Modelling Heat pumps within the Home Energy Model

A technical explanation of the methodology

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Any enquiries regarding this publication should be sent to us at: homeenergymodel@energysecurity.gov.uk

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).

The Home Energy Model is still under development and its first version will be implemented alongside the Future Homes Standard (FHS) in 2025. We are publishing information about the model while it is still at a formative stage to enable industry to participate in the ongoing development process.

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 go into further detail on the methodology and the validation exercises that have been carried out. We intend to update and produce further technical documentation throughout the model development process.

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

What: The Home Energy Model consultation, which explains the overhaul to the SAP methodology and seeks views on the approach taken by the new Home Energy Model.

Audience: The Home Energy Model consultation will be of interest to those who want to understand the proposed changes to the SAP methodology and wider SAP landscape.

The Home Energy Model reference code

What: The full Python source code for the Home Energy Model and the Home Energy Model: FHS assessment has been published as a Git repository. This code is identical to that sitting behind the consultation tool. We are currently considering whether the open-source code could serve as the approved methodology for regulatory uses of the Home Energy Model.

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.

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-06 Ventilation and infiltration (relevant for exhaust air heat pumps)
- 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, please see HEMFHS-TP-02 FHS space heating and cooling demand assumptions.

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

src/core/heating_systems/heat_pump.py

src/core/space_heat_demand/ventilation_elements.py (relevant for exhaust air heat pumps)

src/core/project.py (relevant 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 below.

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 although 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. The EN 14825 test data is used in combination with calculations from EN 15316-4-2 to calculate the performance of the heat pump.

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 assumptions and therefore there are several issues that it does not consider, such as the demand on the heat pump in a specific dwelling under specific weather conditions.

An overview of the calculation steps to be performed is listed below. A simplified flowchart can be seen in Figure 1.

1. HEM will provide the energy requirements for the required service and the required flow temperature during the operational hours.
2. Calculate the performance (COP, thermal capacity and degradation coefficient) for the calculation interval conditions using the flow temperature weighted average of the EN 14825 test data.

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

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

5. Calculate energy for any back–up heating, if required.

6. Calculate ancillary energy and 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. Heating of different space heating zones are currently treated as independent services.

For exhaust air heat pumps requiring overventilation, steps 1-5 are executed for water heating, then steps 1-3 are executed for each space heating service. The total running time for all services is then used to calculate a ventilation throughput factor and the space heating demand is recalculated with a modified ventilation rate calculated by multiplying the original ventilation rate by the throughput factor (see section “”). For each space heating service, steps 1-3 are then repeated based on the new space heating demand, before executing steps 4-5. Finally, step 6 is executed.

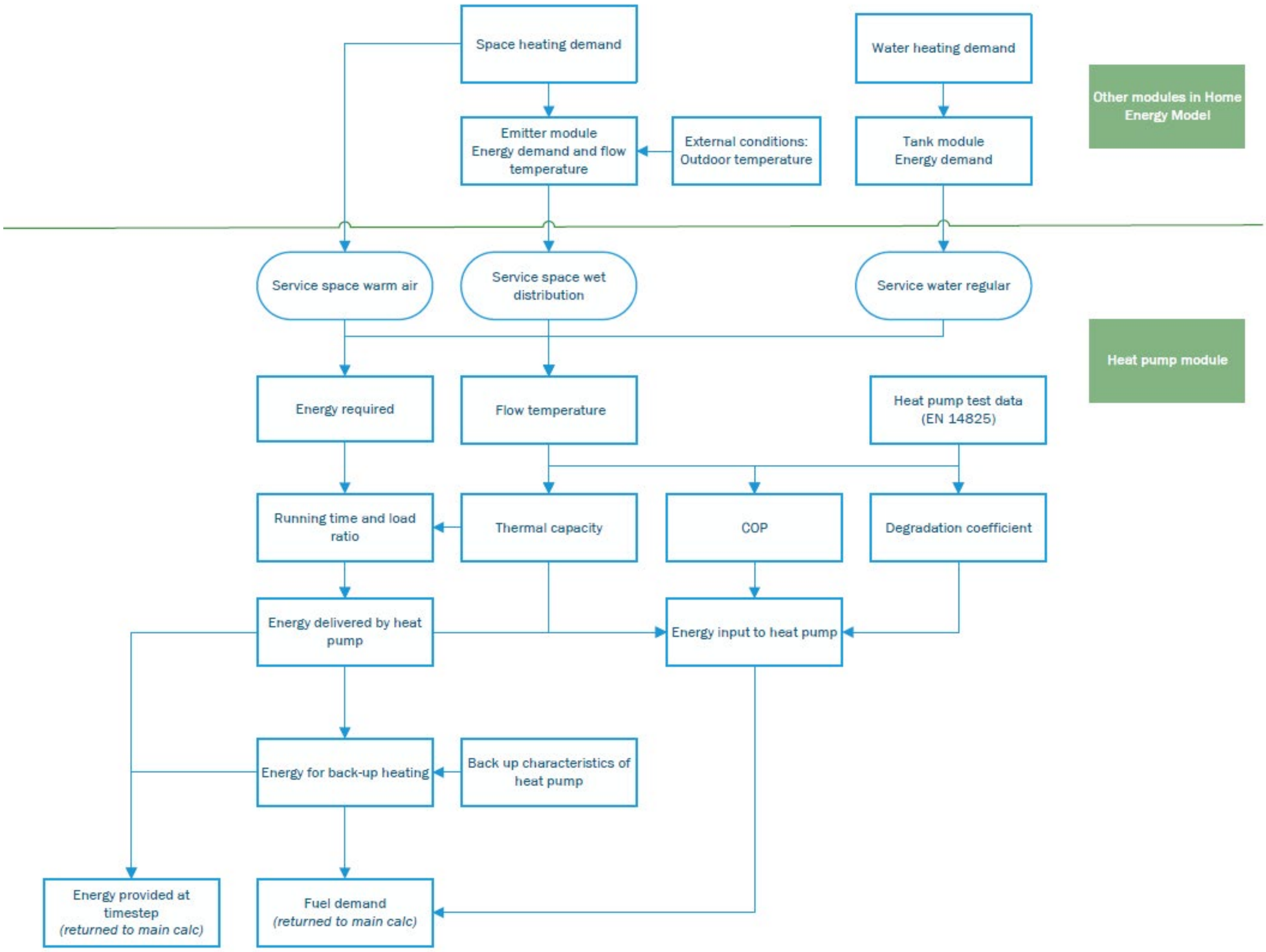

Figure 1 – Simplified flowchart of calculation steps for heat pumps in HEM

1. Energy requirement

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, the energy demand and flow temperature (i.e., sink temperature) during each timestep are calculated by the emitter module and are provided as inputs to the heat pump module. 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 carried out during the test.

2. Performance under operating conditions

2.1 Source temperature

Recognised heat pumps are grouped by heat source:

- Ground (Indirect/Closed-loop)
- Air source
- Exhaust Air MEV[1]
- Exhaust Air MVHR[2]
- Fifth-generation heat network

Ground source heat pumps

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

$$\theta_{gen,in} = T_o \times 0.25806 + 2.8387 \tag{1}$$

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, but amended to UK conditions during the development of previous calculation method.

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.

Exhaust air heat pumps

For exhaust air heat pumps, a fixed source temperature of 20ºC is assumed.

[1] 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.
[2] 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.

Fifth generation heat networks

For fifth generation heat pumps, the source temperature ($\theta_{gen,in}$) is expected provided by the heat network developer, either directly or via a PCDB (or future equivalent) entry for the heat network (to be considered later).

2.2 Coefficient of Performance (COP)

Note: COP figures measured by EN 14825 tests are assumed to be measured at the SEPEMO 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.

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 EN 14825 tests (Measurements A to D and F). If tests are undertaken with fixed outlet (flow) temperature control as well, the calculation steps defined in the section “Other cases” below, 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 and then interpolated based on the flow temperature under operating conditions.

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

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

Where:

a, b and c are the best fit linear 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 interval. This COP is then corrected to account for the operating temperatures as follows (based on EN 15316-4-2:2017, equation D.4).

$$COP_{gen}(\theta_{in};\theta_{out}) = COP_{gen;T_o} \times \frac{(\theta_{gen;out;X}) \times (\theta_{gen;out;ref} - \theta_{gen;in;ref})}{(\theta_{gen;out;ref}) \times (\theta_{gen;out;X} - \theta_{gen;in;X})} \quad \textbf{(3)}$$

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

Other cases

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):

$$COP_{gen,\theta in,\theta out;\Delta\theta}(t) = f_{LR;exer;X(t)} \times COP_{gen,exer}(t) \times f_{COP;\theta in,\theta out;\Delta\theta} \quad \textbf{(4)}$$

Where:

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

$f_{LR;exer;X(t)}$ 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):

$$COP_{gen;exer} = \frac{\theta_{gen;out}(t)}{\theta_{gen;out}(t) - \theta_{gen;in}(t)} \quad \textbf{(5)}$$

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

$$LR_{exer;X} = \frac{COP_{exer;X}}{COP_{exer;cld}} \times \left[\frac{\left(\theta_{out;cld}/\theta_{in;cld}\right)}{\left(\theta_{out;X}/\theta_{in;X}\right)}\right]^{n\_exer} \quad \textbf{(6)}$$

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.

If EN 14825 test data for more than one design flow temperature has been provided then the calculation of exergetic load ratio under operating conditions is done for each design flow temperature and then interpolated based on the flow temperature under operating conditions.

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 operating temperatures using the nearest exergetic load factors (XX and YY) as shown below (based on EN 15316-4-2:2017, equation 38).

$$f_{LR;exer;X}(t) = f_{LR;exer;XX} - \left(f_{LR;exer;XX} - f_{LR;exer;YY}\right) \times \left[\frac{LR_{exer;XX} - LR_{exer}(t)}{LR_{exer;XX} - LR_{exer;YY}}\right] \quad \textbf{(7)}$$

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 calculation of exergetic efficiency under operating conditions is done using interpolated figures, where each figure is calculated for each design flow temperature and then interpolated based on the flow temperature under operating conditions.

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:

$$f_{COP;\theta in,\theta out;\Delta\theta} = \left[1 - \frac{(\Delta\theta_{gen;out;ref} - \Delta\theta_{gen;out}(t))/2}{\theta_{gen;out}(t) - \Delta\theta_{gen;out;ref}/2 + \Delta\theta_{HP;gen;cond;int} - (\theta_{gen;in}(t) - \Delta\theta_{HP;gen;evap;int})}\right] \quad \textbf{(8)}$$

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[3]).

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 operating 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.

[3] 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 correct.

If EN 14825 test data for more than one design flow temperature has been provided then the calculation of the temperature spread correction factor is done for each design flow temperature and then interpolated based on the flow temperature under operating conditions.

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 “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 calculations below of thermal capacity under operating conditions is done for each design flow temperature and then interpolated based on the flow temperature under operating conditions.

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.

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.

2.4 Degradation coefficient

The degradation coefficient represents two effects:

- the power consumption of the unit when the compressor is off.
- refrigerant pressure equalization that reduces the heating capacity when the unit is restarted.

A value is quoted for each test condition (A - D and F). Note that if EN 14825 test data for more than one design flow temperature has been provided then the calculation below of degradation coefficient under operating conditions is done for each design flow temperature and then interpolated based on the flow temperature under operating conditions.

For air-to-water units and water-to-water units, the degradation due to the pressure equalization effect when the unit restarts can be considered as negligible (8.4.3 EN14825-2006). This means it is straightforward to obtain from the degradation coefficient the power consumption of the unit when the compressor is off.

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 degradation coefficient (Cdh) from the EN 14825 tests is used.

Other cases

For variable capacity control heat pumps or fixed capacity control with variable output temperature, the two values of exergetic load factor calculated from the test data either side of the exergetic load factor for the calculation interval operating conditions are obtained. Additionally, the two corresponding values of the degradation coefficients are obtained.

By linear interpolation the degradation coefficient under operating conditions is calculated as follows:

$$Cdh(t) = Cdh_{XX} - (Cdh_{XX} - Cdh_{YY}) \times \left(\frac{LR_{exer;XX} - LR_{exer(t)}}{LR_{exer;XX} - LR_{exer;YY}}\right) \quad \textbf{(11)}$$

Where:

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

Cdh_{XX} and Cdh_{YY} are the degradation coefficients for the test records with the closest exergetic load ratios below and above the exergetic load ratio under operating conditions.

3. Running time and load ratio

Calculate the full-load operating time and load ratio of each service from the energy requirement and the heat capacity of the heat pump using the following (based on EN 15316-4-2:2017, equation 35):

$$t_X = \frac{Q_{\theta in,\theta out;X;out}}{\varphi_{\theta in,\theta out;X}} \text{ and } LR_X = \frac{t_X}{t_{ci}} \quad \textbf{(12)}$$

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[4]

t_{ci} is the calculation timestep.

For the highest-priority service required, the operating time is subject to a maximum of the calculation interval. For lower-priority services, the operating time is subject to a maximum of the calculation interval duration minus the operating time of any higher-priority services.

For exhaust air heat pumps requiring overventilation, the running time is used to calculate the mechanical ventilation throughput factor for the timestep:

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

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.

R_{hp} is the overventilation ratio

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. 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.

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

4. Energy delivered and consumed by heat pump

Note: If the backup heater is operating instead of the compressor (see), 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:

$$Q_{X;gen,out}(t) = \varphi_{\theta in;\theta out;X}(t) \times t_X \quad \textbf{(13)}$$

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):

$$E_{gen;\theta in,\theta out,\Delta\theta,X}(t) = \frac{Q_{X;gen,out}(t)}{COP_{gen;\theta genin;\theta genout}(t)} \quad \textbf{(14)}$$

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.

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, known as the minimum modulation rate ($LR_{cont;min}(t)$). The minimum modulation rates can be calculated at flow temperatures 20°C, 35°C and 55°C. The fixed minimum modulation rate is determined by:

1. Obtaining the heat pump capacity at EN 14825 test condition C (source temperature: 7/6°C) using the relevant flow temperature
2. Obtaining the EN 14511 standard rating condition capacity at the flow temperature (source temperature: 7/6°C)
3. Dividing 1. by 2.

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.

The minimum modulation rate is a flow temperature dependent variable determined for each time-step by linear interpolation between two fixed minimum rates that are declared by the heat pump manufacturer.

Compressor power at full load

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

$$P_{gen,LR100,X}(t) = \frac{\varphi_{\theta in;\theta out;X}(t)}{COP_{gen;\theta genin;\theta genout}(t)} \tag{15}$$

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.

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):

$$P_{gen,LR;comp;min,X}(t) = P_{gen,LR100,X}(t) \times LR_{cont;min,X}(t) \tag{16}$$

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.

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):

$$P_{gen;comp;ONOFF;LR}(t) = P_{gen;LR;comp;min,X}(t) \times \frac{\tau_{eq} \times LR \times (1 - LR)}{\tau_{out,em,type}} \tag{17}$$

LR is the load ratio under operating conditions.

τ_{eq} is a characteristic parameter of the heat pump, due to the inertia of the on/off transient (assumed to be 140 seconds).

$\tau_{out,em,type}$ 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.

Energy input (wet distribution)

For air-to-water, water-to-water and brine-to-water units, the corrected heat pump driving energy can be calculated using:

$$E_{X;gen,in}(t) = \Big((1 + f_{aux}) \times P_{gen,LR100,X}(t) + P_{gen,comp,ONOFF,LR}\Big) \times t_X + E_{X;gen,aux} \tag{18}$$

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[5].

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

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

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

t_X is the running time for the service.

$E_{X;gen,aux}$ 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 (18 and adds it later as part of the ancillary energy calculation.

Energy input (warm air distribution)

For air to air, brine to air and water to air units, the corrected heat pump driving energy can be calculated using Equation 25.

For air to air, brine to air and water to air units the power consumption of the unit when the compressor is off cannot be separated from the pressure equalization effect so Equation (19 does not apply.

The degradation coefficient is obtained according to Equation (11. If the calculated value of $Cdh(t)$ is higher than 0.25 then restrict to 0.25, and if is lower than 0 then restrict to 0.

$$E_{X;gen,in} = \frac{\left((1 + f_{aux}) \times P_{gen;\theta in;\theta out}(t) + P_{gen,comp,ONOFF,LR}\right) \times t_X}{\left(1 - Cdh \times \left(1 - \frac{LR_X}{LR_{cont;min,X}}\right)\right)} \quad (19)$$

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 back-up heater

5.1 Control of backup heater

Back-up energy is required when:

- the source temperature is below the declared temperature operating limit of the heat pump (TOL).
- the required flow temperature is above the maximum operating 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.

5.2 Energy delivered/consumed

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

Back-up energy 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 backup heater. The maximum thermal capacity of the backup heater is calculated as follows:

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

Where:

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

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

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

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

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

If the backup heater is operating in addition to the compressor, then the backup energy 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.

ϕ_X is the energy output provided (in the absence of backup heating) for the service.

t_X is the running time for the service.

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.

6. Ancillary and auxiliary energy

6.1 Ancillary energy

Ancillary energy is the consumption of the compressor unit during the off part of the on/off cycle. This part of the calculation involves the degradation coefficient (Cdh) from EN 14825 tests for heat pumps using wet distribution only. For heat pumps using wet distribution only if the calculated value of $Cdh(t)$ is lower than 0.9 then restrict to 0.9 (default of EN 14825 standard), and if higher than 1 then restrict to 1.

The energy consumption of the compressor unit during the off part of the on/off cycle is calculated as follows:

$$E_{gen,anc,off} = (1 - Cdh(t)) \times P_{gen;LR100,X}(t) \times t_{off,X} \tag{23}$$

Where:

$Cdh(t)$ is the degradation coefficient under operating conditions.

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

$t_{off,X}$ is the time remaining in the calculation timestep after all services have run.

Note that this is only calculated for the lowest-priority service that runs during the timestep:

1. For the first service calculation, if during the calculation interval only this service is provided for the time period (i.e. $t_1 > 0$, t_2 = 0 and t_3=0), then set $t_{off,1} = t_{ci} - (LR_1/LR_{cont,min,1} \times t_{ci})$, subject to a minimum of zero, and calculate $E_{1,gen,aux}$. If either a second or third service is also provided during the calculation interval $E_{1,gen,aux}$ = 0.
2. For the second service calculation, if during the calculation interval the second service is provided for time period, and not the third service (i.e. t_2 > 0 and t_3 = 0), then set $t_{off,2} = t_{ci} - t_1 - (LR_2/LR_{cont,min,2} \times t_{ci})$, subject to a minimum of zero, and calculate $E_{2,gen,aux}$. If the third service is provided $E_{2,gen,aux}$= 0.
3. And so on for subsequent services.

6.2 Auxiliary energy

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 manufacturer declared rated brine circulation pump power multiplied by the heat pump running time. This pump must be included in the heat pump package.

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 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.

When operating at continuous, but part load conditions, the running time is as follows:

- When only a single service is provided in the calculation interval then the running time is set to the calculation interval.
- When two or more services are provided in the same calculation interval, the running time of the lowest priority service is set to the calculation interval minus the time the

other services operate at full load. The running times of other services are set to the times they would operate at full load.

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 (24. 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:

$$E_{gen;in;LR0}(t) = (P_{SB} + P_{CK}) \times \left(t_{ci} - \sum\nolimits_{X} t_X\right) \quad (24)$$

Where:

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

t_X 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:

$$E_{gen;in;LR0}(t) = P_{SB} \times \left(t_{ci} - \sum_{X} t_X\right) \tag{25}$$

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

$$E_{gen;in;LR0}(t) = P_{off} \times t_{ci} \tag{26}$$

Methodology - Heat pumps providing water heating only

For hot water only heat pumps EN 16147 test data to load profile "M" must be available, otherwise an 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:

$$\eta_W = \frac{\left(E_{DTP} + \left(Q_{WS,ls,24} \times 0.6 \times 0.9\right)\right) \times 100}{\left(Q_{elec} - \left(P_{es} \times 24 \times 0.6 \times 0.9\right) + \left(\frac{Q_{WS,ls,24}}{COP_{DHW}} \times 0.6 \times 0.9\right)\right)} \quad \textbf{(27)}$$

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 PCDB (or future equivalent). 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 (kW) measured in the EN 16147 test

COP_{DHW} is the COP measured in the EN 16147 test (System boundary SEPEMO H4)

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 SEPEMO H4, which means it includes the storage tank. As HEM models the storage tank separately, Equation (27 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.

Future development

The following features are being considered for integration into the HEM heat pump methodology:

- Storage operation which can supply heating or hot water at a later time.
- Water source heat pumps where water is extracted from the ground and re-injected into the ground or discharged at the surface or where water is extracted from surface water, such as rivers and lakes.
- Exhaust air mixed which uses either an MEV or MVHR configuration for air supply to the evaporator in conjunction with external air supply.
- Solar assisted heat pumps which use solar PV to heat hot water in the storage tank.
- Buffer tanks which reduce cycling in heat pumps, but may reduce overall system performance in other ways (e.g. by reducing flow temperatures to emitters).

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.

Annex A – System boundaries

Equations used for the determination of heat pump annual performance are shown below. The boundaries were defined during the 2011 SEPEMO-Build project (Seasonal performance factor and monitoring for heat pump systems in the building sector).

$$COP_{\text{SEPEMO H1}} = \frac{\text{heat pump heat output}}{\text{electricity input for heat pump}}$$

$$COP_{\text{SEPEMO H2}} = \frac{\text{heat pump heat output}}{(\text{electricity input for heat pump} + \text{electricity for circulation pump on borehole or ground array})}$$

$$COP_{\text{SEPEMO H3}} = \frac{(\text{heat pump heat output} + \text{backup heater heat output})}{(\text{electricity input for heat pump} + \text{electricity for circulation pump on borehole or ground array} + \text{backup heater electricity use})}$$

$$COP_{\text{SEPEMO H4}} = \frac{(\text{heat pump heat output} + \text{backup heater heat output} - \text{tank losses})}{(\text{electricity input for heat pump} + \text{electricity for circulation pump on borehole or ground array} + \text{backup heater electricity use} + \text{electricity for external circulating pumps})}$$

The COP from the EN 14825 test data has a system boundary of SEPEMO H1.

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

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