

Heat Pump Systems in Existing Multifamily Buildings: A Meta-Analysis of Field Measurement Data Focusing on the Relationship of Temperature and Performance of Heat Pump Systems

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Heat pumps play a central role in decarbonizing the heat supply of buildings. However, in this article, implementing heat pumps in existing buildings, a significant challenge is still presented due to high temperature requirements. In this article, a systematic analysis of the effects of heat source temperatures, maximum heat pump condenser temperatures, and system temperatures on the seasonal performance of heat pump (HP) systems is presented. The quantitative performance analysis encompasses over 50 heat pumps installed in residential buildings, revealing correlations between the building characteristics, observed temperatures, and heat pump type. The performance of an HP system retrofitted to a 30-dwelling multifamily building is presented in more detail. The bivalent HP system combines air and ground as heat sources and achieves a seasonal performance factor of 3.25 with a share of the gas boiler of 27% in its first year of operation. In these findings, the technical feasibility of retrofitting heat pumps is demonstrated in existing buildings and insights are provided into overcoming the challenges associated with high temperature requirements.

1. Introduction

The transition from fossil toward low-carbon heat supply technologies is vital to achieve climate protection targets in the building sector. There is a strong common sense within the research community that the future heat supply in Central Europe will predominantly rely on two technologies: district heating networks^[1] and heat pumps (HP) for residential and commercial buildings.^[2] This must be accompanied by a massive expansion of renewable power generation.^[3] The future role of combustion of biofuels and hydrogen in the heating sector is subject to an ongoing discussion.^[4] The urgency to move away from fossil fuels in buildings has become even clearer against the background of increasing gas prices and the security of energy supply in 2022.

HPs utilize low-temperature ambient heat sources such as air, geothermal heat or ground water, and electricity from preferably renewable energy sources to supply space heating (SH) and domestic hot water (DHW). While HPs are an established technology in new buildings, where low-temperature heating systems leverage a high energy efficiency, retrofitting heating systems in existing buildings is still considered a challenge^[5] and the implementation of HPs is far less widespread.^[6]

However, an analysis of the structure of the German building sector demonstrates that the greatest challenge and the greatest potential lie in the transformation of existing buildings. 62% of the building stock were constructed before 1979 and thus mostly prior to the implementation of the first thermal insulation regulations in 1977. These buildings account for about two-thirds of final energy consumption in the building sector.^[7] It is therefore clear that achieving climate protection targets can only be accomplished by transforming the heat supply in existing buildings. A look at the system technology shows that over 40% of heat generators are more than 20 years old, with an average age of the system technology of 17 years.^[8] This implies that many heat generators have to be replaced in the near future. To prevent lock-in effects, it is essential to replace existing fossil-based heating systems with low-carbon technologies prior to deep building

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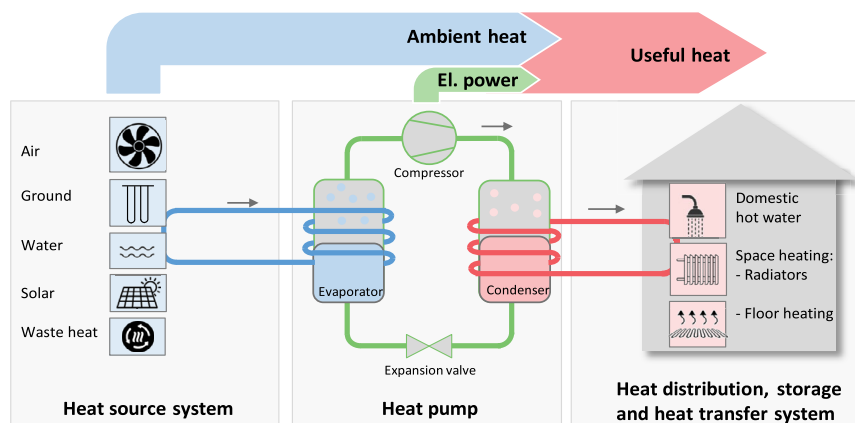


Figure 1. Subsystems of a heat pump (HP) system: heat source; heat pump; and heat distribution, storage, and heat-transfer system.

renovation, as the renovation rate of existing buildings currently lies around 1%.

HP systems consist of three subsystems: the heat source system, the HP and the heat distribution, storage, and transfer system. **Figure 1** illustrates the three subsystems with their components and technologies. Although HPs are also capable of cooling building, we focus on tap water and SH, which are the dominant functions in the Continental European climate. Retrofitting HP systems in existing multifamily houses (MFH) poses specific challenges as in many cases supply temperatures $>60^{\circ}\text{C}$ are required for SH and DHW provision.^[9]

Hydronic radiator systems are the predominant heat-transfer technology for SH in the European building stock. Existing radiators can be often be reused also in conjunction with retrofitting the heating systems with HPs. The required temperature levels in the SH circuit depend on the installed radiator area and heat-transfer capacity and can be determined following a room-wise calculation of heat demand and radiator capacity. Replacing single-bottleneck radiators by radiators of larger capacity might enable a further reduction of heating temperatures, and thus enhance the performance.^[10] Additionally, low insulation standards of existing buildings result in increased transmission heat losses and thus lead to a higher specific heat demand. In combination with a larger heated floor area, this leads to large required HP capacities in existing MFH.

Hot-water preparation and distribution in larger installations are a major challenge due to high temperature requirements to avoid the formation of legionella, a type of bacteria that can cause Legionnaires' disease. Legionella grow and multiply in water systems at medium temperatures, with highest growth rates between 30 and 50°C .^[11] The regulations to technically avoid Legionella in the DHW system differ between European countries. The German requirements, for example, demand a minimum potable hot temperature at the outlet of the DHW generator of $T_{\text{sup,DHW}} = 60^{\circ}\text{C}$ at all times.^[12] Additionally, the constant circulation of potable water has to be guaranteed with a minimum return temperature of $T_{\text{ret,circ}} = 55^{\circ}\text{C}$. A large share of circulation losses, often in the order of magnitude of the actual tap water demand, is observed in existing MFH of older building age, owing to bad thermal insulation of the circulation system.^[13] New technologies such as ultrafiltration might

enable temperature reduction without compromising water hygiene.^[11,14]

The availability and integration of ambient heat sources depends on the specific structural conditions of the building and its location. Especially in densely-built urban environments, the availability of conventional heat sources, air, and ground, can be limited.^[9,15] Photovoltaic–thermal (PVT) collectors, that are specifically designed for low-temperature integration as sole HP source, may be an alternative solution gaining more recent interest.^[16]

Adapted HP components and refrigerant cycles using natural refrigerants may meet these elevated temperature requirements. Propane (R290) is a promising refrigerant with working temperatures of 70°C and higher.^[17] Alternatively, hybrid, bivalent HP systems combine HPs with a secondary heat generator, e.g., a gas boiler, so that temperature and power requirements of existing MFH can be reached.^[18]

From the literature review, it becomes clear that high temperature requirements are central challenges for retrofitting HP in existing MFH, affecting all three subsystems: 1) all-year high heat source temperatures are desired for an efficient HP operation; 2) the maximum supply temperature of the HP is limited by the design of the refrigerant circuit; and 3) temperature requirements for DHW and SH affect HP performance and the necessity to install a backup heater.

As existing multifamily buildings are a relatively new segment for HPs, only limited monitoring data from field measurement is available. The available data is mostly reported for single installations, rarely following a consistent methodological basis that allows a cross comparison of reported indicators.

This article addresses this research gap, by systematically analyzing the relationship of operating temperatures and HP performance based on data from field measurements. The overall objective is to quantitatively assess the influence of heat source temperatures, maximum HP supply temperatures, and heat sink temperatures using statistical methods. Thus, the novel data enables a systematic understanding of the extent to which the studied key factors affect the performance of HPs in existing buildings.

To do so, results from multiple research projects at the Fraunhofer Institute for Solar Energy Systems are combined.^[19–23]

In the individual projects, specific aspects were investigated, developed, and evaluated based on experimental monitoring data. This article now presents a meta-analysis linking different recent projects and reevaluating their results on a consistent methodological basis.

Section 2 establishes the methodological basis by defining the key performance indicators utilized for the comprehensive evaluation of HP systems. In Section 3.1, an analysis of monitored HP systems compares the observed heat source temperatures of air, ground, and PVT collectors as heat sources. Section 3.2 derives the maximum condenser temperatures of currently available HPs in the market based on an analysis of certified products. The performance assessment of HP systems in field tests is conducted in Section 3.3, where the seasonal performance is correlated with various characteristics of the HPS and the building. Finally, in Section 3.4, a thorough analysis is conducted to evaluate the energetic performance and the corresponding equivalent $\text{CO}_{2,\text{equ}}$ emissions of a hybrid HPS in a large multifamily building during its initial operational year.

2. Methodology for Evaluating HPS

2.1. Coefficient of Performance

The coefficient of performance (COP) describes the instantaneous efficiency of HPs and is defined as the ratio of thermal power output $\dot{Q}_{\text{HP}}$ to the electrical power input P_{HP}

$$\text{COP} = \dot{Q}_{\text{HP}} / P_{\text{HP}} \quad (1)$$

From a thermodynamic perspective, the COP is derived from the temperature levels of the heat source $T_{\text{sup},\text{SRC}}$ and the heat supply $T_{\text{sup},\text{HP}}$ while considering the second law efficiency ζ for irreversible and nonideality effects

$$\text{COP} = \zeta \frac{T_{\text{sup},\text{HP}}}{T_{\text{sup},\text{HP}} - T_{\text{sup},\text{SRC}}} \quad (2)$$

Figure 2 plots the COP of HP with a second law efficiency of $\zeta = 0.5$ and source temperature of $T_{\text{sup},\text{SRC}} = 0^\circ\text{C}$. From these thermodynamic considerations, it becomes clear that a high HP efficiency is achieved by a small temperature increase from heat source $T_{\text{sup},\text{SRC}}$ to heat sink $T_{\text{sup},\text{HP}}$. In the example in Figure 2, the HP achieves a COP of 4.4 for a temperature lift

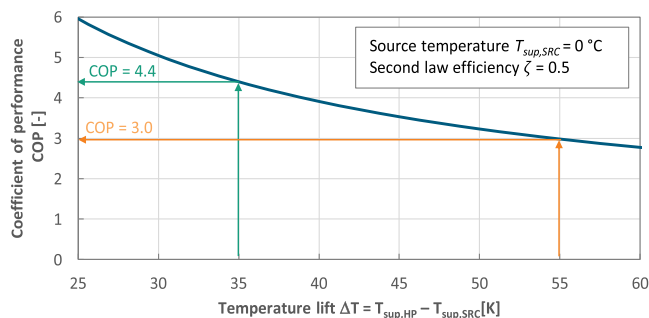


Figure 2. Coefficient of performance coefficient of performance (COP) as a function of the temperature lift from heat source $T_{\text{sup},\text{SRC}}$ to heat supply temperature $T_{\text{sup},\text{HP}}$.

of $\Delta T = 35^\circ\text{C}$ (e.g. floor heating system), while the COP drops to 3.0 for a temperature lift of $\Delta T = 55^\circ\text{C}$ (e.g. radiators).

2.2. Seasonal Performance Factor

The seasonal performance factor (SPF) characterizes the energetic performance of an HPS over a longer period, typically an entire heating period. Analogously to the dynamic COP, the SPF is defined as the ratio of the provided thermal energy Q to the consumed electrical energy E , both given in kWh. In the context of an HPS, the SPF can be evaluated within different system boundaries as shown in Figure 3. For a consistent basis for benchmarking and comparison, the following definitions based on refs. [19,24,25] are used in the following.

SPF_1 defines the efficiency of the HP and considers the electricity E_{HP} for the operation of the compressor, controller, and auxiliaries, in relation to the supplied heat for SH $Q_{\text{HP},\text{SH}}$ and DHW $Q_{\text{HP},\text{DHW}}$

$$\text{SPF}_1 = \frac{Q_{\text{HP},\text{SH}} + Q_{\text{HP},\text{DHW}}}{E_{\text{HP}}} \quad (3)$$

SPF_2 additionally includes the electrical consumers to operate the heat source E_{SRC} , such as fans, pumps, or devices for defrosting

$$\text{SPF}_2 = \frac{Q_{\text{HP},\text{SH}} + Q_{\text{HP},\text{DHW}}}{E_{\text{HP}} + E_{\text{SRC}}} \quad (4)$$

SPF_3 extends the system boundaries to the HPS and also includes the electricity input for electrical resistance heaters as backup units ($E_{\text{BU},\text{SH}}$ and $E_{\text{BU},\text{DHW}}$) and their corresponding heat output ($Q_{\text{BU},\text{SH}}$ and $Q_{\text{BU},\text{DHW}}$). Therefore, the system boundary of SPF_3 balances the useful heat supplied to the buffer storages.

$$\text{SPF}_3 = \frac{Q_{\text{HP},\text{SH}} + Q_{\text{HP},\text{DHW}} + Q_{\text{BU},\text{SH}} + Q_{\text{BU},\text{DHW}}}{E_{\text{HP}} + E_{\text{SRC}} + E_{\text{BU},\text{SH}} + E_{\text{BU},\text{DHW}}} \quad (5)$$

Finally, SPF_4 of the HP heating system also considers heat losses in the storage and distribution system and relates the useful heat $Q_{\text{HP}} + Q_{\text{DHW}}$ to all electrical consumers

$$\text{SPF}_4 = \frac{Q_{\text{SH}} + Q_{\text{DHW}}}{E_{\text{HP}} + E_{\text{SRC}} + E_{\text{BU},\text{DHW}} + E_{\text{BU},\text{SH}}} \quad (6)$$

2.3. Definition of Heat-Weighted Mean Temperatures T_m

The temperature level is of central importance for the analysis, benchmark, and optimization of HP systems. In both monitoring of field installations and simulation of HP systems, the assessment of mean temperatures has proven to be of great practical relevance for HP systems^[24] or PVT systems.^[26] Fraga et al.^[27] use a similar indicator to correlate the SPF with the mean temperature of various types of heat sources.

From our experience, we therefore recommend assessing the mean operating temperature regime by means of a heat-weighted average. This method aims to evaluate the effective temperature

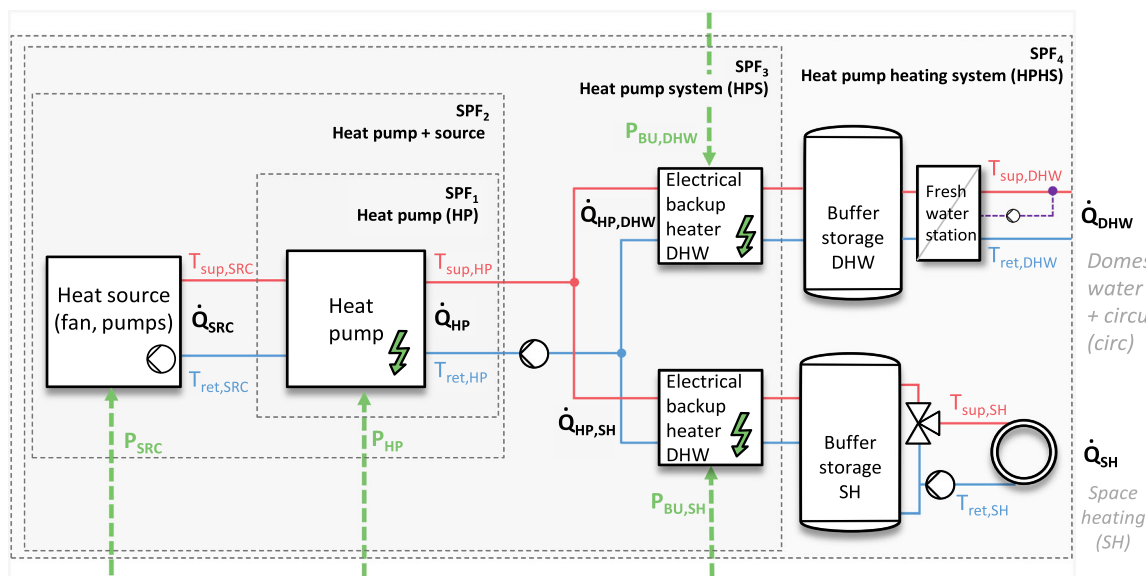


Figure 3. Definition of the seasonal performance factor (SPF) within four system boundaries with corresponding temperatures and electric power (P) and heat flow ($\dot{Q}$).

level with regards to fluctuating heat flow over the temperature node.

The general definition of the heat-weighted mean temperature $T_{m,i}$ is

$$T_{m,i} = \frac{\int_{t_1}^{t_2} T_i \cdot \dot{Q}_i dt}{\int_{t_1}^{t_2} \dot{Q}_i dt} \quad (7)$$

where T_i describes the temperature measured at location i , $\dot{Q}_i$ is the heat flow, and $t_2 - t_1$ the observation period.

3. Results

3.1. Heat Source Temperatures

The different types of heat sources vary in their temperature level and as a consequence in their energetic performances. According to German market figures for 2022, 87% of HPs for SH use air as heat source, followed by ground (10%) and water (3%).^[28] PVT collectors are a novel and upcoming technology that utilize ambient air and solar energy as heat source. To benchmark the performance of heat sources, data from field measurements is evaluated on a consistent methodological basis using the definition of heat-weighted mean temperatures in Equation (7).

Figure 4 plots the mean heat source temperature of 45 air source and 14 ground source HP systems from ref. [24] Four PVT HP systems in ref. [21] complement the analysis. For this purpose, we evaluated the heat-weighted outdoor air temperature for the direct-evaporation air source HPs and the heat-weighted brine temperature for the PVT and ground source HPs.

Ground source HPs achieve the highest performance factors due to relatively high temperatures of the ground in the main heating period with a median brine temperature of 6.2 °C. The median of the heat-weighted air temperature stands at

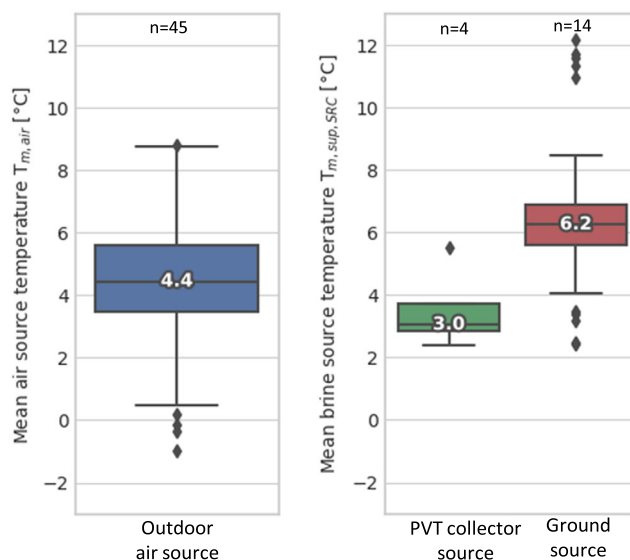


Figure 4. Heat-weighted mean source temperatures for different types of heat sources.

4.4 °C. Comparing these values, it has to be noted that we compare air with brine temperatures and this analysis disregards the heat exchange in the evaporator with a corresponding temperature drop.

Despite the possibility of PVT collectors to utilize solar radiation, their average source temperatures lie 1.4 K below that of air source HPs. The PVT collectors employed in our monitoring campaign are directly connected to the HP source without thermal insulation to maximize the utilization of ambient heat. However, during the winter's core heating period, when solar radiation levels are low, the PVT collectors primarily function as an air-to-brine heat exchanger (AHX), with minimal solar

heating contributions. Despite having higher instantaneous brine temperatures than air temperatures during sunny conditions, the mean brine temperatures remain 1.4 K below the air temperatures over the course of a year. This indicates that the solar utilization aspect of PVT collectors in HP systems is of secondary importance compared to their effectiveness in facilitating air-to-brine heat exchange.

Second, these differences are due to the mentioned methodological inconsistencies from comparing air and brine temperatures. Moreover, the difference might result from different analyzed measurement periods (PVT: 2022, others: 2016–2020). Given the limited number of monitored PVT systems, the reported figures are not representative and should rather serve as an indication of source temperatures.

The availability of ambient heat sources may be a critical issue, especially in densely built urban areas, if the available area is too small for the use of geothermal heat alone or if noise requirements limit the utilization of outdoor air. In these cases, PVT combined collectors, multisource systems, or cold district heating networks can be considered as alternative heat sources.

3.2. Maximum Supply Temperatures of HPs

The need to provide high temperatures in existing MFHs poses a challenge for the technological functionality of HPs to reliably supply heat also at elevated temperatures above 60 °C. The water temperature operating limit (WTOL) defines the maximum supply temperature ($T_{\text{sup,HP}}$) that a HP can achieve. The WTOL is described in EN 14825:2019, Annex E, and it is provided on the Heat Pump Keymark data sheet for HPs certified in the CEN Keymark scheme.

With existing buildings evolving as a growing market segment, the maximum supply temperature receives more relevance. In existing MFH, HPs often operate at or near their WTOL to provide the required temperatures which often lie between 60 and 70 °C depending on the distribution system for SH and hot water. If the HP is unable to do so, the backup unit has to cover the gap, resulting in a drop of efficiency and a corresponding increase of $\text{CO}_{2,\text{equ}}$ emissions.

Given the importance of the WTOL, we carried out an empirical, quantitative analysis of the maximum supply temperatures of commercial HPs in the market. To do so, we used the data basis of the Heat Pump Keymark^[29] to evaluate the WTOL of

existing HP models in dependence of their HP and refrigerant type. The dataset that was also used in ref. [22] was first checked for plausibility, and all models with a $\text{WTOL} < 50$ °C removed from the analysis. Note that the circumstance that the WTOL is typically provided by the manufacturer and not characterized or verified by the test labs leaves the possibility for inaccuracies of the information on the data sheet. In total, $n = 2502$ HP models are evaluated after preprocessing.

The histogram in **Figure 5** shows the number of HPs per 5 K interval, as most values for the WTOL on the datasheet are also given in 5 K steps (55, 60, 65 °C). And, 48% of all HPs reach a WTOL of 55 °C, 34% a WTOL of 60 °C, 13% a WTOL of 65 °C, and less than 6% a WTOL larger than that. Regarding the difference of HP type, there are several brine/water or water/water models reaching a WTOL of 65 °C. At the same time, there are several outdoor air HPs that also achieve a WTOL of 70 °C or even higher. This leads to the conclusion that, while 80 % of HPs in the market are limited to temperatures below 60 °C, there are 135 models available achieving temperatures of 70 °C or higher. Given the recent market trends toward R290 systems, it is likely that this distribution will see a considerable shift toward higher WOTL in the coming years.

The analysis of the dependency of HP type from the refrigerant gives an interesting insight: the two most prevalent refrigerants in the latest years, R32 and R410A, have a median WTOL of 55 and 60 °C, respectively. The category “Other” includes R134a, R452B, R407C, and unspecified refrigerants. Propane (R290) as natural refrigerant gains more attention in recent years, not only due to its low global warming potential of 3. It is also a refrigerant with large application potential due to its high critical temperature of 96.8 °C and the capability to achieve higher supply temperatures. The median WTOL of HPs with R290 lies at 70 °C and thus well above synthetic, fluorinated refrigerants. **Table 1** compares the properties of these refrigerants.

A central technological challenge for a more widespread application of R290 HPs is the flammability of propane, which is why HP circuits utilizing propane have special requirements regarding their installation locations. A charge of 150 g of refrigerant is considered the critical amount regarding restriction-free indoor application. Current developments aim at developing R290 HPs with a reduced charge, by identifying charge and energy efficient components and their combination with different designs of refrigerant cycles. Methler et al.^[30]

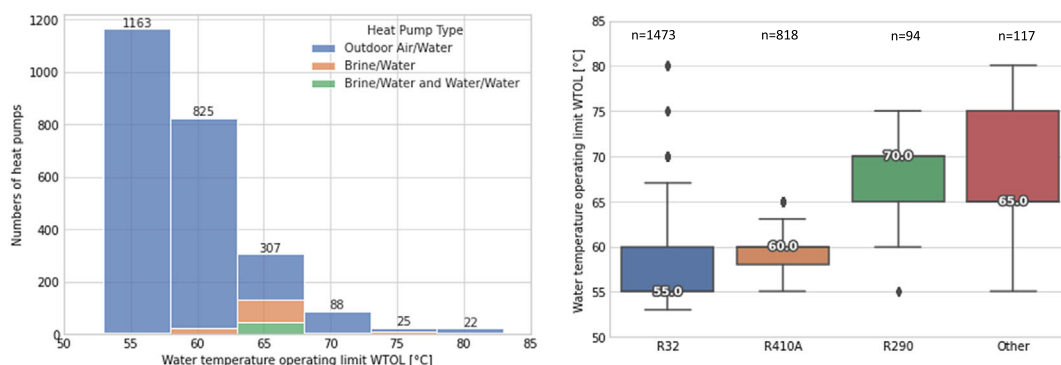


Figure 5. Number of HPs grouped by water temperature operating limit (WTOL) and boxplot per type of refrigerant.

Table 1. Properties of R32, R410A, and R290 refrigerants.^[26]

Refrigerant	Type	Critical temperature [°C]	GWP (IPCC4)	Flammability
R32	HFC	78.2	675	Low
R410A	HFC	72.5	2090	No
R290	Natural	96.8	3	High

report on experimental results of an HP that achieved an overall seasonal coefficient of performance (SCOP) of 5.0 with a minimum efficient charge of 180 g and a maximum heating capacity of 9 kW. This corresponds to a specific charge of 20 g kW⁻¹. A further reduction of charge to 150 g results in a decrease of the SCOP to only 4.7. In more recent developments, a heating capacity of 12.8 kW with an SCOP of 4.7 was achieved with 124 g of propane, which results in a specific refrigerant charge of 10 g kW⁻¹.

Cascading HP systems combine multiple refrigerant stages, so that in total large HP capacities can be achieved while guaranteeing safe operation with low charges per stage.

This has two main reasons: on the one side, comparing not only the WTOL but also COP efficiencies for available R290 HPs with all other market available HPs at higher supply temperatures^[22]; on the other side, designs with two-stage compression (real or quasi two-stage by vapor injection) could lead to higher efficiencies since it is the method of choice and known technique to allow high-temperature lift HPs as it is massively used for cold-climate HPs.^[31]

3.3. Analysis of Monitoring Data of HP Systems in Field Measurements

In the following chapter, we systematically and statistically evaluate the seasonal performance of HP systems based on long-term scientific monitoring and field measurements. A total of 58 HP systems were measured in renovated and unrenovated existing buildings during the period from July 2018 to June 2019. The investigated buildings consisted mainly of single-family

houses (SFHs) and some small-sized MFH. The characteristics of each of the 58 analyzed building and HPS are described in ref. [24]. In the following, we reevaluate the monitoring data focusing on the relationship of temperature and performance with respect to different features of the HPS. Thereby, the presented data focuses on SH and neglects domestic water heating.

3.3.1. HP Performance and Temperature for Different Types of HPs, Buildings, and Heat-Transfer Systems

Two indicators are used for the analysis of HP performance and temperature. The SPF for SH system $SPF_{3,SH}$ (Equation (5)) and the mean HP temperature $T_{m,HP}$. The latter indicator describes the energy-weighted yearly average of the HP supply and return temperature $(T_{m,HP,sup} + T_{m,HP,ret})/2$ following the definition of heat-weighted mean temperatures in Equation (7).

The monitored HP systems are categorized according to their type of HP, type of building, and type of heat-transfer system. **Figure 6** shows the boxplots for $SPF_{3,SH}$ and **Figure 7** shows the corresponding boxplots of $T_{m,HP}$, indicating the range and median values of performance and temperatures for each of these three categories.

Regarding HP type, ground source HPs achieve a higher $SPF_{3,SH} = 4.1$ compared to air source HPs ($SPF_{3,SH} = 3.3$). The median of the mean HP temperature amounts to $T_{m,HP} = 37.3$ °C for the air source HP systems and to $T_{m,HP} = 38.3$ °C for the ground source HP systems, respectively.

SFHs and small MFH were present in the monitoring project. The analyzed MFH consist of buildings with 2, 3, and 4 apartments, with a heated living area ranging from 190 to 420 m² with a median of 250 m². The heated floor area for the SFH ranges between 90 and 310 m² with a median of 170 m². The evaluated $SPF_{3,SH}$ differs between the building types, with the MFH having a value of $SPF_{3,SH} = 3.2$ and the SFH having a value of $SPF_{3,SH} = 3.7$. The median of mean HP temperature amounts to $T_{m,HP} = 36.8$ °C (SFH) and to $T_{m,HP} = 39.3$ °C (MFH), respectively.

Radiator, floor heating and mixed heat-transfer systems were present in the monitoring campaign. The median $SPF_{3,SH}$

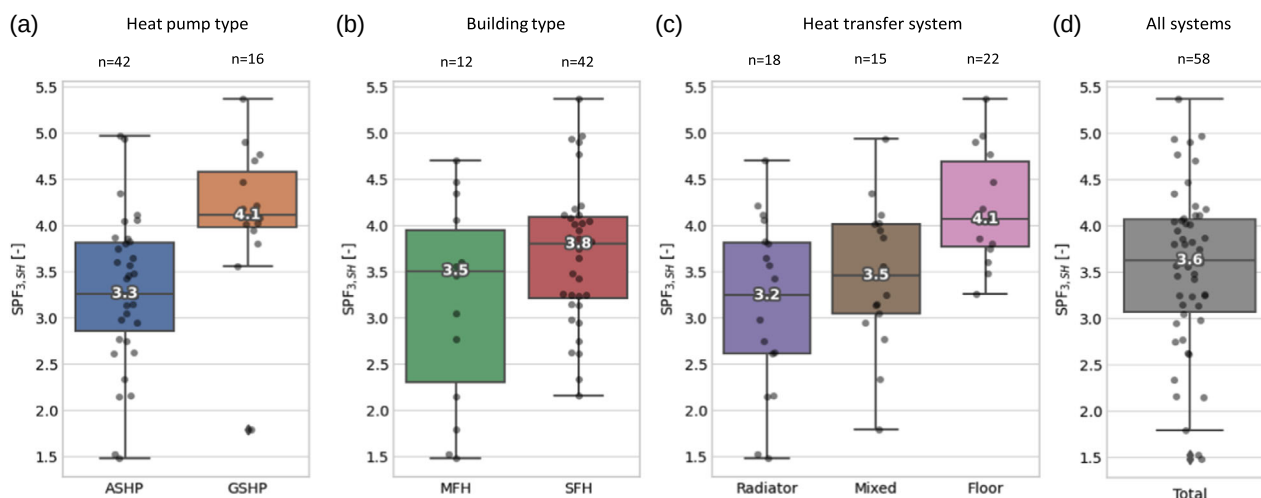


Figure 6. Box plots of the influence of a) HP type, b) building type, c) heat transfer system, and d) all systems on the SPF for space heating (SH) $SPF_{3,SH}$.

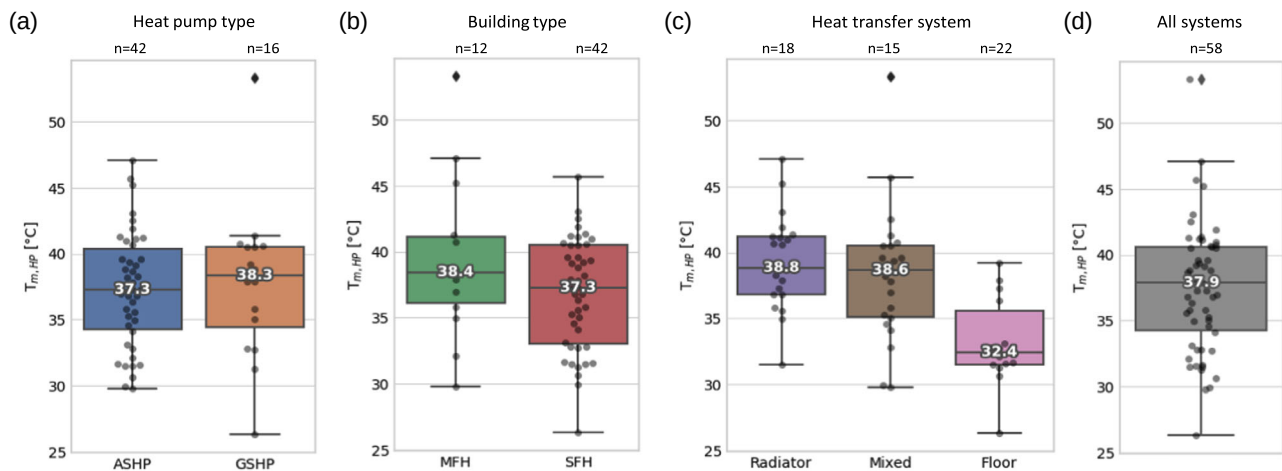


Figure 7. Box plots of the influence of a) HP type, b) building type, and c) heat transfer system, and d) all systems on the mean HP temperature $T_{m,HP}$.

amounts to 3.2 for radiator heating systems, 3.4 for mixed heating systems consisting of radiators and underfloor heating, and 4.2 for underfloor heating. These differences are predominantly ascribed to variations in the mean operating temperature, which is determined by the heating curve. The average mean HP temperature amounts to $T_{m,HP} = 33.2$ °C (floor heating), 38.4 °C (mixed systems), and 38.9 °C (radiators). There clearly seems to be a relationship between temperatures of the heat-transfer system and $SPF_{3,SH}$.

3.3.2. Heating Curve Temperatures of the SH Circuit

Efficient operation of HP systems requires that the SH circuit is operated at the lowest possible temperatures that can still provide the necessary heat and meet the thermal comfort requirements of the user. Heating curves are a common approach to control supply or return temperatures of the SH circuit as a function of the outdoor temperature.^[31] A theoretical equation of the heating curve, derived from the radiator equations, is given in ref. [32].

Heating curves differ in their shape and height: the nominal conditions, i.e., the ambient air temperature with the highest heat demand the system has to provide, determine the nominal supply and return temperatures $T_{sup,SH,nom}$ and $T_{ret,SH,nom}$. The shape of the curve is described by the radiator exponent, which affects the curvature and slope of the heating curve.

In the following, we analyze heating curves as observed during field measurements of the HP systems in ref. [24]. Return temperatures of the SH circuit, which frequently serve as control variables for the heating curve, are analyzed as a function of the outdoor air temperatures for this purpose. Trend lines of $T_{ret,SH}$ were identified on the basis of 1 year of field measurements using statistical analysis to determine the heating curve for each system. The analyzed heating systems are classified into three categories: floor heating ($n=9$), radiators ($n=14$), and mixed systems with both radiators and floor heating systems ($n=23$). In addition to the individual heating curves, the averaged heating curve for each category is plotted as bold line.

The results in **Figure 8** show that floor heating systems achieve the lowest heating curve temperatures, followed by mixed systems, while radiator systems reach the highest return temperatures. An interesting observation is made for the mixed systems: some heating curves have a behavior of floor heating systems, where the remaining radiators do not affect the overall heating curve; other heating curves are clearly dominated by the radiator with higher temperature requirements, resulting in heating curves close to radiator only systems.

A few heating curves show a constant behavior of the heating curve. These systems operate at unnecessary high temperatures even in the periods with a low heat demand, at the cost of a lower HP efficiency. Some heating curves are also characterized by a maximum cutoff temperature. This may be a solution for hybrid systems with multivalent heat generators. Overall, adapting the heating curves to the specific requirements to lower SH temperatures is an important aspect when optimizing the operation and performance of HPs systems.

3.3.3. Relation Between $SPF_{3,SH}$ and Mean HP Temperature, Specific Heat Demand, and Year of Construction

Multiple factors affect the performance of HP systems. Next to the previously discussed categories of HP, building and heat-transfer type, we now take a closer look at the influence of the mean HP temperature $T_{m,HP}$, the specific annual heat demand, and the original year of construction. **Figure 9** plots the SPF for SH $SPF_{3,SH}$ as a function of these parameters along with regression curves and the 95% confidence interval.

Figure 9a conveys the relationship between $SPF_{3,SH}$ and the mean HP temperature $T_{m,HP}$. As discussed in Section 3.3.1, the mean HP temperatures strongly affect $SPF_{3,SH}$. Reducing mean HP temperatures, e.g., by lowering heating curve temperatures, improves the HP performance directly. Especially in existing buildings lower supply temperatures is therefore one key toward an efficient HP operation.

The specific heat demand does not have a direct influence on $SPF_{3,SH}$ in Figure 9b. However, there is an indirect relation

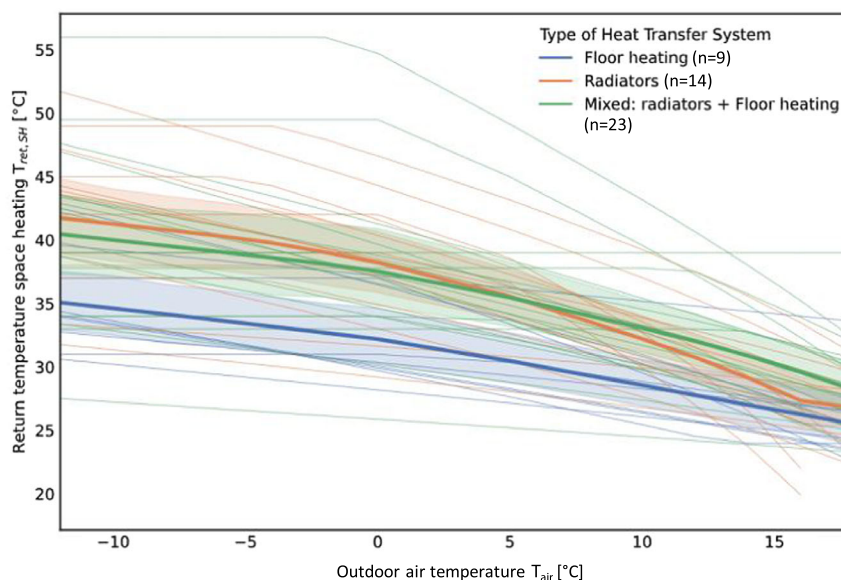


Figure 8. Heating curves of return temperature $T_{\text{ret,SH}}$ of floor heating, radiator, and mixed heat transfer systems with averaged heating curve per heat-transfer system as bold line.

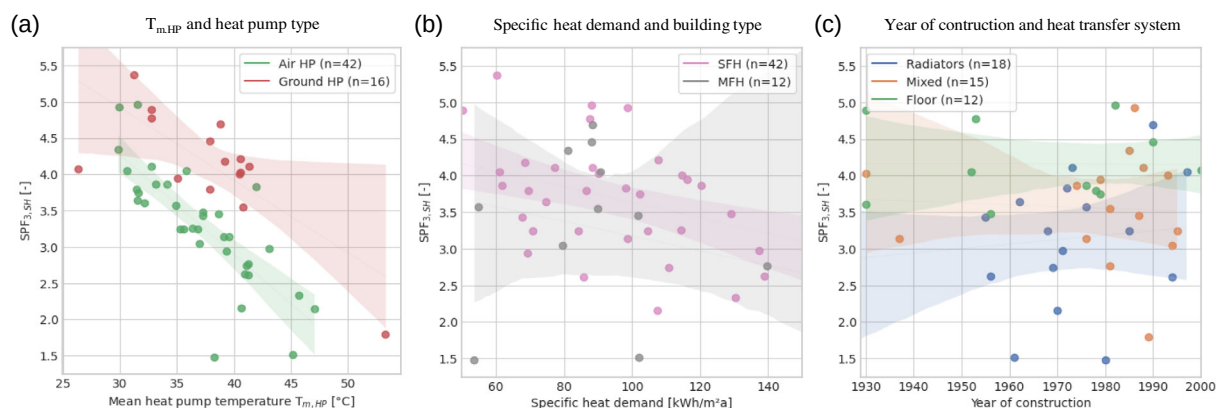


Figure 9. Scatter plots of the SPF for SH $\text{SPF}_{3,\text{SH}}$ related to a) mean HP temperature, b) specific heat demand, and c) year of building construction along with the 95% confidence interval per category.

that becomes evident when analyzing the cross correlation between the specific heat demand and $T_{\text{m,HP}}$. With a lower specific heat demand, also the $T_{\text{m,HP}}$ is significantly lower. Heating systems of better insulated buildings therefore operate at lower temperatures, leading to an increase of $\text{SPF}_{3,\text{SH}}$. On average for all systems, we observe a statistical decrease of $\text{SPF}_{3,\text{SH}}$ by 0.1 for every 10 kWh m^{-2} a higher space heat demand.

Regarding the influence of the original year of construction in Figure 9b, data points are scattered widely with wide confidence intervals. This indicates that no correlation between the year of construction and $\text{SPF}_{3,\text{SH}}$ can be identified. The depth of renovation and insulation measures varies significantly between the analyzed buildings, independent of their year of construction. This underlines the hypothesis that HPs can operate efficiently in buildings of all construction ages, independent of their original year of construction.

3.4. Performance Analysis of a Bivalent HP System in Large MFH

3.4.1. Characteristics of Demonstration Building and Bivalent HPS

The demonstration project “SQ-Durlach” in Karlsruhe, Germany, involves a detailed monitoring of a bivalent HPS in a large multifamily building with 30 apartments and a total heated floor area of 2107 m^2 . The objective is to demonstrate the technical feasibility of installing HPs in buildings without deep renovation. The building was constructed in 1963 and underwent renovation of its building envelope in 1995, leading to a reduction of the specific heat demand to 55 kWh m^{-2} . The specific hot-water demand for DHW and circulation amounts to 29 kWh m^{-2} . Detailed information on the building characteristics prior and after renovation can be found in ref. [10].

As part of the demonstration project, the previous gas boiler heating system was replaced by a bivalent HPS. Temperatures in

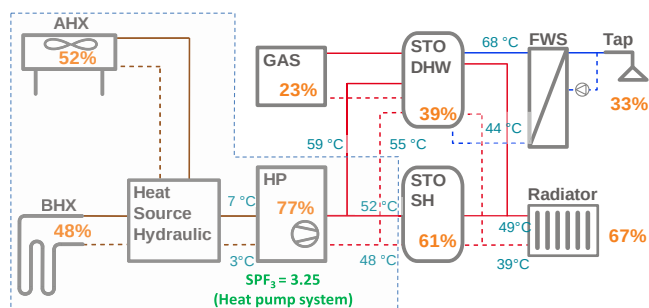


Figure 10. Hydraulic diagram of the bivalent HPS. Reproduced with permission.^[35] Copyright 2023, Fraunhofer IRB Verlag.

the SH circuit were reduced to 55/45 °C by replacing 8% of all radiators, for this purpose. Two parallel HPs with a total power of 71 kW_{th} combine the heat sources air and ground with intelligent control and adapted hydraulics. Compared to a single-source HP, only 53% of the size of a conventional borehole heat exchanger is required, thus reducing investment costs and installation space. The remaining thermal power is supplied by an AHX.^[33]

The maximum supply temperature WTOL of the HP amounts to 65 °C. An auxiliary natural gas condensing boiler supplies high-temperature heat for DHW and peak load power for SH at the coldest days. Three buffer storages with each 900 L are installed, of which two are used in series as a DHW storage and one as a buffer storage for SH. By means of a mixing valve in the SH circuit, heat from the DHW storage can be mixed to the SH circuit, which enables gas boiler support for SH. A fresh water station aims at balancing the hygienic requirements while trying to minimize return temperatures. **Figure 10** shows a simplified hydraulic diagram of the bivalent HPS.

3.4.2. Monitoring Results of the Bivalent HPS

Since its commissioning in February 2022, the HP system is in regular operation with power, temperature and heat flows being monitored. Several optimization measures regarding control were implemented in the first two months of operation.^[34]

The SPF of the HPS SPF₃ considers the electricity input for the HP, source hydraulics, defrosting, and controller. An overall, annual SPF₃ = 3.25 was achieved for the period from March 2022 to February 2023.^[35] Analyzing the heat-weighted mean temperatures and the annual share of heat source, heat generator, storages, and heat supply gives insight into the system performance: the good HP performance can be partly attributed to the multisource hydraulics that enable high source temperatures throughout the year with a mean source temperature of $T_{m,src,sup} = 7^\circ\text{C}$. The share of the gas boiler amounts to 23%. This relatively large share is mostly related to high energy-weighted supply and return temperatures of $T_{m,sup} = 68^\circ\text{C}$ and $T_{m,ret} = 44^\circ\text{C}$ from the fresh water station, which limits the HP share for DHW supply to 45%. The share of the HP for SH amounts to 98%.

The monthly energy balance in **Figure 11** displays the specific heat supply from HP and gas boiler along the monthly performance factor. SPF₃ increases during summer months with a peak of SPF₃ = 3.9 in October, as source temperatures increase

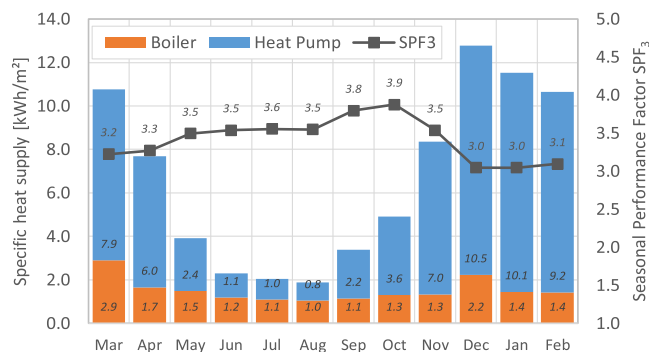


Figure 11. Monthly energy balance of the first year of operation obtained from measurement data. Reproduced with permission.^[35] Copyright 2023, Fraunhofer IRB Verlag.

and supply temperatures decrease. The heat supplied by the gas boiler is relatively constant, corresponding to its main use of supplying DHW at high temperatures. For SH, the gas boiler contributes only during the coldest days in December. In March, when the highest gas boiler share was observed, the optimization of the controller was still ongoing.^[34] The supplied heat of HP and gas boiler together is equal to the heat supplied by the previous gas boiler system. The improvement in terms of energy efficiency the share of heat supply by the HP from the overall heat supply.

To conclude, the demonstrated system shows good performance for SH and proves the functionality and advantages of a multisource system while the DHW heating still bears substantial optimization potentials.

3.4.3. CO_{2,eq} Emissions of Bivalent HP systems

The impact of an HPS on the environment is closely associated with the greenhouse gas emissions it generates. In monovalent HP systems, the CO_{2,eq} intensity of electricity from the power grid $f_{electricity}$ has to be considered in addition to the SPF₃. In bivalent systems, also the share of heat supplied by the gas boiler and the corresponding CO_{2,eq} intensity of natural gas has to be accounted for.

A simple method for balancing the specific carbon intensity of the supplied heat in grams of equivalent CO_{2,eq} emissions per kWh_{th} of useful heat is given by Equation (8)–(10).

The intensity of electricity $i_{electricity}$ per supplied useful heat in kWh_{el} kWh_{th}^{−1} is given by

$$i_{electricity} = r_{HP} / SPF_3 \quad (8)$$

where the HP coverage rate r_{HP} describes the share of heat supplied by the HP in a bivalent system.

The intensity of natural gas i_{gas} per supplied heat in kWh kWh_{th}^{−1} is given by

$$i_{gas} = (1 - r_{HP}) / \eta_{Boiler} \quad (9)$$

assuming a boiler efficiency η_{boiler} of 80% related to the higher heating value.

Table 2. Specific, local, equivalent CO₂ emission factors for electricity $f_{\text{electricity}}$ and gas f_{gas} related to the higher heating value, based on refs. [23,36].

Year	$f_{\text{electricity}}$ [gCO _{2,eq.} kWh ⁻¹]	f_{gas} [gCO _{2,eq.} kWh ⁻¹]	Ratio $f_{\text{electricity}}/f_{\text{gas}}$
2020	402.9	200.8	2.0
2030	193.0	183.9	1.0
2040	107.1	175.2	0.6
2050	21.1	166.4	0.1

Combining the electricity and gas intensity by weighing with emission factors of electricity $f_{\text{electricity}}$ and natural gas f_{gas} yields the carbon intensity $i_{\text{CO}_2,\text{eq}}$ in gCO_{2,eq.} kWh⁻¹ of supplied heat

$$i_{\text{CO}_2} = i_{\text{electricity}} f_{\text{electricity}} + i_{\text{gas}} f_{\text{gas}} \quad (10)$$

Table 2 shows the projected values of the CO_{2,eq.} emission factors $f_{\text{electricity}}$ and f_{gas} for Germany, based on refs. The values refer to equivalent CO₂ emissions which include direct and indirect emissions from CO₂ and other greenhouse gases. Accordingly, the emission factor of electricity $f_{\text{electricity}}$ is projected to decrease from 403 gCO_{2,eq.} kWh⁻¹ in 2020 to 107 gCO_{2,eq.} kWh⁻¹ in 2040 due to an increasing share of renewable electricity in the power grid. In the same period, the emission factors of gas are projected to decrease slightly from 201 to 175 gCO_{2,eq.} kWh⁻¹ owing to a larger share of green gases in the natural gas mix.

Figure 12 depicts the CO₂ intensity i_{CO_2} for the years 2020, 2030, and 2040 as a function of SPF₃ and r_{HP} using Equation (8)–(10). The carbon intensity of any bivalent HPS can be derived from this diagram. With current emissions factors, the CO_{2,eq.} intensity of HP systems is superior to any gas-based system, if the SPF₃ is higher than 1.6. Future development of the emissions factors favors the deployment of HPs. Even an SPF₃ of 1.0, which corresponds to direct electrical heating, is superior to gas boiler heating systems. In contrast, smaller HP shares reduce the impact of declining emission factors of electricity. Improving the HP efficiency and coverage rates then yields to lowest carbon intensity.

The bivalent HPS in Durlach, described in the previous chapter, with SPF₃ = 3.25 and r_{HP} = 77% achieves a CO₂ intensity of $i_{\text{CO}_2,\text{eq.}} = 152 \text{ gCO}_{2,\text{eq.}} \text{ kWh}_{\text{th}}^{-1}$ in 2020. This corresponds to a reduction of the CO₂ emissions of –40% compared to the previous gas boiler system ($r_{\text{HP}} = 0\%$) with a carbon intensity of $i_{\text{CO}_2,\text{eq.}} = 251 \text{ gCO}_{2,\text{eq.}} \text{ kWh}_{\text{th}}^{-1}$. Compared to the original state of construction in 1963 with a specific heating demand of $130 \text{ kWh m}^{-2} \text{ a}^{-1}$, this corresponds to a CO_{2,eq.} reduction of –70%. In 2040, the CO_{2,eq.} reduction potential is projected to be even larger, when the bivalent HPS in Durlach would achieve a carbon intensity of $73 \text{ gCO}_{2,\text{eq.}} \text{ kWh}_{\text{th}}^{-1}$, which corresponds to a reduction of CO_{2,eq.} emissions of –65% compared to the boiler system ($208 \text{ gCO}_{2,\text{eq.}} \text{ kWh}_{\text{th}}^{-1}$).

4. Conclusion

Retrofitting of HPs to existing buildings, particularly residential multifamily buildings, poses technological challenges related to high-temperature requirements for SH and hot-water supply. A systematic analysis of the relationship of heat source temperatures, maximum supply temperatures, and system temperatures on the performance of HP systems is presented in this article.

The heat-weighted mean temperature of different heat sources is analyzed on the basis of field measurements. A mean brine temperature of 3.0 °C is observed for PVT collectors and 6.2 °C for ground source HPs. A mean air temperature of 4.4 °C is found for air-source HPs. Especially in densely built urban areas, the technical availability of ambient heat sources is often limited. In these cases, a combination of multiple heat sources or the application of PVT collectors is an interesting and promising solution.

The findings of a market research on certified HPs indicate that maximum heating temperatures are predominantly restricted to supply temperatures below 60 °C. HPs with the natural refrigerant Propane (R290) are suitable to supply maximum temperatures of 70 °C and beyond. These types of HPs can be seen as a promising solution to provide higher temperature heat to multifamily buildings. However, the findings from long-term field test trials did not include R290 HPs which just recently started to substitute the high-GWP refrigerant R410A.

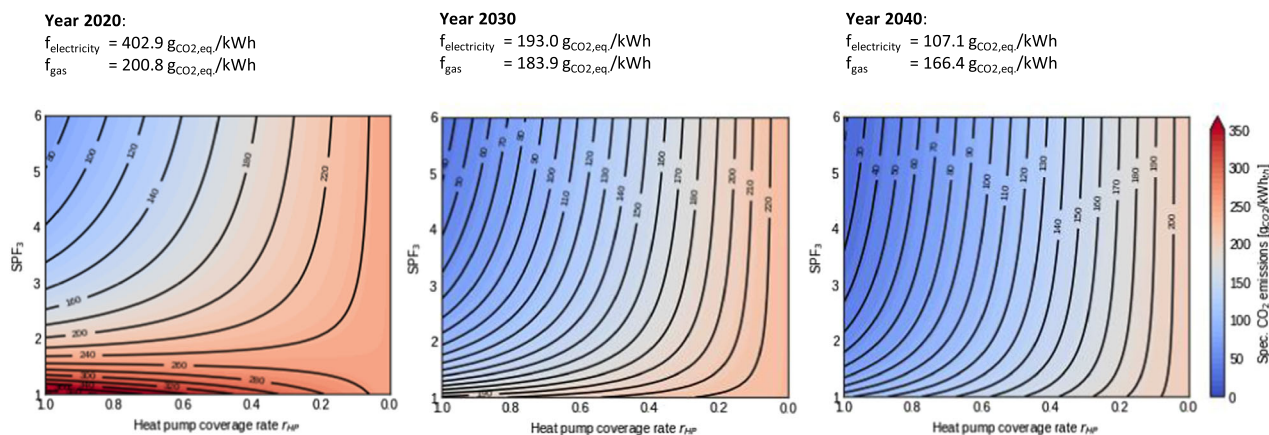


Figure 12. CO_{2,eq.} intensity of bivalent HP systems as a function of the SPF₃, HP coverage rate r_{HP} , and emission factors of electricity and gas.

On the HPS level, we evaluated the performance of 58 HP systems installed in existing buildings in the field and studied the impact of various factors on the seasonal performance. The analysis considers HP type, specific heating demand, building type, year of construction, and heat-transfer system. The findings provide valuable insights into the strengths and limitations of HP systems in different contexts, guiding the development of more efficient and effective systems in the future.

The detailed analysis of a bivalent HPS in a large multifamily building achieved an SPF₃ of 3.25 with a share of the HP of 73% in its first year of operation. With the current carbon intensity of the German electricity mix, this corresponds to a CO_{2,eq} reduction of −40% compared to the previous natural gas-condensing boiler. The same system would even achieve a CO_{2,eq} reduction of −65% in 2040 employing projected grid intensities.

In conclusion, the HP solutions and case studies presented in this article demonstrate the potential of applying HPs in existing multifamily buildings. Successful implementation, however, always requires careful planning and installation for every specific building individually. Standardization of systems and the development of digital planning methods may accelerate the deployment of HPs in this sector. Additionally, further research and development efforts are needed to identify solutions to replace gas-fired apartment-wise heaters in multistory residential buildings and to advance the use of climate-friendly refrigerants such as propane. These steps will be critical to achieving the decarbonization goals of residential multifamily buildings and transitioning to more sustainable heating.

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Conflict of Interest

The authors declare no conflict of interest.

Data Availability Statement

Research data are not shared.

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decarbonization, energy-efficient buildings, existing buildings, heat pumps, renewable heating, seasonal performance factor

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