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Developing Operational Model for Heat Pumps as Space Heating and Domestic Hot Water Provider

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Abstract

In the United Kingdom, means of meeting domestic heating is being electrified to decarbonise in effort to reduce the greenhouse gases emissions from the burning of natural gas. Therefore, the uptake of heat pumps is on the increase. The operation and working principle of heat pumps must be well understood in the investigations of their impacts on the grid and the grid assets, especially distribution transformers which could be overloaded due to higher peak load demand. This work develops an operational model of heat pumps as combined space heating and domestic hot water provider implemented in MATLAB. The developed operational model of heat pumps is adaptable and repeatable for different input parameters. The developed model is used to generate daily average demand profiles of heat pumps for a typical winter weekday and a typical summer weekday. The generated demand profiles of heat pumps by the developed model compared well with the demand profiles of heat pumps generated from actual field projects which are usually expensive and time-tasking.

Keywords

Heat Pumps, Greenhouse Gases, Space Heating, Domestic Hot Water, Transformer, MATLAB

1. Introduction

Wanton emissions of greenhouse gases (GHGs) into the atmosphere are responsible for global warming. Global warming and its consequent climate change impacts portend serious danger for man and the planet earth. Globally and indeed in the United Kingdom (UK), electricity generation, heat production (domestic heating), and transportation are the major contributing sectors of the GHG emissions [1]. **Figure 1** shows the UK's GHG emissions by sector in 2016. Therefore, how electricity is generated and used must change for the sake of environmental sustainability.

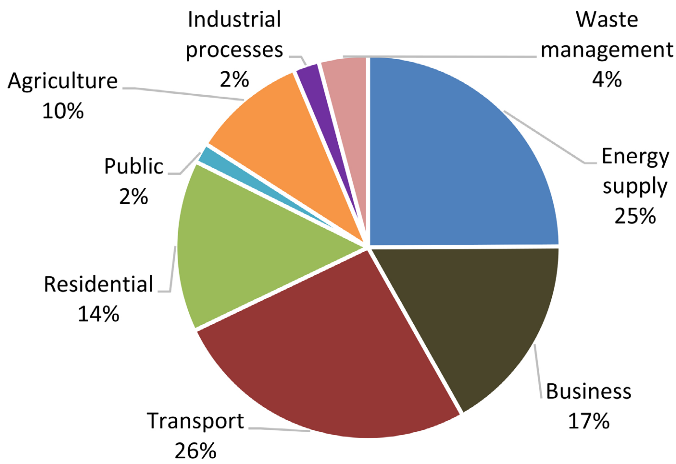

Figure 1. GHG emissions by sector, UK, 2016 [1].

The UK has a policy target of 80% reduction of GHG emissions with respect to the 1990 level by the year 2050 [2]. The realization of the target will involve a transition from fossil fuel based to low carbon-based electricity generation and usage. Decarbonisation of road transport and heat take centre stage considering that in 2016 GHG emissions from transport and domestic sectors accounted for 26% and 14%, respectively, of the total UK GHG emissions as seen in **Figure 1**.

The main source of the GHG emissions in the transport sector is the road transport, in particular passenger cars [1]. In the domestic sector, the use of natural gas for heating is the most significant source GHG emissions in the sector [1]. About 81% of heating demand is basically met by natural gas boilers in the UK [3] [4]. Therefore, there is considerable potential for cutting down on GHG emissions with increasing share of renewable energy sources in the electricity generation, increasing uptake of heat pumps (HPs) for residential heating and electric vehicles (EVs) for road transportation.

Studies have been carried out on the benefit of electrifying the domestic heating demand. Possible energy and GHG emissions savings achievable by using HPs for residential heating in Italy were estimated in [5]. Results highlight that if a fourth of the existing residential buildings are heated by means of air source heat pumps (ASHPs), a saving of about 20% of natural gas can be achieved in 2024, with a corresponding reduction of about 1.7 Mt of GHG emissions [5]. Different technologies for satisfying heat demand in residential buildings were compared in [6] in terms of primary energy consumption. Results showed that electric resistance is practically less favourable than HPs, and the primary energy savings provided by HPs compared to natural gas boilers is about 30% on average [6].

According to [7], replacing 80% of current gas-fired boilers with HPs would enable the UK to meet its target of 80% emissions reduction in the domestic sector by 2050. The caveat according to [7] is that the replacement of gas boilers with HPs must be accompanied by simultaneous decarbonisation of the electricity supply.

In order to meet this target, Government introduced different schemes to promote low-carbon and renewable electricity generation and usage. Amongst the schemes introduced are:

- **Renewable Heat Incentive (RHI)**: This scheme is applicable to both residential and non-residential buildings under domestic RHI and non-domestic RHI. It is a Government financial incentive to encourage landlords and building owners to switch from conventional fossil fuel heating to renewable heating [8]. Eligible technologies include biomass, air-source heat pump, ground-source heat pump and solar thermal for both domestic and non-domestic RHI, and geothermal, biogas and CHP (generating from solid biomass, solid biomass contained in waste, biogas and geotermal) for non-domestic RHI [8].
- **Climate Change Levy (CCL)**: CCL was introduced in 2001 under the Finance Act 2000 [9]. It is a tax on energy supplied from non-renewable sources to non-domestic users. It aims to promote electricity generation from renewable sources.
- **Renewable Obligation**: In this scheme, a mandatory obligation is placed on the UK's electricity suppliers to source a particular proportion of their electricity from renewable sources [10].
- **Feed-in Tariffs (FITs)**: FITs aim to promote rapid and widespread deployment of a range of small-scale renewable and low-carbon distributed generation (DG) of electricity. In the scheme, licensed electricity suppliers are required to pay a generation tariff to these small-scale generators for the electricity generated, whether or not it is exported to the electricity grid [11]. Eligible technologies for this scheme include solar photovoltaic (PV), wind, hydro, and anaerobic digestion up to a maximum installed capacity of 5 MW, and micro combined heat and power (CHP) up to 2 kW [11].
- **Plug-in Car Grant (PICG)**: PICG is a UK Government financial incentive in the form of purchase subsidy for ultra-low emission vehicles. This aims to encourage the uptake of EVs. Vehicles eligible for PICG, must amongst other things, have a zero emission range of at least 70 miles and must emit less than 50 g of CO_2 per kilometre driven [11].

Figure 2 shows the statistics of the total number of different types of renewable heating systems approved under the Domestic RHI scheme between May 2015 and June 2018. The uptake of ASHP under the Domestic RHI scheme increased by about 150% between 2015 and 2018, reaching a total of 32,268 by June 2018. The increasing uptake of HPs as means of domestic heating presents both technical challenges and business opportunities in the electricity industry. Technical challenges arising from the increasing uptake of HPs include higher peak load demand, violation of statutory voltage limits, increased grid losses, and overloading of grid assets especially distribution transformers [12]. While business opportunities include increased electricity generation and a boost in economic activities for the players in the electricity industry.

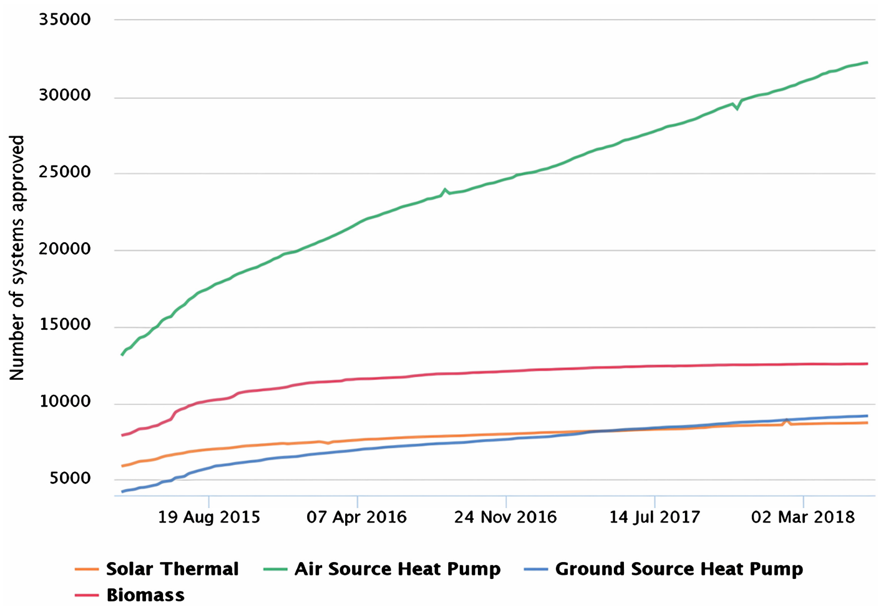

Figure 2. Total number of domestic RHI approved per heating system [13].

Therefore, the operation and working principle of HPs must be well understood in the investigations of their impacts on the grid and the grid assets to adequately tackle the technical challenges and take full advantages of the sprouting business opportunities arising from the increasing uptake of HPs.

Many previous literature on the use of HPs in the UK have used demand profiles either generated from actual field projects (which are expensive and time-tasking) or scenarios based [14]-[16]. This work develops an operational model of heat pumps as combined space heating (SH) and domestic hot water (DHW) provider implemented in MATLAB. The developed operational model of heat pumps is adaptable and repeatable for different input parameters. The developed model is used to generate daily average demand profiles of heat pumps for a typical winter weekday and a typical summer weekday. The generated heat pumps demand profiles by the developed model compared well with heat pumps demand profiles generated from actual field projects which are usually expensive and time-tasking.

2. Heat Pumps: Types and Principle of Operation

Heat pump is a device that transfers heat from a low-temperature source to another location at higher temperature [17]. The heat taken from the low-temperature source is worked upon to raise the temperature before transferring it to another location. In order to transport heat from a heat source to a heat sink, external energy (usually electricity or fuel) is needed to drive the HP. Vapour compression cycle and absorption cycle are two main principles upon which the operation of HPs is based according to [17]. This work focuses on the vapour compression cycle because it is the principle upon which commercially available domestic HPs are operating [18]-[20]. The main components of HPs based on this principle are compressor, expansion valve, evaporator and condenser as shown in Figure 3.

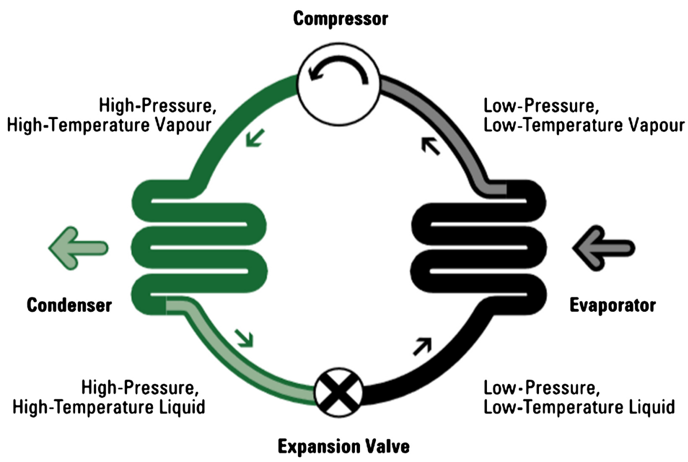

Figure 3. Vapour compression cycle of HP [21].

The working fluid, also called the refrigerant, circulates through the four components. The working fluid, which is a volatile liquid, is usually of lower temperature than the heat source while inside the evaporator. It thus turns into vapour inside the **evaporator** at low pressure and temperature due to the heat received from the heat source. Vapour from the evaporator is then compressed to a higher pressure and temperature inside the **compressor**. The hot vapour then enters the **condenser**, where it condenses and ejects the useful heat. Finally, the high-pressure working fluid is expanded to the evaporator pressure and temperature in the **expansion valve**. The working fluid is returned to its original state and once again enters the evaporator. The compressor is usually driven by an electric motor or sometimes by a combustion engine. HPs with electric motor driven compressors are considered in this work.

The efficiency of an electric compression HP, which is also called coefficient of performance (COP), is defined as the ratio of the heat delivered by the HP and the electricity supplied to the compression [17]. The COP of an ideal HP is inversely related to the difference between the temperature of the heat sink and the heat source. That is the difference between the condensation temperature and the evaporation temperature. While an ideal HP might not exist in reality because of some heat loss to the surroundings, but the COP of a practical HP is always greater than unity and maintains a strong dependence on the difference between the heat sink temperature and heat source temperature. This underscores the importance of deploying an adequate heat source with reasonable temperature level and moderating the heat sink temperature if possible. Presently, modern heat pumps operate at a COP in the range of 4 - 5 at a heat source temperature of 0˚C and 35˚C heat sink temperature [22]. This means that 1-kWh of electricity could be transformed to 4 - 5 kWh of heating. In comparison to modern condensing boilers, HP has better efficiency (always greater 100%) than the condensing boiler and coupled with the fact that a HP produces no direct emissions in its operation makes it the technology of choice in space heating and domestic hot water provision in the quest of cutting down on GHG emissions.

The types and therefore the naming of HPs is based on the following criteria: principle of operation, source of heat energy, medium of heat transfer, and type of compressor. Based on principle of operation, we have vapour compression cycle HPs, absorption cycle HPs. Based on source of heat energy, we have air-source HPs, ground-source HPs, water-source HPs, and hybrid-source HPs which make use of more than one sources of heat. Based on the medium of heat distribution, we have hydronic HPs and air-based HPs. Finally, based on the type of compressor, we have single-speed HPs, two-speed HPs, and variable-speed HPs whose compressors are capable of modulating their operational speeds to adjust capacity in effort to run at the best speed to meet the demand [23] [24].

The three main factors for consideration in the choice of HP for installation are technical, economic and logistic factors. Technically, a good choice of HP for installation is that whose heat source is abundantly available with moderate level of temperature, especially during the heating season. In terms of economics, a good choice of HP is that with moderate investment, installation and operational costs. Logistic factors include nearness of the heat source to the point of installation, obtaining installation permit from the authorities and ease of retrofitting into existing building. In this work, the full description of the type of HP considered based on technical, economic and logistic factors is the Air-to-Water, Variable-speed, electric-compression HP.

3. Model Formulation

The operation of variable speed Air Source Heat Pump (ASHP) providing both space heating (SH) and domestic hot water (DHW) is modelled. The operation of variable speed ASHP is dynamic in that the heat output and the coefficient of performance (COP) of the HP vary with the heating demand of the building in which it is installed and the external temperature respectively. **Figure 4**, adapted from [25], illustrates the block diagram of HP system configuration modelled in this work. The HP system configuration is such that the provision for DHW and SH are mutually exclusive. The DHW provision has priority control in the event of DHW demand and SH demand occurring at the same time. In this event, the DHW demand is met first and then the SH demand. This design configuration is the most common in the market [25]-[27].

3.1. Model Formulation of HP Operation in SH Mode

The formula, as adapted from [28], for the internal air temperature of the building after a time slot t is given as by Equation (1):

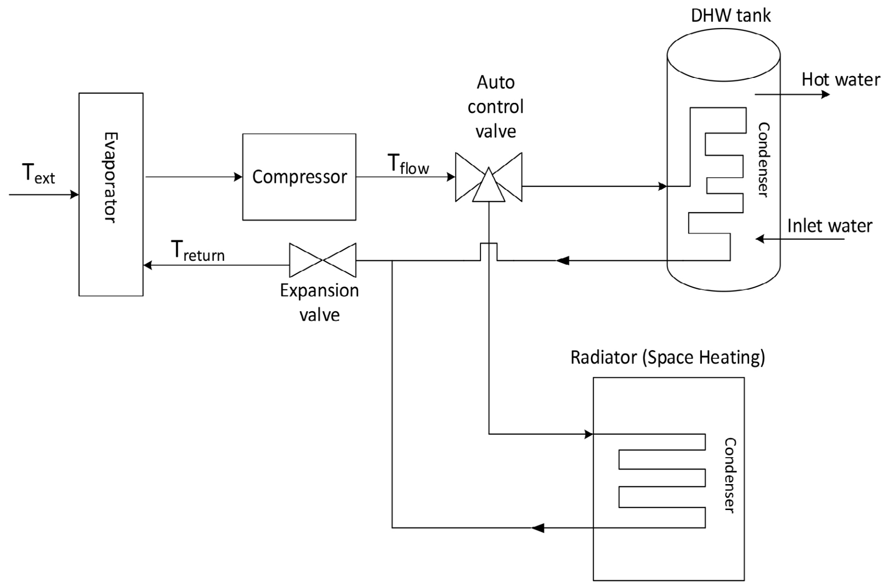

Figure 4. Block diagram of HP system configuration (adapted from [25]).

$$T_{int(t+1)} = T_{int(t)} - \left(Q_{loss(t)} - Q_{gain(t)} - HP_{SH(t)} \cdot y_{(t)}\right)\frac{\Delta t}{\Delta q} \tag{1}$$

where:

$T_{int(t+1)}$ is the internal air temperature (˚C or K) of the building after a time slot t.

$T_{int(t)}$ is the internal air temperature (˚C or K) of the building in time slot t.

$Q_{loss(t)}$ and $Q_{gain(t)}$ are the heat loss (W) and heat gain (W) of the building in time slot t.

$HP_{SH(t)}$ is the heat output (W) of the HP in SH mode in time slot t.

$y_{(t)}$ is binary variable which determines the operational status (ON = 1 or OFF = 0) of the HP in SH mode in time slot t.

Δt is the duration of the time slot in (s).

Δq is the energy needed to change the internal air temperature of the building by 1˚C (J/˚C).

The heat loss of a building is the sum of heat loss through the fabric of the building (floors, walls, roof, windows and doors) and the heat loss due to ventilation/infiltration [29]. The heat loss, $Q_{loss(t)}$, of the building in time slot t is given by Equation (2):

$$Q_{loss(t)} = \left(\Sigma\left(AU\right) + 0.3N_{ac}V\right) \times \left(T_{int(t)} - T_{ext(t)}\right) \tag{2}$$

where:

U is thermal transmittance (W/m²·K).

A is surface area through which heat transfer occurs (m²).

N_{ac} is the number of air changes per hour (ac/h).

V is the volume of the building (m³).

$T_{int(t)}$ is the internal air temperature (˚C or K) of the building in time slot t.

$T_{ext(t)}$ is external air temperature (˚C or K) in time slot t.

The heat gain, $Q_{gain(t)}$, of the building in time slot t is given by Equation (3):

$$Q_{gain(t)} = \left(Q_p \times N_p\right) + \left(A_{SW} \times SHGC \times S_{rad(t)}\right) \tag{3}$$

where:

Q_p is heat gain from one person (W).

N_p is number of occupants.

A_{SW} is area of window facing south (m^2).

$SHGC$ is solar heat gain coefficient of window.

$S_{rad(t)}$ is solar irradiance (W/m^2) in time slot t.

The energy needed to change the internal air temperature of the building is given by Equation (4):

$$\Delta q = C_{air} \times \rho_{air} \times V \tag{4}$$

where:

C_{air} is specific heat capacity of air for typical room condition (J/kg·˚C).

ρ_{air} is density of air (kg/m^3).

V is the volume of the building (m^3).

The operational status, $y_{(t)}$, of the HP in SH mode is represented by Equation (5):

$$y_{(t)} = \begin{cases} 1 = \text{ON}, & T_{int(t)} < T_{set} - T_{sg} \\ 0 = \text{OFF}, & T_{int(t)} > T_{set} + T_{sg} \\ y_{(t-1)}, & T_{set} - T_{sg} \le T_{int(t)} \le T_{set} + T_{sg} \end{cases} \tag{5}$$

where:

$T_{int(t)}$ is the internal air temperature (˚C or K) of the building in time slot t.

T_{set} is the set-point temperature of the internal air (˚C or K).

T_{sg} is the swing temperature (˚C or K).

$y_{(t-1)}$ is the operational status of the HP in previous time slot.

In Equation (5), T_{set} is the desired internal air temperature and therefore the thermostat set-point. If the actual internal air temperature, $T_{int(t)}$, drops below the temperature lower limit, T_{low}, which is the difference between T_{set} and T_{sg}, then the HP is switched ON to raise the internal air temperature. Conversely, when the internal air temperature rises above the temperature upper limit T_{up}, which is the sum of T_{set} and T_{sg}, the HP switches OFF. However, the operational status of the HP remains unchanged if the internal air temperature is between T_{low} and T_{up}.

Ignoring losses, the heat output of the HP in SH mode is equal to the radiator output which is also equal to the condenser output. That is:

$$HP_{SH(t)} = Q_{condenser(t)} = Q_{radiator(t)} \tag{6}$$

where:

$HP_{SH(t)}$ is the heat output (W) of the HP in SH mode in time slot t.

$Q_{condenser(t)}$ is the condenser heat output (W) in time slot t

$Q_{radiator(t)}$ is the radiator heat output (W) in time slot t.

The heat flux inside the condenser of the HP can be expressed as:

$$Q_{condenser(t)} = mc\left(T_{flow} - T_{return(t)}\right) \tag{7}$$

where:

m is the mass flow rate (kg/s) of water.

c is the specific heat capacity (J/kg·˚C) of water.

T_{flow} is the operating temperature (˚C or K) of the working fluid reaching the condenser.

$T_{return(t)}$ is the temperature (˚C or K) of the working fluid leaving the condenser.

The heat output of the radiator can be expressed as:

$$Q_{radiator(t)} = U_{rad} A_{rad}\left(T_{rad(t)} - T_{int(t)}\right) \tag{8}$$

where:

U_{rad} is the heat transmission coefficient (W/m²·K) of the radiator.

A_{rad} is the surface area (m²) of the radiator.

$T_{int(t)}$ is the internal air temperature (˚C or K) of the building in time slot t.

$T_{rad(t)}$ is the radiator temperature (˚C or K).

The radiator temperature, $T_{rad(t)}$, is the average of the temperature of the working fluid reaching the condenser (T_{flow}) and the temperature of the working fluid leaving the condenser ($T_{return(t)}$). That is:

$$T_{rad(t)} = \frac{T_{flow} + T_{return(t)}}{2} \tag{9}$$

From Equations (6) to (9) the return temperature, $T_{return(t)}$, can be expressed as:

$$T_{return(t)} = \frac{T_{flow}\left(2mc - U_{rad} A_{rad}\right) + 2U_{rad} A_{rad} T_{int(t)}}{U_{rad} A_{rad} + 2mc} \tag{10}$$

Based on test data, from the Heat Pump Test Centre WPZ, of 30 different models of ASHPs [22], the expression for the Coefficient of Performance (COP) of HP can be deduced from the plot of COP against "$T_{return} - T_{ext}$" with a coefficient of determination (R^2 value) of 0.9797 by Equation (11):

$$COP_{(t)} = 7.90471\mathrm{e}^{-0.024\left(T_{return(t)} - T_{ext(t)}\right)} \tag{11}$$

where:

$COP_{(t)}$ is the coefficient of performance of the HP at time slot t.

$T_{ext(t)}$ is the external air temperature (˚C or K) at time slot t.

The COP-curve, which is here defined as the plot of COP against "$T_{return} - T_{ext}$" derived from the test data, is shown in **Figure 5**. In **Figure 5**, there are 9 test points and the COP at a point is the average of COPs of 30 ASHPs at that point.

The actual electrical input, $P_{SH_{elect(t)}}$ (W), for the operation of the HP in SH mode is therefore given by Equation (12):

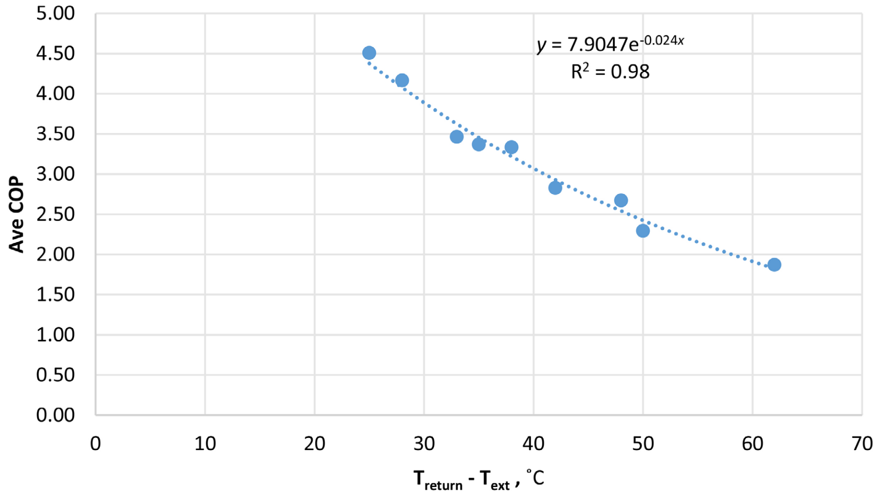

Figure 5. The *COP*-curve adapted from test data at HP Test Centre WPZ [22].

$$P_{SH_{elect(t)}} = \frac{HP_{SH(t)}}{COP_{(t)}} \tag{12}$$

Two temperature regimes were used in the modelling. The set-point temperature, T_{set}, of the HP between 00:00 hours and 10:30 hours is 18˚C with a swing temperature, T_{sg}, of 2˚C. Whereas T_{set} between 11:00 hours and 23:30 hours is 20.5˚C with a T_{sg} of 3˚C.

3.2. Model Formulation of HP Operation in DHW Mode Equations

In the model formulation, single-node state is assumed since there is no occurrence of draw event large enough to trigger the transition from single-node state into two-node state. A hot water tank remains in single-node state and only changes into two-node state when a considerable volume of water is drawn in a usage event which occurs in a short interval of time [30]. In single-node state, the water in the tank is considered as a single mass of body with the heat and temperature of the water uniformly distributed. Therefore, the water in the tank is not stratified after a draw event into upper layer warm water and lower layer cold water from the inlet that replaces the drawn water. **Figure 6** shows the DHW tank in single-node state as modelled in this work.

The temperature, $T_{(t)}$, of the water leaving the tank is the average temperature of the hot water inside the tank. The tank is refilled with inlet water at temperature, T_{in}, to replace the drawn water. The inlet water mixes with the hot water inside the tank and a new average temperature, $T_{(t+1)}$, is formed for the next water draw event. The heat (W) available inside the tank after a water draw event in time slot t can be expressed in terms of heat balance equation as follows:

$$Q_{(t+1)} = Q_{(t)} - Q_{use(t)} - Q_{aml(t)} + HP_{DHW(t)} \cdot z_{(t)} \tag{13}$$

where:

$Q_{(t+1)}$ is the heat (W) remaining after a water draw event.

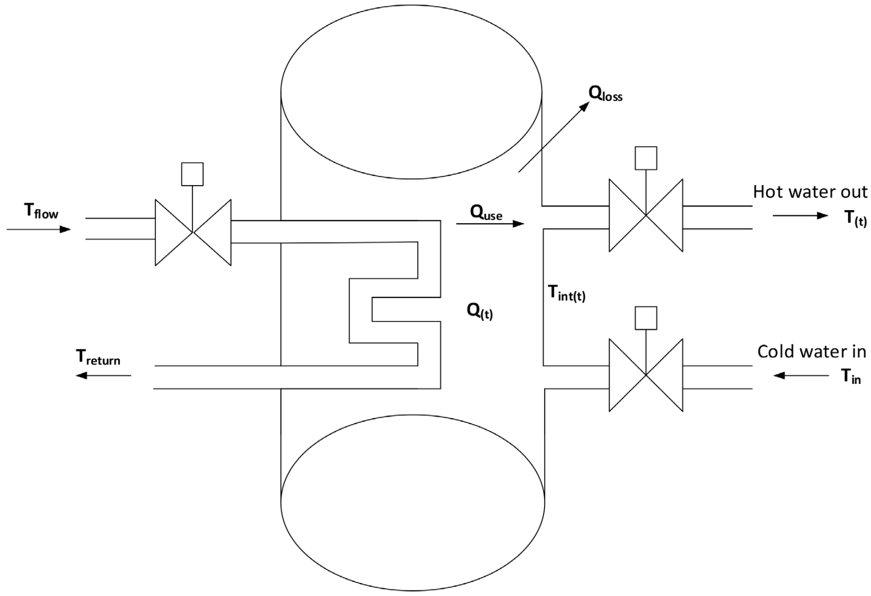

Figure 6. DHW tank in single-node state (adapted from [30]).

$Q_{(t)}$ is the heat (W) available before the water draw event.

$Q_{use(t)}$ is the heat (W) loss due to the water draw event.

$Q_{aml(t)}$ is the heat (W) loss to the ambience due to heat dissipation from the tank the to the environment.

$HP_{DHW(t)}$ is the heat output (W) of the HP in DHW mode in time slot t.

$z_{(t)}$ is binary variable which determines the operational status (ON = 1 or OFF = 0) of the HP in DHW mode in time slot t.

The heat balance equation in (13) can be written in terms of volume and change in temperature as follows:

$$\begin{aligned}\frac{Vc\left(T_{(t+1)}-T_{in}\right)}{60t}&=\frac{Vc\left(T_{(t)}-T_{in}\right)}{60t}-\frac{V_{use(t)}c\left(T_{(t+1)}-T_{in}\right)}{60t}\\&\quad -U_{ta}A_{ta}\left(T_{(t)}-T_{int(t)}\right)+\frac{Vc\left(T_{flow}-T_{return(t)}\right)}{60t}\cdot z_{(t)}\end{aligned} \tag{14}$$

where:

V is the volume (l) of the tank.

$V_{use(t)}$ is the volume (l) of the hot water used in time slot t.

$T_{(t)}$ is the temperature (˚C or K) of hot water inside the tank in time slot t.

$T_{(t)}$ is also equal to the return temperature, $T_{return(t)}$, of the working fluid.

$T_{(t+1)}$ is the temperature (˚C or K) of hot water inside the tank after the water draw event.

T_{in} is the temperature (˚C or K) of the inlet cold water.

U_{ta} is the heat transmission coefficient (W/m²·K) of the tank.

A_{ta} is the surface area (m²) of the tank.

c is the specific heat capacity of water in kJ/kg·˚C *i.e.* 4.184 kJ/kg·˚C.

$T_{int(t)}$ is the internal air (˚C or K) of the building in time slot t.

T_{flow} is the operating temperature (˚C or K) of the working fluid.

t is the duration of the time slot in minutes.

The operational status, $z_{(t)}$, of the HP in DHW mode is represented as follows:

$$z_{(t)} = \begin{cases} 1 = \text{ON}, & T_{(t)} < T_{set(W)} - T_{sg(W)} \\ 0 = \text{OFF}, & T_{(t)} > T_{set(W)} + T_{sg(W)} \\ z_{(t-1)}, & T_{set(W)} - T_{sg(W)} \le T_{(t)} \le T_{set(W)} + T_{sg(W)} \end{cases} \tag{15}$$

where:

$T_{(t)}$ is the temperature (˚C or K) of hot water inside the tank time slot t.

$T_{set(W)}$ is the set-point temperature (˚C or K) of hot water inside the tank.

T_{sg} is the swing temperature (˚C or K).

$z_{(t-1)}$ is the operational status of the HP in previous time slot.

Substituting for constant and solving for $T_{(t+1)}$ in Equation (14) yields:

$$T_{(t+1)} = \frac{VT_{(t)} + V_{use(t)}T_{in} - 0.0143tU_{ta}A_{ta}\left(T_{(t)} - T_{int(t)}\right) + V\left(T_{flow} - T_{(t)}\right)\cdot z_{(t)}}{V + V_{use(t)}} \tag{16}$$

The set-point temperature, $T_{set(W)}$, of the HP for DHW is 50˚C with a swing temperature, $T_{sg(W)}$, of 5˚C. The hot water set-point temperature and the swing temperature are such that will prevent the growth of Legionella bacteria inside the tank. Legionella bacteria mostly thrives in the temperature range between 20˚C and 45˚C [31].

The COP of the HP while working in DHW mode is as expressed in Equation (11) with $T_{return(t)}$ substituted by $T_{(t)}$. The actual electrical input, $P_{DWH_{elect(t)}}(W)$, for the operation of the HP in DHW mode is given by:

$$P_{DWH_{elect(t)}} = \frac{HP_{DHW(t)}}{COP_{(t)}} \tag{17}$$

4. Model Implementation and Results

The model is tested on a 6-kW heat output capacity, variable-speed ASHP with a COP of 2.7 at test condition A-7/W35 and R407C as refrigerant [22]. The HP operational model as SH and DHW provider is implemented in MATLAB for a typical winter weekday and a typical summer weekday. **Figure 7** shows the block diagram of the implementation process of the model. Inputs to the model in the SH mode are time series external air temperature, time series solar radiation, thermostat set-point for the desired internal air temperature, SH swing temperature and the time series internal air temperature which is fed back from the output. These input parameters interact with intrinsic properties of the building (such as size of building, areas of building fabrics and U-values of building fabrics), number of occupants and the COP-curve of the HP to produce outputs in the SH mode.

In the DHW mode, the inputs are time series external air temperature, time series internal air temperature, temperature of inlet water, thermostat set-point for the desired hot water temperature, DHW swing temperature, time series water

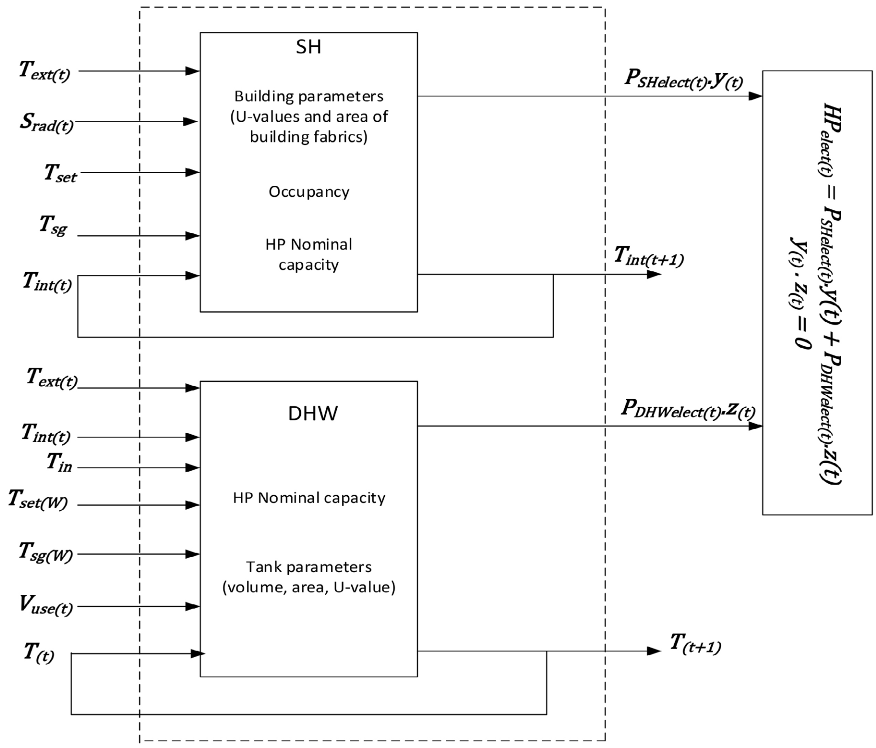

Figure 7. Block diagram of implementation process of HP operation.

usage profile and the hot water temperature which is fed back from the output. The tank parameters like volume, surface area and heat transmission coefficient interact with the input parameters to produce outputs in the DHW mode.

Input data about parameters of buildings used in the model are available in Appendix 1. Parameters of DHW tank are provided in Appendix 2. Weather data and water draw events are available in Appendix 3. Parameters of radiator are provided in Appendix 4.

The outputs of the model depend on the mode of the HP (SH mode or DHW mode) which is active in a time slot. The outputs of the model in SH mode are internal air temperature and the electricity consumption of the HP in that mode while the outputs in DHW mode are hot water temperature and the electricity consumption of the HP in that mode. The electricity consumption of the HP in a time slot t is given by Equation (18):

$$HP_{elect(t)} = P_{SH_{elect(t)}} \cdot y_{(t)} + P_{DWH_{elect(t)}} \cdot z_{(t)} \tag{18}$$

Equation (19) ensures that the HP can only operate either in SH mode or DHW mode at a given time slot.

$$y_{(t)} \times z_{(t)} = 0 \tag{19}$$

The model is run with 100 buildings. In order to achieve diversity in the operation of the HPs in different buildings, the following input parameters of the model are randomized: building size, U-values of building fabrics, number of occupants, solar heat gain coefficient (SHGC) of windows, number of air change, initial internal air temperature and initial hot water temperature.

Figure 8 presents the outcome of the developed model—the average electricity demand profiles of HPs on a typical winter weekday and a typical summer weekday in the UK. Peaks are observed at about 7:30 and 9:30 in the morning for both typical winter weekday and typical summer weekday average electricity demand of the HPs.

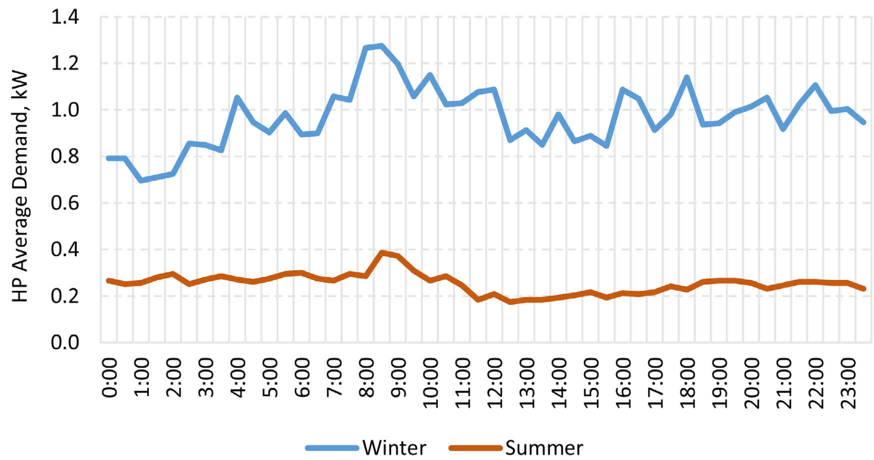

Figure 8. HP daily average demand.

The morning peaks in the demand profiles of HPs on both typical winter weekday and typical summer weekday are due to the increased use of hot water as people are getting prepared for the day's activities. The morning peak in the summer (0.4 kW) is not as pronounced as in the winter (1.3 kW) because less heat energy and hot water are required in the summer.

Model Validation

To ensure validity of the developed HP operational model, empirical data from credible sources were used as inputs to run the model. Decision on the number of occupants per household was based on [32]. Average daily DHW requirement of household in litres/day was estimated in line with technical guidelines from [33] and it is given by Equation (20):

$$HH_{daily_{DHW}} = 25N_p + 36 \tag{20}$$

where:

$HH_{daily_{DHW}}$ is the average daily household DHW requirement in litres.

N_p is the number of occupants in the household.

Normalized DHW tapping profile from [34] was used to estimate the actual DHW draw at any time of the day. Figure 9 illustrates the normalized DHW tapping profile. Data on geometric and constructional characteristics of the hot water tank came from [35]. Data about buildings parameters which consist of building type and size and U-values of building elements were from [36] and [37] respectively. Weather data were from [38].

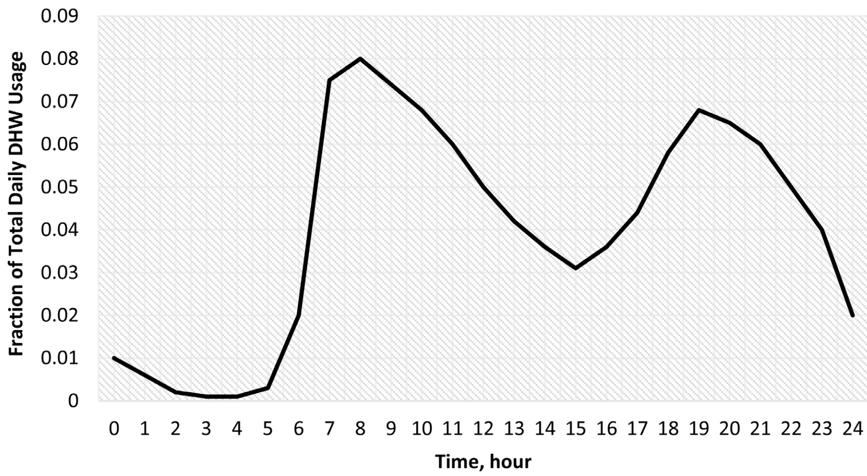

Figure 9. Normalised DHW tapping profile [34].

The model outputs, typical winter weekday and typical summer weekday average electricity demand of HPs expressed in half-hourly intervals as seen in **Figure 8**, were compared with the measured daily average electricity demand of HPs, shown in **Figure 10**, in the Carbon, Control and Comfort (CCC) project [14]. The comparison between the model outputs and the actual measured outputs of the CCC project showed close similarity in trends and kW values of the HPs daily average electricity demand profile. The outputs of both the model and the actual field project have morning demand spikes around 8.30 hours of 1.3 kW and 0.4 kW in both their winter and summer average electricity demand profiles respectively. This gives reasonable credence to the usefulness of the developed model. **Figure 10** is the screenshot from CCC Project of average HP demand. The midnight peak observed in **Figure 10** but not in **Figure 8** is due to the fact that the HPs in the CCC project operate a weekly pasteurization cycle (raising the DHW temperature above 60˚C to kill Legionella bacteria) which always takes place at midnight [14].

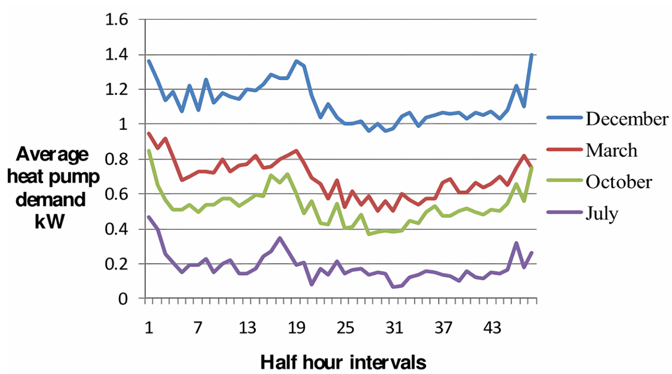

Figure 10. Screenshot from CCC Project of average HP demand [14].

5. Conclusion

This work developed an operational model of heat pumps as combined space heating and domestic hot water provider implemented in MATLAB. The developed model is used to generate daily average demand profiles of heat pumps for a typical winter weekday and a typical summer weekday in the UK. The generated demand profiles of the HPs by the developed model compared well with the demand profiles of HPs generated from actual field projects which are usually expensive and time-tasking. The developed operational model of heat pumps is adaptable and repeatable for different input parameters. The developed model will find use in further studies in the investigations of the impacts of HPs on the grid and the grid assets to adequately tackle the technical challenges and exploit the opportunities arising from the increasing uptake of HPs.

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Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

References

[1] Department for Business Energy & Industrial Strategy (DBEIS) (2018) 2016 UK Greenhouse Gas Emissions, Final Figures. Statistical Release: National Statistics, 1-51.

[2] Committee on Climate Change (CCC) (2015) The Fifth Carbon Budget: The Next Step towards a Low-Carbon Economy.

[3] Connor, P.M., Xie, L., Lowes, R., Britton, J. and Richardson, T. (2015) The Development of Renewable Heating Policy in the United Kingdom. *Renewable Energy*, **75**, 733-744. https://doi.org/10.1016/j.renene.2014.10.056

[4] Chaudry, M., Abeysekera, M., Hosseini, S.H.R., Jenkins, N. and Wu, J. (2015) Uncertainties in Decarbonising Heat in the UK. *Energy Policy*, **87**, 623-640. https://doi.org/10.1016/j.enpol.2015.07.019

[5] Bianco, V., Scarpa, F. and Tagliafico, L.A. (2017) Estimation of Primary Energy Savings by Using Heat Pumps for Heating Purposes in the Residential Sector. *Applied Thermal Engineering*, **114**, 938-947. https://doi.org/10.1016/j.applthermaleng.2016.12.058

[6] Jarre, M., Noussan, M. and Simonetti, M. (2018) Primary Energy Consumption of Heat Pumps in High Renewable Share Electricity Mixes. *Energy Conversion and Management*, **171**, 1339-1351. https://doi.org/10.1016/j.enconman.2018.06.067

[7] Gupta, R. and Irving, R. (2014) Possible Effects of Future Domestic Heat Pump Installations on the UK Energy Supply. *Energy and Buildings*, **84**, 94-110. https://doi.org/10.1016/j.enbuild.2014.07.076

[8] Department for Business, Energy & Industrial Strategy (DBEIS) (2016) The Renewable Heat Incentive: A Reformed Scheme.

[9] Parliament of UK (2000) Finance Act 2000: Part II Climate Change Levy. Statute Law

Database, 622.

[10] Office of Gas and Electrcity Market (OFGEM) (2018) Guidance for Generators: Co-Location of Electricity Storage Facilities with Renewable Generation Supported under the Renewables Obligation or Feed-In Tariff Schemes (Version 2). 1-36.

[11] Office of Gas and Electricity Market (OFGEM) (2018) Feed-In Tariffs: Guidance for Licensed Electricity Suppliers (Version 10). 1-132.

[12] Oliyide, R.O. (2019) Load Management of Electric Vehicles and Heat Pumps in Emerging Electricity System. Master's Thesis, Cardiff University.

[13] (2018) Office of Gas and Electricity Markets. OFGEM, Public Reports and Data: Domestic RHI. https://www.ofgem.gov.uk/environmental-programmes/domestic-rhi/contacts-guidance-and-resources/public-reports-and-data-domestic-rhi

[14] Boait, P. and Stafford, A. (2011) Electrical Load Characteristics of Domestic Heat Pumps and Scope for Demand Side Management. *International Conference on Electricity Distribution*, **21**, 6-9.

[15] Kreuder, L. and Spataru, C. (2015) Assessing Demand Response with Heat Pumps for Efficient Grid Operation in Smart Grids. *Sustainable Cities and Society*, **19**, 136-143. https://doi.org/10.1016/j.scs.2015.07.011

[16] Teng, F., Aunedi, M., Pudjianto, D. and Strbac, G. (2015) Benefits of Demand-Side Response in Providing Frequency Response Service in the Future GB Power System. *Frontiers in Energy Research*, **3**, Article 36. https://doi.org/10.3389/fenrg.2015.00036

[17] International Energy Agency (IEA) (2011) Heat Pumps for Cooling and Heating Subtask 5, Report No. 3.

[18] Nowak, T. (2018) Heat Pumps: Integrating Technologies to Decarbonise Heating and Cooling. European Copper Institute, 1-86.

[19] Baster, M.E. (2011) Modelling the performance of Air Source Heat Pump Systems. Master's Thesis, University of Strathclyde.

[20] Forsén, M. (2005) Heat Pumps: Technology and Environmental Impact. European Heat Pump Association Raphaela (EHPA), (Part 1), 1-120.

[21] Natural Resource Canada's Office of Energy Efficiency (NRC) (2004) Heating and Cooling with a Heat Pump.

[22] Wärmepumpen-Testzentrum, W.P.Z. (2013) Information Sheet of the Heat Pump Test Center WPZ. *WPZ—Bulletin*, **41**, 1-24.

[23] Domitrovic, R. (2014) Variable-Speed Heat Pumps for Energy Efficiency and Demand Response. Electric Power Research Institute (EPRI), 1-84.

[24] Del Col, D., Azzolin, M., Benassi, G. and Mantovan, M. (2015) Energy Efficiency in a Ground Source Heat Pump with Variable Speed Drives. *Energy and Buildings*, **91**, 105-114. https://doi.org/10.1016/j.enbuild.2014.12.048

[25] Wemhoener, C. (2011) Final Report Annex 32 Part 3 Economical Heating and Cooling Systems for Low Energy Houses. IEA Heat Pump Program, 1-72.

[26] Wemhöner, C., *et al.* (2007) Calculation Method for the Seasonal Performance of Heat Pump Compact Units and Validation. ETDEWEB.

[27] Wemhöner, C. and Afjei, T. (2003) Seasonal Performance Calculation for Residential Heat Pumps with Combined Space Heating and Hot Water Production (FHBB Method). Swiss Federal Office of Energy.

[28] Shao, S., Pipattanasomporn, M. and Rahman, S. (2013) Development of Physical-Based Demand Response-Enabled Residential Load Models. *IEEE Transactions on*

Power Systems, **28**, 607-614. https://doi.org/10.1109/tpwrs.2012.2208232

[29] Hall, F. and Greeno, R. (2011) Building Service Handbook. 6th Edition, Elsevier Limited.

[30] Nel, P.J.C., Booysen, M.J. and van der Merwe, B. (2018) A Computationally Inexpensive Energy Model for Horizontal Electric Water Heaters with Scheduling. *IEEE Transactions on Smart Grid*, **9**, 48-56. https://doi.org/10.1109/tsg.2016.2544882

[31] Health and Safety Executive (HSE) (2014) Legionnaires' Disease Part 2: The Control of Legionella Bacteria in Hot and Cold Water Systems. HSE Books.

[32] Office for National Statistics (ONS) (2017) Families and Households: 2017. Statistical bulletin: People, Population and Community, 1-16.

[33] Henderson, J. and Hart, J. (2015) BREDEM 2012—A Technical Description of the BRE Domestic Energy Model, Version 1. BRE.

[34] Hendron, R. and Burch, J. (2008) Development of Standardized Domestic Hot Water Event Schedules for Residential Buildings. *ASME* 2007 *Energy Sustainability Conference*, Long Beach, 27-30 July 2007, 253-272.

[35] Kingspan (UK) (2016) Vented Hot Water Cylinders, Albion cylinder: Technical and Installation Documents.
https://www.kingspan.com/gb/en-gb/products/hot-water-cylinders/vented-hot-water-cylinders

[36] Department for Communities and Local Government (DCLG) (2015) Housing Stock Profile, 2014-15.

[37] HM Government (2010) The Building Regulations 2010: Conservation of Fuel and Power in Existing Dwellings—Approved Document Part L1B.

[38] UK Met Office (2016) Archive Services—Met Office.
https://www.metoffice.gov.uk/learning/library/archive/services

[39] Department for Communities and Local Government (DCLG) (2015) Technical Housing Standards—Nationally Described Space Standard. DCLG, 1-5.

[40] Stein, M. (2008) When Technology Fails: A Manual for Self-Reliance, Sustainability, and Surviving the Long Emergency. Chelsea Green Publishing.

[41] (2019) Roofing Calculator.
https://www.calculator.net/roofing-calculator.html

[42] The Chartered Institution of Building Services Engineers (2008) London, CIBSE Concise Handbook. 3rd Edition, Chartered Institution of Building Services Engineers.

[43] Ainsworth, B.E., Haskell, W.L., Whitt, M.C., Irwin, M.L., Swartz, A.M., Strath, S.J., *et al.* (2000) Compendium of Physical Activities: An Update of Activity Codes and MET Intensities. *Medicine & Science in Sports & Exercise*, **32**, S498-S516.
https://doi.org/10.1097/00005768-200009001-00009

[44] (2019) Met—Metabolic Rate.
https://www.engineeringtoolbox.com/met-metabolic-rate-d_733.html

Appendices

Appendix 1: Parameters of Building

Parameters	Values
Number of bedrooms	Randomised between 2 and 3 bedrooms, since the mean number of bedrooms for all household in GB is 2.8 [36].
Floor area (A_floor)	Randomised between 90 m^2 and 110 m^2. Range of standard floor area for 2/3 bedroom house [39].
Height of building	4.6 m (assuming two storey building). Minimum floor to ceiling height is 2.3 m [39].
Area of door (A_door)	3.02 m^2 (assuming 2 external doors each of size 1981 mm by 762 mm).
Volume of building (V_house)	Calculated from "Floor area" and "height of building".
Number of occupants (N_person)	Randomised between 2 and 3, since the mean number of persons per household is 3 [36].
External wall area (A_wall)	Calculated from the "Floor area" assuming a wall thickness of 362 mm.
Overall area of windows (A_window)	Calculated from "Area of wall", assuming 15% wall-to-window ratio [40].
Overall area of south-facing window (A_sth_window)	Calculated from "Area of wall", assuming 12% wall-to-window ratio [40].
Net external wall area (Net_A_wall)	Calculated from "Area of wall", "Area of door" and "Area of window".
Area of roof (A_wall)	Calculated from "Area of floor", assuming 45 degrees pitch angle [41].
Number of air change (N_air)	Randomised between 0.5 and 1.0 air changes/hr, standard for bedroom and living room respectively [42].
U-value of floor (U_floor)	Randomised between 0.22 and 0.45 $W/m^2 \cdot K$ [37] [42].
U-value of wall (U_wall)	Randomised between 0.28 and 0.45 $W/m^2 \cdot K$ [37] [42].
U-value of roof (U_roof)	Randomised between 0.18 and 0.25 $W/m^2 \cdot K$ [37] [42].
U-value of door (U_door)	Randomised between 1.8 and 2.0 $W/m^2 \cdot K$ [37] [42].
U-value of window (U_window)	Randomised between 1.6 and 2.0 $W/m^2 \cdot K$ [37] [42].
SHGC	Randomised between 0.45 and 0.67 [42].
Heat gain per person (H_person)	93.5 W. Calculated from the average of heat emission from reclining/sleeping (83 W) and seated/relaxed (104 W) [43] [44].
Initial internal space temperature (T_room_initial)	Randomised between 14˚C and 22˚C.

Appendix 2: Parameters of Tank

Parameters	Values
Volume of tank (V_tank)	150 litres [35].
Surface area of tank (A_tank)	2.36 m^2 [35].
Thermal transmittance coefficient of tank (U_tank)	1.13 $W/m^2 \cdot K$ [35].
Initial DHW temperature (T_DHW_initial)	Randomised between 47˚C and 60˚C.

Appendix 3: Temperature, Solar Radiation and Water Draw Events

Time	Temperature (˚C)		Solar radiation (W/m^2)		Water draw (litres)
	summer	winter	summer	winter	
00:00	18.4	5.4	0	0	1.20
00:30	17.5	3.4	0	0	0.80
01:00	16.7	1.4	0	0	0.50
01:30	16.2	−0.2	0	0	0.30
02:00	15.7	−2.3	0	0	0.10
02:30	16.2	−3.6	0	0	0.08
03:00	16.7	−5.0	0	0	0.05
03:30	16.6	−5.7	0	0	0.05
04:00	16.6	−6.4	0	0	0.05
04:30	16.5	−6.7	0	0	0.08
05:00	16.4	−7.0	0	0	0.10
05:30	16.6	−7.2	0	0	0.60
06:00	16.9	−7.3	40	0	1.10
06:30	17.4	−7.0	130	0	2.63
07:00	17.8	−6.6	210	0	4.15
07:30	18.5	−6.3	310	0	4.30
08:00	19.2	−6.0	410	0	4.45
08:30	20.4	−5.6	510	0	4.40
09:00	21.6	−5.1	610	30	4.35
09:30	23.4	−4.1	660	70	4.08
10:00	25.1	−3.0	700	100	3.80
10:30	26.4	−2.1	720	80	3.58
11:00	27.6	−1.3	740	50	3.35
11:30	28.3	−0.7	760	60	3.08

Continued

12:00	29.0	−0.1	780	70	2.80
12:30	29.1	0.9	790	90	2.50
13:00	30.1	2.0	790	100	2.20
13:30	30.1	2.2	770	120	2.15
14:00	30.0	2.4	750	130	2.10
14:30	29.3	2.4	720	80	2.00
15:00	29.7	2.4	690	30	1.90
15:30	29.7	2.4	610	30	1.95
16:00	29.4	2.2	520	0	2.10
16:30	28.8	2.2	420	0	2.22
17:00	28.1	2.1	310	0	2.35
17:30	27.9	2.2	220	0	2.85
18:00	27.7	2.2	120	0	3.35
18:30	26.9	2.2	120	0	3.58
19:00	26.1	2.3	0	0	3.80
19:30	25.4	2.3	0	0	3.73
20:00	24.6	2.3	0	0	3.65
20:30	23.9	2.3	0	0	3.50
21:00	23.2	2.3	0	0	3.35
21:30	22.2	2.3	0	0	3.08
22:00	21.3	2.2	0	0	2.80
22:30	20.7	2.2	0	0	2.58
23:00	20.1	2.1	0	0	2.35
23:30	19.3	3.8	0	0	1.78

Appendix 4: Parameters of Radiator

Parameters	Values
Dimension of radiator	Height 700 mm, Width 2000 mm, and Depth 50 mm
Power output	2966 W
Area (A_rad)	5.6 m^2 (4 units by 1.4 m^2)
Thermal transmittance coefficient (h_rad)	38.5 $W/m^2 \cdot K$
Water mass flow rate (m)	0.078 kg/s (this is variable setting)