Modelling Heat Pumps within the Home Energy Model

A technical explanation of the methodology

January 2026

Acknowledgements

This methodology has been developed for the Department for Energy Security & Net Zero by a number of organisations and individuals, including Sustenic, Quidos, Scene Connect, City Science, Hoare Lea, Oxford Brookes University, University of Bath, 10-x, Building Research Establishment (BRE), AECOM, Kiwa Ltd., Loughborough University Enterprises Limited, Chris Martin and John Tebbit.

Quality assurance has been undertaken by a consortium led by Etude, including Levitt Bernstein, People Powered Retrofit, University of Strathclyde's Energy Systems Research Unit, Julie Godefroy Sustainability, and UCL.

Document reference: HEM-TP-12

Document version: v3.0

Issue date: January 2026

Home Energy Model version: HEM 1.0

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Contents

Background to the Home Energy Model

What is the Home Energy Model?

The Home Energy Model (HEM) is a calculation methodology designed to assess the energy performance of homes, which will replace the government's Standard Assessment Procedure (SAP).

Where can I find more information?

This document is part of a wider package of material relating to the Home Energy Model.

Home Energy Model technical documentation (e.g. this document)

What: This document is one of a suite of technical documents, which explain the calculation methodology in detail. New documents will be added, and the content amended, when necessary to ensure documentation is sufficiently comprehensive. This will usually, but not always, occur alongside the release of a new version of HEM.

Audience: The technical documentation will be of interest to those who want to understand the detail of how the Home Energy Model works and how different technologies are treated.

The Home Energy Model consultation and government response

What: The Home Energy Model consultation introduces the overhaul to the SAP methodology and sought views on the approach taken by the new Home Energy Model. The Home Energy Model consultation summarises the feedback to the consultation and the actions taken subsequently in development, ahead of the initial release of HEM.

Audience: The Home Energy Model consultation will be of interest to those seeking a general introduction to HEM and its role in government policy on domestic energy performance.

The Home Energy Model reference code

What: The full Python source code for the Home Energy Model core engine has been published as a Git repository. Note the reference code for official HEM wrappers is published separately.

Audience: The reference code will be of interest to those who want to understand how the model has been implemented in code, and those wishing to fully clarify their understanding of the new methodology. It will also be of interest to any potential contributors to the Home Energy Model or those wishing to use it within their own projects.

Related content

This paper sets out the methodology for modelling heat pumps within the Home Energy Model core engine. Other relevant papers on the core Home Energy Model include:

- HEM-TP-04 Space heating and cooling demand
- HEM-TP-11 Hot water storage tanks
- HEM-TP-16 Heat emitters
- HEM-TP-17 Controls

For further information on relevant assumptions made within the FHS assessment wrapper, see HEMFHS-TP-02 FHS space heating and cooling demand assumptions.

To understand how this methodology has been implemented in computer code, see:

src/core/heating_systems/heat_pump.py

src/core/project.py (relevant for exhaust air heat pump overventilation calculation – see *Annex B – Overventilation for exhaust air heat pumps*)

Note that there are two separate calculation methodologies for heat pumps in the Home Energy Model, for:

- Heat pumps providing space heating, either alone or in addition to water heating, and tested according to EN 14825
- Heat pumps providing water heating only and tested according to EN 16147

These methodologies are described in separate sections in the paper.

Overview

A heat pump is a device that uses a refrigeration cycle to transfer heat energy from a heat source (e.g. external air) to a higher-temperature heat sink (e.g. a radiator circuit or hot water cylinder), typically for space or hot water heating purposes. Most heat pumps use an electrically driven compressor. Thermally driven heat pumps also exist but are not covered by this methodology at present.

Heat pumps operate more efficiently when the source temperature is higher and/or the sink temperature is lower, and their performance also depends on the quality of the refrigeration cycle (i.e., the performance of the system compared to an ideal system). Some heat pumps incorporate a direct-electric heater to supplement space or hot water heating, but use of any backup heaters may mean more electricity is used for heating than if the heat pump itself were providing all the heat.

Methodology - Heat pumps providing space heating (with or without water heating)

This methodology is for heat pumps that have been tested to EN 14825:2018. The EN 14825 test data is used in combination with calculations from BS EN 15316-4-2:2017 to calculate the performance of the heat pump. For heat pumps tested to EN 16147:2017 rather than EN 14825:2018, see *Methodology - Heat pumps providing water heating only*.

The EN 14825 data provides measurements of the heat pump's coefficient of performance (COP), but these are measured under fixed test conditions, which are of limited use for evaluating performance in a specific dwelling over a longer period of time such as a year, where source temperatures may change significantly. Sink temperatures may also change significantly throughout the year due to weather compensation controls, although these are handled in the emitters module (see HEM-TP-16 Heat emitters). The standard does allow calculation of a seasonal coefficient of performance (SCOP), but this makes further assumptions and does not consider the demand on the heat pump in a specific dwelling under specific weather conditions.

An overview of the calculation steps[1] to be performed is listed below, and a simplified flowchart can be seen in Figure 1. Steps 1-5 are executed for each service that the heat pump provides, and step 6 is executed after steps 1-5 have been run for all services.

[1] Note that the placement of the auxiliary energy calculation among these steps may differ from BS EN 15316-4-2:2017, which in section 6.5 places the calculation of auxiliary energy before the calculation of energy input. As HEM calculates water heating demand and system response before calculating space heating demand, and the auxiliary energy depends on both, it is calculated after the calculation of energy input for each service and is added to the total energy input at the end.

1. HEM provides the energy requirements for the required service (space heating or hot water) and the required flow temperature during the operational hours. See section *1. Energy requirement*.

2. Calculate the performance (COP, thermal capacity) for the calculation interval conditions using the flow temperature weighted average of the EN 14825 test data. See section *2. Performance under operating conditions*

3. Calculate the running time and load ratio of the heat pump in different operation modes. See section *3. Running time and load ratio*

4. Calculate the energy delivered by the heat pump system and energy input to deliver that energy depending on climatic conditions and energy requirements. See section *4. Energy delivered and consumed by heat* pump

5. Calculate energy delivered by any secondary heat source (e.g. direct electric back-up heating) and energy input to deliver that energy, if required. See section *5. Energy delivered and consumed by secondary heat* source

6. Calculate auxiliary energy. See section 6. *Auxiliary energy*

Steps 1-5 are executed for each service that the heat pump provides, and step 6 is executed after steps 1-5 have been run for all services. HEM assumes that water heating is a higher-priority service than space heating, so the heat pump will not provide any space heating until water heating demand is satisfied (services are run sequentially, not concurrently). Heating of different space heating zones are currently treated as independent services.

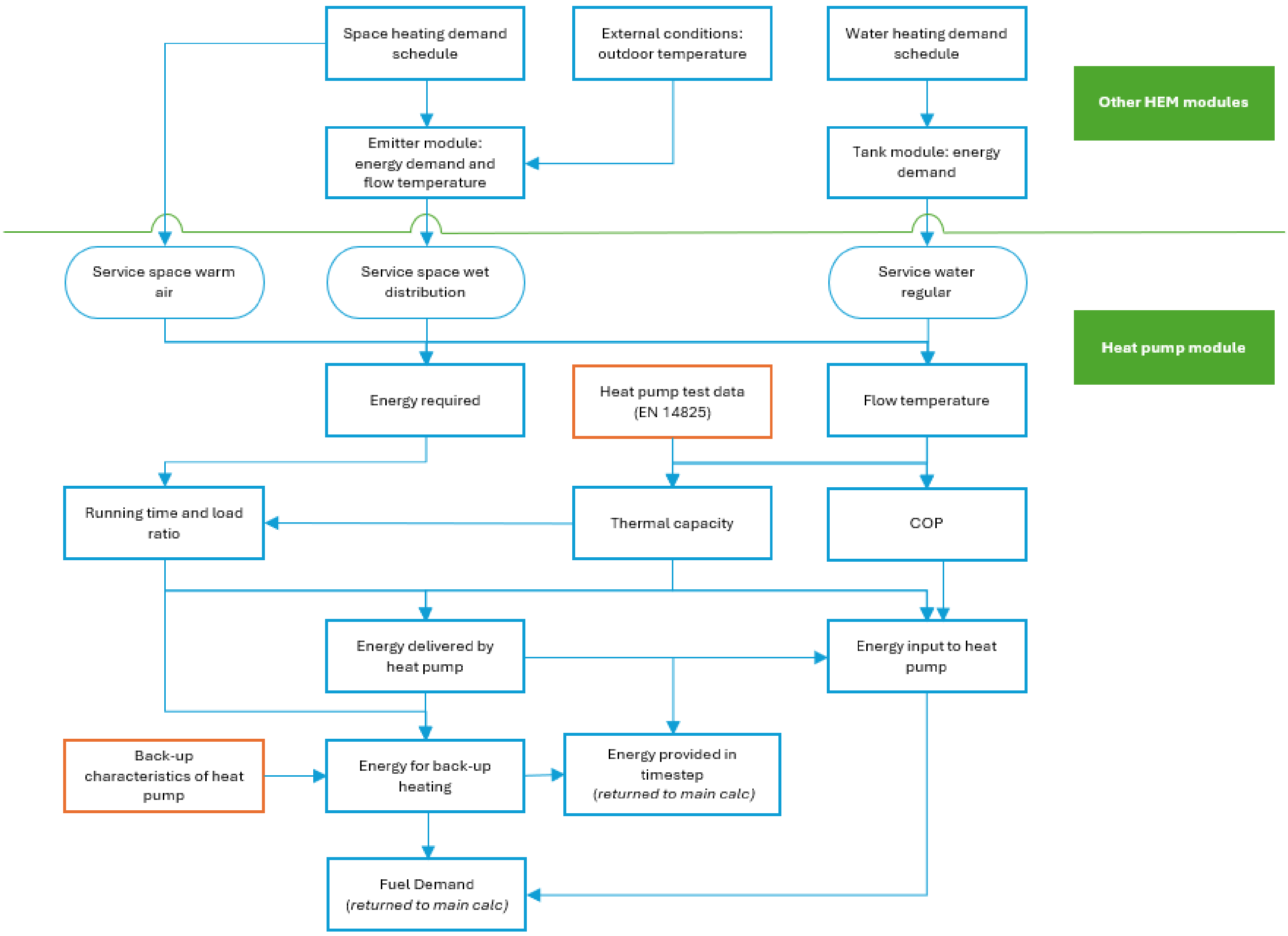

Figure 1 – Simplified flowchart of calculation steps for heat pumps in HEM

1. Energy requirement

The energy required from the heat pump for each service is an input to the heat pump calculation. This is calculated differently depending on the type of service.

1.1 Water heating

For water heating services, the energy demand and temperature of hot water to be provided is calculated by the storage tank module and is provided as an input to the heat pump module.

1.2 Space heating

For space heating services with wet distribution, the energy demand and flow temperature (i.e., sink temperature) required during each timestep are calculated by the emitter module and are provided as inputs to the heat pump module. See HEM-TP-16 for how these are calculated.

For warm air distribution, the energy demand from the space heating demand calculation is used directly (rather than being modified by the emitter module) and the flow temperature is set as the temperature used during the EN 14825 test.

When buffer tanks are included in the system (specifically four-port buffer tanks or low-loss headers for wet distribution), they create hydraulic separation. This enables the use of a secondary circulating pump, decoupling the primary and secondary circuits and improving system flexibility. Buffer tanks can provide significant additional volume to reduce heat pump cycling by mitigating rapid fluctuations in heating demand; however, the volumetric capacity of low-loss headers is typically negligible. Both types can help achieve desired flow rates and operational temperatures. The buffer volume will need to be heated to a slightly higher temperature than if the heat pump were connected to the emitters directly, resulting in higher flow/return temperature and therefore a reduced efficiency. The methodology for incorporating buffer tanks into the heat pump system is detailed in *Annex C – Buffer tanks*.

2. Performance under operating conditions

1. The performance of the heat pump under specific operating conditions is determined based on the operating conditions and performance test data collected according to BS EN 14825:2018. The operating conditions are the source temperature (see section *2.1 Source temperature*) and the sink temperature (see section 1. *Energy requirement*).

2.1 Source temperature

Recognised heat pumps are grouped by heat source:

- Ground (Indirect/Closed-loop)
- Air source
- Exhaust Air Mechanical Extract Ventilation (MEV)[2]
- Exhaust Air Mechanical Ventilation with Heat Recovery (MVHR)[3]
- Exhaust Air Mixed
- Ground water (Direct/Open-loop)
- Surface water (Direct/Open-loop)
- Fifth-generation heat network[4]

The calculation of the source temperature is detailed in the following sections for each heat source type.

2.1.1 Ground source heat pumps

For ground source heat pumps, the assumed source temperature ($\theta_{gen,in}$) is determined as follows:

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[2] Note: For validity of this calculation method, the heat pump installation must exclusively satisfy the “System 3” definition provided in Building Regulations – Approved Document Part F (Ventilation), i.e. no supplementary ventilation systems should be required to satisfy a dwelling’s ventilation requirements.
[3] Note: For validity of this calculation method, the heat pump installation must exclusively satisfy the “System 4” definition provided in Building Regulations – Approved Document Part F (Ventilation), i.e. no supplementary ventilation systems should be required to satisfy a dwelling’s ventilation requirements.
[4] Fifth generation heat networks are heat networks which operate at temperatures close to ambient ground temperatures (20-25°C) with heat pumps in each end user system to extract heat from the network as defined in CIBSE: Heat networks Design Guide.

$$\theta_{gen,in} = T_o \times 0.25806 + 2.8387 \quad \textbf{(1)}$$

Where:

T_o is the external air temperature (dry bulb), in Celsius, from the relevant weather data used for the simulation.

$\theta_{gen,in}$ is subject to a maximum value of 8ºC and a minimum of 0ºC.

Equation (1) was developed from air vs brine temperature relationship presented in EN15316-4-2:2008, amended to UK conditions during the development of the calculation method in SAP 2009 and SAP 2012 Appendix N.

2.1.2 Air source heat pumps

For air source heat pumps, the source temperature is taken to be the external air temperature (dry bulb) from the relevant weather data used for the simulation.

2.1.3 Exhaust air MEV/MVHR heat pumps

For exhaust air heat pumps, the source temperature is taken to be the average internal air temperature from the previous timestep, weighted by zone volume.

Exhaust air heat pumps are tested with a specific mechanical ventilation system as one unit, and so the source temperatures quoted in the test data are for the air entering the mechanical ventilation system and not the heat pump unit itself. This means that for exhaust air MVHR heat pumps, the source temperature seen by the heat pump itself is dependent on the heat recovery efficiency of the MVHR unit and there may be some inaccuracy in the calculation if the test data is used when the heat pump is installed with a different MVHR system than the one that it was tested with.

The source temperature seen by an exhaust air heat pump also depends in practice on the flow rate of the extract ventilation system. If the flow rate in practice is lower than in the test data, then the heat pump is likely to achieve a lower COP due to the heat pump seeing a lower source temperature in practice than in the test. HEM accepts test data at different air flow rates which will then be linearly interpolated based on the flow rate of the relevant extract ventilation system. If this flow rate is higher than any of the test data, then the test data with the highest air flow rate will be used to avoid extrapolation. This will most likely lead to an underestimate of the heat pump COP. If this flow rate is lower than any of the test data, then the calculation will stop as the test data is not valid for the operating conditions specified and using it risks overestimating the heat pump COP (in previous iterations of HEM, an overventilation adjustment was made, as described in *Annex B – Overventilation for exhaust air heat pumps*).

2.1.4 Exhaust air mixed heat pumps

Exhaust air mixed heat pumps may utilise either an MEV or MVHR system to supply air to the evaporator in conjunction with an external air supply. In this case the source temperature is determined for each calculation interval based on the proportion of the air flow volume passing through the MEV or MVHR system (at the source temperature defined above for these systems) and the proportion of air flow volume coming directly from outside (at the external air temperature for the timestep). The proportion of air flow coming from each source is an input to the calculation and does not vary by timestep, unless one of the following conditions is met:

- The external air temperature exceeds the external air temperature limit[5].
- The mixed air temperature calculated based on these proportions would fall below the mixed air temperature limit[6]

If one of the above conditions is met, then all air passing over the evaporator is assumed to come from the MEV or MVHR system (i.e. the source temperature is the same as it would be for a non-mixed exhaust air heat pump).

2.1.5 Ground water source heat pumps

Where water is extracted from the ground and re-injected into the ground or discharged at the surface, the source temperature is assumed to be constant and equal to the annual average air temperature[7]. Unlike ground source heat pumps, it is assumed that ground water source heat pumps do not significantly cool the ground, as the water is constantly flowing.

2.1.6 Surface water source heat pumps

Where water is extracted from surface water, such as rivers and lakes, the source temperature is assumed to be equal to the monthly average air temperature. It is assumed that this extraction does not substantively affect the average temperature of the water volume, and thus the body of water must have sufficient thermal capacity for this assumption to be valid.

2.1.7 Fifth generation heat networks

For fifth generation heat networks, the source temperature ($\theta_{gen,in}$) is expected to be provided by the heat network developer, either directly or via a HEM database entry for the heat network (evidence requirements to be considered separately).

2.2 Coefficient of Performance (COP)

Note: COP figures measured by EN 14825 tests are assumed to be measured at the H1 system boundary (see Annex A – System boundaries). If other energy consumption (e.g. circulation pumps) were to be included in the tests, then the corresponding energy/power should not be entered separately in this calculation.

2.2.1 When test data is measured at variable source and/or sink temperatures

The COP under operating conditions is calculated from the interpolation of the exergetic efficiency using a modified load factor.

The achieved COP under operating conditions is calculated as follows (based on EN 15316-4-2:2017 equations 39 and D.8):

[5] This is defined as part of the input data for the product.
[6] This is defined as part of the input data for the product.
[7] British Geological Survey Report No GR_999999/1 (Temperature and Thermal properties) states that “at a depth of 10 to 15 m the ground temperature is equal to the mean annual air temperature”.

$$COP_{gen,\theta in,\theta out;\Delta\theta}(t) = f_{LR;exer}(t) \times COP_{gen,exer}(t) \times f_{COP;\theta in,\theta out;\Delta\theta}(t) \quad \textbf{(2)}$$

Where:

$COP_{gen;exer}$ is the Carnot COP under operating conditions (see below).

$f_{LR;exer}$ is the exergetic efficiency under operating conditions (see below).

$f_{COP;\theta in,\theta out;\Delta\theta}$ is the temperature spread correction factor (see below).

The minimum COP is set at 1 since values less than this are likely to be extrapolation errors. The effective COP for a complete time interval may be less than 1 due to on/off operation at low heat loads.

The Carnot COP under operating conditions is calculated as follows (based on EN 15316-4-2:2017, equation 36):

$$COP_{gen;exer} = \frac{\theta_{gen;out}(t)}{\theta_{gen;out}(t) - \theta_{gen;in}(t)} \quad \textbf{(3)}$$

Where:

$\theta_{gen;in}$ is the source temperature at the heat pump, in Kelvin.

$\theta_{gen;out}$ is the sink (flow) temperature at the heat pump, in Kelvin.

The exegetic load ratio under operating conditions is calculated as follows (based on EN 15316-4-2:2017, equation 37):

$$LR_{exer;X} = \frac{COP_{exer;X}}{COP_{exer;cld}} \times \left[\frac{\left(\theta_{out;cld}/\theta_{in;cld}\right)}{\left(\theta_{out;X}/\theta_{in;X}\right)}\right]^{n\_exer} \quad \textbf{(4)}$$

Where:

$COP_{exer;X}$ is the Carnot COP under operating conditions.

$COP_{exer;cld}$ is the Carnot COP for the coldest test condition.

$\theta_{in;X}$ and $\theta_{out;X}$ are source and sink temperatures under operating conditions, in Kelvin.

$\theta_{in;cld}$ and $\theta_{out;cld}$ are source and sink temperatures for the coldest test condition, in Kelvin.

n_exer is the Exergy Exposure Factor, which has a value of 3 from EN 15316-4-2:2017, Table 9.

The exergetic efficiency under operating conditions is calculated by the linear interpolation of the nearest test exergetic efficiencies (points XX and YY) to that of the source and sink temperatures using the nearest exergetic load factors (XX and YY) as shown below (based on EN 15316-4-2:2017, equation 38).

$$f_{LR;exer}(t) = f_{LR;exer;XX} - \left(f_{LR;exer;XX} - f_{LR;exer;YY}\right) \times \left[\frac{LR_{exer;XX} - LR_{exer}(t)}{LR_{exer;XX} - LR_{exer;YY}}\right] \quad \textbf{(5)}$$

Where:

$LR_{exer}(t)$ is the exergetic load ratio under operating conditions.

$LR_{exer;XX}$ and $LR_{exer;YY}$ are the closest exergetic load ratios below and above the exergetic load ratio under operating conditions in the test data.

$f_{LR;exer;XX}$ and $f_{LR;exer;YY}$ are the exergetic efficiencies for the test records with the closest exergetic load ratios below and above the exergetic load ratio under operating conditions.

If EN 14825 test data for more than one design flow temperature has been provided, then the exergetic efficiency under operating conditions is calculated for each design flow temperature in the test data and then the final value is interpolated based on the design flow temperature specified for the installation.

The precise evaluation of the thermodynamic process involves the temperature spread at the evaporator and condenser which depends on the refrigerant temperature and other properties which are too complex to include in the formulae above. Instead, EN15316-4-2:2017 equation D8 gives a correction factor:

$$f_{COP;\theta in,\theta out;\Delta\theta} = \left[1 - \frac{(\Delta\theta_{gen;out;ref} - \Delta\theta_{gen;out}(t))/2}{\theta_{gen;out}(t) - \Delta\theta_{gen;out;ref}/2 + \Delta\theta_{HP;gen;cond;int} - (\theta_{gen;in}(t) - \Delta\theta_{HP;gen;evap;int})}\right] \quad \textbf{(6)}$$

Where:

$\Delta\theta_{gen;out;ref}$ is the temperature spread on the condenser under standard test conditions.

$\Delta\theta_{gen;out}(t)$ is the temperature spread on the condenser in operation due to the design of the heat emitter system. This is currently set as a user input.

$\theta_{gen;out}(t)$ is the temperature, in Kelvin, at the outlet of the condenser (sink temperature) under operating conditions.

$\theta_{gen;in}(t)$ is the temperature, in Kelvin, at the inlet of the evaporator (source temperature) under operating conditions.

$\Delta\theta_{HP;gen;cond;int}$ is the average temperature difference between the heat transfer medium and refrigerant in the condenser (assumed to be 5 K according to EN 15316-4-2:2017 Table D1).

$\Delta\theta_{HP;gen;evap;int}$ is the average temperature difference between the heat transfer medium and the refrigerant in the evaporator (assumed to be 15 K for air and exhaust air heat pumps and 10 K for water and ground source heat pumps according to EN 15316-4-2:2017 Table D1[8]).

This correction factor is applied to the COP (as shown above) and is only applicable when the temperature spread at the condenser during the calculation interval differs from that at the test condition during space heating operation. No correction is necessary for hot water heating, since EN15316-4-2:2017 assumes source and sink temperature spread is the same as the test conditions during these modes of operation. Therefore, for water heating the correction factor is set to 1.

2.2.2 When test data is measured at fixed source and sink temperatures

Heat pumps other than air source heat pumps are tested with a single source temperature during EN 14825 tests (test conditions A to D, and F). If tests are undertaken with fixed flow (sink) temperature control[9] as well as fixed source temperature, the calculation steps defined in the section *2.2.1 When test data is measured at variable source and/or sink temperatures*, which are based on EN 15316-4-2:2017 section 6.7.3, are not possible. An alternative approach is defined below. Note that if EN 14825 test data for more than one design flow temperature has been provided then the calculation below is done for each design flow temperature in the test data and then linearly interpolated based on the design flow temperature specified for the installation.

The best fit quadratic equation for COP at test conditions A to D and F against the outside temperature (T_0), is calculated, i.e.:

$$COP_{gen;T_0} = \left(a \times {T_0}^2\right) + (b \times T_0) + c \quad \textbf{(7)}$$

Where:

a, b and c are the quadratic regression coefficients.

T_0 is the external air temperature (dry bulb) from the relevant weather data used for the simulation.

The quadratic regression equation is then used to calculate the COP at the outside temperature for the calculation timestep This COP is then corrected to account for the source and sink temperatures as follows (based on EN 15316-4-2:2017, equation D.4).

$$COP_{gen}(\theta_{in};\theta_{out}) = COP_{gen;T_o} \times \frac{(\theta_{gen;out;X}) \times (\theta_{gen;out;ref} - \theta_{gen;in;ref})}{(\theta_{gen;out;ref}) \times (\theta_{gen;out;X} - \theta_{gen;in;X})} \quad \textbf{(8)}$$

[8] Figures in BS EN ISO 15316-4-2:2017 Table D1 are -15 K and -10 K, but figures in BS EN ISO 15316-4-2:2008 page 36 were +15 K and +4 K, and signs in the temperature spread correction equation have not changed. Using negative numbers leads to divide-by-zero errors in the calculation which do not occur when using positive numbers. Given that the equation that uses these figures already has a minus sign in front of this variable (as written in the standard) this would seem to suggest that using positive numbers is intended..

[9] Note that this refers to the flow (sink) temperature across test conditions A to D and F for a given design flow temperature. If test data for more than one design flow temperature is provided, this does not mean that the heat pump has been tested with variable flow (sink) temperature control.

Where:

$\theta_{gen;in;X}$ and $\theta_{gen;out;X}$ are source and sink temperatures under operating conditions, in Kelvin.

$\theta_{gen;in;ref}$ and $\theta_{gen;out;ref}$ are source and sink temperatures used in the EN 14825 tests (in this case the source and sink temperatures do not vary between conditions), in Kelvin.

Note that: $\theta_{gen;out;X} - \theta_{gen;in;X}$ is subject to $\Delta\theta_{min}$, which is a global minimum temperature difference of 5 K applied throughout the calculation method.

2.3 Thermal capacity

These calculations are based on the temperature at the evaporator and the temperature at the condenser and are calculated separately for each operating mode.

If the thermal capacity of the heat pump is higher than the energy requirement, then:

- Fixed Capacity Control heat pumps cycle on and off in proportion to the energy demand and thermal capacity.
- Variable Capacity Control heat pumps (invertor type) adapt the capacity to the heat load. However, below a certain capacity limit (determined by $LR_{cont;min}$ – see section *4.3.1 Minimum continuous load ratio*) the heat pump will cycle on and off between the minimum rate and zero rate.

If EN 14825 test data for more than one design flow temperature has been provided, then the calculation below of thermal capacity under operating conditions is done for each design flow temperature in the test data and then linearly interpolated based on the design flow temperature specified for the installation.

2.3.1 When test data measured at fixed source and sink temperatures

Heat pumps other than air source heat pumps are tested with a single source temperature during EN14825 tests (Measurements A to D and F). If tests are undertaken with fixed outlet (flow) temperature control as well, then the mean capacity from the EN 14825 tests is used.

2.3.2 Other cases

For fixed capacity control heat pumps the heat output under operating conditions is calculated as follows (based on EN 15316-4-2:2017, equation 33):

$$\Phi_{\theta in;\theta out;X}(t) = \Phi_{cld;ref} + (\Phi_{D;ref} - \Phi_{cld;ref}) \times \frac{(\Delta\theta_{in;out;cld} - \Delta\theta_{in;out;X}(t))}{(\Delta\theta_{in;out;cld} - \Delta\theta_{in;out;D})} \quad \textbf{(9)}$$

Where:

$\Phi_{cld;ref}$ is the thermal capacity of the heat pump at the coldest test condition.

$\Phi_{D;ref}$ is the thermal capacity of the heat pump at test condition D.

$\Delta\theta_{in;out;cld}$ is the difference (in Kelvin) between the source and sink temperatures at the coldest test condition.

$\Delta\theta_{in;out;D}$ is the difference (in Kelvin) between the source and sink temperatures at test condition D.

$\Delta\theta_{in;out;X}(t)$ is the difference (in Kelvin) between the source and sink temperatures under operating conditions.

For variable capacity control heat pumps the heat output under operating conditions is calculated as follows (based on EN 15316-4-2:2017, equation 34):

$$\Phi_{\theta in;\theta out;X}(t) = \Phi_{cld} \times \left[\frac{\left(\theta_{gen;out;cld} / \theta_{gen;in;cld}\right)}{\left(\theta_{gen;out;X(t)} / \theta_{gen;in;X(t)}\right)}\right]^{n\_exer} \quad \textbf{(10)}$$

Where:

Φ_{cld} is the thermal capacity of the heat pump at the coldest test condition.

$\theta_{gen;in;cld}$ and $\theta_{gen;out;cld}$ are the source and sink temperatures at the coldest test condition, in Kelvin.

$\theta_{gen;in;X(t)}$ and $\theta_{gen;out;X(t)}$ are the source and sink temperatures under operating conditions, in Kelvin.

n_exer is the Energy Exposure Factor, which has a value of 3 from EN 15316-4-2:2017, Table 9.

Note: EN 15316-4-2:2017 refers to output at the bivalent point, which is defined in accordance with that standard and not the EN 14825 definition. “biv” is replaced with “cld” to avoid confusion.

3. Running time and load ratio

The operating/running time for each service is calculated at this stage because it is needed throughout the rest of the calculation. The load ratio is closely related to the running time and is used to determine whether the heat pump is operating in on/off mode and the amount of additional energy used in this case (see section *4.3 Driving energy during on/off operation*).

The full-load operating time (t_X) and load ratio (LR_X) of each service are calculated as follows, using the energy requirement and the heat capacity of the heat pump (based on EN 15316-4-2:2017, equation 35):

$$t_X = \frac{Q_{\theta in,\theta out;X;out}}{\varphi_{\theta in,\theta out;X}} \text{ and } LR_X = \frac{t_X}{t_{ci}} \tag{11}$$

Where:

$Q_{\theta in,\theta out;X;out}$ is the energy requirement.

$\varphi_{\theta in,\theta out;X}$ is the thermal capacity of the heat pump for the service under operating conditions[10].

t_{ci} is the calculation timestep.

It is assumed that water heating always runs at full load and therefore $LR_W = 1$.[11]

The part-load operating time of each service is calculated as follows:

$$t_{X:partload} = \frac{t_X}{\max(LR_X, LR_{cont;min})} \tag{12}$$

The operating time calculation must account for the operating time of any higher-priority services. For each service a period of non-operation at the start of the timestep can also be accounted for, if relevant (e.g. to account for emitters cooling down at the start of the timestep), for the capacity calculation. Therefore, the operating time calculated above is subject to a maximum value of the time available for the service, which is calculated as follows (note: this is an addition to the method in BS EN 15316-4-2:2017):

$$t_{available,X} = (t_{ci} - t_{X,prev}) \times \left(1 - \frac{t_{X,start}}{t_{ci}}\right) \tag{13}$$

Where:

t_{ci} is the calculation timestep.

$t_{X,prev}$ is the operating time for higher-priority services, in hours.

$t_{X,start}$ is the length of any period of non-operation for the current service at the start of the timestep.

[10] Assuming maximum capacity at a given source temperature; ignoring that heat pump control may modulate capacity in practice.

[11] The definition of load ratio in equation 15 from EN 15316-4-2 (along with the definition of hot water running time in equation 11) implies that water heating always runs at full load. This is likely to be true in practice as the goal will be to heat the water as quickly as possible and then return to space heating operation. This calculation follows Path B from the standard, and the definition of load ratio in equation 35 from EN 15316-4-2 implies that the combined load ratio for both water and space heating should be used in the calculations. However, this does not account for any variation in heat pump output between the two services (e.g. running at full load for water heating and then in on-off mode to meet low space heating demand). It also affects equation 28, which requires the time constant of the particular service under consideration.

When accounting for the period of non-operation at the start of the timestep, the calculation above assumes that time spent on other services is evenly spread throughout the timestep, and therefore the adjustment for start time is a proportional reduction of the overall time available, not simply a subtraction. This avoids assuming that higher-priority services must have run in the early part of the timestep, which is not necessarily the case.

Note that different values of time available are calculated for calculations related to capacity and calculations related to load ratio and cycling. For calculations related to capacity, the time available accounts for the time required for emitters to cool to the maximum flow temperature at the start of the timestep to account for the effect of needing to allow the emitters to cool down on the heat pump's usable capacity. If there is a requirement to allow emitters to cool down, then the heat pump capacity will be limited by this requirement. However, in reality the maximum flow temperature may be adjusted more frequently than just once per modelling timestep, so the real effect is likely to look more like an overall reduction in demand (and therefore load ratio) across the whole timestep, and so when calculating the load ratio we do not want to adjust the time available as if there were a cool-down period only at the start of the timestep, but instead consider the load ratio over the whole timestep.

4. Energy delivered and consumed by heat pump

The energy delivered and consumed by the heat pump (excluding any secondary heat source such as direct electric backup – see section *5. Energy delivered and consumed by secondary heat source*) is based on the performance under operating conditions (see section *2. Performance under operating conditions*) and the running time and load ratio (see section *3. Running time and load ratio*).

Note: If a secondary heat source (e.g. a direct electric backup heater) is operating instead of the compressor (see section 5.1 Control of secondary heat source), then the energy delivered and consumed by the heat pump (in the absence of contribution from backup) is zero.

4.1 Energy delivered

The energy delivered by the heat pump (excluding any contribution from a backup heater) is calculated as follows:

$$Q_{X;gen,out}(t) = \varphi_{\theta in;\theta out;X}(t) \times t_X \tag{14}$$

Where:

$\varphi_{\theta in;\theta out;X}(t)$ is the thermal capacity of the heat pump for the service under operating conditions.

t_X is the operating time during the calculation interval for each service (a separate value is calculated for each service).

4.2 Driving energy during continuous operation

The compressor driving energy input for continuous operation during each service is derived from the values for COP, operating time and thermal capacity as follows (based on EN 15316-4-2:2017, equation 40):

$$E_{gen;\theta in,\theta out,\Delta\theta,X}(t) = \frac{Q_{X;gen,out}(t)}{COP_{gen;\theta genin;\theta genout}(t)} \quad \textbf{(15)}$$

Where:

$Q_{X;gen,out}(t)$ is the energy delivered by the heat pump (excluding any contribution from a backup heater).

$COP_{gen;\theta genin;\theta genout}(t)$ is the COP under operating conditions for the service.

4.3 Driving energy during on/off operation

This mode occurs for:

- Fixed capacity control heat pumps.
- Variable capacity control heat pumps when the load ratio under operating conditions is lower than the lowest possible load ratio ($LR_{cont;min}$) applicable to the compressor.

The calculation of driving energy during on/off operation is undertaken separately for each service (X) provided.

Note that although this methodology follows Path B from BS EN 15316-4-2:2017, some of the equations referenced are defined as part of Path A. This is because the text in Path B states that part of Path A is to be used if the heat pump is operating in on/off mode.

4.3.1 Minimum continuous load ratio

Variable capacity control heat pumps can adapt their capacity to the heat load. However, below a certain capacity limit, they cycle on and off. This limit is known as the minimum modulation rate ($LR_{cont;min}(t)$) and is an input to the calculation. The minimum modulation rates can be provided at the following flow temperatures:

- 20°C for heat pumps with warm air distribution (required input)
- 35°C for heat pumps with wet distribution (required input)
- 55°C if EN 14825:2018 test data has been provided for design flow temperature of 55°C

The minimum modulation rate used in the calculation varies depending on the flow temperature. It is determined for each time-step by linear interpolation between the two minimum modulation rates provided in the inputs (each associated with a specific flow temperature). For flow temperatures higher than the highest flow temperature provided, the minimum modulation rate at the highest flow temperature will be used. For flow temperatures below the lowest flow temperature provided, the minimum modulation rate at the lowest flow temperature will be used.

4.3.2 Compressor power at full load

The power of the compressor at full load is calculated as follows:

$$P_{gen,LR100,X}(t) = \frac{\varphi_{\theta in;\theta out;X}(t)}{COP_{gen;\theta genin;\theta genout}(t)} \quad \textbf{(16)}$$

Where:

$\varphi_{\theta in;\theta out;X}(t)$ is the thermal capacity of the heat pump for the service under operating conditions.

$COP_{gen;\theta genin;\theta genout}(t)$ is the COP under operating conditions for the service.

4.3.3 Compressor power at lowest continuous load

The power of the compressor at the lowest possible continuous load is calculated as follows (based on EN 15316-4-2:2017, equation 26):

$$P_{gen,LR;comp;min,X}(t) = P_{gen,LR100,X}(t) \times LR_{cont;min,X}(t) \quad \textbf{(17)}$$

Where:

$LR_{cont;min,X}(t)$ is the minimum continuous load ratio at the operating flow temperature for the delivered service.

$P_{gen,LR100,X}(t)$ is the power of the compressor at full load.

4.3.4 Compressor power consumption due to inertia

The power used due to non-reversibility of the heat pump (inertia) is $P_{gen;comp;ONOFF;LR}(t)$. This is equal to 0 when the load ratio (LR) is greater than or equal to $LR_{cont;min;X}(t)$. For lower load ratios, it is equal to the following (based on EN 15316-4-2:2017, equation 28):

$$P_{gen;comp;ONOFF;LR}(t) = P_{gen;LR;comp;min,X}(t) \times \frac{\tau_{eq} \times LR \times (1 - LR)}{\tau_{out,em,type}} \quad \textbf{(18)}$$

Where:

LR is the load ratio under operating conditions. The load ratio used is the combined load ratio from all services of a given type.

τ_{eq} is a characteristic parameter of the heat pump, due to the inertia of the on/off transient.

$\tau_{out,em,type}$ represents the operating time to reach the required conditions of the emitter distribution system. This value depends on the category of emitters for heating and the temperature of the domestic hot water. Default time characteristic values are defined in EN15316-4-2:2017 Table 13. The current methodology uses the value for “Light embedded systems" for all heat pumps with wet distribution, the value for "Very low" for warm air heat pumps and the value for "Domestic hot water & storage" for water heating.

4.3.5 Energy input

The corrected heat pump driving energy can be calculated using:

$$E_{X;gen,in}(t) = \left((1 + f_{aux}) \times P_{gen,LR100,X}(t) + P_{gen,comp,ONOFF,LR}\right) \times t_{available,X} + E_{X;gen,aux} \quad \textbf{(19)}$$

Where:

f_{aux} is the fraction of auxiliary energy that is not implicitly included in the COP measurements when operating continuously. This is zero for electric heat pumps[12].

$P_{gen,LR100,X}(t)$ is the compressor power at full load.

$P_{gen,comp,ONOFF,LR}$ is the power consumption due to inertia.

$t_{available,X}$ is the time available for the service to run.

$E_{X;gen,aux}$ is the energy consumption of the compressor unit during the off part of the on/off cycle. The calculation of this requires information on whether any lower-priority services are running, which is not known at this stage of the calculation. Therefore, HEM omits it from Equation **(19)** and adds it later as part of the ancillary energy calculation.

4.4 Energy extracted from heat network

Where the heat pump is part of a fifth-generation heat network, the energy extracted from the heat network is calculated by subtracting the driving energy from the energy delivered by the heat pump (excluding any contribution from backup heating), as described in EN 15316-4-2:2017 section 6.9. The energy extracted from the heat network by the heat pump is then reported in the overall energy consumption of the dwelling.

5. Energy delivered and consumed by secondary heat source

A heat pump unit may have a secondary heat source controlled by the same control unit to supplement the heat pump. This can take the form of either a direct electric backup heater or a boiler. Table 1 shows how the system arrangement affects service provision. In hybrid heat pump and boiler packages ("hybrid heat pumps"), the heat pump may be combined with either a regular boiler or a combi boiler.

[12] It would be non-zero for non-electric heat pumps, should they be added to the model in the future.

Table 1 - Service provision by system arrangement

System arrangement	Hot water provided by:	Space heating provided by:
Heat pump with combi boiler	Combi boiler	Heat pump and/or combi boiler (depending on control)
Heat pump with regular boiler	Heat pump and/or regular boiler (depending on control)	Heat pump and/or regular boiler (depending on control)
Heat pump with direct electric backup	Heat pump and/or direct electric backup (depending on control)	Heat pump and/or direct electric backup (depending on control)

5.1 Control of secondary heat source

5.1.1 Back-up requirement

Back-up energy is required when:

- the source temperature is below the declared temperature operating limit (TOL) of the heat pump.
- the required flow temperature is above the maximum flow temperature limit of the heat pump.
- energy provided by the heat pump is insufficient to meet the total demand for all the required services.

If the backup-heater is used because the conditions are outside the heat pump's operating limits, it is assumed that the backup heater runs instead of the compressor. When the requirement for backup energy is due to demand exceeding the capacity of the heat pump, two alternative operating modes are recognised in the methodology:

- Top-up: The backup heater operates in addition to the compressor, providing a boost to the energy output of the unit.
- Substitute: The backup heater operates instead of the compressor if the backup heater has a higher maximum output under the operating conditions for the timestep.

The primary operation of any supplementary water heater (e.g., electric immersion) must be controlled by the heat pump controller to be counted as backup. This ensures that the timing of supplementary heating is coordinated with the heat pump to prevent unnecessary operation of the supplementary heater. Local occupant control to provide additional boosting may be provided, but this should automatically reset once the required hot water temperature is achieved in the vessel, requiring further manual intervention for any subsequent boosting.

There is an additional input for the backup heater delay time. This is the time that the system will run the compressor at full power before activating the backup heater in the case where back-up energy is required due to insufficient capacity. This can ensure that the system does not use the backup heater excessively in trying to meet the entire demand in a single timestep.

5.1.2 Cost-effectiveness

For a hybrid heat pump, the boiler is assumed to act as a backup heater according to the logic above (unless the backup control type is set to “None”). In addition to this, the boiler is also assumed to operate at times when it has lower running costs[13] than the heat pump, in which case the heat pump does not run at these times. In order for the boiler to have lower running costs, the ratio of the heat pump COP to the boiler efficiency must be greater than the ratio of the unit costs of the heat pump’s energy source (typically electricity) and the boiler’s energy source (e.g. mains gas). The heat pump COP and boiler efficiency are dependent on the flow and return temperatures for the service in question, which are inputs to the hybrid heat pump calculation. The energy unit costs are entered in the input file and may be different for each timestep.

5.2 Running time

Where the secondary heat source is a direct electric backup, the time available for the backup heater to run is the length of the timestep minus the part-load running time of the higher-priority services. When only the backup heater is running, the running time of the backup heater is calculated assuming that the backup heater always operates at full power, so the backup heater’s part-load and full-load running times for the service are the same.

For a hybrid heat pump, the running time logic depends on the back up control type:

- For Top-up mode, the boiler and heat pump can run independently. Therefore, the available time for the boiler is independent of the heat pump running time.
- For Substitute mode, either the boiler or the heat pump will be running at any one time, but not both. In this case, the boiler’s running time is calculated accounting for the time that the heat pump has already run in the timestep. The boiler and heat pump running times are then both added to the total running time for the timestep.
- If no back up mode is specified, the running time for the boiler is calculated as it is for Substitute mode.

5.3 Thermal capacity

The maximum thermal capacity of a direct electric backup heater is calculated as follows:

$$Q_{gen,bu,out,max} = P_{gen,bu,out,max} \times t_{available,X} \quad \textbf{(20)}$$

Where:

$P_{gen,bu,out,max}$ is the maximum power output of the backup heater, in kW.

[13] The “cost” in question would typically be financial, but the model does not require this. For example, if the cost schedules provided to the calculation are in terms of emissions, then the boiler will operate when this leads to lower emissions.

$t_{available,X}$ is the time available for the service to run, as calculated in equation (13).

Where the secondary heat source is a boiler, the maximum thermal capacity is calculated in the boiler module (see HEM-TP-14).

5.4. Energy required/delivered

The energy required from the secondary heat source is the difference between the energy required for the service and the heat energy produced by the heat pump, limited by the maximum thermal capacity of the secondary heat source.

If the secondary heat source is operating instead of the compressor, then the energy delivered and consumed by the heat pump (in the absence of contribution from the secondary heat source) is zero and the energy required from the secondary heat source for the service is calculated as follows:

$$Q_{X,gen,bu,out} = Q_{X,gen,dis,out} \tag{21}$$

If the secondary heat source is operating in addition to the compressor, then the energy required from the secondary heat source for the service is calculated as follows:

$$Q_{X,gen,bu,out} = Q_{X,gen,dis,out} - (\phi_X \times t_X) \tag{22}$$

Where:

$Q_{X,gen,dis,out}$ is the energy output required to meet the demand for the service (see section *1. Energy requirement*).

ϕ_X is the thermal capacity of the heat pump in the absence of backup heating (see section *2.3 Thermal capacity*).

t_X is the full-load running time for the service (see section *3. Running time and load ratio*).

If the result of calculating $Q_{X,gen,bu,out}$ is negative, then $Q_{X,gen,bu,out}$ is set to zero (i.e., no back-up energy is required). If the result is higher than the maximum thermal capacity of the backup heater, then $Q_{X,gen,bu,out}$ is set to $Q_{gen,bu,out,max}$ (i.e., backup heater is running at maximum capacity.

5.5. Energy consumed

It is currently assumed that a direct electric backup heater has a COP of 1, and therefore the energy consumed by the backup heater equals the energy delivered by the backup heater.

Where the secondary heat source is a boiler, the energy input to the boiler is calculated in the boiler module (see HEM-TP-14), based on the energy required/delivered calculated in section *5.4. Energy required/delivered.*

6. Auxiliary energy

To account for all energy consumption by the heat pump, there are some calculations that are required after the calculation of energy consumption by individual services.

The auxiliary electrical consumption of a heat pump during heating is already incorporated into the electrical input test measurement and hence the COP of the test data. This includes the effect of cycling on and off when there is a small heat demand.

The calculation for ground source heat pumps includes brine circulation pump energy, which is additional to that measured during EN 14825 tests. This energy is taken as an input brine circulation pump power multiplied by the heat pump running time.

6.1. Additional auxiliary electrical consumption during running

This section is concerned with auxiliary electrical consumption not measured during standard tests. The electrical energy consumed by auxiliary pumps is determined by the full-load running time of each service which is multiplied by the sum of the pump powers. When the heat pump operates in on/off mode, the running time is the time needed to operate at full load; this is full load for constant output systems. Note that pump power values defined in the inputs must include any secondary pumps that are part of the distribution system, as pump energy is not calculated in the emitters module.

6.2. Additional auxiliary electrical consumption at zero load

Auxiliary electrical consumption when the demand for a heating service is zero occurs when heat demand is satisfied or outside the operating hours.

EN 14825 test data contains information about the power consumption during off mode (P_{off}), standby mode (P_{SB}) and crankcase heater mode (P_{CK}). These are used in conjunction with the operating hours at zero heat load and non-operating hours to calculate the energy consumption.

EN14825 test data has two crankcase heater test scenarios:

1. "If the crankcase heater is on during standby measurements, then the power consumption due to the crankcase heater mode shall be considered equal to the standby power consumption." Here the crankcase heater is included in the standby consumption measurement and the power consumption of both modes are reported as equal in the test report. In such cases, the calculation should set P_{CK} to zero to prevent double counting.

2. Separate test of crankcase heater required. "If the crankcase heater is not operating during standby measurement then a test shall be performed as follows: After the "B" temperature conditions test in heating mode is finished, the unit is stopped with the control device, and the energy consumption of the unit shall be measured for 8 h. Average of 8-hour power input shall be calculated. The standby power consumption is deducted from this measured energy consumption to determine the crankcase heater operation consumption." This crankcase heater consumption measurement excludes

the standby consumption measurement. These different values are both reported in the test report.

For the avoidance of doubt, the crankcase heater consumption in this method assumes it excludes standby consumption, which is therefore a separate term in Equation **(23)**. If this is not the case the crankcase heater and standby consumption measurements must be adjusted accordingly.

When the calculation interval coincides with space heating operational hours:

$$E_{gen;in;LR0}(t) = (P_{SB} + P_{CK}) \times \left(t_{ci} - \sum_{X} t_{X;partload}\right) \quad \textbf{(23)}$$

Where:

t_{ci} is the length of the calculation timestep, in hours.

$t_{X;partload}$ is the running time for each service, in hours.

When the calculation interval is outside the space heating operational hours and within water heating operational hours:

$$E_{gen;in;LR0}(t) = P_{SB} \times \left(t_{ci} - \sum_{X} t_{X;partload}\right) \quad \textbf{(24)}$$

When the calculation interval is outside the space heating and water heating operation hours:

$$E_{gen;in;LR0}(t) = P_{off} \times t_{ci} \quad \textbf{(25)}$$

Methodology - Heat pumps providing water heating only

This methodology is for heat pumps that have been tested to EN 16147:2017 rather than EN 14825:2018. For heat pumps tested to EN 14825:2018, see *Methodology - Heat pumps providing space heating (with or without water heating)*.

For hot water only heat pumps EN 16147:2017 test data to load profile "M" must be available, otherwise a Seasonal Performance Factor (SPF) value cannot be devised. If the EN 16147 test data to load profile "L" is also available, the HEM hot water SPF is determined by interpolating or extrapolating an SPF (efficiency) using both “M” and “L” test measurements in accordance with the daily hot water requirement of the dwelling being assessed.

The SPF is calculated from the following equation:

$$\eta_W = \frac{\left(E_{DTP} + \left(Q_{WS,ls,24} \times 0.6 \times 0.9\right)\right) \times 100}{\left(Q_{elec} - \left(P_{es} \times 24 \times 0.6 \times 0.9\right) + \left(\frac{Q_{WS,ls,24}}{COP_{DHW}} \times 0.6 \times 0.9\right)\right)} \quad \textbf{(26)}$$

Where:

$Q_{WS,ls,24}$ is the daily hot water vessel heat loss (kWh/day) for a 45K temperature difference between the vessel and its surroundings. It is tested in accordance with BS 1566 or EN 12897 or any equivalent standard, though is not recorded in the HEM product database. The vessel must be the same as that used during the EN 16147 test.

Q_{elec} is the electrical input energy (kWh) measured in the EN 16147 test (defined as $\frac{Q_{ref}}{Q_{LP}} \times W_{EL-LP}$ in EN 16147) over 24 hours.

P_{es} is the standby power (W) measured in the EN 16147 test.

COP_{DHW} is the COP measured in the EN 16147 test.

E_{DTP} is the total daily energy in kWh/day for the tapping profile.

0.6 is a temperature factor to reflect the daily temperature variation of the vessel contents.

0.9 is a factor applied as there is typically a separate time control of domestic hot water.

The COP_{DHW} included in the EN 16147 has a system boundary of H4 (see *Annex A – System boundaries*), which means it includes the storage tank. As HEM models the storage tank separately, Equation (26) removes the effect of the losses associated with the storage tank, as these will already be included in the demand figure which is input to the heat pump module and will be specific to the tank installed in the dwelling rather than the tank used for the EN 16147 test.

No seasonal weather calculation is applied to hot water only heat pumps; EN 16147 results are accepted as suitably representing annual performance, despite using a fixed source temperature. It is assumed that the vessel is located within the dwelling heated envelope.

As the SPF is based on lab tests in which the heat pump was combined with a specific hot water vessel, the SPF is multiplied by an “in-use factor” if the characteristics of the hot water vessel installed are likely to lead to a lower SPF being achieved in practice, i.e. if one or more of the following are true:

- Installed hot water vessel has a lower volume than the vessel used in the test
- Installed hot water vessel has greater standing losses under standardised conditions than the vessel used in the test
- Installed heat exchanger area is less than the heat exchanger area in the test

The value of the in-use factor is currently an input to the core model.

The water heating demand on the heat pump is divided by the SPF calculated according to equation (26) (after application of the in-use factor, if applicable) to give the heat pump’s electricity demand.

Future development

The following features may be added to the HEM heat pump methodology in the future:

- Storage operation which can supply heating or hot water at a later time.
- Solar assisted heat pumps which use solar PV to heat hot water in the storage tank.
- Ability for a heat pump to accept electricity from a PV diverter, to power its operation.
- Cooling provision (reversible heat pumps and more detailed treatment of air conditioning). Note that heat pump systems capable of providing cooling can still be entered into HEM as heating systems, even if their cooling functionality is not yet explicit within the simulation.

Test data for EN16147 may be requested from manufacturers alongside the EN14825 data. The additional data may be used in the core methodology or used for validation purposes.

The methodology currently uses default time characteristic values defined in EN15316-4-2:2017 Table 13, but time characteristic values could potentially be calculated based on the characteristics of the particular distribution and storage systems specified.

Annex A – System boundaries

Equations used for the determination of heat pump annual performance are shown below:

$$COP_{\mathrm{H1}} = \frac{Q_{HP}}{E_{HP}} \quad \textbf{(27)}$$

$$COP_{\mathrm{H2}} = \frac{Q_{HP}}{E_{HP} + E_{sourcepump}} \quad \textbf{(28)}$$

$$COP_{\mathrm{H3}} = \frac{Q_{HP} + Q_{secondary}}{E_{HP} + E_{sourcepump} + E_{secondary}} \quad \textbf{(29)}$$

$$COP_{\mathrm{H4}} = \frac{Q_{HP} + Q_{secondary}}{E_{HP} + E_{sourcepump} + E_{secondary} + E_{heatingpump}} \quad \textbf{(30)}$$

$$COP_{\mathrm{H5}} = \frac{Q_{HP} + Q_{secondary} - Q_{tankloss}}{E_{HP} + E_{sourcepump} + E_{secondary} + E_{heatingpump}} \quad \textbf{(31)}$$

Where:

Q_{HP} is the heat output from the heat pump

$Q_{secondary}$ is the heat output from any secondary heat source

$Q_{tankloss}$ is the heat loss from any storage tank

E_{HP} is the energy input to the heat pump (not including energy extracted from the environment, e.g. the outside air for an air source heat pump)

$E_{sourcepump}$ is the electricity input to any circulation pump on the borehole or ground array

$E_{secondary}$ is the energy input to any secondary heat source

$E_{heatingpump}$ is the electricity input to any circulation pump on the heating circuit

The COP from the EN 14825 test data corresponds to the H1 boundary.

The COP from the EN 16147 test data corresponds to the H4 boundary. The H4 boundary definition merges both space and hot water heating SPF into a single value. However, as the EN 16147 test is a hot water test the space heating system is ignored.

The system boundaries used above are based on those defined during the 2011 SEPEMO-Build project (Seasonal performance factor and monitoring for heat pump systems in the building sector). However, the H4 boundary may not match due to ambiguity around whether the hot water storage vessel should be considered within the H4 boundary, i.e. whether the output for water heating should be based on the energy content of the hot water drawn off of the storage vessel (as in EN 16147 tests) or the energy provided by the heat pump and any secondary heat source to the hot water storage vessel. The SEPEMO report[14] includes the hot water vessel in the H4 boundary in Figure 1 but excludes it in subsequent figures and equations. HEM excludes the hot water storage vessel from the H4 boundary but includes the hot water storage vessel within the H5 boundary so that the H5 boundary aligns with the COP calculated in the EN 16147 tests.

[14] D4_2_D2_4_Concept_for_evaluation_of_SPF_Hydronic_Version_2_2_2012-05-31.pdf

Annex B – Overventilation for exhaust air heat pumps

Note: from HEM v0.29 onwards, the overventilation calculation has not been used, but most of the implementation is still in the code repository, either commented out or simply not called. To bring it back into use, modifications may be required to make it compatible with the new ventilation model (introduced in v0.29) and the handling of multiple heating/cooling systems in the same zone (introduced in v0.31).

Overview

For exhaust air heat pumps requiring overventilation, the following calculations are carried out (with reference to the steps defined in the section Methodology - Heat pumps providing space heating (with or without water heating)):

- Steps 1-5 are executed for water heating, then a throughput factor (see equation (32)) is calculated accounting for just the water heating service. If there is more than one water heating service, then the throughput factors for each service are combined (see equation (33)).
- Steps 1-3 are then executed for the first space heating service and a throughput factor for the space heating service is calculated. This is then combined with the throughput factor for water heating to calculate a throughput factor for the zone. Note that this means that any additional ventilation required due to water heating demand is apportioned to the zones in proportion to their volume, while any additional ventilation due to space heating is assigned to the zone that the heating demand originates from, in order to simplify the calculation.
- If the throughput factor is greater than 1 then the space heating demand is recalculated with a modified ventilation rate calculated by multiplying the original ventilation rate by the throughput factor, and all calculation steps up to this point are repeated based on the new space heating demand. Then, steps 4-5 are executed.
- The above calculation is then repeated for each subsequent space heating service.
- Finally, step 6 is executed.

Note the additional ventilation throughput is the result of the heat pump running to satisfy both space and water heating demand but only space heating demand needs to be recalculated. Recalculating the space heating demand leads to a different running time than the one used to calculate the throughput factor, so there is a circularity in the calculation; to resolve this, the space heating demand is only recalculated once.

Throughput factor

For exhaust air heat pumps requiring overventilation, the running time is used to calculate the mechanical ventilation throughput factor (F_{mv}) for the timestep:

$$F_{mv} = 1 + \frac{(R_{hp} - 1) \times \sum_X t_X}{t_{ci}} \quad \textbf{(32)}$$

Where:

t_{ci} is the calculation timestep.

$\sum_X t_X$ is the sum of the running times for each service provided by the heat pump that is either the service currently being considered or a higher-priority service.

R_{hp} is the overventilation ratio, which is the ratio of the lowest air flow rate in the test data to the air flow rate of the extract ventilation system the heat pump is connected to, subject to a minimum value of 1.

Throughput factors for different services or heating systems can be combined as follows to calculate an overall throughput factor:

$$F_{mv,combined} = 1 + \sum_i (F_{mv,i} - 1) \quad \textbf{(33)}$$

This formula is used whenever throughput factors are described as being combined in the section *Overview* above.

Annex C – Buffer tanks

The buffer tank methodology, as implemented in HEM, is encapsulated in a sequence of calculations that simulate the tank's role in supporting the heat pump system. The methodology starts with the primary functionality of the buffer, which is to mediate between the heat pump and emitters by managing temperature stability and flow rate mismatches. Auxiliary processes incorporated in the methodology include calculating thermal losses (based on the tank's specific heat loss rate and operating conditions) and maintaining energy balances by accounting for both recoverable and non-recoverable heat losses as internal gains to the system.

Heat demand and temperature adjustment under load

The buffer tank calculation determines how it meets heat demands from the emitters. The following steps occur during each timestep:

Evaluating demand and flow temperature adjustment

To determine the flow and return temperatures on the heat pump-buffer side, the temperature difference across the buffer tank (ΔT_{buffer}) is calculated as:

$$\Delta T_{buffer} = \frac{\dot{P}_{emitters}}{\dot{m}_{flow} \times c_p \times \rho} \quad \textbf{(34)}$$

Where:

$\dot{P}_{emitters}$ is the power demand from the emitters (kW).

$\dot{m}_{flow}$ is the mass flow rate of the buffer tank-emitter loop (kg/s)[15].

c_p is the specific heat capacity of the fluid (water or glycol25) in the buffer tank (kWh/kg·K).

ρ is the density of the fluid (water or glycol25) in the buffer tank (kg/l).

Flow and Return temperatures between the heat pump and the buffer tank are subsequently calculated as follows:

$$T_{flow} = T_{req} + \frac{\Delta T_{buffer}}{2} \quad \textbf{(35)}$$

$$T_{return} = T_{req} - \frac{\Delta T_{buffer}}{2} \quad \textbf{(36)}$$

Where:

T_{req} is the required emitter flow temperature (°C).

T_{flow} is the calculated flow temperature (°C).

T_{return} is the calculated return temperature (°C).

[15] The mass flow rate is determined by the flow rate of the fixed rate pump circulating water from the buffer tank to the emitters.

Thermal losses

The buffer tank's thermal losses are calculated as a function of the average tank temperature, the room temperature from the previous timestep, and the specific loss characteristics of the tank. These losses are added to the overall heat demand.

The specific heat loss rate of the buffer tank (in W/K) is calculated as follows:

$$H_{buffer} = \frac{1000 \times L_{daily}}{24 \cdot (T_{set} - T_{amb})} \quad \textbf{(37)}$$

Where:
1000 is the conversion between W and kW
24 is the number of hours in a day
L_{daily} is the daily standing losses of the buffer tank under standard test conditions (kWh/day), which is an input to HEM, see Standard BS EN 12897:2016.
$T_{set} = 65$ is the reference tank set temperature (°C) as stated in the Standard.
$T_{amb} = 20$ is the reference ambient temperature (°C) as stated in the Standard.

The absolute heat loss for the buffer tank (in kWh) is then calculated as follows:

$$Q_{loss} = \frac{(T_{ave} - T_{room}) \times H_{buffer}}{1000} \cdot \Delta t \quad \textbf{(38)}$$

Where:
T_{ave} is the current average buffer tank temperature (°C), which equals the temperature required by the emitters in this case[16].
T_{room} is the room temperature (°C) from the previous timestep.
Δt is the timestep duration (hours).
1000 is the conversion factor between Wh and kWh.

Note that Q_{loss} is divided by the number of zones in the code to avoid counting the full loss more than once as the heating service is sequentially called by each zone.

Flow balancing

The methodology checks that the heat pump's flow rate matches system requirements without exceeding the design flow rate of the tank-emitter loop. If a mismatch is detected, (e.g. If the flow rate between the heat pump and buffer tank is greater than the flow rate between the buffer tank and emitters), then the calculation will be stopped.

[16] A different assumption is needed when there is no demand – see Annex C, *Handling zero-load conditions*

Handling zero-load conditions

When there is no heat demand:

- The buffer tank cools passively due to thermal losses (see equation **(40)** below). The cooling rate is derived from the tank's volume, the thermal properties of its contents, and the calculated heat loss for the timestep.
- The updated average temperature is calculated and stored for the next timestep (see equation **(41)** below).

Buffer temperature update due to heat loss

After accounting for thermal losses, the average temperature of the buffer tank is updated to reflect the system's thermal dynamics using the equations below.

The heat capacity of the buffer tank, c_{buffer}, is calculated as:

$$c_{buffer} = V \times \rho \times c_p \tag{39}$$

Where:

V is the volume of the buffer tank (litres),
ρ is the density of the tank's contents (kg/litres),
c_p is the specific heat capacity of the tank's contents (kWh/kgK)

The temperature decrease, ΔT_{loss}, due to the thermal losses is then given by:

$$\Delta T_{loss} = \frac{Q_{loss}}{c_{buffer}} \tag{40}$$

Q_{loss} is calculated using equation **(38)**, but as there is no heat load T_{ave} in that equation is the average buffer tank temperature (°C) from the previous timestep.

The new average buffer tank temperature (to be passed to the next timestep) is calculated as:

$$T_{avg,new} = T_{avg} - \Delta T_{loss} \tag{41}$$

Heat loss and recovery

Thermal losses are a key aspect of the buffer tank's operation. Heat losses are divided into recoverable and non-recoverable components. The recoverable portion (based on a standardised recovery factor of 0.75 from BS EN 15316-5:2017 Table B.3), is treated as internal gains to the building, reducing the effective heating demand in the next timestep.

Ancillary energy use

The circulation pump associated with the buffer tank consumes energy based on its fixed flow rate and the associated power requirement. This energy is included in the system’s total ancillary energy use.

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