15TH IEA HEAT PUMP CONFERENCE
MAY 26–29, 2026 | HOFBURG
VIENNA 2026
DECARBONISATION THROUGH INNOVATION

From Refrigerant Cycle to Product Testing: Fostering Innovation in Heat Pump Controllers through Real-Life Testing

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Abstract

At the EU policy level, it is discussed to revise the test method for heat pumps within the framework of the EU Energy Labelling regulation for central space heating appliances. The proposed change aims to replace current stationary measurements (component testing), conducted according to EN 14511:2022 and requiring manufacturer support to force steady-state operation, with a load-based testing approach (product testing) known as the compensation method.

The new compensation testing method couples a building model to hydronic test benches to emulate a realistic response of the hydronic system, resulting in dynamic operating behavior. The heat pump is assessed in its normal operating mode, without any additional inputs from the manufacturer, paving the way towards independent testing and rating. The resulting evaluation parameter (COP) accounts for the efficiency of the refrigerant cycle, the backup heater, and the heat pump controller, which is closer to reality than steady-state measurements. Moreover, this method is technologically neutral and enables a level playing field for all technologies, including future compressor-less heat pumps.

While the emulator was already introduced in the literature, this work introduces new features of the method designed to test dynamic hydronic flow characteristics. Specifically, it incorporates a perfect hydraulic switch to allow variable flow rates of the heat pump within one test point. The target parameter (setpoint) of the method is the average heat flow provided by the heat pump. This allows the manufacturer to resolve the trade-off between mass flow and supply temperature in a system-specific manner. The method is performed with different heat pumps in several laboratories to prove reproducibility.

Implementing the proposed load-based test via future Energy Labelling regulations will incentivize better controls through more representative testing, allow for a fair comparison of different products and technologies, and enable independent market surveillance.

Selection and/or peer-review under the responsibility of the organizers of the 15th IEA Heat Pump Conference 2026.

Keywords: Standardization; representative testing; compensation method; load-based testing; building emulator;

1. Introduction

At the EU policy level, it is discussed to change the test method for heat pumps in the scope of Ecodesign and Energy Labelling for lot 1 (central space heating appliances). In the future, stationary measurements according to EN 14511:2022, which can only be carried out with the manufacturer's support, should be replaced by a load-based test (compensation method) to ensure independent testing and real-life operating

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behavior. The compensation method developed by the Federal Institute for Materials Research and Testing (BAM) in collaboration with independent research and testing institutes is currently under discussion by CEN/TC 113/WG 8. The method aims to introduce an assessment metric that takes a holistic view of the heat generation system. It comprises the heating devices (heat pump, supplementary heater) and the underlying control systems: heat pump controller, including the compressor control, superheat control and volume flow control during partial load and cyclic operation, the defrost controller, and the supplementary heater control.

Previous studies show that the test stand inertia must be aligned to ensure similar operating behaviour in different labs [1], [2] . To overcome this problem, physical and virtual definitions for the inertia were discussed. While the physical implementation results in:

- specific buffer tanks depending on the heat pump's rated heat output,
- different stratification and mixing behavior of tanks and
- additional deviation due to the test benches' control settings (PID),

the virtual inertia implementation calculates directly the set return temperature $\vartheta_{\mathrm{r,set}}$ and does not require standardized test rigs. Through the virtual implementation, testing labs can determine their control quality by comparing measured and calculated return temperature.

The emulator approach is well known from Hardware-in-the-Loop (HiL) tests widely applied in research institutes and increasingly also in manufacturers' R&D labs. However, no standardised methodology was available for product comparison. Different groups worldwide developed very similar load-based testing methodologies for air-to-air units independently [3]. They all rely on a room emulator to align the inertia of different test rigs.

Hence, BAM, RWTH, and AIT developed a straightforward model to emulate the response of a representative building. The initial model was then refined and parameterised by RWTH. The model was required to be in line with EN 14825 [4], which already assumes a building for the test (e.g., prescribed load line) and requires the manufacturer to declare a design capacity (P_{design}). Thus, coupling the heat pump with a virtual building model through the test stand is in line with the spirit of EN 14825 [5]. Recently, the German Environment Agency (UBA) incorporated a reference to the load-based test including the building emulator within its award criteria for the 'Blue Angel' quality label for heat pumps [6].

This work introduces the method, its implementation, and its parametrization using building performance simulations (Section 2). Section 3 shows the building emulator's application with experimental results. Potential improvements are discussed with respect to the trade-off between accuracy and simplicity in Section 4.

2. Model implementation

The objective of the building emulator is to test heat pump systems under realistic and reproducible conditions. The heat pump is directly connected to a radiator or underfloor heating (UFH) system. Further hydraulic components, such as buffer tanks or domestic hot water tanks, are neglected. The building is considered as one zone with one heating cycle. The heat pump delivers the water mass flow with a supply temperature. A return temperature results from the heating cycle in the building.

A building emulator (two-mass model) calculates, and a hydraulic test bench emulates the return temperature depending on mass flow and supply temperature. The following section describes the implementation of the model that current version can be used to test according the most relevant temperature application of EN14825: medium temperature (equivalent to radiator) and low temperature (equivalent to UFH) [5].

2.1. Implementation

Figure 1 shows the schematic structure of the developed model. In approximation, "Mass H" represents the heating system (e.g., radiator or UFH), and "Mass B" the building envelope. "Heat Flow I" describes the heat provided by the heat pump system. "Heat Flow II" represents the heat transfer between the heating system and the building, and "Heat Flow III" describes the heat losses from the building to the ambient.

To emulate the inertia of the virtual building, the return temperature $\vartheta_{\mathrm{R,emu}}$ (and the building temperature $\vartheta_{\mathrm{B,emu}}$, where applicable) is controlled by the test stand according to the two-mass building model. For each time step Δt_{step}, the model calculates the return temperature $\vartheta_{\mathrm{R,calc}}$ and the building temperature $\vartheta_{\mathrm{B,calc}}$ (output variables) based on the measured supply temperature ϑ_S and the heating capacity $\dot{Q}_{\mathrm{HP}}$ (input variables) of the unit under test. Through the measured heating output, the model considers the measured supply temperature ϑ_s, emulated (measured) return temperature $\vartheta_{\mathrm{R,emu}}$, and measured mass flow rate $\dot{m}_{\mathrm{W}}$.

Thus, the test stand dynamically adapts the compensation load to match the calculated return temperature $\vartheta_{\mathrm{R,calc}}$. The heat pump responds in accordance with its heating curve, its indoor air temperature control, or both as it tries to maintain the required supply temperature ϑ_s. The outdoor room shall maintain constant conditions over a temperature range associated with different climate zones for testing air-to-water heat pumps; for ground-sourced heat pumps the outdoor heat exchanger inlet water temperature $\vartheta_{\mathrm{w,in}}$ shall be maintained constant according to the referenced standards. All heat pumps under test (i.e., single-speed, multiple-speed, and variable-speed) shall be operated with their onboard control system (native control) active and not in a fixed-speed mode. The heat pumps are thus permitted to switch into on/off operation of the compressor.

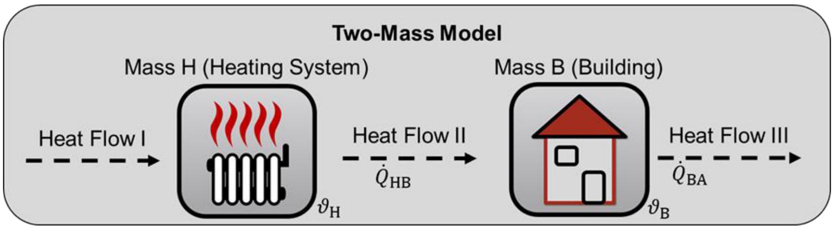

Figure 1. Schematic overview of two-mass building model.

"Heat Flow I" describes the actual heat provided by the heat pump system. Therefore, the heat output is measured directly at the machine's in- and outlet:

$$\dot{Q}_{\mathrm{HP}}(t) = \dot{m}_{\mathrm{W}} \cdot c_{\mathrm{p}} \cdot \left(\vartheta_{\mathrm{s}}(t) - \vartheta_{\mathrm{R,emu}}(t)\right) \tag{1}$$

The mass flow $\dot{m}_{\mathrm{W}}$, the supply temperature $\vartheta_{\mathrm{s}}(t)$ and the return temperature $\vartheta_{\mathrm{R,emu}}(t)$ are measured values from the test bench. To calculate the heat flow, the heat capacity c_p is needed.

"Heat Flow II" represents the heat flow between the heating system and the building (eq. 2). The heating system's implementation approximates all pipes and the radiator or UFH as one homogeneous mass, assuming that the temperature of the heating system corresponds to the return temperature ($\vartheta_{\mathrm{R,emu}} = \vartheta_{\mathrm{H}}$).

$$\dot{Q}_{\mathrm{HB}}(t) = (UA)_{\mathrm{HB}} \cdot \left(\frac{(\vartheta_{\mathrm{S}}(t)+\vartheta_{\mathrm{R,calc}}(t)}{2} - \vartheta_{\mathrm{B,calc}}(t)\right) \tag{2}$$

The thermal conductance between the heating system and the building is described by the parameter $(UA)_{\mathrm{HB}}$. $\vartheta_{\mathrm{R,calc}}(t)$ and $\vartheta_{\mathrm{B,calc}}(t)$ are the current temperatures of the Mass H and the building, respectively. The present implementation uses the arithmetic average between the temperature at the heat transfer system's inlet $\vartheta_{\mathrm{S}}(t)$ and outlet $\vartheta_{\mathrm{R,calc}}(t)$.

The heat flow heats the building represented by mass B. The dissipated heat flow from the building into the environment (Heat flow III) is described below.

$$\dot{Q}_{\mathrm{BA}}(t) = (UA)_{\mathrm{BA}} \cdot \left(\vartheta_{\mathrm{B,calc}}(t) - \vartheta_{\mathrm{i,A}}\right) \tag{3}$$

The parameter $(UA)_{\mathrm{BA}}$ describes the conductance between the building and the ambient and considers all types of heat losses (heat flow through building envelope, air exchange losses, etc.). The ambient temperature $\vartheta_{\mathrm{i,A}}$ is represented by a constant value for each specific test point.

The temperature of the heating system $\vartheta_{\mathrm{R,calc}}(t)$ represents the return temperature ($\vartheta_{\mathrm{R,calc}}(t) = \vartheta_H(t)$) and is the set value for the test bench. The temperature can be calculated by solving an energy balance around mass H in each time step. The change of the internal energy of mass H, is calculated by the following equation:

$$\frac{dU_{\mathrm{H}}}{dt} = \dot{Q}_{\mathrm{HP}}(t) - \dot{Q}_{\mathrm{HB}}(t) \tag{4}$$

The measured heat pump heat output $\dot{Q}_{\mathrm{HP}}(t)$ and the optional heat of the virtual backup heater $\dot{Q}_{\mathrm{BUH,calc}}(t)$ flow into the mass while $\dot{Q}_{\mathrm{HB}}(t)$ is the heat flow from mass H into the building. The internal energy can be described using the temperature $\vartheta_{\mathrm{R,calc}}$ and capacity C_{H} of mass H:

$$C_{\mathrm{H}} \frac{d\vartheta_{\mathrm{R,calc}}}{dt} = \frac{dU_{\mathrm{H}}}{dt} \tag{5}$$

By integrating the equation over time, the temperature of mass H for the next time step $\vartheta_{\mathrm{R,calc}}(t + \Delta t_{\mathrm{step}})$ is calculated.

$$\vartheta_{\mathrm{R,calc}}(t + \Delta t_{\mathrm{step}}) = \vartheta_{\mathrm{R,calc}}(t) + \frac{\left(\dot{Q}_{\mathrm{HP}}(t) - \dot{Q}_{\mathrm{HB}}(t)\right) \cdot \Delta t_{\mathrm{step}}}{C_{\mathrm{H}}} \tag{6}$$

Equivalent to mass H, the equations for mass B can be formulated:

$$\vartheta_{\mathrm{B}}(t + \Delta t_{\mathrm{step}}) = \vartheta_{\mathrm{B}}(t) + \frac{\left(\dot{Q}_{\mathrm{HB}}(t) - \dot{Q}_{\mathrm{BA}}(t)\right) \cdot \Delta t_{\mathrm{step}}}{C_{\mathrm{B}}} \tag{7}$$

In total, four parameters are needed to define the model: The Thermal conductance between heating system and building UA_{HB}, the thermal conductance between building and ambient U, A_{BA} and the thermal capacities C_{H} and C_{B}. The definitions of the parameters are explained in the following section.

2.2. Parametrization using building performance simulation

The model parameters can be deduced from the design heating capacity and the system's thermal inertia, defined by the time constant τ. To do this, the thermal conductance is determined in the first step, followed by the capacities.

2.2.1. Thermal conductivities UA_{HB} and UA_{BA}

In terms of steady-state conditions, all temperature gradients are zero, resulting from equal heat flows between the heat pump, the two masses, and the ambient.

$$\dot{Q}_{\mathrm{HP}}(t) = \dot{Q}_{\mathrm{HB}}(t) = \dot{Q}_{\mathrm{BA}}(t) = \mathrm{Ph}(\mathrm{T}_i) = \dot{Q}_{\mathrm{designh}} \tag{8}$$

For the heat flow between the heating system and the building, we obtain the following equation:

$$\dot{Q}_{\mathrm{HB}}(t) = UA_{\mathrm{HB}} \cdot \Delta T = UA_{\mathrm{HB}} \cdot \left(\frac{\vartheta_{\mathrm{S,design}} + \vartheta_{\mathrm{R,design}}}{2} - \vartheta_{\mathrm{B,design}}\right) = \mathrm{Ph}(\mathrm{T}_i) \tag{9}$$

which can be formulated as

$$UA_{\mathrm{HB}} = \frac{Ph(T_i)}{\vartheta_{i,\mathrm{s,design}} - 0.5\, \Delta\vartheta_{\mathrm{i,cond,design}} - \vartheta_{\mathrm{B,design}}} \tag{10}$$

The design parameters for each test point i, $\vartheta_{i,\mathrm{s,design}}$ and $\Delta\vartheta_{\mathrm{i,cond,design}}$ can be taken from the standard EN 14825. If the mass flow of the heat pump is constant for all test points, the temperature difference over the condenser for the test point i is defined as:

$$\Delta\vartheta_{\mathrm{i,cond,design}} = \frac{\mathrm{Ph}(\mathrm{T_i})}{\dot{m}_{\mathrm{W,design}} \cdot c_p} \tag{11}$$

The design building temperature $\vartheta_{\mathrm{B,design}}$ is equal to 20 °C. The design mass flow $\dot{m}_{\mathrm{W,design}}$ is defined as:

$$\dot{m}_{\mathrm{W,design}} = \frac{\mathrm{Ph}(\mathrm{T_i})}{\Delta T_{\mathrm{LT,MT}} \cdot c_{p,\mathrm{w}}} \tag{12}$$

The temperature difference is defined as 8 K for medium-temperature application (MT) and 5 K for low-temperature application (LT).

Similar to the thermal conductance between mass H and mass B, the thermal conductance between mass B and the ambient (A) can be obtained as follows:

$$UA_{\mathrm{BA}} = \frac{Ph(T_i)}{(\vartheta_{\mathrm{B,design}} - \vartheta_{i,\mathrm{A}})} \tag{13}$$

With the given conductivities, the thermal capacities can be calculated.

2.2.2. Thermal capacities C_H and C_B

The thermal capacities are relevant for describing the system's inertia. Since the model is used for different thermal powers, the capacities have to scale with the heat pump's rated heat output. To guarantee the same behavior for different building energy systems (heating system and building envelope), the time constants τ_{H} and τ_{B} are introduced to describe the inertia of the system.

For a simple step response (heat pump turns off, $\vartheta_{\mathrm{S}}(t) = \vartheta_{R,\mathrm{calc}}(t)$) the energy balance around the heating system is:

$$C_{\mathrm{H}} \cdot \frac{d\vartheta_{\mathrm{H}}(t)}{dt} = -(UA)_{\mathrm{HB}} \cdot \left(\vartheta_{\mathrm{R,calc}}(t) - \vartheta_{\mathrm{B,calc}}(t)\right) \tag{14}$$

By integrating the equation over time, the following equation is obtained:

$$\vartheta_{\mathrm{H}}(t) = \vartheta_{\mathrm{B}}(t_0) + \left(\vartheta_{\mathrm{H}}(t_0) - \vartheta_{\mathrm{B,calc}}(t)\right) \cdot e^{-\frac{(UA)_{\mathrm{HB}}}{C_{\mathrm{H}}} \cdot t} \tag{15}$$

The simple time constant is defined when (1-e^{-1}) of the whole step is complete or when $t = C_{\mathrm{H}}/(UA)_{\mathrm{HB}}$. Therefore, the time constant for mass H τ_{H} is defined as $C_{\mathrm{H}}/(UA)_{\mathrm{HB}}$. Since the time constants of the building and the heating system differ by more than one order of magnitude, the differential equations are considered separately. Therefore, the time constant for the building is determined equivalent to that of the heating system as $C_{\mathrm{B}}/(UA)_{\mathrm{BA}}$.

With the known time constants of both masses τ_{H} and τ_{B}, both capacities are defined:

$$C_{\mathrm{H}} = \tau_{\mathrm{H}} \cdot (UA)_{\mathrm{HB}_{\mathrm{PLC=E}}};\ C_{\mathrm{B}} = \tau_{\mathrm{B}} \cdot (UA)_{\mathrm{BA,PLC=E}} \tag{16, 17}$$

The capacities are only calculated once in the design test point (PLC-E) by using the time constants and the conductance of Eq. (10,13).

The following parameters design the model:

- $Ph(T_i)$: Design part load condition
- $\vartheta_{\mathrm{i,s,design}}$: Design supply temperature
- τ_{H}: Time constant for mass H
- τ_{B}: Time constant for mass B

The following section contains derivations of the time constants for the heating mass H and the building mass B. We distinguish between low and medium temperature levels, linked to underfloor and radiator heating systems, respectively.

2.3. Derivation of the time constants

The model's time constants describe the system's inertia and become relevant during dynamic measurements, particularly during on-off cycling and defrost operation. Since different temperature levels in the current standard EN 14825 represent different system types, we link the medium temperature level to a radiator system and the low-temperature level to a UFH. To get representative time constants for the system, we use detailed models for

- The building to define the time constant τ_{B}
- The radiator to define the time constant $\tau_{\mathrm{H,MT}}$
- The UFH to define the time constant $\tau_{\mathrm{H,LT}}$

2.3.1. Representative time constant for the building mass τ_B

To obtain the time constant of the building mass τ_{B}, we use a building performance model written in Modelica. Using the open-source library Teaser [6], we generate typical buildings from the Tabula database [7]. The building from 1975 with refurbished windows has a net floor area of 150 m^2. In each orientation (north, east, south, west) windows of 7.4 m^2 are considered. The resulting heat load for an ambient temperature of -10 °C is 12.680 kW.

The step response shown in Figure 2 is performed by shutting off the heat source after a steady-state operation. The Figure shows the cooling process of the detailed Modelica model benchmarked to the simplified model. The simplified model is only the second mass (Mass B) of the two-mass model. The ambient temperature is kept constant and the heat flux from the heating system is either equal to the heat load or zero.

The simplified model is parametrized to have an intersection point with the detailed model at 15 °C. The intersection at 15 °C is chosen to minimize the error at the beginning of the cooling phase, since during the standard test, no long cooling phases without heating are expected.

At the beginning of the step response, the temperature of the building's indoor air volume decreases fast until it reaches the concrete temperature. Afterward, the building's concrete and air temperature decrease at the same speed. A more accurate description of the building's behavior is only possible with a more detailed model and is discussed Section 4.1. The resulting time constant of the building τ_{B} is 240563 s.

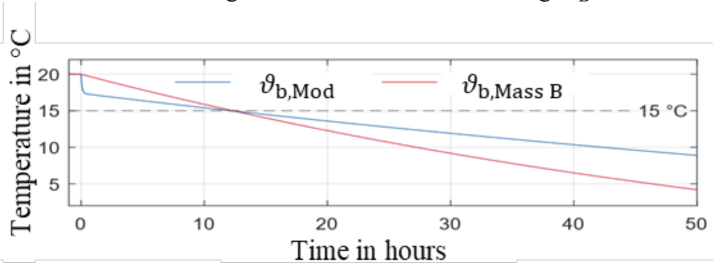

Figure 2. Step response of building envelope compared with simplified model.

2.3.2. Representative time constant for the radiator heating system

The time constant for the radiator heating system is determined by a step response using a detailed radiator model [8] written in Modelica. The radiator model can be used as a dynamic or steady-state model. The required parameters are data that are typically available from manufacturers that follow the European Standard EN 442-2 [9].

However, to allow for varying mass flow rates, the transferred heat is computed using a discretization along the water flow path, and heat is exchanged between each compartment and a uniform room air and radiation temperature. This discretization is different from the computation in EN 442-2, since the standard may yield water outlet temperatures that are below the room temperature at low mass flow rates. Furthermore, rather than using only one room temperature, this model uses a room air and room radiation temperature. The model is parametrized for a heat load of 7 kW, nominal inlet temperature of 55 °C and outlet temperature of 47 °C. The nominal water mass flow results from the previous parameter. The radiator is coupled to an infinite room with a constant temperature of 20 °C.

The step response is carried out as follows:

First, the radiator is operated in a stationary state until all variables no longer show any change over time. The temperature of the flow is then set equal to that of the return. The water flow then cools down. Now the time is determined at which 63.21 % $(1-e^{-1})$ of the jump is completed. With the given start temperature of 47 °C and the constant room temperature of 20 °C, 63.21 % of the step is fulfilled when the return temperature reaches 37.07 °C. The resulting time constant for the radiator system is identified: $\tau_{\mathrm{H,MT}} = 2384\ \mathrm{s}$

Figure 3 (top) shows the step response of the radiator system in Modelica (blue line) compared with the step response of the one-mass model (red line). In general, the curves match well. The main difference can be determined at the beginning of the step response. Here, the detailed model shows higher temperatures and a later start of the step. This results from the implemented discretized pipes in the model, resulting in a dead time, which the one-mass model does not have.

2.3.3. Representative time constant for the underfloor heating system

To determine the representative time constant for the UFH $\tau_{\mathrm{H,MT}}$ the same approach as for the radiator is used. A detailed UFH is used to perform a step response. The UFH model [10] automatically parametrizes the UFH system according to prEN 1264-3:2020-02 [11]. Since the model represents a whole underfloor system, it consists of the system level of the distributor and several heating circuits and takes, at the smallest level, the heat transfer from a pipe element through different floor layers into account. The model is verified for the system requirements according to prEN 1264-2:2020-02.

The step response is determined with the same principle as for the radiator system. The supply temperature at the beginning is 35.97 °C, resulting in a return temperature of 31.01 °C. These design parameters are part of the model resulting from inputs like pipe spacing, pipe material and pipe thickness. All used parameters are summarized in Table 1. The wall layers are parametrized according EnEV2009.
The result of the step response is demonstrated in Figure 3 (bottom). The resulting time constant is $\tau_{\mathrm{H,LT}} = 21380\ \mathrm{s}$. The higher deviations at the beginning of the step response result from non-linear heat transfer in particular from higher ratio of radiation.

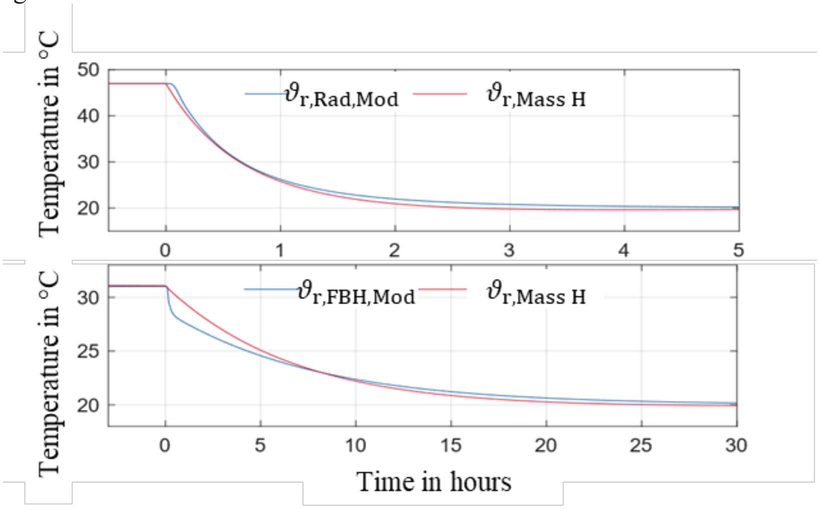

Figure 3. Step responses of radiator system (top) and underfloor heating system (bottom) compared with one mass behavior.

The performed step responses deliver the needed time constant to parametrize the model. By adding the heating load, the model is fully parameterized. The following time constants are defined:

- The building to define the time constant $\tau_{\mathrm{B}} = 240{,}563\ \mathrm{s}$
- The radiator to define the time constant $\tau_{\mathrm{H,MT}} = 2{,}384\ \mathrm{s}$
- The UFH to define the time constant $\tau_{\mathrm{H,LT}} = 21{,}380\ \mathrm{s}$

Table 1: Parameters of underfloor heating system

Parameter	**Value**
Wall layers	2
Conductance layer 1 and 2	1.2 W/(mK) and 0.45 W/(mK)
Specific heat capacity of layer 1/2	1000 J/(kgK)
Emissivity of inner wall surface	0.95
Thickness of layer 1 and 2	0.06 m and 0.0045 m
Density of layer 1 and 2	2000 kg/m^3 and 140 kg/m^3
Pipe material	Polyethylen PE-RT
Spacing between pipes	0.2 m
Inner and outer diameter of pipe	0.016 m and 0.02 m

2.4. Additional features

Section 2.1 and 2.2 describe the general implementation and the parametrization of the building emulator. However, further features are implemented and can be used optionally for different cases. First, tests below the bivalence point can be performed with the physical backup heater or with a virtual one. Second, a hydronic switch test can be optionally integrated to easily enable dynamic volume flow behavior.

2.4.1. Auxiliary electric heater integration

Besides the refrigerant cycle, heat pump systems usually include an additional backup heater to meet the design power below the bivalence point. While the current standard does not consider the backup heater (BUH), we integrated two representative ways to consider the BUH:

a. The physical BUH and the internal control logic are used.

b. The physical BUH is deactivated, and a virtual BUH is used.

If the physical BUH is used, the model can be used as presented in 2.1.

If no physical BUH is installed or cannot be tested, we propose to use a virtual BUH to guarantee representative efficiency values:

The virtual heater fills the gap between the supply temperature setpoint $\vartheta_{i,\text{s,set}}$ and the actual supply temperature $\vartheta_\text{s}(t)$:

$$\dot{Q}_\text{BUH,calc}(t) = \dot{m}_\text{w} \cdot c_p \cdot \left(\vartheta_{i,\text{s,set}} - \vartheta_\text{s}(t)\right) \tag{18}$$

The heat flow of the virtual BUH is limited to a maximum power and zero:

$$\max\left(\dot{Q}_\text{BUH,calc}(t)\right) = \dot{Q}_\text{BUH,real,max};\ \min\left(\dot{Q}_\text{BUH,calc}(t)\right) = 0 \tag{19, 20}$$

The power limitation shall be equal to the declared maximum power of the real BUH.

In case the backup heater is turned off or the physical backup heater is used, there is no additional temperature lift through the backup heater ($\Delta\vartheta_\text{BUH,calc} = 0$). If the model should consider a virtual backup heater, the resulting temperature lift must be considered:

In case the calculated virtual power of the BUH is equal to the maximum ($\dot{Q}_\text{BUH,calc}(t) = \dot{Q}_\text{BUH,real,max}$) the temperature gap $\Delta\vartheta_\text{BUH,calc}(t)$ is calculated by the following equation:

For $\dot{Q}_\text{BUH,calc}(t) < \dot{Q}_\text{BUH,real,max}$:

$$\Delta\vartheta_\text{BUH,calc}(t) = \left(\vartheta_{i,\text{s,set}} - \vartheta_\text{s}(t)\right) \tag{21}$$

For $\dot{Q}_\text{BUH,calc}(t) = \dot{Q}_\text{BUH,real,max}$:

$$\Delta\vartheta_\text{BUH,calc}(t) = \frac{\dot{Q}_\text{BUH,real,max}}{\dot{m}_\text{w} \cdot c_p} \tag{22}$$

For activated virtual BUH, the heat pump's heating capacity $\dot{Q}_\text{HP}(t)$ is replaced by the heat system's heat capacity $\dot{Q}_\text{HPS}(t)$:

$$\dot{Q}_\text{HPS}(t) = \dot{Q}_\text{HP}(t) + \dot{Q}_\text{BUH,calc}(t)$$

Additionally, Eq. (9) changes to consider the higher supply at the heat pump's outlet:

$$\dot{Q}_\text{HB}(t) = (UA)_\text{HB} \cdot \left(\frac{(\vartheta_\text{S}(t) + \Delta\vartheta_\text{BUH,calc}(t)) + \vartheta_\text{R,calc}(t)}{2} - \vartheta_\text{B,calc}(t)\right) \tag{23}$$

2.4.2. Hydronic integration

In many heat pump systems, an internal control logic is integrated to manage the circulating pump effectively. It is essential that the tests include this pump operation to accurately assess system performance. Two typical hydraulic schemes are discussed below:

Direct connection: In this configuration, the heat pump is directly connected to the heating system. If the mass flow provided by the heat pump deviates from the design mass flow, either being higher or lower, the resulting heat load will not align with the intended design specifications.

Hydraulicc separator: In this setup, a hydronic separator is placed between the heat pump and the heating system. When the mass flow of the heat pump exceeds the design mass flow, water flows through the hydraulic separator from the supply to the return pipe, leading to an increase in return temperature. If the heat pump maintains its supply temperature during this process, it indicates that the heat load aligns with the design specifications. When the mass flow from the heat pump is below design levels, the hydraulic separator enables mixing before reaching the heating system's inlet (water flows from the return to the supply pipe).

The virtual temperature at the heating system's inlet can be used to reduce test repetitions: By emulating the virtual temperature at the heating system's inlet and linking it back to the heat pump system, we can establish a target value for operation. If the heat pump maintains this inlet temperature, the heat load corresponds with the design load.

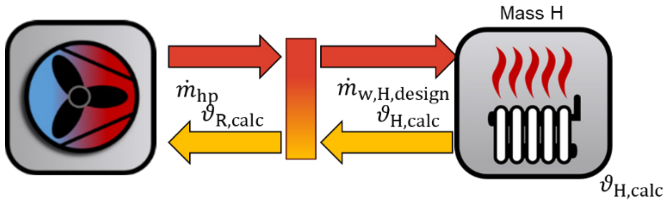

Figure 4. Simplified illustration of a hydraulic connection with a hydraulic separator.

For both solutions mentioned above, it is crucial to design the heating system's mass flow appropriately. For nominal load conditions, this can be calculated by defining a design temperature spread across the heating system, typically 5 K for UFH and 8 K for radiators. It should be noted that under part-load conditions, this temperature spread may differ.

The general model implementation with the introduced features leads to a simple building emulator that can be quickly parametrized. The following section presents exemplary results obtained with an air-to-water heat pump.

3. Results

In this section, exemplary results are shown for a variable compressor speed air-to-water heat pump with a rated capacity of about 8 kW, tested according to the described methodology. We parametrized the model and tested the heat pump according to EN 14825 [5]. Therefore, test condition D, medium temperature application (equivalent to radiator) represents an ambient temperature of 12 °C and supply temperature of 30 °C (A12W30). Part-load condition (PLC) B represents A2W42 and PLC-A stands for A-7W52.

Figure 5 shows cyclic behavior during the D test condition with variations in supply temperature and heating capacity driven by the on-off operation of the compressor. The test can emulate the modeled return temperatures stand with maximum deviances of less than ±0.3 K. The method captures the impact of the unit's controller, which directly influences the performance of the heat pump and cannot be correctly represented in standard testing.

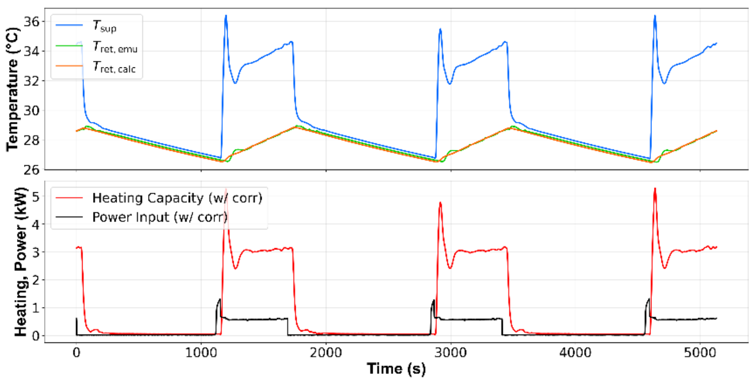

Figure 5. Compressor cycling behavior in condition D.

Figure 6 demonstrates the ability of the method to capture defrost cycles. The dynamic load response of the building emulator during this transient process enables capturing the controlling strategy of the unit and the impact on performance in a more representative operating condition. While the current standard can capture defrost operation provided that the unit undergoes more frequent defrost cycles, the proposed approach assesses the controller's performance during defrost under dynamic building load conditions

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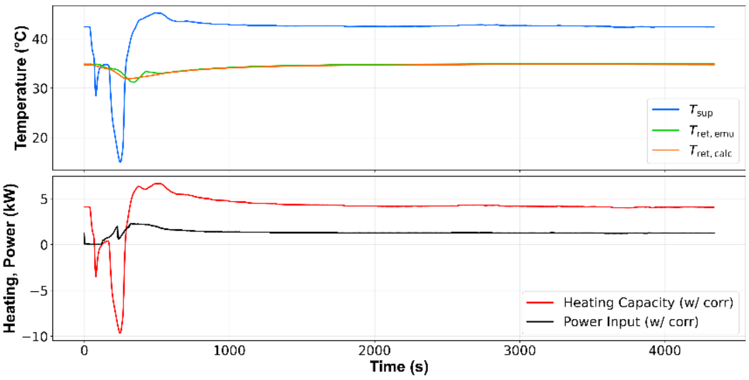

Figure 6. Defrost cycle captured during condition B.

Figure 7 represents an example of test condition A, which requires the parallel operation of the heat pump and its integrated auxiliary electric heater. The oscillatory behavior reflects the unit controller's logic in handling the supplementary heating by the auxiliary electric heater, which is not in line with the idealized introduction of the electrical heater in SCOP calculations of the current standard.

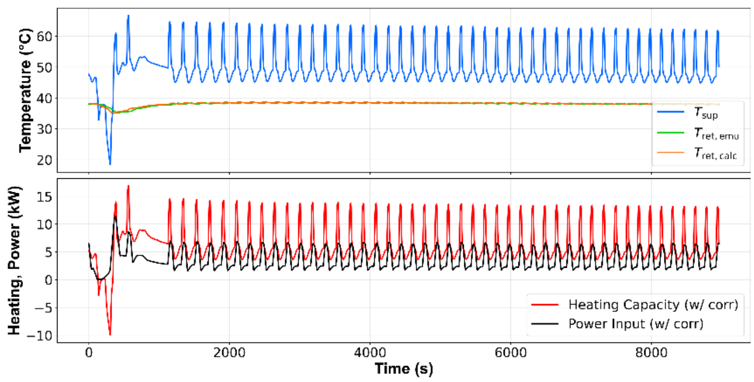

Figure 7. auxiliary electric heater activation in A condition.

Test results demonstrate that the load-based testing coupled with a building emulator provides a more representative assessment approach for the heat pump performance including not only the refrigeration cycle but also integrated controller and auxiliary electric heating under more realistic dynamic operating conditions.

4. Discussion

The described two-mass model approximates a building energy system. However, different possible changes are discussed.

4.1. General model structure

The two-mass model approximates the water temperature of the heating system with the first mass, and the building air temperature with the second mass. The building envelope has an inertia three or two orders of magnitude higher than the radiator or UFH, respectively. Therefore, the influence of the building on the return temperature is low in the context of a typical test time below 12 hours. Therefore, the second mass can be neglected for short tests. Nevertheless, the virtual building temperature is useful for modern control systems that take the indoor air temperature into account. The step response in Figure 2 shows differences from the one-mass behavior at the beginning of the step. The building envelope, made of concrete and air volume, results in two media with different behaviors. First, the air volume cools down until the concrete temperature is reached. Afterward, the air temperature and the concrete temperature move at the same speed. To further improve the temperature behavior, a third mass representing the air volume is needed.

We recommend either a) limiting the approach to one mass or b) using a third mass to represent the air temperature more accurately.

4.2. Definition of target value

Currently, heat pumps under test must meet both the supply temperature and the heat load setpoints. However, some heat pumps control their water flow rate dynamically, leading to an infinite number of potential solutions to achieve the correct heat load, each characterized by different supply and return temperatures.

For example, a heat pump system may choose to heat the building using a supply temperature of 35 °C with a return temperature of 30 °C. Alternatively, it could use a lower mass flow rate while providing a supply temperature of 36 °C and a return temperature of 29 °C.

For a representative application, we propose that the target value should focus solely on achieving the heating load, allowing the heat pump to determine how to best meet this requirement. This approach provides greater flexibility in operation and can lead to more efficient performance across varying conditions. Additionally, the chosen formulation leads to a proportional relation between dissipated heat $\dot{Q}_{\mathrm{HB}}(t)$ and the arithmetic mean temperature (from the supply and return). This means that replacing the supply by the arithmetic mean temperature would be equivalent to specifying the design heat load for each test point. This approach ensures representative operation, freedom for manufacturers to optimize their controls and creates a level-playing field among different control strategies.

4.3. Modification of the inertia

Discussions with various stakeholders revealed that some heat pumps require a higher minimum amount of heating water than is currently considered. The two-mass model with the described time constants represents a simple building energy system in which the heat pump is directly connected to a transfer system (radiator or UFH, depending on the temperature application. The current parametrization corresponds to an equivalent water mass of 14.6 kg of water per kW design heating capacity (P_{design}) in the design test point (PLC-E) for medium temperature application.

Some manufacturers might prescribe a higher minimum water volume or an additional buffer tank in their technical documentation. The additional needed water volume could be easily integrated into the model by increasing the thermal capacity of the heating system.

5. Conclusion

This work presents a building emulator for representative and reproducible heat pump testing. The two-mass model scales with P_{design} of the tested heat pump and is parameterized with sophisticated Modelica models designed according to relevant standards. A radiator system or an UFH system is assumed, depending on the temperature application.

The two-mass model allows the model to be implemented on any test stand control. The model can be further simplified (e.g. neglecting the building influence (one-mass model)) or adjusted to better fit EN 14825 prescriptions. The changes should be discussed, considering the trade-off between simplicity for the sake of standardisation and accuracy to mimic real-life response.

In physical tests, it has been proven that the two-mass model can be applied to typical heat pump controls (fixed and variable flow, supply and return temperature controlled), resulting in realistic operating behaviour. It has also been demonstrated in a round robin test that the methodology can yield reproducible test results comparable to the existing fixed frequency test.

Nomenclature

$\dot{Q}_{\mathrm{designh}}$	Declared heating load at T_{designh}	PLR_i	Part load ratio at test condition
$P_h(T_j)$	Part-load capacity at temperature i	$\dot{Q}_{\mathrm{HP}}$	Measured heating capacity of heat pump
$\vartheta_{i,\mathrm{A}}$	Outdoor (ambient) temperature at test condition i	$\vartheta_{\mathrm{B,set}}$	Indoor temperature set point of 20 °C
ϑ_s	Indoor heat exchanger outlet water temperature (supply temperature)	$\vartheta_{i,\mathrm{s,set}}$	Supply temperature set point at test condition i according to EN 14825:2022
$\vartheta_{\mathrm{B,calc}}(t)$	Calculated building temperature	$\vartheta_{\mathrm{R,calc}}$	Calculated return temperature

$\Delta\vartheta_{\mathrm{BUH,calc}}(t)$	Calculated supply temperature increase due to virtual backup heater	$\Delta\vartheta_{\mathrm{i,cond,design}}$	Design temperature difference across the condenser
$\dot{Q}_{\mathrm{BUH,calc}}$	Heat flux of the virtual backup heater.	$\dot{Q}_{\mathrm{BUH,real,max}}$	Maximum heat flux of the backup heater of the tested heat pump.
$\dot{Q}_{\mathrm{HB}}$	Heat flux between mass H and mass B	$\dot{Q}_{\mathrm{BA}}$	Heat flux between mass B and the ambient
$\mathrm{UA_{HB}}$	Thermal conductance between mass H and B	$\mathrm{UA_{BA}}$	Thermal conductance between mass B and ambient
$\mathrm{C_H}$	Thermal capacity of mass H	$\mathrm{C_B}$	Thermal capacity of mass B
$\dot{m}_{\mathrm{W,hp}}$	Water mass flow delivered by the heat pump	$\dot{m}_{\mathrm{w,H,design}}$	Design mass flow of the heating system

Acknowledgements

The authors are grateful for the support and funding by the Deutsche Bundesstiftung Umwelt and the non-profit organization CLASP. The partners thank the heat pump manufacturer for providing the unit for testing. We also appreciate the fruitful discussions with Alexandre Royer and experts from CEN/TC 113/WG 8 to. .

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