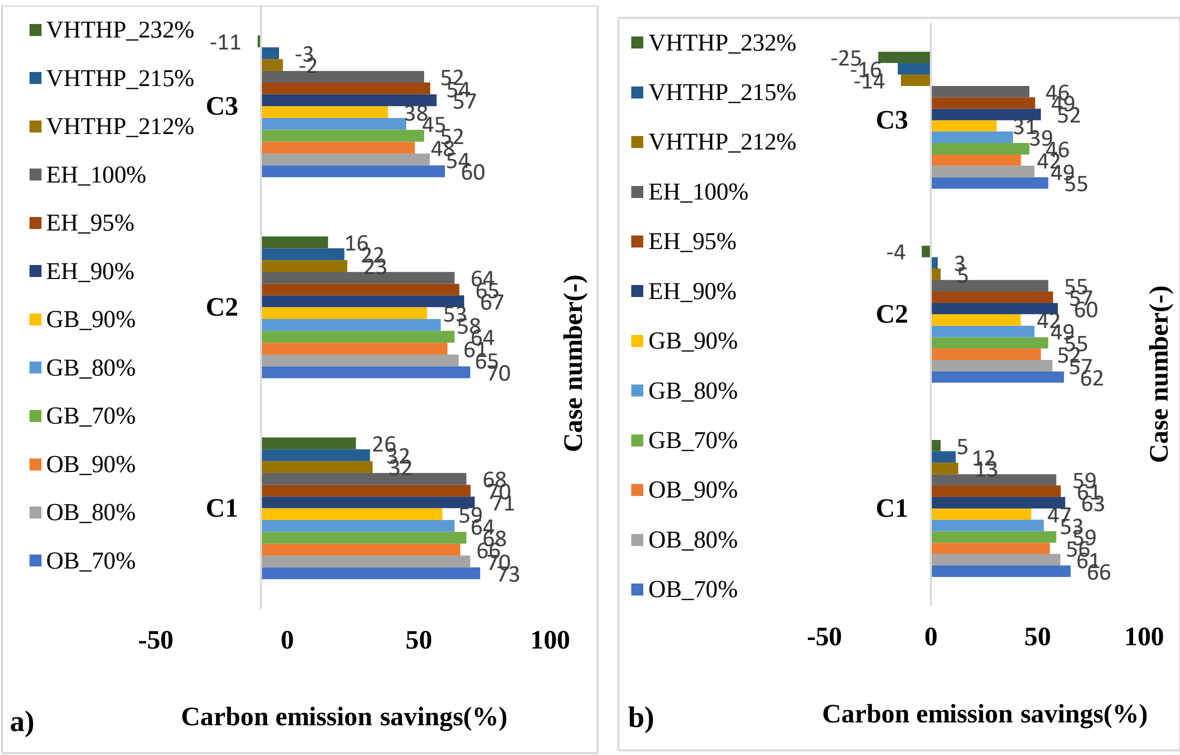


Figure 6.61 HP carbon emission savings in Detached type with age period of 1991-2007, a) VSM, b) FSM

6.6.2 Other climatic conditions retrofit Assessment (Valentia, Dublin, Aviemore)

The HP (VSM) retrofit assessment for other three climatic conditions of Valentia, Dublin, and Aviemore three age period, oil/gas boilers heating technologies, and five property types have been conducted in terms of annual running cost, carbon emissions. The oil/gas boilers have been compared for at three different efficiencies of 90%, 80%, and 70% representing the range of boilers, i.e., from modern condensing boilers to old heavy weight, for the five property types, i.e., Flats, Mid terrace, End terrace, Semi-detached and Detached and has been reported in Table 6.11-6.15. The heat demand for these property types, age period and climatic conditions have been mentioned in the performance tables in introduction. Among the three heating technologies considered, the climatic conditions of Valentia have the least cost, and carbon emission at all percentage efficiency with the Aviemore responsible for highest values for the respective property type, age period. The HP (C1) annual running cost for the Flat type (1900 -1949) is £757, £839, *& £*1012 for Valentia, Dublin, & Aviemore, with the corresponding values of *£* 394, *£*439, & *£*529 for third age period (1976-1990). Similar trend have been shown by other property types with Detached building type the highest annual running cost, for the HP (C1), age period (1900-1949) with the value of *£*1735, *£*1985, *£*2515, for three climatic conditions with the corresponding values of *£*1296, 1453, and 1785 for third age period (1976-1990). The annual running cost, carbon emissions for other heating technologies of oil/gas boilers at 90%, 80%, 70%, three climatic conditions and five property types have been shown in Table 6.11-6.15. The variation in running cost and carbon emissions due to climatic conditions for the individual property, is because of the variation in annual heat demand for oil/gas boilers as the efficiency is fixed but in case of HP the additional impact of climatic variation is due to HP annual COP variation, as well. These calculations are based on the methodology explained earlier by the authors own calculations with it base of experimental results from the HP system.

Table 6.11: HP (VSM only) retrofit assessment for Flat property type and three age period in others climatic conditions

	Valentia			Dublin			Aviemore		
Variable speed mode (VSM) (1900-1950)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	4325	5133	6876	4795	5640	7459	5781	6708	8688
Annual running cost (£)	757	898	1203	839	987	1305	1012	1174	1520
Annual C02 emissions(kg)	1254	1489	1994	1390	1636	2163	1677	1945	2520
2) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	14782	16630	19005	15674	17633	20152	17716	19930	22778
Annual running cost (£)	1005	1131	1292	1066	1199	1370	1205	1355	1549
Annual C02 emissions(kg)	3592	4041	4618	3809	4285	4897	4305	4843	5535
3) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	14782	16630	19005	15674	17633	20152	17716	19930	22778
Annual running cost (£)	695	782	893	737	829	947	833	937	1071
C02 emissions(kg)	3001	3376	3858	3182	3579	4091	3596	4046	4624
1951-1975									
1) Variable speed mode (VSM)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	3323	3942	5350	3688	4331	5775	4440	5128	6658
Annual running cost (£)	582	690	936	645	758	1011	777	897	1165
Annual C02 emissions(kg)	964	1143	1552	1069	1256	1675	1287	1487	1931
2) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	10968	12339	14102	11630	13084	14953	13145	14788	16901
Annual running cost(£)	746	839	959	791	890	1017	894	1006	1149
Annual C02 emissions(kg)	2665	2998	3427	2826	3179	3634	3194	3594	4107
3) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	10968	12339	14102	11630	13084	14953	13145	14788	16901
Annual running cost (£)	516	580	663	547	615	703	618	695	794
C02 emission(kg)	2227	2505	2863	2361	2656	3035	2668	3002	3431
1976-1990									
1) Variable speed mode (VSM)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	2250	2671	3717	2510	2946	4009	3025	3484	4561
Annual running cost (£)	394	467	650	439	516	702	529	610	798
Annual C02 emissions(kg)	653	775	1078	728	854	1163	877	1010	1323
2) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	7332	8248	9427	7774	8746	9995	8787	9885	11298
Annual running cost (£)	499	561	641	529	595	680	598	672	768
Annual C02 emissions(kg)	1782	2004	2291	1889	2125	2429	2135	2402	2745
3) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used(KWh)	7332	8248	9427	7774	8746	9995	8787	9885	11298
Annual running cost (£)	345	388	443	365	411	470	413	465	531
C02 emissions(kg)	1488	1674	1914	1578	1775	2029	1784	2007	2293

Table 6.12: HP (VSM only) retrofit assessment for Mid-terraced property type and three age period in others climatic conditions

	Valentia			Dublin			Aviemore		
1) Variable speed mode (VSM) (1900-1949)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	5850	6918	9158	6511	7634	10012	7901	9134	11794
Annual running cost (£)	1024	1211	1603	1139	1336	1752	1383	1598	2064
Annual C02 emissions(kg)	1697	2006	2656	1888	2214	2903	2291	2649	3420
2) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	21227	23880	27292	22507	25321	28938	25440	28620	32709
Annual running cost (£)	1443	1624	1856	1531	1722	1968	1730	1946	2224
Annual C02 emissions(kg)	5158	5803	6632	5469	6153	7032	6182	6955	7948
3) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	21227	23880	27292	22507	25321	28938	25440	28620	32709
Annual running cost (£)	998	1122	1283	1058	1190	1360	1196	1345	1537
C02 emissions(kg)	4309	4848	5540	4569	5140	5874	5164	5810	6640
1950-1975									
1) Variable speed mode (VSM)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	4871	5777	7710	5404	6356	8385	6527	7574	9809
Annual running cost (£)	852	1011	1349	946	1112	1467	1142	1325	1717
Annual C02 emissions(kg)	1412	1675	2236	1567	1843	2432	1893	2196	2845
2) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	16999	19124	21856	18025	20278	23175	20373	22920	26194
Annual running cost (£)	1156	1300	1486	1226	1379	1576	1385	1559	1781
Annual C02 emissions(kg)	4131	4647	5311	4380	4928	5631	4951	5570	6365
3) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	16999	19124	21856	18025	20278	23175	20373	22920	26194
Annual running cost (£)	799	899	1027	847	953	1089	958	1077	1231
C02 emissions(kg)	3451	3882	4437	3659	4116	4704	4136	4653	5317
1976-1990									
1) Variable speed mode (VSM)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual heat output (KWh)	11681	11681	11681	12385	12385	12385	13999	13999	13999
Annual Input power (KWh)	3870	4582	6360	4290	5030	6850	5165	5966	7891
Annual COP	3.02	2.55	1.84	2.89	2.46	1.81	2.71	2.35	1.77
Annual running cost (£)	677	802	1113	751	880	1199	904	1044	1381
Annual C02 emissions(kg)	1122	1329	1845	1244	1459	1986	1498	1730	2288
2) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	12979	14601	16687	13762	15482	17693	15555	17499	19999
Annual running cost (£)	883	993	1135	936	1053	1203	1058	1190	1360
Annual C02 emissions(kg)	3154	3548	4055	3344	3762	4299	3780	4252	4860
3) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	12979	14601	16687	13762	15482	17693	15555	17499	19999
Annual running cost (£)	610	686	784	647	728	832	731	822	940
C02 emissions(kg)	2635	2964	3387	2794	3143	3592	3158	3552	4060

Table 6.13: HP (VSM only) retrofit assessment for End-terraced property type and three age period in others climatic conditions

		Valentia			Dublin			Aviemore	
1) Variable speed mode (VSM) (1900-1949)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	7435	8697	11404	8338	9653	12564	10248	11644	14960
Annual running cost (£)	1301	1522	1996	1459	1689	2199	1793	2038	2618
Annual C02 emissions(kg)	2156	2522	3307	2418	2799	3643	2972	3377	4338
3) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	28263	31796	36338	29968	33714	38530	33873	38107	43551
Annual running cost (£)	1922	2162	2471	2038	2293	2620	2303	2591	2961
Annual C02 emissions(kg)	6868	7726	8830	7282	8193	9363	8231	9260	10583
4) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	28263	31796	36338	29968	33714	38530	33873	38107	43551
Annual running cost (£)	1328	1494	1708	1409	1585	1811	1592	1791	2047
C02 emissions(kg)	5737	6455	7377	6084	6844	7822	6876	7736	8841
1950-1975									
Developed HP		**Valentia**			**Dublin**			**Aviemore**	
1) Variable speed mode (VSM)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	5656	6694	8880	6290	7382	9696	7624	8823	11405
Annual running cost (£)	990	1172	1554	1101	1292	1697	1334	1544	1996
Annual C02 emissions(kg)	1640	1941	2575	1824	2141	2812	2211	2559	3307
3) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	20370	22916	26189	21598	24298	27769	24413	27464	31388
Annual running cost (£)	1385	1558	1781	1469	1652	1888	1660	1868	2134
Annual C02 emissions(kg)	4950	5569	6364	5248	5904	6748	5932	6674	7627
4) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	20370	22916	26189	21598	24298	27769	24413	27464	31388
Annual running cost (£)	957	1077	1231	1015	1142	1305	1147	1291	1475
C02 emissions(kg)	4135	4652	5316	4384	4933	5637	4956	5575	6372
1976-1990									
Developed HP		**Valentia**			**Dublin**			**Aviemore**	
1) Variable speed mode (VSM)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	4812	5662	7781	5341	6230	8425	6449	7421	9799
Annual running cost (£)	842	991	1362	935	1090	1474	1129	1299	1715
Annual C02 emissions(kg)	1395	1642	2256	1549	1807	2443	1870	2152	2842
3) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	16763	18858	21552	17774	19996	22852	20090	22601	25830
Annual running cost (£)	1140	1282	1466	1209	1360	1554	1366	1537	1756
Annual C02 emissions(kg)	4073	4583	5237	4319	4859	5553	4882	5492	6277
4) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	16763	18858	21552	17774	19996	22852	20090	22601	25830
Annual running cost (£)	788	886	1013	835	940	1074	944	1062	1214
C02 emissions(kg)	3403	3828	4375	3608	4059	4639	4078	4588	5243

Table 6.14: HP (VSM only) retrofit assessment for Semi-detached property type and three age period in others climatic conditions

	Valentia			Dublin			Aviemore		
1) Variable speed mode (VSM)(1900-1949)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	7875	9174	11989	8854	10201	13244	10933	12341	15834
Annual running cost (£)	1378	1605	2098	1550	1785	2318	1913	2160	2771
Annual C02 emissions(kg)	2284	2660	3477	2568	2958	3841	3170	3579	4592
2) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	30185	33958	38809	32006	36006	41150	36176	40698	46512
Annual running cost (£)	2053	2309	2639	2176	2448	2798	2460	2767	3163
Annual C02 emissions(kg)	7335	8252	9431	7777	8750	9999	8791	9890	11302
3) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	30185	33958	38809	32006	36006	41150	36176	40698	46512
Annual running cost (£)	1419	1596	1824	1504	1692	1934	1700	1913	2186
C02 emission(kg)	6128	6893	7878	6497	7309	8353	7344	8262	9442
1950-1975									
Developed HP	**Valentia**			**Dublin**			**Aviemore**		
1) Variable speed mode (VSM)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	5966	6955	9424	6643	7683	10278	8065	9207	12069
Annual running cost (£)	1044	1217	1649	1163	1344	1799	1411	1611	2112
Annual C02 emissions(kg)	1730	2017	2733	1926	2228	2980	2339	2670	3500
3) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	21730	24446	27938	23040	25920	29623	26042	29298	33483
Annual running cost (£)	1478	1662	1900	1567	1763	2014	1771	1992	2277
Annual C02 emissions(kg)	5280	5940	6789	5599	6299	7198	6328	7119	8136
4) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	21730	24446	27938	23040	25920	29623	26042	29298	33483
Annual running cost (£)	1021	1149	1313	1083	1218	1392	1224	1377	1574
C02 emissions(kg)	4411	4962	5671	4677	5262	6014	5287	5947	6797
1976-1990									
Developed HP	**Valentia**			**Dublin**			**Aviemore**		
1) Variable speed mode (VSM)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	4837	5689	7815	5367	6258	8461	6482	7456	9843
Annual running cost (£)	846	996	1368	939	1095	1481	1134	1305	1723
Annual C02 emissions(kg)	1403	1650	2266	1556	1815	2454	1880	2162	2854
3) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	16881	18991	21704	17899	20137	23013	20232	22761	26012
Annual running cost (£)	1148	1291	1476	1217	1369	1565	1376	1548	1769
Annual C02 emissions(kg)	4102	4615	5274	4350	4893	5592	4916	5531	6321
4) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	16881	18991	21704	17899	20137	23013	20232	22761	26012
Annual running cost (£)	793	893	1020	841	946	1082	951	1070	1223
C02 emissions(kg)	3427	3855	4406	3634	4088	4672	4107	4620	5280

Table 6.15: HP (VSM only) retrofit assessment for Detached property type and three age period in others climatic conditions

	Valentia			Dublin			Aviemore		
1) Variable speed mode (VSM) (1900-1949)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	9917	11289	14674	11341	12706	16504	14370	15685	20283
Annual running cost (£)	1735	1976	2568	1985	2224	2888	2515	2745	3549
Annual C02 emissions(kg)	2876	3274	4255	3289	3685	4786	4167	4549	5882
2) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	38655	43487	49699	40987	46110	52697	46327	52118	59564
Annual running cost (£)	2629	2957	3380	2787	3135	3583	3150	3544	4050
Annual C02 emissions(kg)	9393	10567	12077	9960	11205	12805	11258	12665	14474
3) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	38655	43487	49699	40987	46110	52697	46327	52118	59564
Annual running cost (£)	1817	2044	2336	1926	2167	2477	2177	2450	2799
C02 emissions(kg)	7847	8828	10089	8320	9360	10698	9404	10580	12091
1950-1975									
1) Variable speed mode (VSM)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	8506	9704	12783	9613	10872	14195	11971	13314	17106
Annual running cost (£)	1489	1698	2237	1682	1903	2484	2095	2330	2993
Annual C02 emissions(kg)	2467	2814	3707	2788	3153	4117	3472	3861	4961
2) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	32934	37051	42344	34921	39286	44898	39471	44405	50749
Annual running cost (£)	2240	2519	2879	2375	2671	3053	2684	3020	3451
Annual C02 emissions(kg)	8003	9003	10290	8486	9547	10910	9591	10790	12332
3) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	32934	37051	42344	34921	39286	44898	39471	44405	50749
Annual running cost (£)	1548	1741	1990	1641	1846	2110	1855	2087	2385
C02 emissions(kg)	6686	7521	8596	7089	7975	9114	8013	9014	10302
1976-1990									
1) Variable speed mode (VSM)									
Case number (-)	1	2	3	1	2	3	1	2	3
Annual Input power (KWh)	7404	8527	11361	8301	9485	12510	10199	11482	14884
Annual running cost (£)	1296	1492	1988	1453	1660	2189	1785	2009	2605
Annual C02 emissions(kg)	2147	2473	3295	2407	2751	3628	2958	3330	4316
3) Oil Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual oil used (KWh)	28115	31630	36148	29811	33538	38329	33696	37908	43323
Annual running cost (£)	1912	2151	2458	2027	2281	2606	2291	2578	2946
Annual C02 emissions(kg)	6832	7686	8784	7244	8150	9314	8188	9212	10528
4) Gas Boilers									
Efficiency (%)	90%	80%	70%	90%	80%	70%	90%	80%	70%
Annual gas used (KWh)	28115	31630	36148	29811	33538	38329	33696	37908	43323
Annual running cost (£)	1321	1487	1699	1401	1576	1801	1584	1782	2036
C02 emission(kg)	5707	6421	7338	6052	6808	7781	6840	7695	8795

6.6.2.1 Retrofit assessment in other locations for all properties (Valentia, Dublin, Aviemore)

The cost and carbon emissions savings (%) when the oil/gas boiler is retrofitted with the developed HP for three age period, five property types (illustrated in **Appendix B**). The positive values indicate that developed HP is beneficial in terms of cost and/or carbon emissions, while the negative values represent that the HP is more expensive in terms of cost, and/or carbon emissions compared to oil/gas boilers. The factor of age period, property types three cases, control mode studied for HP and different % efficiency for oil/gas boiler have been investigated in the earlier sections with fixed climatic conditions of Belfast. The variation in cost and carbon emission saving with climatic conditions for the developed HP when retrofitted instead of GB/OB have been investigated for five property types, three age period. The cost and carbon emission savings for the individual property have been divided into two sections (A & B) respectively. The HP running cost increases at higher HDDs of Aviemore compared to lower HDDs value of Valentia, due to poor performance, and higher energy consumption. The cost and carbon emission savings (%) reduces for fixed age period, case considered in all property types as the climatic conditions changes from Valentia to Aviemore. For example, Flats type, age period (1900-1949) with GB (90%) retrofitting with the HP(C1) the cost savings values are -9%, -14%, and -22% and carbon emission savings are 58%, 56%, 53% for Valentia, Dublin, and Aviemore. The cost savings become 3%, -1%, and -8% at 80% of GB. The OB even at highest efficiency of 90% have shown positive values at all three climatic conditions (Valentia, Dublin, and Aviemore) with value of 25%, 21 %, 16%. The Detached type, age period (1900-1949) with GB (90%) retrofitted with the developed HP (C1) cost savings are 4%, -3%, and -15% for the Valentia, Dublin, and Aviemore climatic conditions and become 15%, 8%, and -3% at 80% efficiency. The carbon emission savings are always positive in all cases when the OB/GB is retrofitted with the developed HP in all age property type. Similar trends can be seen for other property types and age period.

6.6.2.2 Payback period analysis for the developed HP with other heating technologies

The payback period analysis has been performed in case of running cost savings for the VSM and C1 only with other heating technologies and the results have been presented in Table 6.16. The total cost for the developed HP is assumed to be 13000 pounds including the HP installation and heating distribution system while the upfront cost of 6000 pounds is considered for comparing with high temperature heat pump (HP). The payback period for developed HP have been shown only in case of VSM and C1 retrofitted with 80% and 70%

efficiency oil and gas boilers as there is either no operational savings or very small with 90% oil and gas boilers. The minimum payback period is 23 years when oil boilers is at 70% efficiency for oil and gas boilers. The minimum payback period in case of electric heater is 12 years and become 14 years for high temperature HP. The payback period was calculated by dividing the upfront cost for the HP system by annual operating cost savings.

Table 6.16: Payback period analysis (VSM-C1 considered only)

Locations	**Belfast**	**Valentia**	**Dublin**	**Aviemore**
Oil Boiler-80%	29	29	29	29
Oil Boiler-70%	23	24	24	23
Gas Boiler-80%	176	92	135	919
Gas Boiler-70%	57	49	54	71
Electric heater-100%	15	16	15	14
Electric heater-95%	14	16	15	13
Electric heater-90%	13	14	14	12
HTHP--232%	17	16	17	19
HTHP-215%	14	14	14	15
HTHP-212%	14	14	14	14

6.7 Chapter summary

In literature, ASHP system experimental development with variable speed mode of control in comparison to fixed speed mode of control at low to medium and high heat supply temperature in the context of Ireland was found with very limited number of studies, but without considering retrofit assessment instead of oil/gas boilers, electric heating option, and very high temperature ASHP. The HP designed, developed, controlled, and tested at constant heat load, simulating the real domestic heat demand under the controlled laboratory conditions and numerical modelling is utilized for the analysis purposes. The HP performance, energy demand, carbon emissions, and cost varies significantly due to changing heat supply temperature (35 ^{o}C, 45 ^{o}C, and 55 ^{o}C), and control mode. The oil & gas boilers considered in the analysis ranges from conventional to highly efficient type and a comparative evaluation is performed in terms of annual running cost, energy consumptions, and carbon emissions with the developed HP. Also, a comparative study with the existing retrofitted very high temperature ASHP inside the test house is conducted. The developed HP at 55 ^{o}C could not defeat the very high temperature HP system (75 ^{o}C supply temperature) in performance and cost savings but become attractive at low supply temperature (55 ^{o}C). The HP performance for other property types including flats, mid terraced, end terraced, semi-detached, and detached with different insulation characteristics have also been discussed. The question of best operating point for the variable speed compressor-based heat pump technology at lower to medium and high heat supply temperature for domestic retrofit have been investigated with the design consideration for the Northern Ireland housing stock. The aim was efficiency improvement potential, using the control techniques with heat supply temperature. The impact on the carbon emissions and cost savings were studied based on the real prototype design and development. The energy savings with the system was dependent on several parameters: including property types, climatic conditions, house thermal inertia, control mode of operations and heat supply temperature. All these factors are interconnected to each other and needs to be considered at the same time to draw solid conclusion. The maximum improvement possible for the Mid terraced 1900s test house with change of control (VSM vs. FSM) was at lower heat supply temperature, up to 33% in milder climatic of conditions of Valentia and reduces to 25% in case of severe climatic conditions of Aviemore. The developed HP as a domestic retrofit technology have shown promising results in terms of carbon emissions reduction in comparison to fossil fuel-based heating technologies for all considered locations. The respective carbon emission savings of 62%, and 68% were obtained when the HP (VSM, C1) was retrofitted instead of modern GB and OB at 90% efficiency in the milder climatic conditions of Valentia. In case of heavy weight condensing GB and OB assumptions (70% efficiency) the carbon emission could reduce even more with a value of up to 71% and 75% respectively. The HP (VSM) was able to show superior characteristics than the VHTHP system at low supply temperature but become less effective at high supply temperature of 55 ^{o}C. This could be concluded that if the existing radiators is being used without heating distribution replacement with either modern radiators, or underfloor heating, then the VHTHP is more favourable due to its higher performance at high supply temperature and lower ambient temperature. The annual running cost for the developed HP in Belfast climatic conditions was not economical compared to GB at all percentage efficiencies with higher heat supply temperature (C1, C2) in both control mode but become effective at lower heat supply temperature(C1) during VSM when the GB was operating at 80%, and 70% efficiency and a respective cost savings of 6% and 17% were obtained. However, an increase of 6% annual running cost with the HP during VSM(C1) was observed when compared to GB at 90% efficiency. The HP during VSM(C1) was able to defeat the GB at all percentage efficiencies (90%,80%,70%) and respective cost savings of 1%, 12%, and 23% was achieved in Valentia climatic conditions. The HP was unable to defeat the GB at all percentage efficiency in terms of annual running cost in Valentia climatic conditions with high supply temperature (C2, C3). However, was able to show superior performance than OB at all percentage efficiencies in all cases (C1, C2, C3) during VSM of control with the only exception of OB (90% efficiency) at C3. The question of operating the variable speed HP best operating point have been investigated. According to the above retrofit assessment for the developed HP the result shows that it is a good candidate for oil/gas boilers replacement specifically in off gas grid area where most of the properties are old aged with higher heat loss coefficients, installed with old heavy weight oil boilers operating with lower efficiency. The cost savings are also possible when the system is operated at lower heat supply temperature and VSM for most of the operating efficiency considered for oil/gas boilers. The only issue is with the higher upfront cost, and beside that the developed ASHP have shown good potential as a renewable energy technology uptake in future. The study could be summarized in the following points.

a) Retrofit assessment of the developed 9kW domestic air source heat pump (ASHP) system for the test house instead of fossil fuel-based heating technologies.
b) The HP performance analysis at three fixed heat supply temperature with two operating modes for capacity control: a) variable speed mode (VSM), b) fixed speed mode (FSM).
c) Building type, age period with different insulation properties, climatic conditions simultaneous considerations and its influence on the HP annual performance.
d) A comparative retrofit analysis for the developed HP with the existing installed very high temperature ASHP (VHTASHP) and carbon emission savings potential.
e) The developed HP (VSM) at low heat supply temperature instead of gas boiler (90% efficiency) could cut the annual carbon emissions by 59% but with additional 6% running cost for Belfast climatic conditions.

Chapter 7: Conclusions and Future work recommendations

In this research work the variable speed compressor based ASHP successful development and its performance assessment as a domestic retrofit technology instead of oil/gas boilers, electric heating, and very high heat supply temperature ASHP. The HP performance improvement with challenges was investigated by simulating range of fixed heat load demands at range of experienced ambient and low to medium and high heat supply temperature inside the laboratory conditions. The numerical model was used to predict its performances for domestic heat load applications in UK considering different property types, and age period factor with insulation thermal characteristics. The air source heat pump (ASHP) with varying compressor-based system is more suitable for retrofit applications due its small size, lower initial capital cost and with minimum house disruptions requirements compared to ground-source heat pump (GSHP). However, the heat load demands inverse relationship with the ASHP heating capacity poses a challenge, and the system designed for the peak demand leads to the poor performances with thermal comfort issues inside the built environment and hence capacity control via variable speed compressor becomes crucial. The performance assessment and the role of the variable speed compressor-based air source heat pump (ASHP) system could play for the implementations of the demand side management programs was investigated. The benefits associated with the variable speed compressor-based heat pump system for the utility and network stability were evaluated by savings energy consumptions and improved performance. The study highlights the importance of the heat load demand, property types, climatic conditions impact, heat supply temperature, and operating mode of control for the performance improvement and retrofit assessment. The annual performance comparison for the operating mode of control (VSM vs. FSM) is more valuable in contrast to single operating conditions and must be considered. The VSM could meet the peak load demands for almost all property types without the needs for auxiliary heaters while in case of FSM the percentage of the load demand meet by the heat pump only depends on the property type load demand, and that causes increase in grid peak load demand. The developed ASHP with variable speed compressor performance was established inside the laboratory conditions at three level of supply temperature (low to medium and high heat supply temperature) with a range of heat loads demands commonly required for domestic purposes at experienced ambient conditions in UK climatic conditions. The testing results were utilized for the performance predictions in different property type and retrofit assessment. The heat pump for domestic heat load application was considered as a domestic retrofit technology instead of fossil fuel-based heating technologies. The decarbonization with the heat pump system and carbon emission savings were systematically investigated. The heat pump technology has shown promising results for carbon emission savings. The performance improvement due to heat supply temperature and control mode were thoroughly investigated. The variable speed compressor-based heat pump system has shown a great potential for improvement in performance and energy savings in comparison to fixed speed mode of operations but with challenges to compressor efficiencies and inverter losses. The challenges/drawbacks in combination with benefits associated with variable speed compressor for domestic demand side management were very rarely investigated in the literature. In literature almost all the studies are dealing with the benefits associated with the variable speed compressor HP system for demand side management and no study in the open literature was found investigating both the pros and cons of variable speed compressor specially in the context of UK. The network stability, smart grid idea and the services provided by the HP were discussed at three levels with different control strategy, but the impact on the compressor itself were recommended for further studies [32]. The only study in the open literature discussing the drawbacks with the variable speed compressor was experimentally established, but the aim was not the heat pump system for domestic heat load application and performance evaluation [82]. The following other conclusions could be drawn:

a) The variable speed compressor-based heat pump system for capacity control could assist in operating the real vapor compression cycle as close to the Carnot cycle for efficiency improvement due to better part load performance, reduced ON/OFF cycles, and less requirements for auxiliary heater.
b) The variable speed compressor-based heat pump system could play a significant role in the implementations of the DSM programmes by making energy savings and providing flexibility to the utility, but the challenges need to be considered.
c) The size of the heat pump system is very crucial for its selection and utilizations in this context with the load demand range according to the locations.
d) Part load better performed i.e., controlling, and matching the heating capacity to the varying heat load demand resulting in peaks clipping- DSM program. Avoids auxiliary heater that causes increase in electric peak loads during winters and

become a challenge for electricity production and distribution companies and reduces defrosting, and ON/OFF cycles.

e) Variable speed compressor designed for 50% of load at design outdoor temperature (DOT) could meet 100% annual heat load demand.

f) The HP result shows that decarbonization and carbon neutrality target from domestic heat sector could be achieved by 2050 in UK but needs government incentives for the initial capital cost. The comparative results with the existing fossil fuel-based technology shows significant savings. The additional control devices and heating distributions system installations in the existing housing stock needs social acceptance and the funding support from government.

g) The potential improvement for the system is higher with variable speed mode (VSM) of operation in cold climatic conditions at lower heat supply temperature compared to the fixed speed mode (FSM) of operations.

h) Old age period (1900-1949) property types including Detached, Semi-detached, Mid terraced, end terraced, and Flat were considered in the analysis.

i) The HP (VSM vs. FSM) annual performance improves by 27% at 35 ^{o}C heat supply temperature for Belfast climatic conditions. The HP(VSM) annual performance degrades by 51% with the change in supply temperature from 35 ^{o}C to 55 ^{o}C in Belfast climatic conditions.

j) The very high heat supply temperature heat pump (VHTHP was found more beneficial in performance with the existing heating distribution system but with increased energy consumptions and carbon emissions.

k) The annual running cost, for the developed HP was higher compared to advanced GB but proved to be advantageous in terms of carbon emissions savings.

The research questions that were posed at the start could be answered as follows:

❖ The HP control is one of the key factors for providing the flexibility to the power system in addition to the HP size, dynamic properties, building physics, consumptions behavior, insulations characteristics, Auxiliary heat requirements, climatic conditions, heating distribution system,

❖ The variable speed compressor-based heat pump could play a significant role in the implementations of DSM programmes, but the challenges need to be considered.

- ❖ The HP performance improves and valuable for future smart grid –by operating the real cycle as close to the Carnot cycle (isentropic compression-adiabatic, reversible)
- ❖ The potential improvement for the system is higher with variable speed mode (VSM) of operation in warm climatic conditions at lower heat supply temperature compared to the fixed speed mode (FSM) of operations

7.1 Research work limitations

Due to unavailability of the performance data for very high temperature ASHP in other climatic conditions, Belfast region was considered only for comparative retrofit assessment. The data utilized for the DHW consumption calculation into five property types were based on assumptions as the real experimental data were not available and depends on actual occupancy schedule and the number of users. The field trials results could produce more valuable results by installing the system in the different property types rather relying on the lab-based testing results.

7.2 Future work recommendations

The lab-based testing results provide a strong base for the HP system performance optimization for domestic heat load applications. However, the transient losses result with the heat pump could be more valuable from the real installations with different heating distribution systems. The efficiency improvement with supply temperature and control mode could be used as a technique for the performance improvement and cost saving. The additional installations of heating distribution system need very detailed economic analysis. In this study very high level of installations cost have been considered. The payback period with low heat supply-based heating distribution system feasibility needs to be studied in more details for justifications inside the existing housing stock, balancing the benefits with the carbon emission savings simultaneously. The following work is suggested in future:

a) The heat pump could be installed in housing stock for the trial purposes with different property type and age period for the actual transient losses' calculations.

b) More robust model could be developed based on the lab testing results and validated with field trial results to predict the system performance

c) The HP testing with thermal energy storage investigating the load shifting potential inside the test house for demand side management and network stability could be investigated as part of SPIRE 2 project.

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APPENDIX A

MODEL VALIDATION

```
(* Import data in following format; Twexit,Tamb,COP*);

In[256]:= data = Drop[data1, 1]; (*drop data first nontext entry *)
          l = Length[data]; (* drop the first non-numeric values, define length of data*)

In[257]:= Clear[a, b, c, d, e, f, g, h, T, Tamb, fr] (* Clear variables*)
          model = a T^2 + b Tamb^2 + c fr^2 + d T Tamb + e Tamb fr + f T fr + g T Tamb fr + h
            (* define the model, a,....,f are unknowns, we can play with this model*)

Out[258]= c fr^2 + h + f fr T + a T^2 + e fr Tamb + d T Tamb + fr g T Tamb + b Tamb^2

In[259]:= sol = NonlinearModelFit[data, model, {a, b, c, d, e, f, g, h}, {T, Tamb, fr},
            MaxIterations → 1000] (* Nonlinear fitting, data means experimental data,
           model is defined above, T,Tamb,and fr stands for independent variables*)

Out[259]= FittedModel[ 3.6453 - 0.000242138 fr^2 + 0.000739652 fr T - 0.000975593 T^2 +
            0.00182599 fr Tamb + 0.000907696 T Tamb - 0.0000438202 fr T Tamb + 0.00237521 Tamb^2 ]

In[260]:= sol[{"RSquared", "BestFitParameters"}]
          (* Once the fit is obtained as above, we can extract are the needed data*)
          sol["ParameterTable"] (* Table of statistical parameters*)
          res = sol["FitResiduals"]; (* Residuals for ploting*)
          res2 = res // TableForm;
          (* Residual in table form for later in table usage*)

Out[260]= {0.993498, {a → -0.000975593, b → 0.00237521, c → -0.000242138, d → 0.000907696,
            e → 0.00182599, f → 0.000739652, g → -0.0000438202, h → 3.6453}}
```

Out[261]=

	Estimate	Standard Error	t-Statistic	P-Value
a	-0.000975593	0.000104239	-9.35922	3.2024×10^{-12}
b	0.00237521	0.00121034	1.96243	0.0557778
c	-0.000242138	0.0000378451	-6.39814	7.30988×10^{-8}
d	0.000907696	0.000505628	1.79518	0.0791958
e	0.00182599	0.000456013	4.00425	0.000224874
f	0.000739652	0.000114443	6.46309	5.83808×10^{-8}
g	-0.0000438202	0.000010733	-4.08273	0.000175721
h	3.6453	0.164838	22.1144	3.77818×10^{-26}

```
In[264]:= Clear[fit]
          fit[T_, Tamb_, fr_] = Normal[sol]
           (* extracting the fitted model in neat form for ploting etc
            and defining it as a function of T and Tamb and frequency (fr)*)

Out[265]= 3.6453 - 0.000242138 fr^2 + 0.000739652 fr T - 0.000975593 T^2 +
            0.00182599 fr Tamb + 0.000907696 T Tamb - 0.0000438202 fr T Tamb + 0.00237521 Tamb^2

In[266]:= compr = Table[{data[[i, 1]], data[[i, 2]], data[[i, 3]], data[[i, 4]],
              fit[data[[i, 1]], data[[i, 2]], data[[i, 3]]], res2[[1, i]]}, {i, 1, l}];
          (*Tam, Twesit, freq, COP_actual,COP_model,Residual*)
          Text@
           Grid[Prepend[compr, {"T_wexit", "T_amb", "Freq", "COP_actual", "COP_Model", "Residual"}],
            Frame → All, ItemStyle → 14, Spacings → {1.2, 1.2}] (*Viewing data in table*)
```

Table A1: COP predicted value with residuals

T_{water}	T_{amb}	Freq	COP_{actual}	COP_{Model}	Residual
35.	−2.33	102.84	2.81	2.42047	0.389534
45.	−2.28	105.17	2.41	2.44622	−0.0362175
45.	−2.25	75.63	2.62	2.74696	−0.126963
35.	−2.23	74.32	3.05	2.92926	0.120736
45.	−2.23	48.83	2.51	2.65427	−0.144265
35.	−2.22	48.64	3.05	3.04614	0.00386384
35.	−1.85	23.8	2.62	2.86566	−0.245659
45.	−1.79	23.07	2.15	2.24924	−0.0992436
55.	1.88	44.83	2.22	2.08424	0.135757
35.	1.92	43.48	3.29	3.21219	0.0778105
55.	2.02	69.37	2.21	2.37963	−0.169631
45.	2.04	44.39	2.54	2.75009	−0.210087
55.	2.05	24.34	2.14	1.62403	0.515968
45.	2.08	20.79	2.2	2.34597	−0.145975
45.	2.1	93.84	2.61	2.72837	−0.118366
35.	2.12	92.25	3.04	2.91292	0.127075
35.	2.13	67.19	3.43	3.21674	0.213257
45.	2.15	21.95	2.2	2.37557	−0.175566
45.	2.18	67.64	2.73	2.89207	−0.162069
35.	2.19	21.1	2.79	2.9831	−0.193102
55.	2.21	93.52	2.21	2.38207	−0.172071
55.	6.2	63.22	2.41	2.47008	−0.0600784
55.	6.36	40	2.43	2.19893	0.231067
55.	6.59	19.85	2.28	1.76198	0.518025
35.	6.76	58.77	3.84	3.57472	0.265277
35.	6.79	37.78	3.8	3.48283	0.317171
35.	6.82	85.27	3.35	3.39419	−0.0441902
45.	6.82	108.33	2.64	2.71507	−0.0750714
45.	6.83	38.95	3.05	2.94976	0.100236
45.	6.86	59.69	3.11	3.12599	−0.0159857
35.	6.9	107.69	3.12	2.97942	0.140577
45.	6.95	19.15	2.43	2.59751	−0.167512
55.	7.08	89.69	2.43	2.49656	−0.0665637
35.	7.32	21.73	2.81	3.30472	−0.494718
55.	7.62	1109	2.43	2.25234	0.17766
45.	7.71	88.5	2.76	3.07545	−0.315453
55.	14.43	68.64	2.87	2.98206	−0.112059
55.	14.61	50.52	2.94	2.93656	0.00343711
55.	14.8	32.49	2.83	2.73851	0.0914918
55.	14.81	97.42	2.78	2.77678	0.00321929
45.	14.87	97.02	3.43	3.54182	−0.111822
45.	14.87	65.27	3.83	3.8016	0.028396
45.	14.87	15.52	2.68	3.22688	−0.546883
45.	14.88	47.46	3.75	3.73465	0.0153536
45.	14.88	30.99	3.65	3.53507	0.114926
55.	14.89	15.97	2.18	2.41312	−0.233122
35.	14.91	101.35	3.64	4.03012	−0.390115
35.	14.92	15.21	3.5	3.857	−0.357001
55.	14.92	114.23	2.63	2.45964	0.170364
35.	14.95	45.6	4.78	4.33226	0.447736
35.	15.11	30.04	4.64	4.16436	0.475643
35.	15.16	93.1	4.11	4.20163	−0.0916301
35.	15.19	62.87	4.96	4.43043	0.529565
45.	15.25	112.58	3.14	3.27272	−0.132725

```
In[257]:= pldata = Table[{data[[i, 4]], fit[data[[i, 1]], data[[i, 2]], data[[i, 3]]]},
    {i, 1, 1}](*Actual COP, Predicted COP,
  in all document semicolon is used for hiding the output*);
 pldata2 = SortBy[pldata, 1](* sorting the plot data for figue below*)

Out[258]= {{2.14, 1.62403}, {2.15, 2.24924}, {2.18, 2.41312}, {2.2, 2.34597}, {2.2, 2.37557},
  {2.21, 2.37963}, {2.21, 2.38207}, {2.22, 2.08424}, {2.28, 1.76198}, {2.41, 2.44622},
  {2.41, 2.47008}, {2.43, 2.19893}, {2.43, 2.25234}, {2.43, 2.49656}, {2.43, 2.59751},
  {2.51, 2.65427}, {2.54, 2.75009}, {2.61, 2.72837}, {2.62, 2.74696},
  {2.62, 2.86566}, {2.63, 2.45964}, {2.64, 2.71507}, {2.68, 3.22688},
  {2.73, 2.89207}, {2.76, 3.07545}, {2.78, 2.77678}, {2.79, 2.9831}, {2.81, 2.42047},
  {2.81, 3.30472}, {2.83, 2.73851}, {2.87, 2.98206}, {2.94, 2.93656},
  {3.04, 2.91292}, {3.05, 2.92926}, {3.05, 2.94976}, {3.05, 3.04614},
  {3.11, 3.12599}, {3.12, 2.97942}, {3.14, 3.27272}, {3.29, 3.21219},
  {3.35, 3.39419}, {3.43, 3.21674}, {3.43, 3.54182}, {3.5, 3.857}, {3.64, 4.03012},
  {3.65, 3.53507}, {3.75, 3.73465}, {3.8, 3.48283}, {3.83, 3.8016}, {3.84, 3.57472},
  {4.11, 4.20163}, {4.64, 4.16436}, {4.78, 4.33226}, {4.96, 4.43043}}

In[269]:= Show[ListPlot[pldata2, PlotStyle → Red, PlotRange → {{1.0, 6}, {1, 6}},
    AspectRatio → 0.8, Frame → True, FrameLabel → {"COP_Actual", "COP_Predicted"},
    LabelStyle → Directive[Black, Bold]], Plot[x, {x, 1, 6}, PlotStyle → Black,
    PlotLegends → Placed[{"x=y line; R²=0.993, 3 Var, Biquard"}, {Left, Top}]],
  Plot[1.05 x, {x, 1, 6}, PlotStyle → Green], Plot[0.95 x, {x, 0.25, 7.5},
    PlotStyle → Green, PlotLegends → Placed[{"±5%"}, {Left, Top}]]]
 ListPlot[res, Filling → Axis, Frame → True, FrameLabel → "Residuals",
```

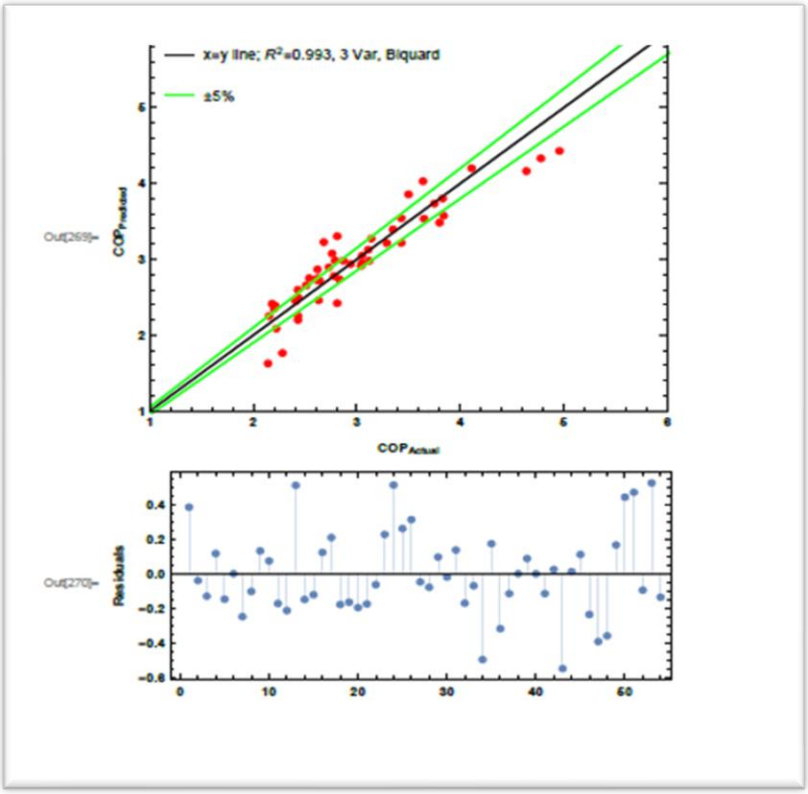

Figure A1: COP Model validation

MODEL CALIBRATION - Ambient temperature > 7 °C

```
(* Import data in following format; Twexit,Tamb,COP*);

In[272]:= data = Drop[data1, 34]; (*drop data first nontext entry *)
l = Length[data]; (* drop the first non-numeric values, define length of data*)

In[273]:= Clear[a, b, c, d, e, f, g, h, T, Tamb, fr] (* Clear variables*)
model = a T^2 + b Tamb^2 + c fr^2 + d T Tamb + e Tamb fr + f T fr + g T Tamb fr + h
  (* define the model, a,....,f are unknowns, we can play with this model*)

Out[274]= c fr^2 + h + f fr T + a T^2 + e fr Tamb + d T Tamb + fr g T Tamb + b Tamb^2

In[275]:= sol = NonlinearModelFit[data, model, {a, b, c, d, e, f, g, h}, {T, Tamb, fr},
  MaxIterations → 1000] (* Nonlinear fitting, data means experimental data,
 model is defined above, T,Tamb,and fr stands for independent variables*)

Out[275]= FittedModel[ 2.8299 - 0.000420739 fr^2 + 0.000966125 fr T + 0.000395929 T^2 +
  0.00207921 fr Tamb - 0.00922303 T Tamb - 0.0000309887 fr T Tamb + 0.0213245 Tamb^2 ]

In[276]:= sol[{"RSquared", "BestFitParameters"}]
(* Once the fit is obtained as above, we can extract are the needed data*)
sol["ParameterTable"] (* Table of statistical parameters*)
res = sol["FitResiduals"]; (* Residuals for ploting*)
res2 = res // TableForm;
(* Residual in table form for later in table usage*)

Out[276]= {0.997189, {a → 0.000395929, b → 0.0213245, c → -0.000420739, d → -0.00922303,
  e → 0.00207921, f → 0.000966125, g → -0.0000309887, h → 2.8299}}
```

Out[277]=

	Estimate	Standard Error	t-Statistic	P-Value
a	0.000395929	0.000873315	0.453363	0.657762
b	0.0213245	0.00772414	2.76077	0.016201
c	−0.000420739	0.0000620951	−6.77572	0.0000130997
d	−0.00922303	0.00524397	−1.75879	0.102113
e	0.00207921	0.000741084	2.80564	0.0148683
f	0.000966125	0.000335257	2.88174	0.0128504
g	−0.0000309887	0.0000243875	−1.27068	0.226111
h	2.8299	0.384403	7.36182	5.48878×10^{-6}

```
In[280]:= Clear[fit]
fit[T_, Tamb_, fr_] = Normal[sol]
 (* extracting the fitted model in neat form for ploting etc
  and defining it as a function of T and Tamb and frequency (fr)*)

Out[281]= 2.8299 - 0.000420739 fr^2 + 0.000966125 fr T + 0.000395929 T^2 +
 0.00207921 fr Tamb - 0.00922303 T Tamb - 0.0000309887 fr T Tamb + 0.0213245 Tamb^2

In[282]:= compr = Table[{data[[i, 1]], data[[i, 2]], data[[i, 3]], data[[i, 4]],
    fit[data[[i, 1]], data[[i, 2]], data[[i, 3]]], res2[[1, i]]}, {i, 1, l}];
(*Tam, Twesit, freq, COPactual,COPmodel,Residual*)
Text@
 Grid[Prepend[compr, {"Twexit", "Tamb", "Freq", "COPactual", "COPModel", "Residual"}],
  Frame → All, ItemStyle → 14, Spacings → {1.2, 1.2}] (*Viewing data in table*)
```

Table A2: Model calibration

T_{wexit}	T_{amb}	Freq	COP_{actual}	COP_{Model}	Residual
35.	7.32	21.73	2.81	2.78892	0.0210824
55.	7.62	110.9	2.43	2.43546	−0.00546196
45.	7.71	88.5	2.76	2.71882	0.0411828
55.	14.43	68.64	2.87	3.18432	−0.314321
55.	14.61	50.52	2.94	3.05549	−0.11549
55.	14.8	32.49	2.83	2.65349	0.176506
55.	14.81	97.42	2.78	2.91653	−0.136526
45.	14.87	97.02	3.43	3.42077	0.0092343
45.	14.87	65.27	3.83	3.88509	−0.055086
45.	14.87	15.52	2.68	2.9067	−0.226704
45.	14.88	47.46	3.75	3.77669	−0.0266918
45.	14.88	30.99	3.65	3.43647	0.213535
55.	14.89	15.97	2.18	2.03272	0.147281
35.	14.91	101.35	3.64	3.85081	−0.21081
35.	14.92	15.21	3.5	3.88832	−0.38832
55.	14.92	114.23	2.63	2.4248	0.205198
35.	14.95	45.6	4.78	4.60017	0.179833
35.	15.11	30.04	4.64	4.39353	0.246467
35.	15.16	93.1	4.11	4.22721	−0.117207
35.	15.19	62.87	4.96	4.74456	0.215438
45.	15.25	112.58	3.14	2.99914	0.140861

```
In[283]:= pldata = Table[{data[[i, 4]], fit[data[[i, 1]], data[[i, 2]], data[[i, 3]]]},
      {i, 1, l}] (*Actual COP, Predicted COP,
    in all document semicolon is used for hiding the output*);
   pldata2 = SortBy[pldata, 1] (* sorting the plot data for figue below*)
```

```
Out[284]= {{2.18, 2.03272}, {2.43, 2.43546}, {2.63, 2.4248}, {2.68, 2.9067}, {2.76, 2.71882},
    {2.78, 2.91653}, {2.81, 2.78892}, {2.83, 2.65349}, {2.87, 3.18432},
    {2.94, 3.05549}, {3.14, 2.99914}, {3.43, 3.42077}, {3.5, 3.88832},
    {3.64, 3.85081}, {3.65, 3.43647}, {3.75, 3.77669}, {3.83, 3.88509},
    {4.11, 4.22721}, {4.64, 4.39353}, {4.78, 4.60017}, {4.96, 4.74456}}
```

```
In[285]:= Show[ListPlot[pldata2, PlotStyle → Red,
     PlotRange → {{1.0, 6}, {1, 6}}, AspectRatio → 0.8, Frame → True,
     FrameLabel → {"COP_Actual", "COP_Predicted"}, LabelStyle → Directive[Black, Bold]],
    Plot[x, {x, 1, 6}, PlotStyle → Black, PlotLegends →
      Placed[{"x=y line; R²=0.997, 3 Var, Biquard, Tamb>7"}, {Left, Top}]],
    Plot[1.05 x, {x, 1, 6}, PlotStyle → Green], Plot[0.95 x, {x, 0.25, 7.5},
     PlotStyle → Green, PlotLegends → Placed[{"±5%"}, {Left, Top}]]]
   ListPlot[res, Filling → Axis, Frame → True, FrameLabel → "Residuals",
    AspectRatio → 0.45, LabelStyle → {Black, Bold}]
```

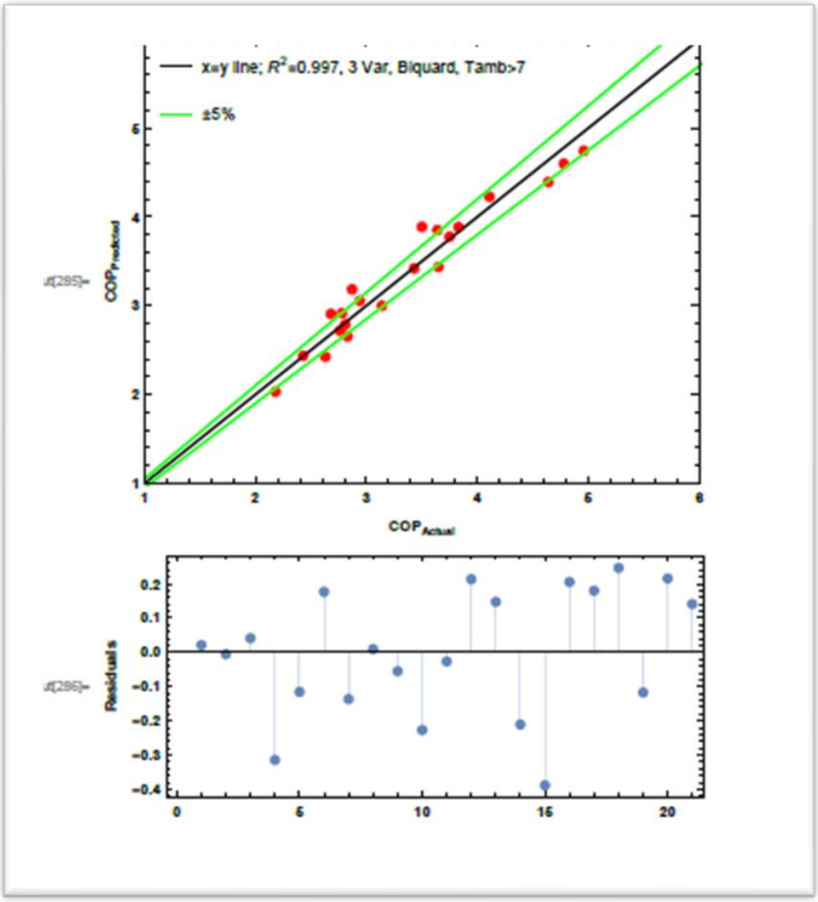

Figure A2: COP Model calibration results

POWER CONSUMPTIONS MODEL VALIDATION

```
(* Import data in following format; Twexit,Tamb,P*);
In[862]:= data = Drop[data1, 1]; (*drop data first nontext entry *)
l = Length[data]; (* drop the first non-numeric values, define length of data*)
In[863]:= Clear[a, b, c, d, e, f, g, h, T, Tamb, fr] (* Clear variables*)
model = a T^2 + b Tamb^2 + c fr^2 + d T Tamb + e Tamb fr + f T fr + g T Tamb fr + h
  (* define the model, a,....,f are unknowns, we can play with this model*)
Out[864]= c fr^2 + h + f fr T + a T^2 + e fr Tamb + d T Tamb + fr g T Tamb + b Tamb^2
In[865]:= sol = NonlinearModelFit[data, model, {a, b, c, d, e, f, g, h}, {T, Tamb, fr},
  MaxIterations → 1000] (* Nonlinear fitting, data means experimental data,
 model is defined above, T,Tamb,and fr stands for independent variables*)
Out[865]= FittedModel[ 0.385425 + 0.000131936 fr^2 + 0.000662867 fr T + 0.000101528 T^2 +
   0.000367014 fr Tamb + 0.000288784 T Tamb - 6.33776×10^-6 fr T Tamb - 0.000943088 Tamb^2 ]
In[866]:= sol[{"RSquared", "BestFitParameters"}]
(* Once the fit is obtained as above, we can extract are the needed data*)
sol["ParameterTable"] (* Table of statistical parameters*)
res = sol["FitResiduals"]; (* Residuals for ploting*)
res2 = res // TableForm;
(* Residual in table form for later in table usage*)
Out[866]= {0.997851, {a → 0.000101528, b → -0.000943088, c → 0.000131936, d → 0.000288784,
  e → 0.000367014, f → 0.000662867, g → -6.33776 × 10^-6, h → 0.385425}}
```

Out[867]=

	Estimate	Standard Error	t- Statistic	P- Value
a	0.000101528	0.000066878	1.5181	0.135832
b	−0.000943088	0.000776539	−1.21448	0.230764
c	0.000131936	0.0000242809	5.43375	2.02423×10^{-6}
d	0.000288784	0.000324404	0.890198	0.377992
e	0.000367014	0.000292572	1.25444	0.216019
f	0.000662867	0.0000734247	9.02785	9.48281×10^{-12}
g	-6.33776×10^{-6}	6.88617×10^{-6}	−0.92036	0.362186
h	0.385425	0.105758	3.64441	0.000679353

```
In[870]:= Clear[fit]
fit[T_, Tamb_, fr_] = Normal[sol]
 (* extracting the fitted model in neat form for ploting etc
  and defining it as a function of T and Tamb and frequency (fr)*)
Out[871]= 0.385425 + 0.000131936 fr^2 + 0.000662867 fr T + 0.000101528 T^2 +
  0.000367014 fr Tamb + 0.000288784 T Tamb - 6.33776 × 10^-6 fr T Tamb - 0.000943088 Tamb^2
In[872]:= compr = Table[{data[[i, 1]], data[[i, 2]], data[[i, 3]], data[[i, 4]],
    fit[data[[i, 1]], data[[i, 2]], data[[i, 3]]], res2[[1, i]]}, {i, 1, l}];
(*Tam, Twesit, freq, P_actual,P_model,Residual*)
Text@Grid[Prepend[compr, {"T_wexit", "T_amb", "Freq", "P_actual", "P_Model", "Residual"}],
  Frame → All, ItemStyle → 14, Spacings → {1.2, 1.2}] (*Viewing data in table*)
```

Table A3: Power predicted values with residual

T_{wexit}	T_{amb}	Freq	P_{actual}	P_{Model}	Residual
35.	−2.33	102.84	4.26	4.22763	0.0323715
45.	−2.28	105.17	4.94	5.1333	−0.193301
45.	−2.25	75.63	3.43	3.55371	−0.123715
35.	−2.23	74.32	2.95	2.9115	0.0384999
45.	−2.23	48.83	2.39	2.31958	0.070423
35.	−2.22	48.64	1.96	1.90764	0.0523608
35.	−1.85	23.8	1.14	1.10838	0.0316202
45.	−1.79	23.07	1.39	1.31973	0.0702676
55.	1.88	44.83	2.7	2.62018	0.0798175
35.	1.92	43.48	1.82	1.79603	0.0239742
55.	2.02	69.37	4.07	3.88734	0.18266
45.	2.04	44.39	2.36	2.2051	0.154899
55.	2.05	24.34	1.37	1.68761	−0.317608
45.	2.08	20.79	1.36	1.29468	0.0653217
45.	2.1	93.84	4.6	4.59125	0.00874654
35.	2.12	92.25	3.94	3.8184	0.121599
35.	2.13	67.19	2.63	2.70228	−0.0722842
45.	2.15	21.95	1.39	1.33677	0.0532253
45.	2.18	67.64	3.29	3.2482	0.0418025
35.	2.19	21.1	1.07	1.08238	−0.0123848
55.	2.21	93.52	5.43	5.29029	0.139709

55.	6.59	19.85	1.31	1.53434	−0.224343
35.	6.76	58.77	2.34	2.41189	−0.0718905
35.	6.79	37.78	1.57	1.63702	−0.0670175
35.	6.82	85.27	3.57	3.5569	0.0131002
45.	6.82	108.33	5.56	5.47593	0.0840693
45.	6.83	38.95	1.96	2.01955	−0.0595492
45.	6.86	59.69	2.89	2.91986	−0.0298564
35.	6.9	107.69	4.8	4.67106	0.128945
45.	6.95	19.15	1.23	1.26628	−0.0362813
55.	7.08	89.69	4.93	5.10066	−0.17066
35.	7.32	21.73	1.06	1.12279	−0.0627881
55.	7.62	110.9	6.17	6.44022	−0.270216
45.	7.71	88.5	4.34	4.3642	−0.0242044
55.	14.43	68.64	4.18	3.8677	0.312305
55.	14.61	50.52	3.06	2.91548	0.144516
55.	14.8	32.49	2.11	2.05369	0.056309
55.	14.81	97.42	5.39	5.5514	−0.161396
45.	14.87	97.02	4.37	4.82967	−0.459673
45.	14.87	65.27	3.13	3.16415	−0.034145
45.	14.87	15.52	1.11	1.08933	0.0206666
45.	14.88	47.46	2.4	2.34622	0.0537807
45.	14.88	30.99	1.64	1.66441	−0.0244133
55.	14.89	15.97	1.38	1.34022	0.039785
35.	14.91	101.35	4.94	4.37683	0.56317
35.	14.92	15.21	0.85	0.867011	−0.0170113
55.	14.92	114.23	6.84	6.63714	0.202857
35.	14.95	45.6	1.88	1.88138	−0.00138032
35.	15.11	30.04	1.29	1.3291	−0.0391034
35.	15.16	93.1	3.65	3.95473	−0.30473

```
In[873]:= pldata = Table[{data[[i, 4]], fit[data[[i, 1]], data[[i, 2]], data[[i, 3]]]},
          {i, 1, l}](*Actual P, Predicted P,
        in all document semicolon is used for hiding the output*);
       pldata2 = SortBy[pldata, 1](* sorting the plot data for figue below*)
```

```
Out[874]= {{0.85, 0.867011}, {1.06, 1.12279}, {1.07, 1.08238}, {1.11, 1.08933}, {1.14, 1.10838},
  {1.23, 1.26628}, {1.29, 1.3291}, {1.31, 1.53434}, {1.36, 1.29468}, {1.37, 1.68761},
  {1.38, 1.34022}, {1.39, 1.31973}, {1.39, 1.33677}, {1.57, 1.63702}, {1.64, 1.66441},
  {1.82, 1.79603}, {1.88, 1.88138}, {1.96, 1.90764}, {1.96, 2.01955}, {2.11, 2.05369},
  {2.34, 2.41189}, {2.36, 2.2051}, {2.39, 2.31958}, {2.4, 2.34622}, {2.42, 2.56449},
  {2.46, 2.42951}, {2.63, 2.70228}, {2.7, 2.62018}, {2.89, 2.91986}, {2.95, 2.9115},
  {3.06, 2.91548}, {3.13, 3.16415}, {3.29, 3.2482}, {3.43, 3.55371}, {3.57, 3.5569},
  {3.65, 3.95473}, {3.72, 3.59417}, {3.94, 3.8184}, {4.07, 3.88734}, {4.18, 3.8677},
  {4.26, 4.22763}, {4.34, 4.3642}, {4.37, 4.82967}, {4.6, 4.59125}, {4.8, 4.67106},
  {4.93, 5.10066}, {4.94, 4.37683}, {4.94, 5.1333}, {5.39, 5.5514}, {5.43, 5.29029},
  {5.56, 5.47593}, {5.72, 5.74068}, {6.17, 6.44022}, {6.84, 6.63714}}
```

```
In[875]:= Show[ListPlot[pldata2, PlotStyle → Red, PlotRange → {{0.25, 8}, {0.25, 8}},
     AspectRatio → 0.8, Frame → True, FrameLabel → {"P_Actual", "P_Predicted"},
     LabelStyle → Directive[Black, Bold]], Plot[x, {x, 0.25, 8}, PlotStyle → Black,
     PlotLegends → Placed[{"x=y line; R²=0.997, 3 Var, Biquard"}, {Left, Top}]],
    Plot[1.05 x, {x, 0.25, 8}, PlotStyle → Green], Plot[0.95 x, {x, 0.25, 8},
     PlotStyle → Green, PlotLegends → Placed[{"±5%"}, {Left, Top}]]]
   ListPlot[res, Filling → Axis, Frame → True, FrameLabel → "Residuals",
    AspectRatio → 0.45, LabelStyle → {Black, Bold}]
```

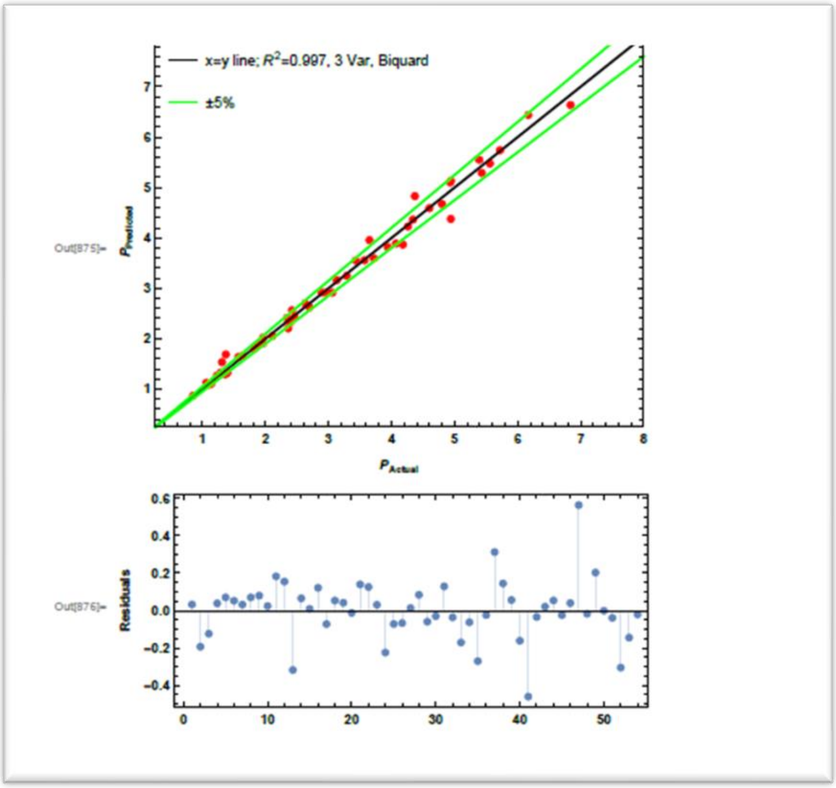

Figure A3: Power model validation results

POWER CONSUMPTIONS MODEL CALIBRATION Tamb>7 °C

```
(* Import data in following format; Twexit,Tamb,P*);
In[878]:= data = Drop[data1, 34]; (*drop data first nontext entry *)
l = Length[data]; (* drop the first non-numeric values, define length of data*)
```

```
In[879]:= Clear[a, b, c, d, e, f, g, h, T, Tamb, fr] (* Clear variables*)
model = a T^2 + b Tamb^2 + c fr^2 + d T Tamb + e Tamb fr + f T fr + g T Tamb fr + h
  (* define the model, a,....,f are unknowns, we can play with this model*)
```

Out[880]= $c\,fr^2 + h + f\,fr\,T + a\,T^2 + e\,fr\,Tamb + d\,T\,Tamb + fr\,g\,T\,Tamb + b\,Tamb^2$

```
In[881]:= sol = NonlinearModelFit[data, model, {a, b, c, d, e, f, g, h}, {T, Tamb, fr},
  MaxIterations → 1000] (* Nonlinear fitting, data means experimental data,
 model is defined above, T,Tamb,and fr stands for independent variables*)
```

Out[881]= FittedModel[$0.325642 + 0.000145178\,fr^2 + 0.000423026\,fr\,T + 0.000465983\,T^2 + 0.000672889\,fr\,Tamb - 0.00103254\,T\,Tamb + 3.7395\times10^{-7}\,fr\,T\,Tamb + 0.000111063\,Tamb^2$]

```
In[882]:= sol[{"RSquared", "BestFitParameters"}]
(* Once the fit is obtained as above, we can extract are the needed data*)
sol["ParameterTable"] (* Table of statistical parameters*)
res = sol["FitResiduals"]; (* Residuals for ploting*)
res2 = res // TableForm;
(* Residual in table form for later in table usage*)
```

Out[882]= {0.997209, {a → 0.000465983, b → 0.000111063, c → 0.000145178, d → -0.00103254, e → 0.000672889, f → 0.000423026, g → 3.7395×10^{-7}, h → 0.325642}}

Out[883]=

	Estimate	Standard Error	t-Statistic	P-Value
a	0.000465983	0.000937557	0.497019	0.627475
b	0.000111063	0.00829233	0.0133934	0.989517
c	0.000145178	0.0000666628	2.17779	0.0484309
d	-0.00103254	0.00562971	-0.183409	0.857307
e	0.000672889	0.000795598	0.845764	0.412981
f	0.000423026	0.000359919	1.17534	0.260938
g	3.7395×10^{-7}	0.0000261814	0.014283	0.988821
h	0.325642	0.412679	0.789091	0.444212

```
In[886]:= Clear[fit]
fit[T_, Tamb_, fr_] = Normal[sol]
 (* extracting the fitted model in neat form for ploting etc
  and defining it as a function of T and Tamb and frequency (fr)*)
```

Out[887]= $0.325642 + 0.000145178\,fr^2 + 0.000423026\,fr\,T + 0.000465983\,T^2 + 0.000672889\,fr\,Tamb - 0.00103254\,T\,Tamb + 3.7395\times10^{-7}\,fr\,T\,Tamb + 0.000111063\,Tamb^2$

```
In[888]:= compr = Table[{data[[i, 1]], data[[i, 2]], data[[i, 3]], data[[i, 4]],
    fit[data[[i, 1]], data[[i, 2]], data[[i, 3]]], res2[[1, i]]}, {i, 1, l}];
(*Tam, Twesit, freq, Pactual,Pmodel,Residual*)
Text@Grid[Prepend[compr, {"Twexit", "Tamb", "Freq", "Pactual", "PModel", "Residual"}],
  Frame → All, ItemStyle → 14, Spacings → {1.2, 1.2}] (*Viewing data in table*)
```

Table A4: Power predicted values with residuals

T_{wexit}	T_{amb}	Freq	P_{actual}	P_{Model}	Residual
35.	7.32	21.73	1.06	1.13728	−0.0772848
55.	7.62	110.9	6.17	6.26073	−0.0907254
45.	7.71	88.5	4.34	4.21001	0.129991
55.	14.43	68.64	4.18	3.90675	0.273249
55.	14.61	50.52	3.06	2.98704	0.0729559
55.	14.8	32.49	2.11	2.16171	−0.0517094
55.	14.81	97.42	5.39	5.56351	−0.173508
45.	14.87	97.02	4.37	4.81137	−0.441369
45.	14.87	65.27	3.13	3.13328	−0.00328165
45.	14.87	15.52	1.11	1.09248	0.0175214
45.	14.88	47.46	2.4	2.32001	0.0799941
45.	14.88	30.99	1.64	1.64987	−0.00986882
55.	14.89	15.97	1.38	1.48776	−0.10776
35.	14.91	101.35	4.94	4.41075	0.52925
35.	14.92	15.21	0.85	0.796459	0.0535414
55.	14.92	114.23	6.84	6.64661	0.193395
35.	14.95	45.6	1.88	1.82569	0.0543103
35.	15.11	30.04	1.29	1.26292	0.0270827
35.	15.16	93.1	3.65	3.97909	−0.329091
35.	15.19	62.87	2.42	2.53294	−0.112936
45.	15.25	112.58	5.72	5.75376	−0.0337565

```
In[889]:= pldata = Table[{data[[i, 4]], fit[data[[i, 1]], data[[i, 2]], data[[i, 3]]]},
      {i, 1, l}](*Actual P, Predicted P,
    in all document semicolon is used for hiding the output*);
   pldata2 = SortBy[pldata, 1](* sorting the plot data for figue below*)

Out[890]= {{0.85, 0.796459}, {1.06, 1.13728}, {1.11, 1.09248}, {1.29, 1.26292}, {1.38, 1.48776},
    {1.64, 1.64987}, {1.88, 1.82569}, {2.11, 2.16171}, {2.4, 2.32001},
    {2.42, 2.53294}, {3.06, 2.98704}, {3.13, 3.13328}, {3.65, 3.97909},
    {4.18, 3.90675}, {4.34, 4.21001}, {4.37, 4.81137}, {4.94, 4.41075},
    {5.39, 5.56351}, {5.72, 5.75376}, {6.17, 6.26073}, {6.84, 6.64661}}

In[891]:= Show[ListPlot[pldata2, PlotStyle → Red,
     PlotRange → {{0.25, 8}, {0.25, 8}}, AspectRatio → 0.8, Frame → True,
     FrameLabel → {"P_Actual", "P_Predicted"}, LabelStyle → Directive[Black, Bold]],
    Plot[x, {x, 0.25, 8}, PlotStyle → Black, PlotLegends →
      Placed[{"x=y line; R^2=0.997, 3 Var, Biquard, Tamb>7"}, {Left, Top}]],
    Plot[1.05 x, {x, 0.25, 8}, PlotStyle → Green], Plot[0.95 x, {x, 0.25, 8},
     PlotStyle → Green, PlotLegends → Placed[{"±5%"}, {Left, Top}]]]
   ListPlot[res, Filling → Axis, Frame → True, FrameLabel → "Residuals",
    AspectRatio → 0.45, LabelStyle → {Black, Bold}]
```

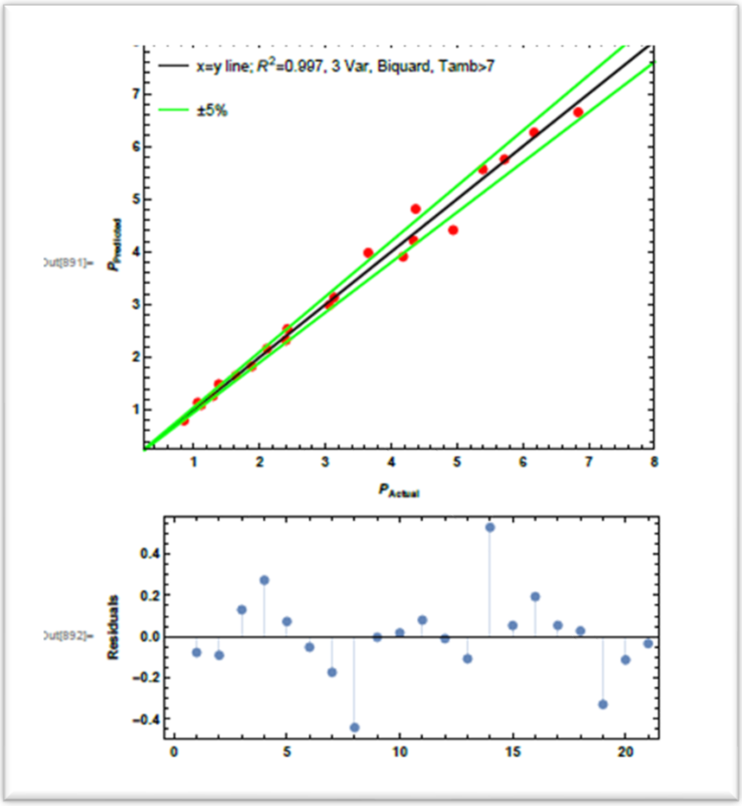

Figure A4: Power model calibration results

APPENDIX B

Cost and Carbon Emission Savings with Other Climatic Conditions with Three Age Period-Variable Speed Mode (VSM) Only

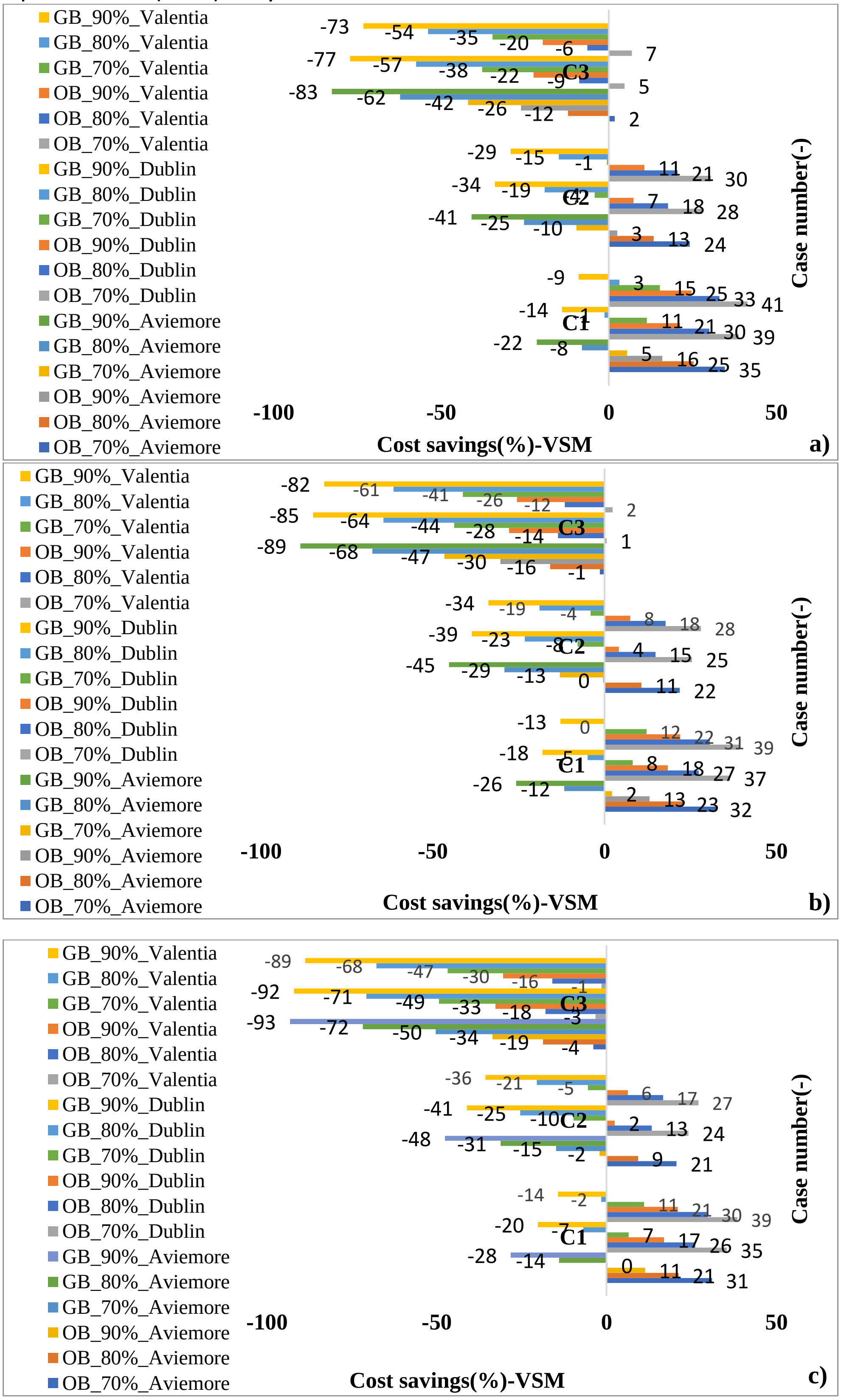

Figure B1. Flat type with three age period, a) 1900-1949, b) 1950-1975, c) 1976-1990

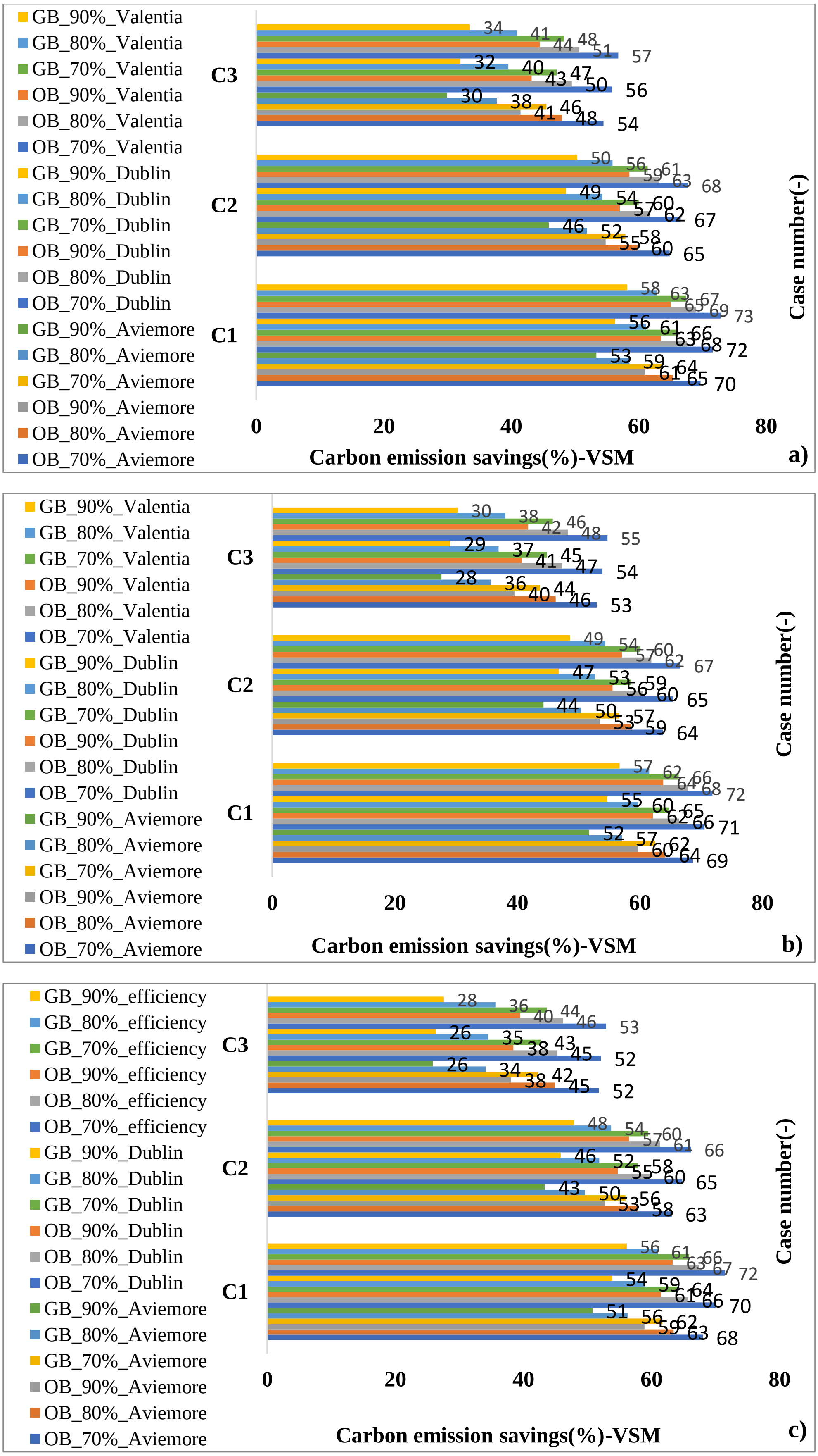

Figure B2. Flat type with three age period of a) 1900-1949, b) 1950-1975, c) 1976-1990

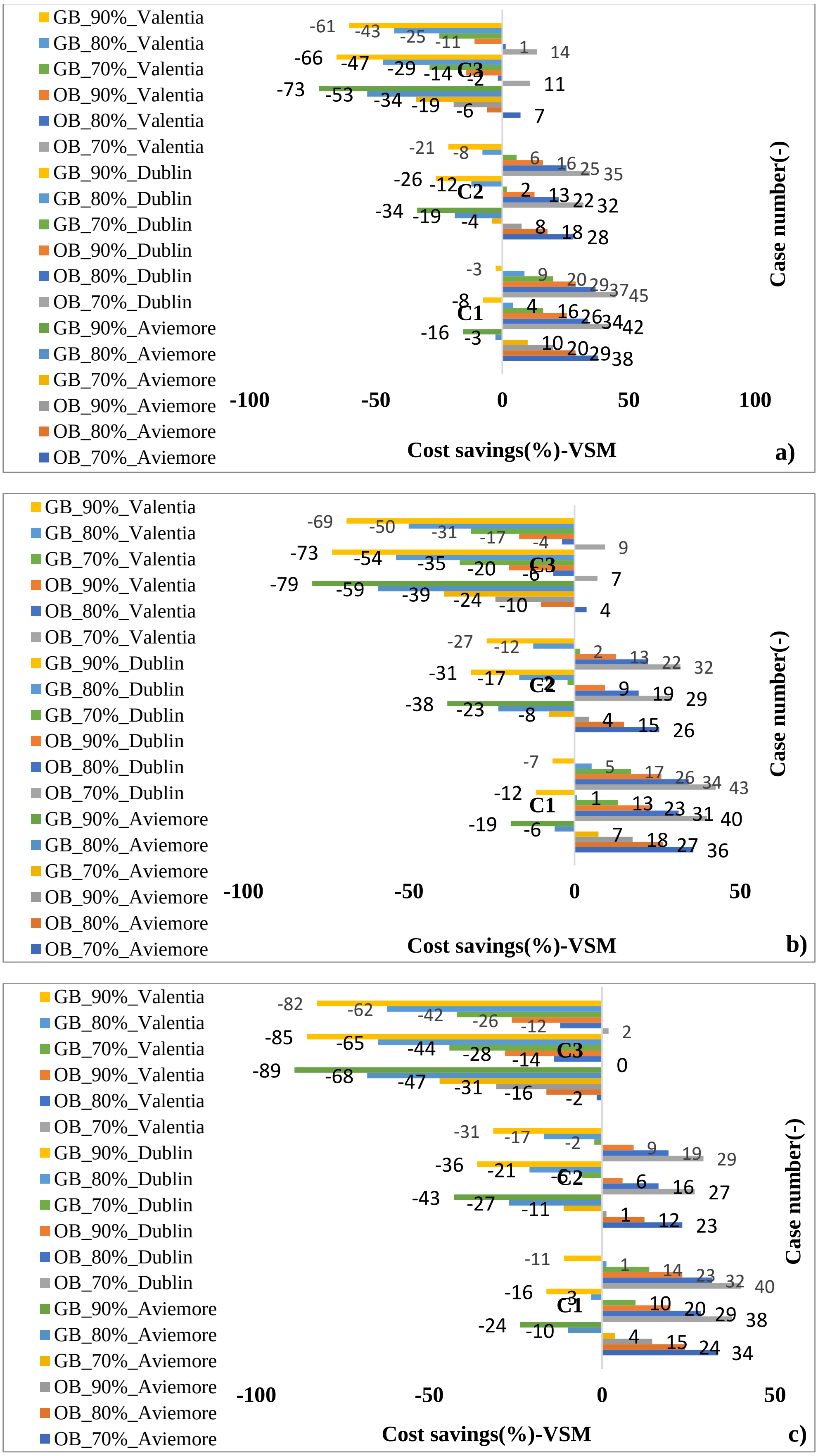

Figure B3. Mid terraced type with age period a) 1900-1949, b) 1950-1975, c) 1976-1990

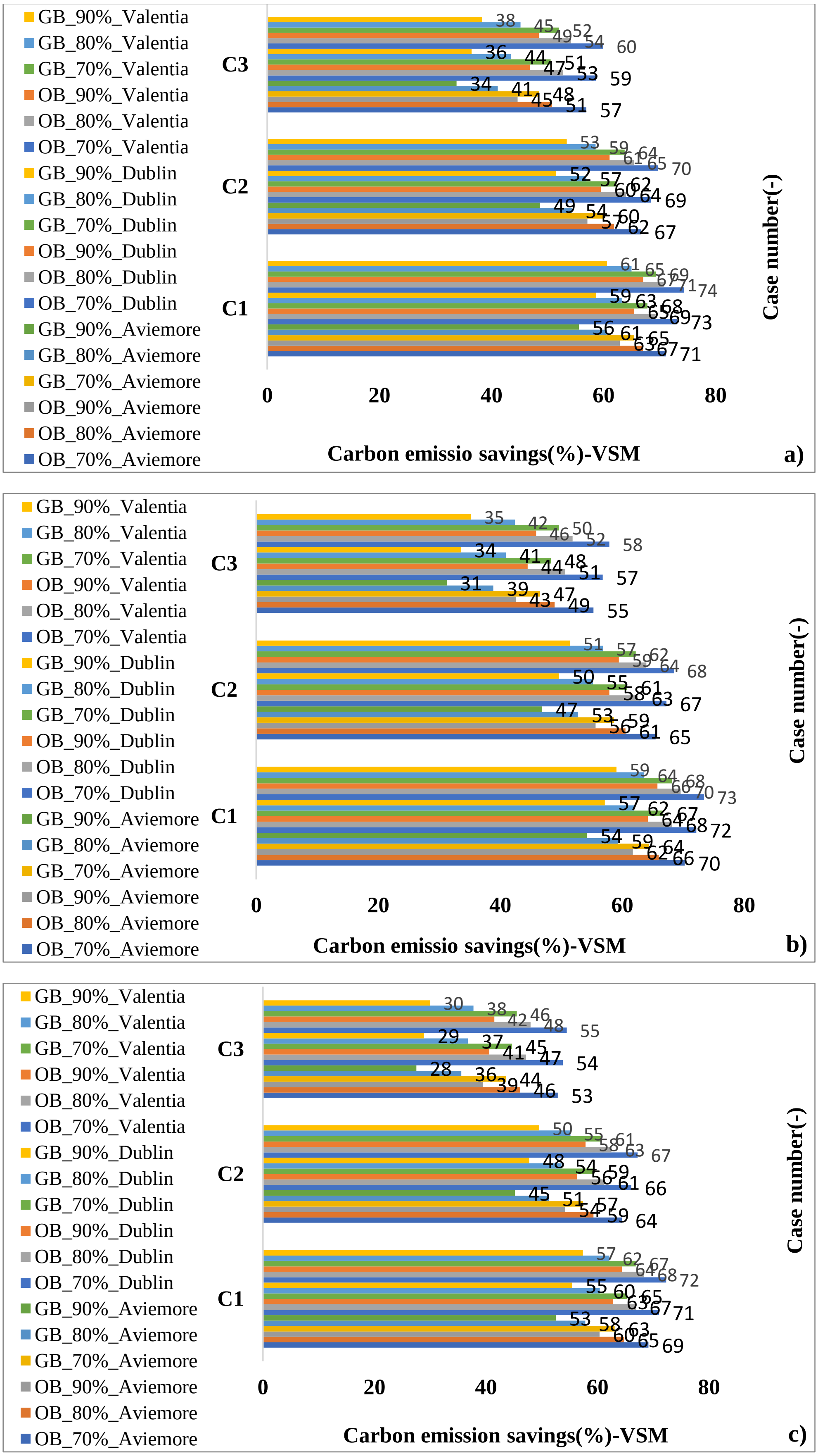

Figure B4. Mid terraced type with age period, a) 1900-1949, b) 1950-1975, c) 1976-1990

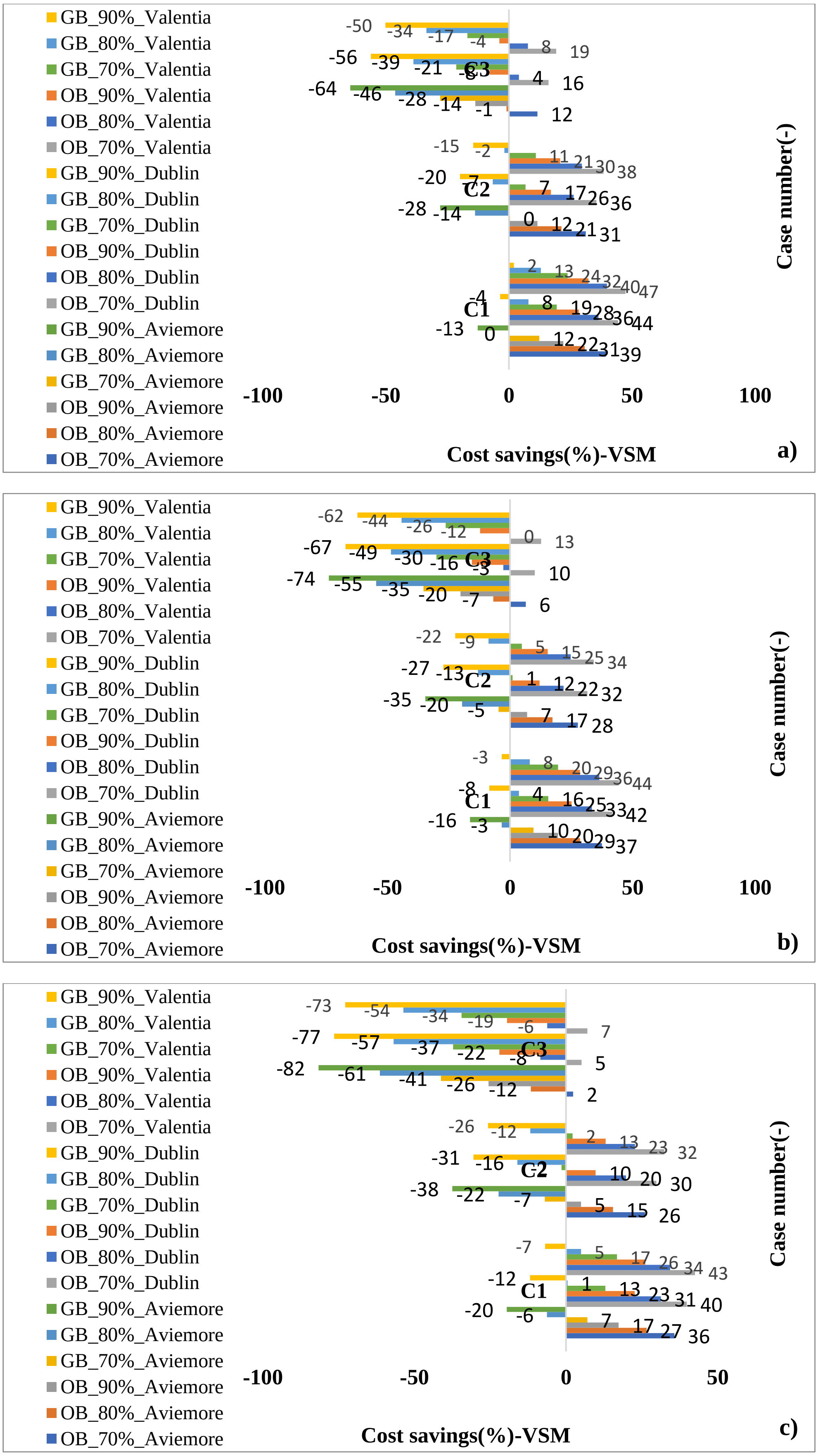

Figure B5. End terraced type with age period, a) 1900-1949, b) 1950-1975, c) 1976-1990

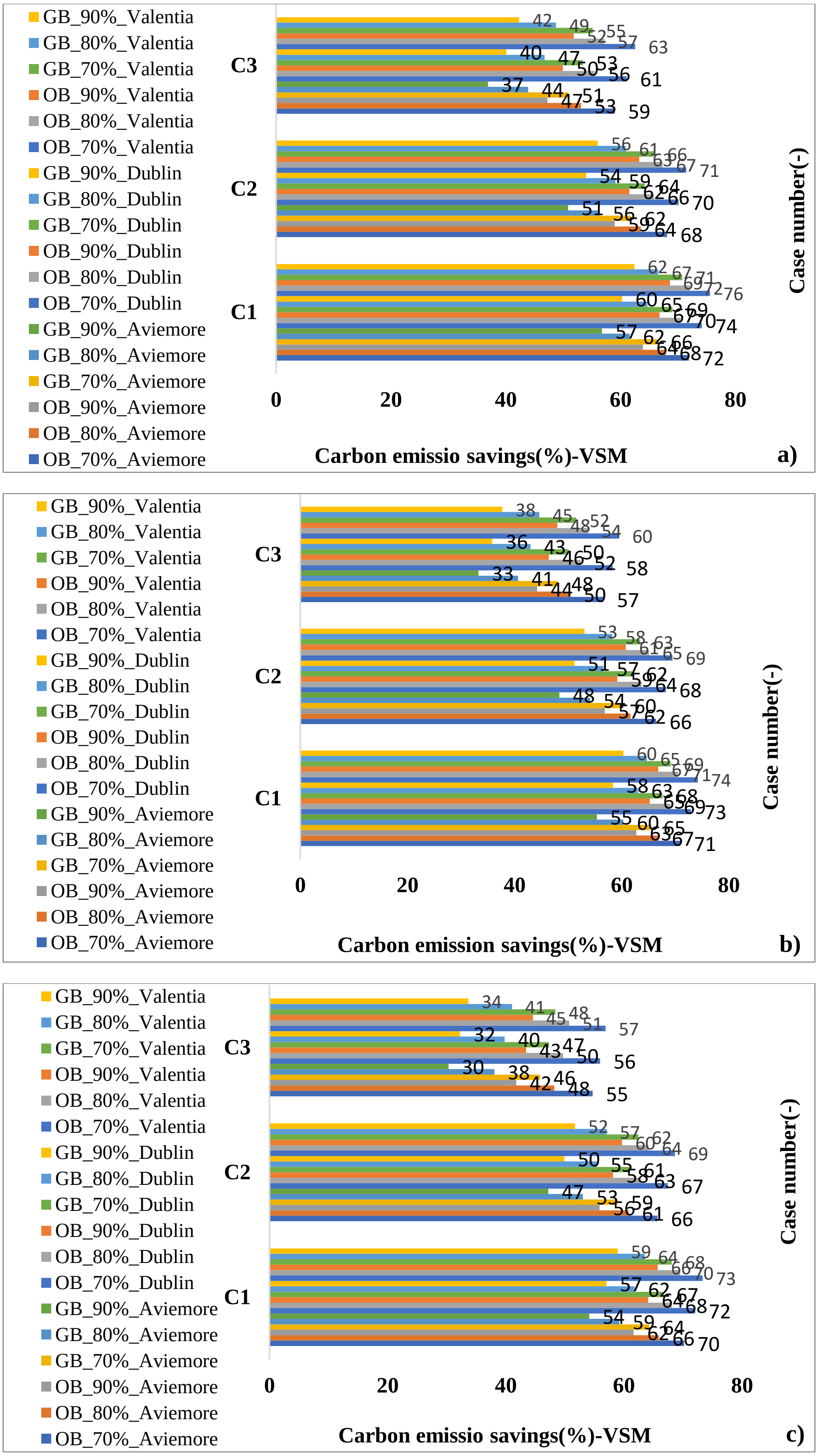

Figure B6. End terraced type with age period, a) 1900-1949, b) 1950-1975, c) 1976-1990

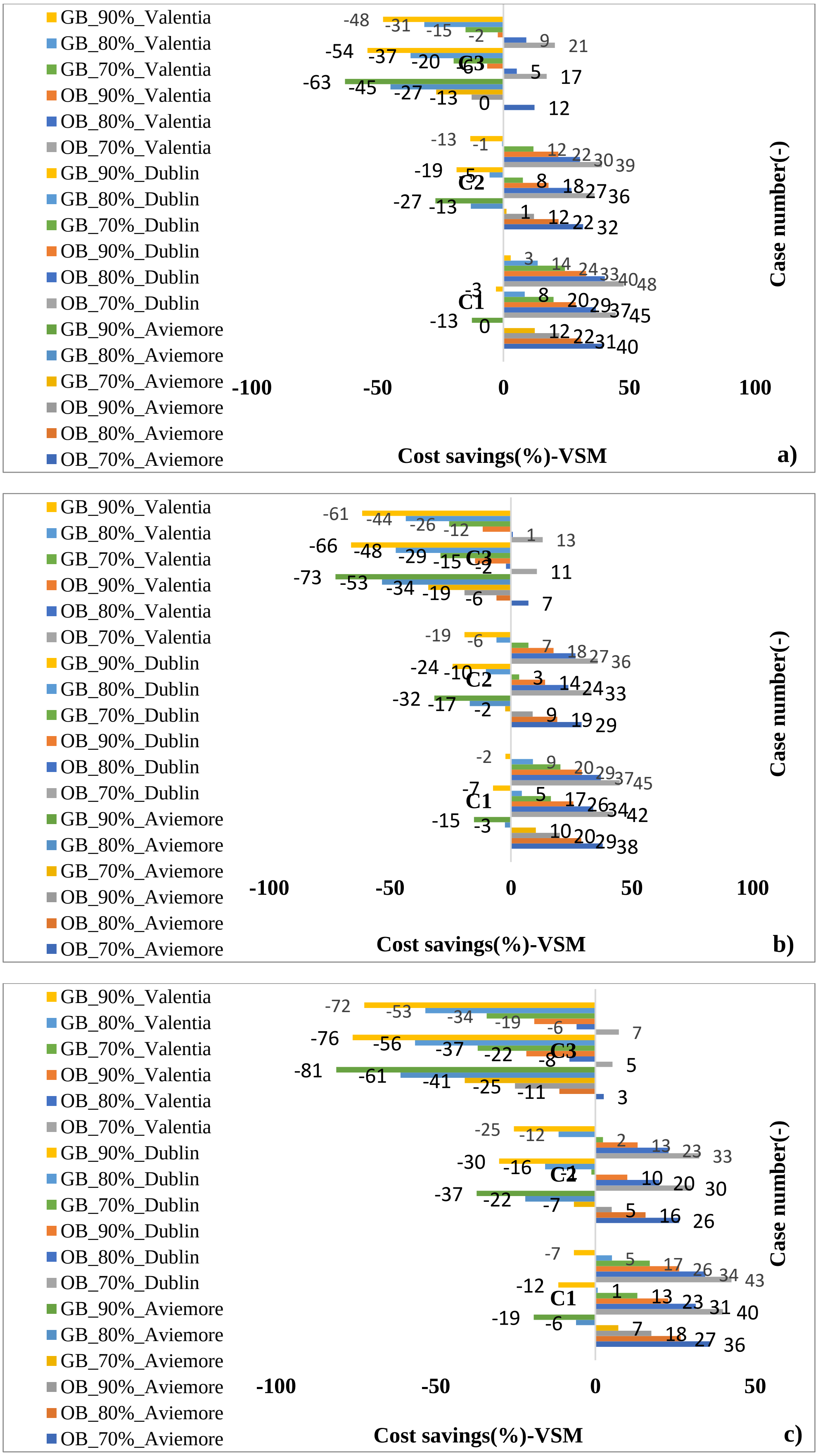

Figure B7. Semi-detached type with age period a)1900-1949, b)1950-1975, c) 1976-1990

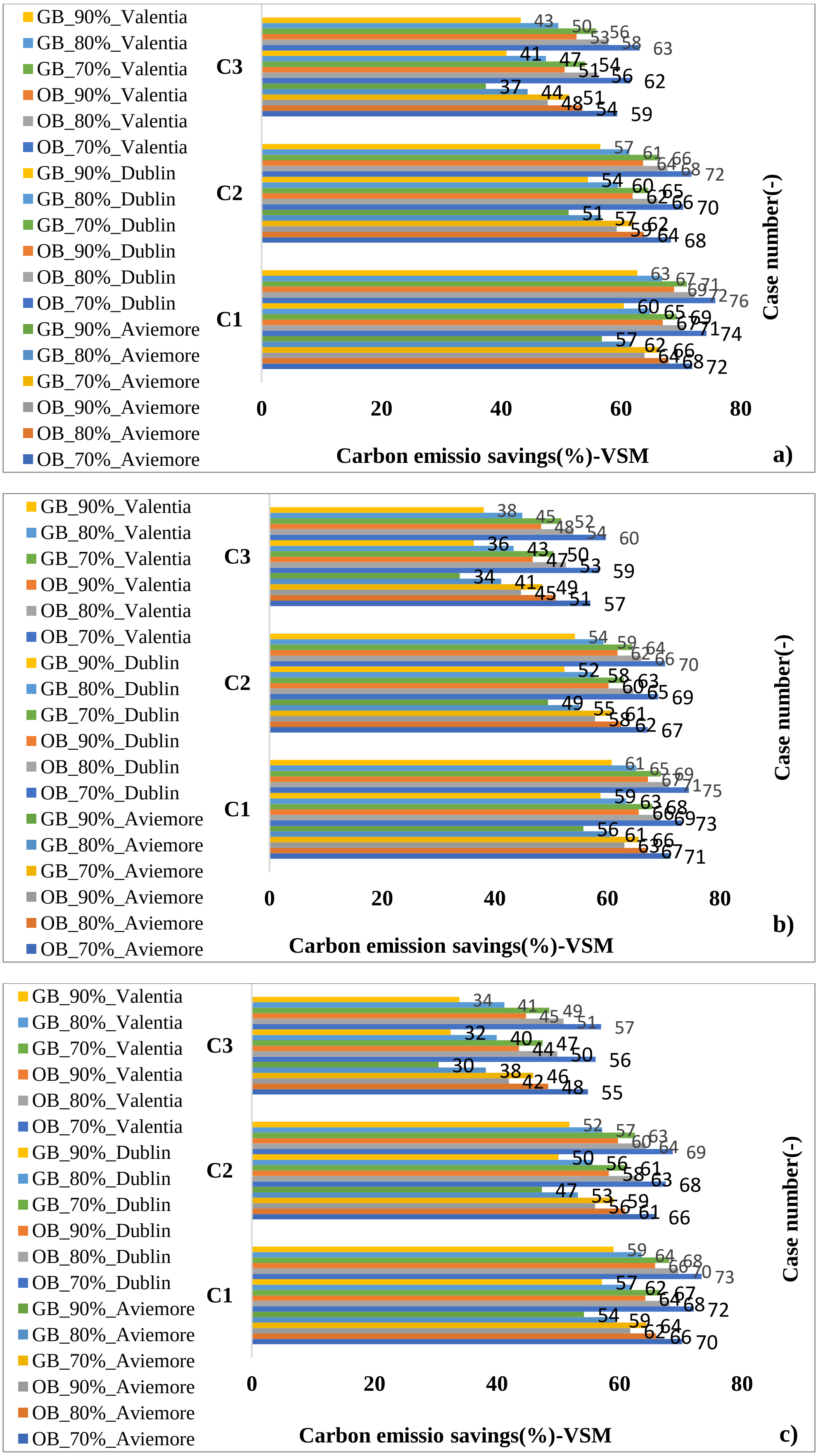

Figure B 8. Semi-detached type with age period, a)1900-1949, b)1950-1975, c)1976-1990

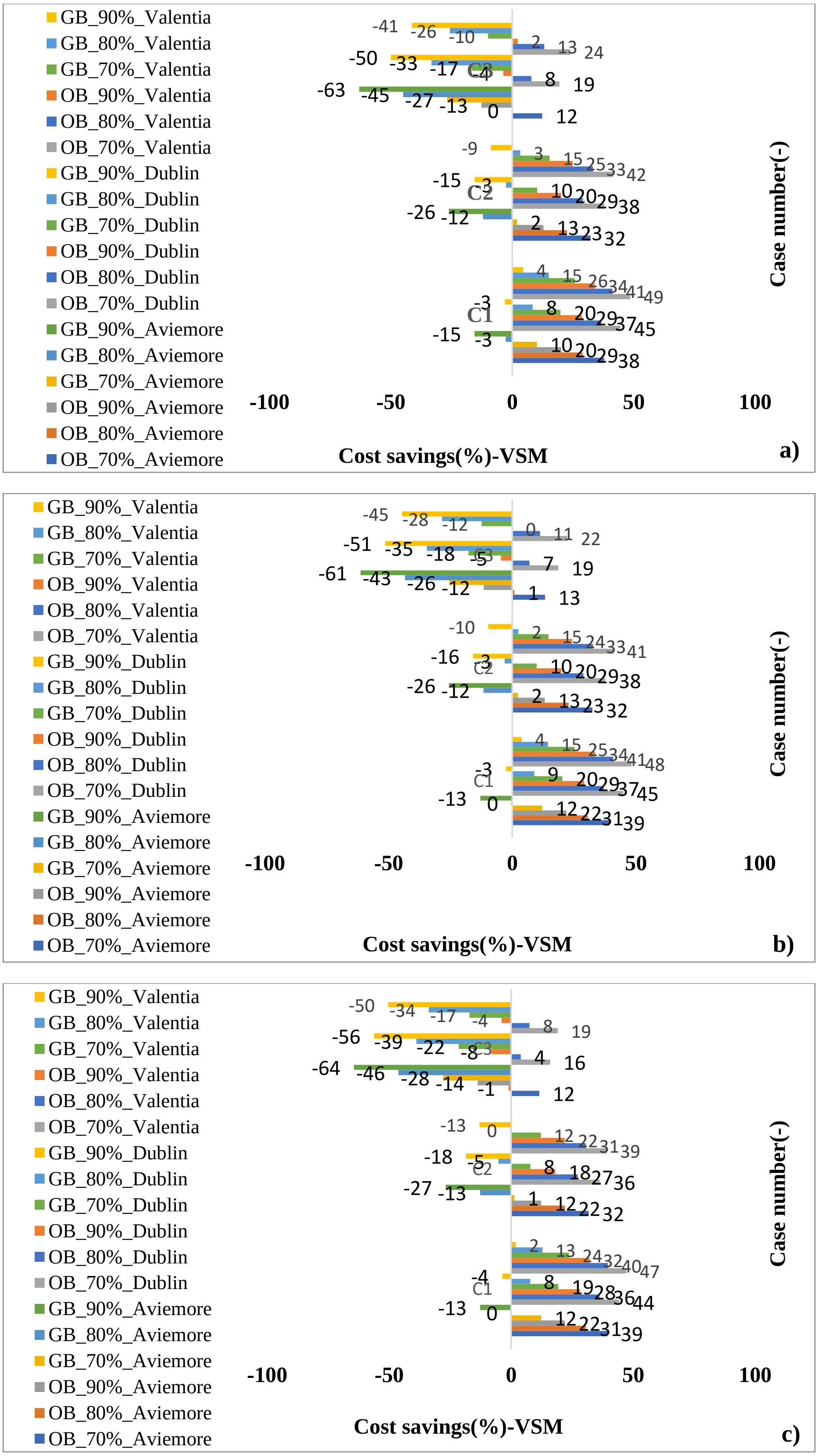

Figure B9. Detached type with age period, a) 1900-1949, b) 1950-1975, c) 1976-1990

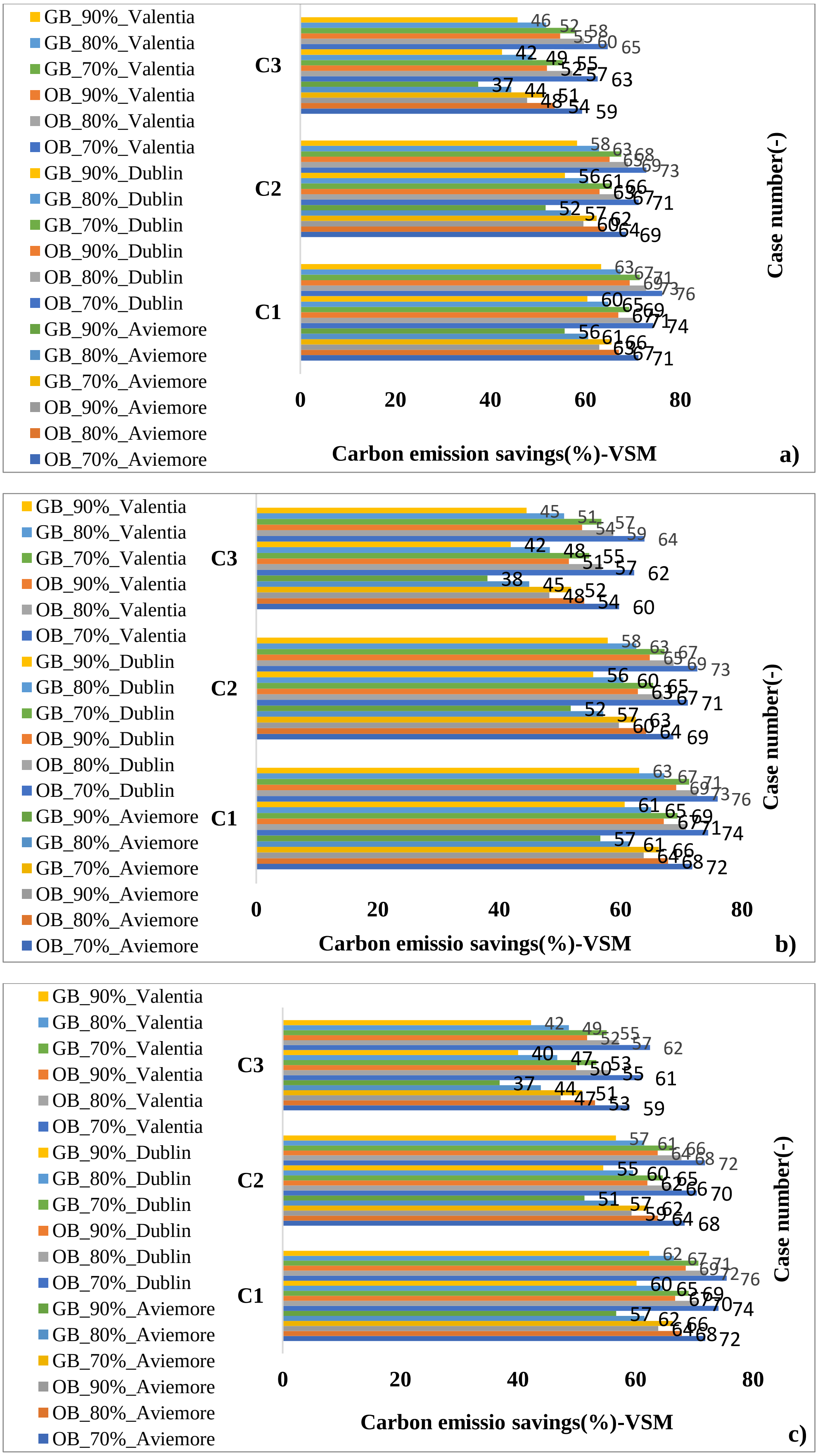

Figure B10. Detached type with age period, a) 1900-1949, b) 1950-1975, c) 1976-1990