Modelling the Electrical Power Demand of Different Heat Pump Systems: Approaches for Simplified and Detailed Load Assessment

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Abstract: The growing share of heat pumps will lead to a substantial increase in decentral loads in electrical distribution networks. This paper analyses factors influencing their power demand and presents two modelling approaches to assess the electrical load increase due to heat pump systems. An easy-to-apply simplified heating system model for average load assessment and a high-resolution heating system model for detailed load forecasting are developed. Electric load profiles for typical buildings and system configurations are shown, and the results of the modelling approaches are compared.

Keywords: heat pump, load modelling, average load profile, distribution grid

1 Motivation and Objectives

According to the German government's legislative resolution, the heat supply in Germany is to be completely climate-neutral by 2045. To achieve this goal, the German government is currently introducing several measures. For example, from 2024, a minimum share of 65 % renewable energy will be mandatory for heating systems in new buildings in development areas in Germany. [1] In the field of decentralized heat supply, the transition will particularly accelerate the ramp-up of electrical heat pumps (HPs), with the German government aiming to achieve at least 500,000 newly installed HPs per year from 2024 [2].

The growing share of HPs will lead to a substantial increase in decentral loads in electrical distribution networks. Especially in urban areas, where load density is high, this can pose significant challenges to distribution system operators (DSOs), which is why a reliable planning basis is required. The objective of this paper is to present approaches for determining the electrical power demand of HPs for individual buildings and thus estimate the resulting additional grid load caused by them. Both a simplified model for average load assessment and a detailed model for generating high-resolution daily load profiles are presented. Different factors influencing the electric power demand of HPs are considered, and exemplary results for typical buildings are shown.

2 Methodology

There are multiple factors (cf. Figure 2) influencing the electrical power demand of HP systems. These are analysed in the first step, and typical values are determined. For factors that vary locally (e.g., norm outdoor temperature), characteristic values for southern Germany are utilized using the city of Munich as an example. The analyses are described in detail in section 3.

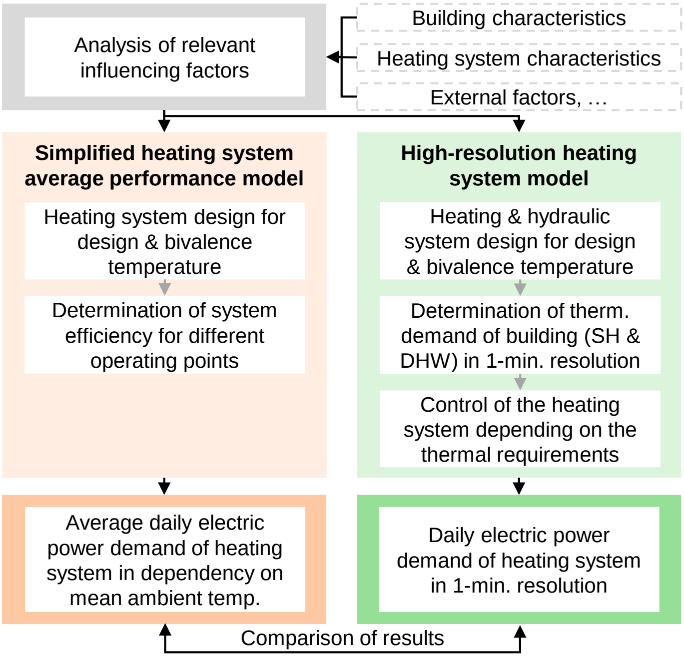

Figure 1: Schematic methodology for determining the electrical power demand of heat pump systems

Two models are then developed in order to assess the electrical load of HP-based heating systems. Since high-resolution modelling requires a comparatively high computational effort and detailed information about the heating object, both an easy-to-apply simplified heating system model (cf. section 4.1) and a high-resolution heating system model (cf. section 4.2) are presented. The simplified heating system model determines the system efficiency of the HP (and, if applicable, the heating rod) for different operating points and allows the estimation of the average daily electrical power demand of the heating system in dependency on the mean ambient temperature. Grid operators can utilize this information as a reference value for grid dimensioning.

The high-resolution heating system model further enables a building-specific assessment of the minute-by-minute load profile. For this, the hydraulic system and a heat-driven operating mode are modelled. A simplified thermal building model and a tap profile generator (for generating hot water withdrawal profiles) are used to derive the thermal space heating (SH) and domestic hot water (DHW) demand of the building. Characteristic load profiles can be determined for individual buildings. A comparison with the results for the average daily power demand of the simplified heating system model is then carried out (cf. section 5). The entire procedure is summarized in Figure 1.

3 Analysis of Relevant Factors Influencing the Electrical Power Demand of Heat Pump Systems

Several factors influence the electrical power demand of HP systems. These can be assigned to four categories (external, building-related, heating system-related, and control-related factors) and are summarized in Figure 2. The individual factors are examined in more detail below.

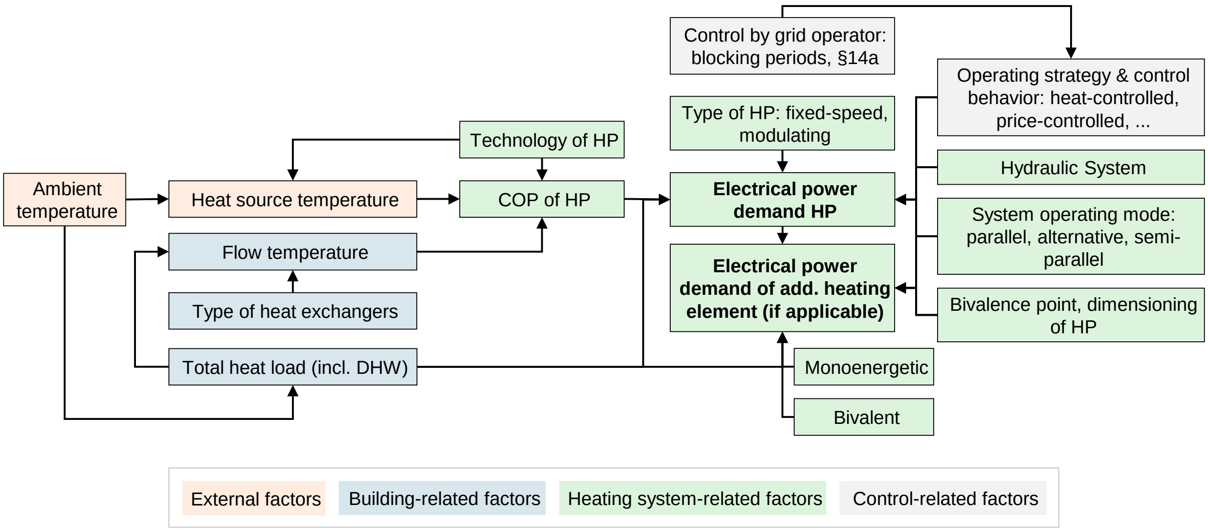

Figure 2: Overview of factors influencing the electrical power demand of heat pump systems

3.1 External Factors

Relevant external factors are ambient and heat source temperature. While the ambient temperature is required in particular to determine the standard heat load and the thermal space heating load profile of the building, the heat source temperature affects the coefficient of performance (COP) of the HP.

Ambient Temperature

The standard outdoor temperature is used to design heating systems and determine the standard heat load (see also section 3.2). According to DIN EN 12831, the standard outdoor temperature is the lowest two-day average ambient air temperature reached or fallen below ten times in 20 years. The standard outdoor temperature varies between different locations. In the city of Munich, the standard outdoor temperature lies, e.g., between -11 °C and -14 °C (cf. [3]).

The standard outdoor temperature represents a stationary average value, but the temperature development over the course of the day is also relevant for determining the dynamic space heating demand. In order to use realistic temperature curves for modelling, the ten coldest two-day average periods in the city of Munich between 2003 and 2022 were examined (based on data from [4]). These and the resulting average temperature curve are shown in Figure 3.

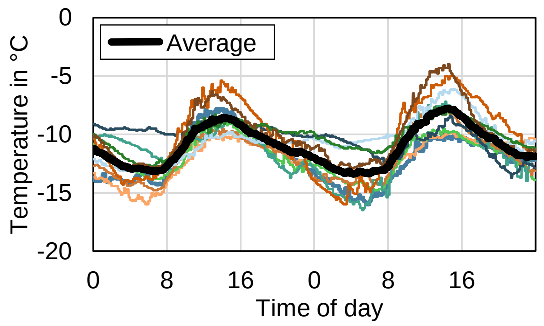

Figure 3: Coldest two-day ambient temperature courses in Munich

Heat Source Temperature

In addition to the ambient temperatures, the heat source temperatures, which differ depending on the technology of the HP, are also relevant. In the case of an air-source HP, the heat source temperature is the same as the ambient temperature, but a separate consideration is required for water- and ground-source HP. Generally, groundwater and soil have much smaller temperature fluctuations than the ambient air. In this work, a constant groundwater temperature of 10 °C and a soil temperature of 1 °C are presumed since the coldest temperature periods are particularly important for assessing the additional load on the grid. These assumptions represent rather conservative values but are in line with current design recommendations (cf. [5], [6]).

3.2 Building-related Factors

Building-related factors include the total heat load resulting from SH and DHW, their influencing factors, the type of heat exchangers used, and the required flow temperatures.

Total Heat Load

Numerous influencing factors must be taken into account to determine the exact thermal heat load of a building. It depends, for example, on the type of building usage, the type of building (e.g., multi-family or single-family house), the size of the building, and the number of people typically present in the building. The heat load varies depending on the transmission heat losses to the outside air and the ground as well as the air exchange rate of the building through infiltration or ventilation. The indoor temperature in the respective rooms and potential thermal gains also influence the SH demand. Thermal gains include, e.g., inputs from solar radiation, people inside the building, and electrical consumption. It is referred to DIN EN 12831-1 for further information on precisely determining the (standard) heat load. Buildings only need to be heated if the temperature falls below the heating limit temperature. The limit temperature varies between individual buildings and, according to VDI 4650-1, lies typically between 15 °C (existing buildings) and 10 °C (passive houses). The heating limit temperature corresponds to the 2-day average temperature.

In addition to the SH demand, there is a heating demand for DHW. Here, the power demand is primarily determined by the building usage and the number of people in the building. In residential buildings, an average daily hot water consumption of 25 l at 60 °C per person is expected following DIN EN 15450. In addition to the exact withdrawal quantity and temperature, the withdrawal time (daily tap profile) is also relevant for the resulting DHW demand.

The approaches chosen in this work for modelling the heating load for SH and DHW are described in more detail in section 4.2.

Type of Heat Exchangers and Flow Temperatures

A heating curve is defined in the control of the heating system to meet the heating load. This curve describes the correlation between ambient temperature and required flow temperature in the heating circuit. The flow temperature depends on both the required heating demand and the size of the heat exchangers. Surface heating systems generally require lower flow temperatures than radiators.

3.3 Heating System-related Factors

Relevant heating system-related factors are the technology and type of the HP, the dimensioning, design, and operating mode of the heating system, as well as the type and size of the storage systems utilized.

Heat Pump Technology

The COP of the HP can vary greatly depending on the technology utilized. The maximum physically achievable COP for a specific operating point can be determined as the reciprocal of the Carnot efficiency. If the real COP $\varepsilon_{\mathrm{OP}}$ is to be determined, this value must be multiplied by a "quality factor" η_{HP} (≤ 1; see equation (1)). To determine the technology-dependent quality factors, both the real COP of HPs based on the database of Eurovent [7] and typical values according to the DIN/TS 18599-12 standard were analysed. For the Eurovent database, the median COP values of the respective HP technologies were used as reference values. The quality factors were then determined using the mean squared error method. Figure 4 summarizes these evaluations. It can be seen that a quality factor of 0.36 for air-source HPs, 0.46 for water-source HPs, and 0.47 for ground-source HPs provides a realistic assessment of the COP. These values are representative for the situation in 2023, whereby an increase of the quality factors can be expected in the future.

$$\varepsilon_{\mathrm{OP}} = \eta_{\mathrm{HP}} \cdot \frac{T_{\mathrm{flow}}}{T_{\mathrm{flow}} - T_{\mathrm{source}}} \tag{1}$$

T_{flow}	flow temperature in K
T_{source}	heat source temperature in K
$\varepsilon_{\mathrm{OP}}$	COP at the corresponding operating point
η_{HP}	quality factor

A further distinguishing criterion is the operating mode of the HP, whereby fixed-speed and modulating HPs can be differentiated. While fixed-speed HPs only enable cyclical on/off operation with a constant compressor speed, a modulating HP with a variable-speed compressor is able to flexibly adjust its power to the respective heating requirements within a certain range.

Design of the Heating System

Several parameters must be taken into account when designing HP systems. In general, the system can be designed as follows:

- Monovalent: HP acts as the sole heat generator across all temperature ranges.
- Monoenergetic: HP acts as the sole heat generator up to the bivalence point, after which an additional electric heating element (typically a heating rod) is also used to cover the heating demand.
- Bivalent: HP is the sole heat generator up to the bivalence point, and then another non-electric heating element (e.g., gas heater) is used to cover the heating demand.

Monovalent operation is particularly suitable for HPs, which can still achieve high efficiency levels even at very low ambient temperatures. These are water-source and ground-source HPs due to the comparatively low fluctuations of the heat source temperatures. In the case of air-source HPs, a monoenergetic design is often used. [8] Different operating modes can be selected for bivalent and monoenergetic systems:

- Parallel: below the bivalence point, heat is provided by the HP and the additional heating element simultaneously.

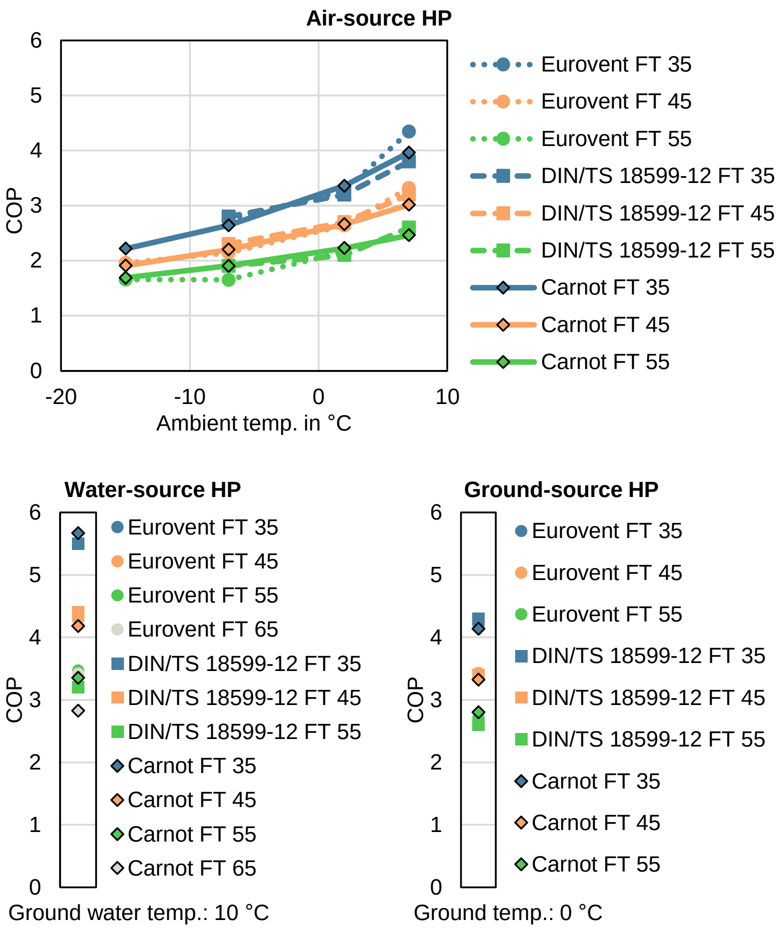

Figure 4: Overview of COP of different heat pump technologies

Note: FT is used as an abbreviation for flow temperature. Carnot FT 35 thus represents, e.g., the COP, which is calculated at the specified source temperature and a flow temperature of 35 °C using equation (1).

- Semi-parallel: in a specific temperature range, heat is provided both by the HP and the additional heating element. Below a specific limit temperature, heat is then provided exclusively by the additional heating element.
- Alternatively: heat is provided exclusively by the additional heating element below the bivalence point.

Figure 5 illustrates the design of a HP system using the example of a fixed-speed air-source HP for the provision of SH and DHW in monoenergetic parallel operation. It is also possible that only SH is provided by the HP, and DHW is supplied by an additional heating system (e.g., continuous-flow water heater). However, surveys show that around 90 % of HPs are designed as heating systems for both SH and DHW. [10]

The entire heating system must be dimensioned so that it is able to cover the standard heat load (heat load at standard outdoor temperature) and the DHW demand of the building. In a monovalent heating system design, the standard heat load plus the DHW demand is to be taken as the reference design value. The HP must be able to provide this thermal power at standard outdoor temperature. The nominal electrical power of the HP can, therefore, also be determined by dividing this thermal power by the respective COP (see also section 4.1, equation (3)).

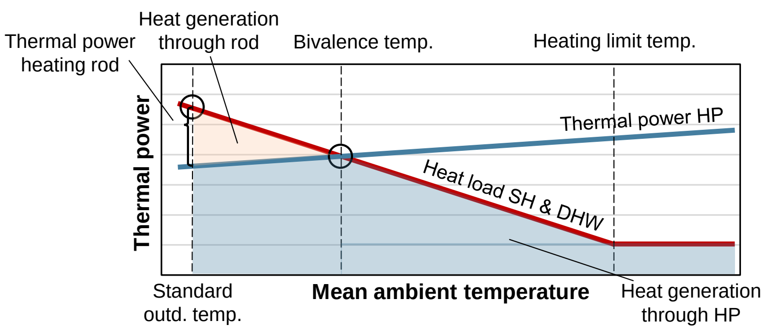

Figure 5: Design example of a fixed-speed air-source HP for the provision of SH and DHW in monoenergetic parallel operation

In monoenergetic parallel operation (see also Figure 5), the power of the HP is sufficient to provide the necessary thermal demand of the building up to the bivalence point. If the temperature falls below the bivalence temperature, the thermal power of the heating rod is also required to meet the heating demands. Here, the thermal power demand (SH & DHW) at the bivalence point is to be taken as the reference design value for the dimensioning of the HP. The HP must be able to supply this thermal power solely at bivalence temperature. The nominal electrical power of the HP can again be determined by dividing this thermal power by the respective COP (see also section 4.1, equation (3)). The thermal power output of the HP at standard outdoor temperature must be subtracted from the total thermal power demand at standard outdoor temperature to determine the additional nominal thermal power of the heating rod. This resulting power difference is to be supplied by the heating rod and, therefore, equals to the nominal thermal power of the rod. The nominal electrical power of the heating rod can be obtained by dividing the nominal thermal power by the efficiency of the rod.

If the thermal power of the fixed-speed HP is higher than the power demand of the building, it starts to cycle. As soon as the average ambient temperature reaches the heating limit temperature, SH is no longer required, and only the DHW demand remains. The choice of bivalence temperature varies and is usually selected depending on the standard outside temperature. In Germany, the values are typically between -2 °C and -7 °C. A typical bivalence temperature for bivalent-parallel or monoenergetic operation is -5 °C. At this bivalence point, the HP provides approx. 98 % of the annual heating energy and only 2 % is covered by the additional heating element. [9] According to DIN EN 15450, the share of the annual heating energy covered by the heating rod should be less than 5 %. The design shown above represents an exact, needs-based system design. In practice, however, the heating system may also be oversized due to, e.g., blocking periods of the DSO, applied safety margins by the installers of the heating systems, or the limited availability of HP sizes (discrete power steps).

Another factor influencing the load profile of the HP is the design of the hydraulic system, whereby several options exist. A comprehensive overview of possible designs is given in [11]. For further investigations in the scope of this work, the hydraulic system shown in Figure 6 is selected as it is assumed that this will be used frequently in practice. HP and heating rod are used to provide both SH and DHW. The buffer storage tank integrated in parallel is used, e.g., to increase the running time of the HP under partial load. The integration of buffer storage tanks is common in practice [12].

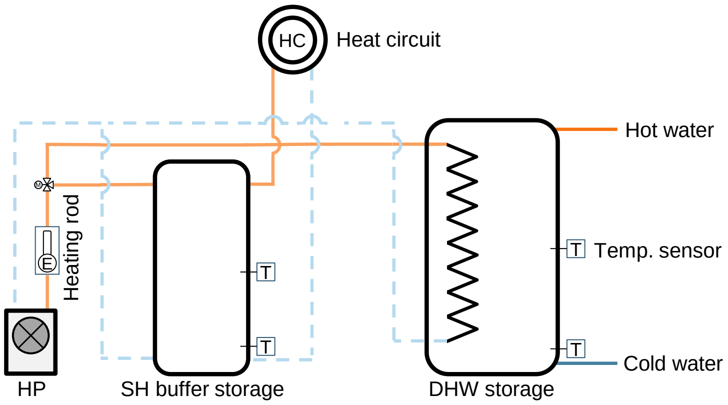

Figure 6: Hydraulic system with DHW heating and parallel buffer storage tank for SH

3.4 Control-related Factors

Control-related factors include the operation strategy and control behaviour of the HP as well as externally triggered control measures like blocking periods of the DSO. A fundamental distinction between heat-driven and non-heat-driven (e.g., price-driven or efficiency-driven) operating modes can be made. In this work, the focus is on the heat-driven operating mode of the system, which means that the power of the heating system is set as a function of the thermal energy demand of the building.

There are several options for controlling a heat-driven heating system, such as weather- or room-temperature-dependent control and (or in combination with) differential temperature control (spread measurement between flow and return temperature). In practice, a sensor measuring the return temperature is often used as a reference variable for the heat generation system since the return temperature provides reliable information about the actual heat demand of the building. Depending on the ambient temperature, a return setpoint is then determined as a switch-on condition based on the individual specifications of the heating curve. [13] Due to the thermal inertia of the building, the setpoint temperature does not follow the ambient temperature directly but typically takes into account the average temperature values of the past 24 h up to 72 h.

However, the heat-driven mode of operation can be influenced by external factors. These include, e.g., the DSO's blocking hours or interventions in accordance with § 14a EnWG. This enables grid operators to switch off HP systems or reduce their power to avoid grid overloads, even if there is a thermal power demand within the building. Due to the thermal inertia of the building and the widespread use of buffer storage tanks, short-term interruptions have little effect on the comfort requirements of the residents. However, in the event of prolonged interruptions or interruptions at times of extremely cold temperatures, catch-up effects are to be expected once the full HP power is available again.

4 Heat Pump System Modelling Approaches

Approaches for modelling the electrical power demand of HP systems are presented below. Since the power demand depends on numerous influencing factors, as presented in the previous sections, both a simplified heating system average performance model (SHSAP-Model) for an easy-to-apply but simplified load estimation and a high-resolution heating system model (HRHS-Model) for the generation of detailed daily load profiles are developed.

4.1 Simplified Heating System Average Performance Model (SHSAP-Model)

The SHSAP-Model determines the system efficiency of the HP (and, if applicable, heating rod) for different operating points. The model allows, therefore, to estimate the average daily electrical power demand of the heating system in dependency on the mean ambient temperature. To apply the simplified model, the following information about the building is required: the standard heat load, the standard outdoor temperature, the heating limit temperature, and the number of occupants.

Figure 7 summarizes the procedure of the model. The first step is determining the total heat load for SH and DHW depending on the average ambient temperature. For the purpose of simplification, a constant power demand of 0.2 kW per person is assumed for DHW, irrespective of the ambient temperature. This corresponds to established guidelines (cf. [9], [10]). Equation (2) can be applied to get the temperature-dependent total thermal heat load $\dot{Q}_{\mathrm{THL}}$. The SH heat load $\dot{Q}_{\mathrm{SHL}}$ is determined based on VDI directive 4645.

The methodology presented in [14] is used to calculate the required flow temperature in the next step. This enables the determination of flow temperatures required for different heat exchange systems, from low-temperature floor heating to medium-temperature radiator heating. The COP $\varepsilon_{\mathrm{OP}}$ of the HP can then be calculated for the relevant operating points

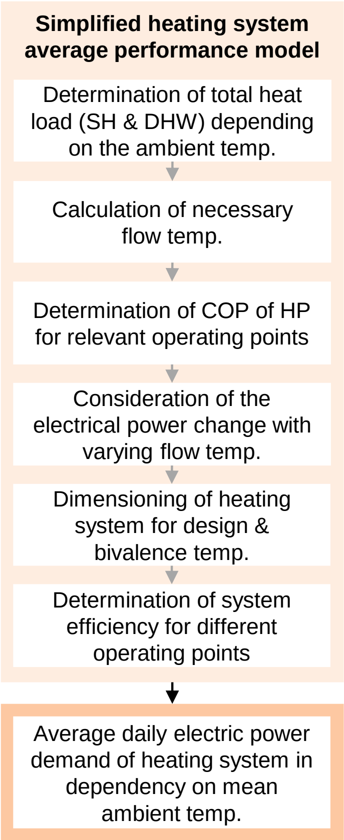

Figure 7: Overview of the work steps of the simplified heating system avg. performance model

according to equation (1). The COP represents the ratio of thermal power output $\dot{Q}_{\mathrm{HP}}$ to electrical power input of the HP P_{HP}. Equation (3) applies. The COP varies depending on the operating point. In addition, P_{HP} is not constant but changes as the flow temperature T_{flow} varies. For the purpose of simplification, it is assumed that only the flow temperature and not the heat source temperature influences the electrical power consumption of the HP. The validity of this simplification is confirmed by data sheets from several manufacturers (cf. [15], [16], [17]). Based on information from [10], [18], [19], and [20], it is assumed that an increase in the flow temperature by 1 K results in an increase in electrical power demand of 2.5 %.

The nominal electrical power of the HP can then be determined for the design point according to equation (3), as both the required thermal power (for monovalent systems: standard heat load, for bivalent systems: heat load at bivalence point) and the COP (according to respective operating point) are known. If the flow temperature T_{flow} now deviates from the flow temperature at the design point $T_{\mathrm{flow\_design}}$ as a result of changing heat load or ambient temperature, the new electrical power consumption of the HP P_{HP} can be determined according to equation (4). This allows the electrical power demand, COP, and thermal power output of the HP to be calculated for all relevant operating points.

If the heating system is designed as a monoenergetic system, the electrical power demand of the heating rod P_{rod} must also be considered if the ambient temperature falls below the bivalence temperature T_{biv}. The thermal output of the rod closes the resulting gap between the thermal power demand of the building and the thermal power output of the HP starting from the bivalence point (cf. Figure 5) and is accounted for with an efficiency of 99 %. The composition of the total heat load $\dot{Q}_{\mathrm{THL}}$ can be described using equation (5).

$$\dot{Q}_{\mathrm{THL}}(T_{\mathrm{amb\_avg}}) = \max\left\{\frac{\dot{Q}_{\mathrm{SHL}} \cdot (T_{\mathrm{lim}} - T_{\mathrm{amb\_avg}})}{T_{\mathrm{lim}} - T_{\mathrm{SO}}} + \dot{Q}_{\mathrm{DHW}};\ \dot{Q}_{\mathrm{DHW}}\right\} \tag{2}$$

$$\dot{Q}_{\mathrm{HP}} = \varepsilon_{\mathrm{OP}} \cdot P_{\mathrm{HP}} \tag{3}$$

$$P_{\mathrm{HP}} = P_{\mathrm{HP\_design}} + (T_{\mathrm{flow}} - T_{\mathrm{flow\_design}}) \cdot 0.025\frac{1}{\mathrm{K}} \cdot P_{\mathrm{HP\_design}} \tag{4}$$

$$\dot{Q}_{\mathrm{THL}} = \begin{cases} (P_{\mathrm{HP}} \cdot \varepsilon_{\mathrm{OP}} + P_{\mathrm{rod}} \cdot 0.99), \text{for } (T_{\mathrm{amb\_avg}} < T_{\mathrm{biv}}) \wedge \text{monoenergetic design} \\ (P_{\mathrm{HP}} \cdot \varepsilon_{\mathrm{OP}}), \text{for } (T_{\mathrm{amb\_avg}} \geq T_{\mathrm{biv}}) \vee \text{monovalent design} \end{cases} \tag{5}$$

P_{HP}	Electrical power consumption of HP in kW
$P_{\mathrm{HP\_design}}$	Electrical power input of HP at design point in kW
P_{rod}	Electrical power of heating rod in kW
$\dot{Q}_{\mathrm{DHW}}$	Heat load for DHW in kW
$\dot{Q}_{\mathrm{HP}}$	Thermal power output of HP in kW

$\dot{Q}_{\mathrm{THL}}$	Total heat load in kW
$\dot{Q}_{\mathrm{SHL}}$	Standard heat load in kW
$T_{\mathrm{amb\_avg}}$	2-day average ambient temperature in K
T_{biv}	Bivalence temperature in K
T_{flow}	Flow temperature in K
$T_{\mathrm{flow\_design}}$	Flow temperature at design point in K
T_{lim}	Heating limit temperature in K
T_{SO}	Standard outdoor temperature in K
$\varepsilon_{\mathrm{OP}}$	COP at the corresponding operating point

4.2 High-resolution Heating System Model (HRHS-Model)

The HRHS-Model enables a building-specific assessment of the minute-by-minute load profile. Figure 8 summarizes the procedure of the model. The first steps are identical to the SHSAP-Model (cf. section 4.1). These include determining total heat load (SH & DHW) and necessary flow temperatures, calculating COP and electrical power consumption of the HP for relevant operating points, and dimensioning the heating system. In the next step, the hydraulic system is designed. The basic layout of the hydraulic system is as described in section 3.3, Figure 6, with a parallel SH buffer storage tank and an additional DHW storage tank. The DHW storage tank is dimensioned according to [14] using equation (6). It is further checked whether the storage volume V_{DHW} is sufficient to meet specific comfort requirements. For instance, in the case of a house built to passive house standard, a HP with a low installed power can be used due to the low SH demand. However, this can result in the DHW storage tank taking a comparatively long time to heat up and, therefore, a loss of comfort for the residents has to be expected if the dimensions are too small. For this reason, the DHW storage capacity is increased if the comfort requirements are unmet. In the case of a 4-person household, the storage capacity must, e.g., be sufficient to allow one bath and three showers in a row, even with a storage capacity of only 50 %. For apartment buildings, simultaneity factors following VDI 2072 are taken into account when checking the comfort requirements.

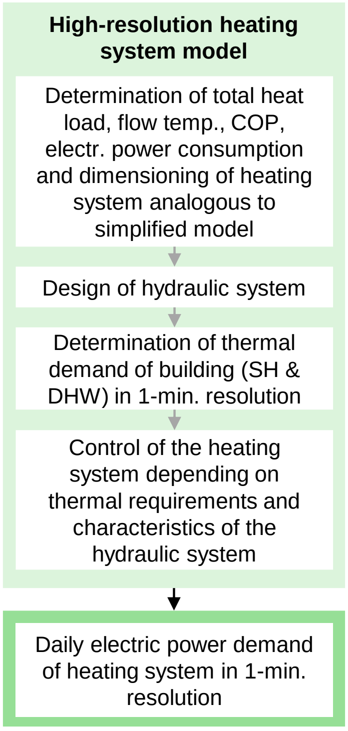

Figure 8: Overview of the work steps of the high-resolution heating system model

When dimensioning SH storage tanks, there are reference values of 12 l to 35 l of storage volume per $\mathrm{kW_{thermal}}$ at the design point (cf. VDI 4645 and DIN EN 15450). In the scope of this work, a storage volume of 20 l per $\mathrm{kW_{thermal}}$ at the design point is assumed. Both DHW and SH storage tanks are rounded up to typical purchasable volumes.

$$V_{\mathrm{DHW}} = S \cdot 65\,\mathrm{l} \cdot n_{\mathrm{persons}}^{0.72} \tag{6}$$

V_{DHW}	Volume of DHW storage tank in l
S	Safety factor between 1.0 and 1.25 (in this work 1.125)
n_{persons}	Number of occupants

The next step is to determine the thermal demand of the building in 1-min. resolution. For the modelling of the DHW demand, the tool DHWcalc [21] is utilized. It can be used to generate drinking water tap profiles. 1,000 tap profiles are generated and saved for all relevant household sizes. The average hot water consumption per person and day is 25 l at a water temperature of 60 °C (cf. DIN EN 15450). However, the withdrawal quantities must be adjusted since a typical hot water withdrawal only takes place at around 45 °C (cf. DIN EN 15450 and [22]). At a typical cold-water temperature of 10° C (cf. DIN EN 15450), 25 l at 60 °C corresponds to 35.7 l at 45 °C. From the resulting 1,000 tap profiles per household size, one is then randomly selected to define the thermal DHW demand for the respective household and day.

The modelling of the SH demand is based on the methodology presented in [23]. This simplified approach requires few input parameters and enables simple parametrization of thermal equivalent circuits for modelling the SH demand of individual buildings. The systems of equations shown in [23] are transformed and simplified to enable implementation even with a limited amount of data. The aim is to apply the methodology when only the standard heat load, the standard outdoor temperature, and the heated area of the respective building are known. Therefore, equation (7) is converted to equation (8), assuming a constant indoor temperature ($\mathrm{d}T_{\mathrm{in}}/\mathrm{d}t = 0$). By this transformation, the thermal capacity C of the building is reduced from the equation. However, since the determination of the heat load in winter is, in contrast to the summer cooling load, a stationary transmission, it is permissible to neglect the thermal energy transfer processes into and out of the thermal mass of the building (represented here by the capacitance) [24].

$$\frac{\mathrm{d}T_{\mathrm{in}}}{\mathrm{d}t} = \frac{1}{RC} \cdot \left(T_{\mathrm{amb}}(t) - T_{\mathrm{in}}(t)\right) + \frac{1}{C} \cdot \left(\dot{Q}_{\mathrm{SH}}(t) + \dot{Q}_{\mathrm{gains}}(t)\right) \tag{7}$$

$$\dot{Q}_{\mathrm{SH}}(t) = -\frac{T_{\mathrm{amb}}(t) - T_{\mathrm{in}}(t)}{R} - \dot{Q}_{\mathrm{gains}}(t) \tag{8}$$

$$R = -\frac{T_{\mathrm{SO}} - T_{\mathrm{in}}}{\dot{Q}_{\mathrm{SHL}}} \tag{9}$$

$$T_{\mathrm{amb}}(t) = T_{\mathrm{amb\_course}}(t) \cdot \frac{T_{\mathrm{SO}}}{T_{\mathrm{avg\_course}}} \tag{10}$$

C	Thermal capacity in kWh/K
$\dot{Q}_{\mathrm{gains}}$	Heat gains in kW
$\dot{Q}_{\mathrm{SH}}$	Heating power demand for space heating in kW
$\dot{Q}_{\mathrm{SHL}}$	Standard heat load in kW
R	Thermal resistance in K/kW
t	Time in h
T_{amb}	Ambient temperature in K
$T_{\mathrm{amb\_course}}$	Ambient temperature of the selected temperature course in K
$T_{\mathrm{avg\_course}}$	Average temperature of the selected temperature course in K
T_{in}	Indoor temperature in K
T_{SO}	Standard outdoor temperature in K

Since the building-specific thermal resistance R is not yet known, this must be determined in the next step. If the standard heat load $\dot{Q}_{\mathrm{SHL}}$ and standard outdoor temperature T_{SO} of the building are known, the thermal resistance can be calculated assuming a typical indoor temperature T_{in} of 20 °C (cf. DIN EN 12831-1) according to equation (9). Potential heat gains are disregarded here ($\dot{Q}_{\mathrm{gains}}(t) = 0$) since when determining the standard heat load, the consideration of heat gains is only optional and is often neglected (cf. DIN EN 12831-1). After determining R, the SH power $\dot{Q}_{\mathrm{SH}}$ demand can be calculated using equation (8). A constant value of 3 W/m^2 of heated area from internal heat sources is assumed as heat gains $\dot{Q}_{\mathrm{gains}}$ (cf. [25]). Solar inputs are neglected as these are volatile and cannot be guaranteed. A temperature course is randomly selected from the curves shown in Figure 3 for each day to determine the SH demand in the high-resolution heating system model. The selected temperature curve is then scaled according to the standard outdoor temperature T_{SO} of the respective building (see equation (10)).

Now that the thermal demands are determined, the next step is to define the response of the heating system to them. In this work, a heat-driven operating mode is modelled, which means that the power of the heating system is set as a function of the thermal energy demand. A weather-dependent control in combination with a differential temperature control is simulated. The required flow temperature for providing the SH is determined according to the heating curve (based on the average ambient temperature of the last 48 h), and the SH buffer storage tank is loaded directly from the flow of the heating system (see also Figure 6). If an SH demand occurs, it is covered by the SH buffer storage tank. The goal is to supply as much energy from the storage tank as is required for SH at any given time ($Q_{\mathrm{SH\_supply}}(t) = Q_{\mathrm{SH\_demand}}(t)$). This corresponds to an energy equilibrium, which results in an energy balance $Q_{\mathrm{SH\_bal}}$ equal to zero (cf. equation (11)). If the energy stored in the SH storage tank is insufficient to meet the SH demand, a heating deficit results ($Q_{\mathrm{SH\_supply}}(t) < Q_{\mathrm{SH\_demand}}(t)$). In reality, this would lead to a slight reduction in the indoor temperature. The objective is then to create an energy deficit equal to zero, which corresponds to an indoor temperature equal to the target indoor temperature, as fast as possible.

$$Q_{\mathrm{SH\_bal}}(t) = Q_{\mathrm{SH\_supply}}(t) - Q_{\mathrm{SH\_demand}}(t) \tag{11}$$

$$Q_{\mathrm{SH\_bal\_agg}} = \sum_{t=t_0}^{t_n} Q_{\mathrm{SH\_bal}}(t) = 0 \tag{12}$$

$Q_{\mathrm{SH\_bal}}$	Energy balance in kWh
$Q_{\mathrm{SH\_bal\_agg}}$	Aggregated SH energy balance in kWh
$Q_{\mathrm{SH\_demand}}$	SH energy demand in kWh
$Q_{\mathrm{SH\_supply}}$	Energy supplied from the tank for SH in kWh
t	Time in h
t_n	Time at which energy equilibrium is restored
t_0	Time at which energy imbalance is created

This means that as soon as there is surplus energy in the storage tank, the surplus energy is used to compensate for the thermal deficit ($Q_{\mathrm{SH\_supply}}(t) > Q_{\mathrm{SH\_demand}}(t)$). Equation (12) applies whereby the aggregated SH energy balance shall be zero at time step t_n. Time step

t_0 represents the time at which the imbalance was created, t_n shall be as close to t_0 as possible. This procedure imitates the differential temperature control.

While the SH storage tank is charged directly with the required flow temperature from the flow of the heating system, the DHW storage tank requires a heat exchanger for reasons of drinking water hygiene so that heating water and drinking water do not get mixed. A temperature difference of 7 K is assumed between the DHW storage temperature and the required flow temperature through the heat exchanger based on [16]. While according to DVGW W 551, there are no specific requirements for DHW storage for single-family houses (SFHs), in most multi-family houses (MFHs), the entire DHW storage content must be heated to 60 °C at least once a day for reasons of hygiene. For the modelling later in this work (see section 5), a target DHW storage temperature of 50 °C is used for SFHs, and a target DHW storage temperature of 60 °C is assumed for MFHs. When the DHW storage tank is loaded, the heating flow temperature, which runs through the heat exchanger, does not start directly at maximum temperature but is constantly increased. In this model, the temperature increases constantly from 30 °C at a state of charge (SoC) of the tank that is lower or equal to 50 % up to the maximum flow temperature (target temperature of DHW storage tank + 7 K). The maximum flow temperature is reached when the SoC of the storage tank is higher or equal to 90 %.

In the modelling presented here, only fixed-speed HPs are considered, which is why the thermal power fed into the storage tank after switching on the HP can be determined according to equation (3) for the required flow and given heat source temperature (see also section 3.1). In the case of a monoenergetic system design, the thermal power of the heating rod is also fed into the storage tank when the 2-day average temperature falls below the bivalence point (see also equation (16)). The energy capacity Q_{sto} of the storage tank can be determined according to equation (13). For the DHW storage tank, ΔT is the difference between the target temperature of storage and the cold-water temperature (e.g., for a SFH in this work: 50 °C – 10 °C); for the SH storage tank, ΔT is the spread between the flow and return temperature of the heating system. A spread of 5 K is assumed here, which is typical for HPs [16].

$$Q_{\mathrm{sto}} = V \cdot \mathrm{c} \cdot \rho \cdot \Delta T \tag{13}$$

Q_{sto}	Thermal capacity of storage in Wh
ΔT	Temperature difference between target temperature of storage and incoming cold water temperature in K
V	Storage volume in m³
c	Specific heat capacity of water (= 1.163 Wh/(kg · K)
ρ	Density of water (= 997 kg/m³)

The resulting thermal losses $\dot{Q}_{\mathrm{loss}}$ of the storage tanks are considered following Commission Regulation (EU) No 814/2013. For the purpose of a conservative estimation and to cover potential further losses occurring in the hydraulic system, the maximum permissible losses are assumed as storage losses according to equation (14).

$$\dot{Q}_{\mathrm{loss}} = 16.66\ \mathrm{W} + 8.33\ \mathrm{W} \cdot \left(\frac{V}{1\ \mathrm{l}}\right)^{0.4} \tag{14}$$

$\dot{Q}_{\mathrm{loss}}$	Standing thermal loss of storage tank in W
V	Storage volume in l

Equation (15) can be used to determine the change in the SoC of the storage tank. The thermal power introduced into the storage tank $\dot{Q}_{\mathrm{HS}}$ can be determined according to equation (16) if the heating system is switched on (see switching conditions below). Otherwise, $\dot{Q}_{\mathrm{HS}}$ equals to zero. $\dot{Q}_{\mathrm{demand}}$ represents, depending on the storage tank type, the SH demand (see equation (8)) or the DHW demand determined by the utilized tap profile generator. At the start of modelling, the storage tanks are randomly assigned a SoC between 50 % and 90 %. In this model, both the SH storage tank and the DHW storage tank are assumed to be cylindrical tanks consisting of two layers with ideal layering (upper layer: warm water; lower layer: cold water) and two temperature sensors: the lower sensor is located at 10 % and the upper sensor at 50 % of the tank height (see also Figure 6). The SH demand is covered by the SH storage tank, the DHW demand by the DHW storage tank. As soon as the cold water layer reaches the position of the upper temperature sensor (storage SoC ≤ 50 %), the HP receives a switch-on command and recharges the corresponding storage tank until the warm water layer reaches at least the lower temperature sensor (storage SoC ≥ 90 %). For reasons of comfort, the reloading of the DHW storage tank always has priority over the SH storage tank.

The HP (and below the bivalence point in the case of a monoenergetic design also the heating rod) are therefore activated depending on the thermal demand of the building and the SoC of the storage tanks. By calculating the electrical power required for this, high-resolution electrical load profiles can be created for individual buildings. If the heating system is activated, the electrical power demand P_{HS} can be determined according to equation (17). $P_{\mathrm{rod\_nom}}$ equals the nominal electrical power of the heating rod, P_{HP} can be determined according to equation (4).

$$\frac{\mathrm{d}SoC}{\mathrm{d}t} = \frac{\left(\dot{Q}_{\mathrm{HS}} - \dot{Q}_{\mathrm{demand}} - \dot{Q}_{\mathrm{loss}}\right) \cdot t}{Q_{\mathrm{sto}}} \tag{15}$$

$$\dot{Q}_{\mathrm{HS}} = \begin{cases} (P_{\mathrm{HP}} \cdot \varepsilon_{\mathrm{OP}} + P_{\mathrm{rod_{nom}}} \cdot 0.99), \text{for } (T_{\mathrm{amb\_avg}} < T_{\mathrm{biv}}) \wedge \text{monoenergetic design} \\ (P_{\mathrm{HP}} \cdot \varepsilon_{\mathrm{OP}}), \text{for } (T_{\mathrm{amb\_avg}} \geq T_{\mathrm{biv}}) \vee \text{monovalent design} \end{cases} \tag{16}$$

$$P_{\mathrm{HS}} = \begin{cases} (P_{\mathrm{HP}} + P_{\mathrm{rod\_nom}}), \text{for } (T_{\mathrm{amb\_avg}} < T_{\mathrm{biv}}) \wedge \text{monoenergetic design} \\ (P_{\mathrm{HP}}, \text{for } (T_{\mathrm{amb\_avg}} \geq T_{\mathrm{biv}}) \vee \text{monovalent design} \end{cases} \tag{17}$$

P_{HP}	Electrical power consumption of HP in kW
P_{HS}	Electrical power demand of heating system in kW
$P_{\mathrm{rod\_nom}}$	Nominal electrical power of heating rod in kW
$\dot{Q}_{\mathrm{demand}}$	Thermal power drawn from storage in kW
$\dot{Q}_{\mathrm{HS}}$	Thermal power of heating system fed in storage in kW
$\dot{Q}_{\mathrm{loss}}$	Standing thermal loss of storage tank in kW
Q_{sto}	Thermal capacity of storage in kWh
SoC	State of charge of the storage tank
t	Time step in h
$T_{\mathrm{amb\_avg}}$	2-day average ambient temperature in K
T_{biv}	Bivalence temperature in K
$\varepsilon_{\mathrm{OP}}$	COP at the corresponding operating point

The following section shows exemplary modelling results of the presented approaches for typical buildings.

5 Modelling Results for Typical Buildings and System Configurations

In the following, the modelling results of the two approaches presented are shown using the example of a typical German SFH and MFH. Both a modern and an old house are modelled for each building type. Table 1 summarizes the characteristics of the analysed buildings.

Table 1: Characteristics of the analysed buildings

Characteristic	SFH modern	SFH old	MFH modern	MFH old
Heated area in m^2	152[1]	152[1]	521.4[2]	521.4[2]
Heat load in kW	4.56[3]	12.16[4]	15.64[3]	41.71[4]
Household sizes	2- or 4- or 6-pers. household	2- or 4- or 6-pers. household	3 x 1-pers. & 2 x 2-pers. & 1 x 3-pers. & 1 x 4-pers. household[5]	3 x 1-pers. & 2 x 2-pers. & 1 x 3-pers. & 1 x 4-pers. household[5]
Heating curve	A-10/W31[6]	A-10/W51[6]	A-10/W31[6]	A-10/W51[6]
Heating limit temperature in °C	12[7]	15[7]	12[7]	15[7]

The standard outdoor temperature is assumed to be -13 °C, and in the case of a monoenergetic design, the bivalence temperature equals -5 °C. A 2-, 4-, and 6-person household is considered for SFHs, allowing the effects of a change in DHW demand to be evaluated. A monoenergetic parallel system design is presumed if an air-source HP is utilized. A monovalent design is chosen for ground-source and water-source HPs. Figure 9 shows the modelling results for the period of standard outdoor temperature. In the case of the HRHS-Model (see also section 4.2), 10,000 2-day periods are modelled, and the average electrical load profile is displayed. The SHSAP-Model shows the determined stationary average power consumption. Only one iteration of calculation is required here. It should be noted that the heating system is dimensioned precisely in accordance with the thermal requirements in the analyses presented below and that no safety buffer for the dimensioning of the heating system is taken into account.

It can be seen that at standard outdoor temperature, an electrical power demand of the heating system of approx. 0.7 kW to 3.3 kW must be expected for a typical modern SFH, depending

[1] average value for SFHs; modern SFHs typically have a larger heated area than old SFHs, but for reasons of better comparability, the same area is used for both buildings in this work [26]

[2] average value for MFHs [28]

[3] heat load of 30 W/m^2, which corresponds to a typical value of a new building with a German standard of approximately KfW 70 (cf. [9] and [10])

[4] heat load of 80 W/m^2, which corresponds to a typical value of an older building with insulation according to German WSchVO 1982 (cf. [9] and [10])

[5] based on the typical allocation of household sizes in Germany (cf. [28] and [29])

[6] a heating curve of A-10/W51 corresponds to a building with lower thermal insulation and without surface heat exchange systems; a heating curve of A-10/W31 represents a building with very good insulation in which surface heat exchange systems are installed (cf. [14] and [27])

[7] based on VDI 4650-1

on the HP technology. For older SFHs, the electrical power demand is approx. between 3.6 kW and 9.2 kW. In particular, the HRHS-Model shows that the household size (and thus the DHW demand) only plays a subordinate role in the period of the standard outdoor temperature, as the thermal power demand for SH clearly predominates here. For a typical modern MFH, the electrical power demand is between 2.8 kW and 11.3 kW. For older MFHs, the power demand ranges from 12.5 kW to 31.6 kW.

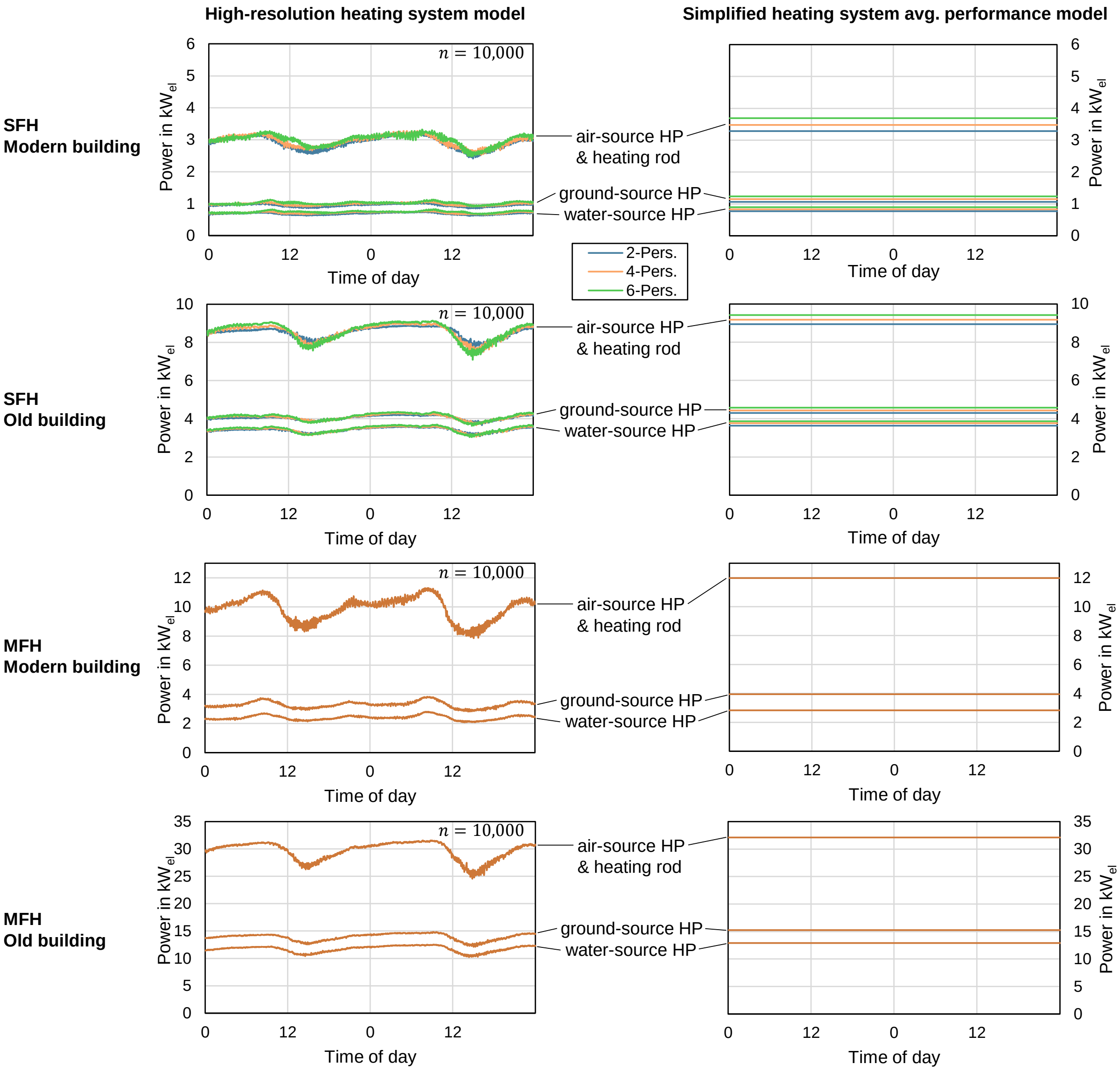

Figure 9: Comparison of the results of both modelling approaches for typical single- and multi-family houses at standard outdoor temperature

Air-source HPs in monoenergetic design have a significantly higher power demand than the other two technologies, with water-source HPs having the lowest power demand. Depending on the heating curve, there is a factor of about 2.5 to 4.3 between the power consumption of water-source and air-source HPs (HRHS-Model). As higher flow temperatures are required in the older buildings, ground- and water-source HPs also operate with slightly lower COP values. The fluctuations over the course of the day are also more significant in the case of air-source HPs, as in addition to the varying thermal demand, the heat source temperature also changes (typically highest in the afternoon), and therefore, the efficiency of the heating system also varies over the course of the day. For ground- and water-source HPs, constant heat source temperatures are assumed (see also section 3.1). In general, the fluctuations during the course of the day are also relatively small in the HRHS-Model. The peak load typically occurs in the morning at around 8 am, while the lowest load is to be expected in the afternoon. In the case of MFHs, the influence of DHW provision is more significant due to the higher number of residents, which is reflected in a slightly more distinct morning load peak. A minor evening peak is also recognizable here. It can further be shown that although the SHSAP-Model cannot generate a high-resolution daily profile, it does provide a good assessment of the expected average peak load with only minor deviations from the HRHS-Model (maximum overestimation of the load for the cases examined using the SHSAP-Model: 11 %, average overestimation of the load for the cases analysed using the SHSAP-Model: 4 %).

For a better contextualization of the depicted values, a comparison is made with the assessment basis for main lines in the low-voltage main distribution board of residential buildings (without electric heating and electric vehicle) according to the German standard DIN 18015-1. For a SFH, a rated power of approx. 9 kW, and for a MFH with seven apartments a rated power of about 48 kW are assumed here. This implies that, depending on the building and the HP technology, a load increase compared to the current rated power of up to 100 % (old SFH with air-source HP) has to be expected.

An overview of further modelling results of the SHSAP-Model is given below to enhance the transferability of the outcomes to other buildings. Figure 10 shows the results of the SHSAP-Model for different system configurations. The result is standardized here to the total heat load at standard outdoor temperature. Equation (18) applies. The normalized electric power demand $P_{\mathrm{HS\_norm}}$ is illustrated for a system with a typical standard outdoor temperature (design temp.) of -13 °C, a bivalence temperature of -5 °C, and a heating limit temperature of 15 °C. DHW power demand equals 0.2 kW, which is 20 % of the total heat load at standard outdoor temperature. Air-source HPs are in monoenergetic design, while water- and ground-source HPs are in monovalent design.

$$P_{\mathrm{HS\_norm}} = \frac{P_{\mathrm{HS}}}{\dot{Q}_{\mathrm{THL\_SO}}} \tag{18}$$

P_{HS}	Electrical power demand of heating system in kW
$P_{\mathrm{HS\_norm}}$	Normalized electric power demand of heating system in kW
$\dot{Q}_{\mathrm{THL\_SO}}$	Total heat load at standard outdoor temperature in kW

If the total heat load $\dot{Q}_{\mathrm{THL\_SO}}$ of a building at a comparable standard outdoor temperature is known, the electrical power demand P_{HS} of different heating technologies can be derived by multiplying the total heat load of this building by the respective power demands shown in Figure 10. In the case of monoenergetic systems, a significant increase in power demand can be observed below the bivalence temperature, which is caused by the additional utilization of the heating rod. It can be seen that a significant reduction in electrical power demand can be achieved by lowering the flow temperature (e.g., by replacing the heat exchanger system). It can further be shown that the use of water-source HPs generally results in the lowest electrical power demand, followed by ground-source HPs. The electrical power demand of air-source

HPs in monoenergetic design is here 2.4 to 4.1 times higher than that of water-source HPs in monovalent design at standard outdoor temperature, depending on the flow temperature.

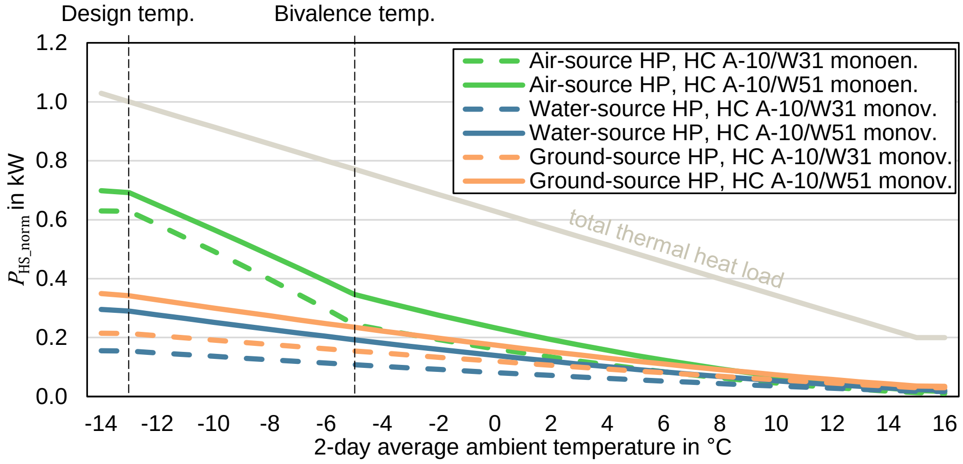

Figure 10: Normalized electrical power demand of different heating system configurations

Note: The abbreviation HC A-10/W31 means a flow temperature of 31 °C is required at an ambient temperature of -10 °C. This heating curve represents a low-temperature floor heating system. HC A-10/W51 indicates a flow temperature of 51 °C is required at an ambient temperature of -10 °C. This heating curve represents a medium-temperature radiator heating system.

It should be noted that the power demand shown in Figure 10 represents a stationary daily average load dependent on the mean outside temperature, in which the hydraulic system is not taken into account, and DHW heating is accounted for via a fixed increase of the heating load. Therefore, this modelling approach does not consider storage losses and the higher flow temperatures required for DHW provision. Nevertheless, the previous analysis (see Figure 9) show that the results of the SHSAP-Model enable a good assessment of the expected power demand.

6 Conclusion & Outlook

The growing share of HPs will lead to a substantial increase in decentral loads in electrical distribution networks. This paper analyses factors influencing the power demand of HPs, whereby external, building-related, heating system-related, and control-related factors are examined. Furthermore, two modelling approaches are developed to assess the electrical load increase due to HP systems. Since high-resolution modelling requires a comparatively high computational effort and detailed information about the heating object, both the easy-to-apply SHSAP-Model and the more detailed HRHS-Model are presented. The SHSAP-Model determines the efficiency of the HP system for different operating points and allows the estimation of the average daily electrical power demand of the heating system in dependency on the mean ambient temperature. The HRHS-Model further enables a building-specific assessment of the minute-by-minute load profile. For this, the hydraulic system and a heat-driven operating mode are modelled. A simplified thermal building model and a tap profile generator are utilized to derive the thermal SH and DHW demand.

The models are subsequently used to create characteristic HP load profiles for typical buildings and system configurations. It can be seen that at standard outdoor temperature, an electrical power demand of the heating system of approx. 0.7 kW to 3.3 kW must be expected for typical modern SFHs, depending on the HP technology. For older SFHs, the electrical power demand is approx. between 3.6 kW and 9.2 kW. A typical modern MFH shows an electrical power demand between 2.8 kW and 11.3 kW, while older MFHs have a power demand from 12.5 kW to 31.6 kW. Compared with current design recommendations for the rated power of buildings without electric heating and electric vehicles, this represents up to a doubling of the electric power demand depending on the building and system type. Air-source HPs in monoenergetic design have a significantly higher power demand than the other two technologies, with water-source HPs having the lowest power demand. It is further shown that although the SHSAP-Model cannot generate a high-resolution daily profile, it does provide a good assessment of the expected average peak load with only minor deviations from the HRHS-Model.

Network operators can use the models and results presented to estimate the future load increase due to HP-based heating systems. The next step in further research is to analyse sensitivities to the electrical load profile of HP systems. For example, the effects of over-dimensioning or under-dimensioning the HP, the influence of interventions by the grid operator, and different operation strategies and control behaviours should be evaluated. The modelling approaches are further used to forecast the electrical load increase resulting from the heat transition using the example of an urban distribution grid.

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