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Field Experience with Residential Heat Pumps in Switzerland: Potential for Improvement and Future Developments

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Abstract

This study summarizes the main findings from field measurements with 14 air-to-water heat pumps (AWHP) and 12 brine-to-water heat pumps (BWHP) in Switzerland. The focus is on heat pump control optimization and possible future performance developments until 2050. The following statements can be derived so far:

- AWHPs with variable-speed compressors are particularly suitable for new buildings with low supply temperatures (e.g., floor heating).
- At a temperature lift of 25 K, AWHPs with inverter technology are 22 % more efficient on average than those with fixed speed. At 40 K, both compressor types operate equally efficiently.
- For retrofits and higher supply temperatures, BWHPs are preferable due to their higher efficiency and more stable source temperature. However, proper sizing, integration, and parameterization are key to the efficient and durable operation.
- AWHPs currently achieve a measured Seasonal Performance Factor (SPF) of 3.5 in new single-family houses (combined heating and hot water mode), while BWHPs achieve an SPF of 4.9.
- In an average-case scenario (new house, 30 °C to 35 °C floor heating, 60 % Carnot efficiency), SPFs of 6.3 and 7.9 are reachable by 2050.

Keywords: field measurements; domestic heat pumps; air-to-water heat pumps; brine-to-water heat pumps; efficiency; optimization

1. Introduction

Heat pumps (HP) for heating and hot water production are on the rise in Swiss households. In 2021, 33,704 units were sold, corresponding to a growth rate of 20 % compared to 2020 (28,064 units) [1]. About 56 % of the HPs fall in the heating capacity range between 5 and 13 kW and over 86 % below 20 kW. Furthermore, 73 % are air-to-water heat pumps (AWHP), 25.6 % are brine-to-water heat pumps (BWHP), and 1.4 % are groundwater-to-water heat pumps (GWHP).

At the same time, estimating the field performance of such HP systems is becoming increasingly important, as the efficiency responds to their integration into the heating system and the settings of the HP control. Therefore, the Heat Pump Test Center (WPZ) and the Institute for Energy Systems (IES) at the Eastern Switzerland University of Applied Sciences in Buchs (SG) have been conducting field measurements of HPs on behalf of EnergieSchweiz since 2015.

The main objectives of the field measurements are to investigate the real performance of HP systems and identify the systems' optimization potential, which can then be implemented. Typically, five new HPs are included in the measurement series every year.

Until 2020, the field measurement campaign included mainly new HP systems installed in single-family houses (new buildings or renovations). However, from 2021 on, HP systems in multi-family homes with a heating capacity of approximately 20 to 30 kW have been included. Before on-site installation, the HPs are

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evaluated in the laboratory at the Heat Pump Test Center (WPZ), and the measuring equipment (e.g., PT-100 sensors with four-wire technology and flow sensors) are calibrated [2], [3].

Compared to former field studies in the 1990s and early 2000s like FAWA (Field Analysis of Heat Pump Installations) [4], the measurement methodology and data acquisition technology have changed considerably. Thanks to digitalization, much more data is available and can be monitored. In addition, precise and automated measuring equipment and high sampling rates of 10 Hz enable meaningful measurement data. Between 30 and 40 sensors are installed in each HP system, and mean values are stored every 10 s. The goal is an overall uncertainty of the target values (e.g., COP, SPF, etc.) of <10 % [3], [5].

The most important results of the field measurements are published in the annual reports of EnergieSchweiz. In addition, many of the results are also presented in Swiss technical journals for planners and installers so that new findings can be implemented directly. The reports and publications are publicly available for download on the website [6] of the University of Applied Sciences Eastern Switzerland.

Moreover, the results from the field monitoring study from 2015 to 2019 have already been presented at the 13th IEA Heat Pump Conference 2020 [3] and the Purdue Conferences 2021 [5], [7], and 2022 [8]. The results clearly show the expected dependence of the seasonal performance factor (SPF) on the supply temperature (T_{supply}) and the selected heat source (e.g., air and brine) [8].

AWHPs in new buildings achieved an average SPF of 3.7 with floor heating (35 °C), while BWHPs had an average SPF of 5.7 [7]. At higher supply temperatures, such as about 50 °C in old buildings with radiator heating, average SPF values of about 2.9 for AWHPs and 4.4 for BWHPs were measured (see Table 1). Combined heating and hot water production systems showed 3 % to 9 % lower SPF due to increased supply temperatures. Typical optimization measures identified were adjusting the heating curve and the heating limit, legionella routines, increasing the charging time at midday for AWHPs and preheating the hot water with the compressor before starting the legionella program with an auxiliary heater. BWHPs were recommended for refurbished buildings.

As of August 2022, 26 HP systems are included in the field measurements campaign. This comprises 14 AWHPs, 9 of which are speed-controlled, and 12 BWHPs with 7 speed-controlled models. In 4 installations, hot water is heated with a separate domestic hot water heat pump (DHWHP). Cooling mode is activated at 6 objects. Meanwhile, up to 6 heating periods, i.e., 2016/17, 2017/18, 2018/19, 2019/20, 2020/21, and 2021/22, can be evaluated per system [9].

So far, the monitoring results have shown that most HP systems are efficient and run robustly. Severe deficiencies were found only rarely. However, the greatest optimization potential was identified in the HP control systems. Therefore, this paper summarizes important findings concerning the optimization of the controller settings.

2. System boundaries and key performance indicators

For the characterization of HP systems in the field monitoring study, EnergieSchweiz has defined different system boundaries and key performance indicators [2], [10]. Figure 1 shows an example of a BWHP with direct heating and domestic hot water storage.

The system boundaries are drawn not only in terms of sensor position but also in terms of time. In addition, a distinction is made between the operating modes “heating,” “hot water charging,” and “cooling.” The electrical standby power consumption (standby here means compressor standstill and no cooling operation) is assigned to the "space heating (SH)" or "domestic hot water charging (DHW)" operation, depending on the position of the three-way valve.

The seasonal performance factor (SPF^{+}) (Eq. 1) is the key indicator for the efficiency of the HP unit. Only the electrical energy of the compressor, fan (for AWHPs), source pump (for BWHPs), and control electronics of the HP are considered in this indicator. In contrast to the COP^{+}, the SPF^{+} value also includes the energy demand of the source circulation pump ($E_{CP,source}$) for BWHPs.

The heat utilization ratio (HUR) (Eq. 2) also includes the electrical energies of the sink circulating pump ($E_{CP,sink}$) and all electrical auxiliary heaters ($E_{ext,HE}$). In this way, the efficiency of the entire heat generation, including the electricity consumption for the distribution system, is considered and thus made comparable to other heating systems.

Finally, the system utilization ratio of domestic hot water (SUR_{DHW}) (Eq. 3) defines the efficiency of the overall hot water generation concerning the used domestic hot water from the storage tank outlet. It includes all storage and distribution losses. A smaller DHW demand generally leads to lower energy demand and efficiency since the losses are more significant. In addition, the SUR_{DHW} can also be determined for DHWHPs, allowing comparison with DHW charging of combined HPs.

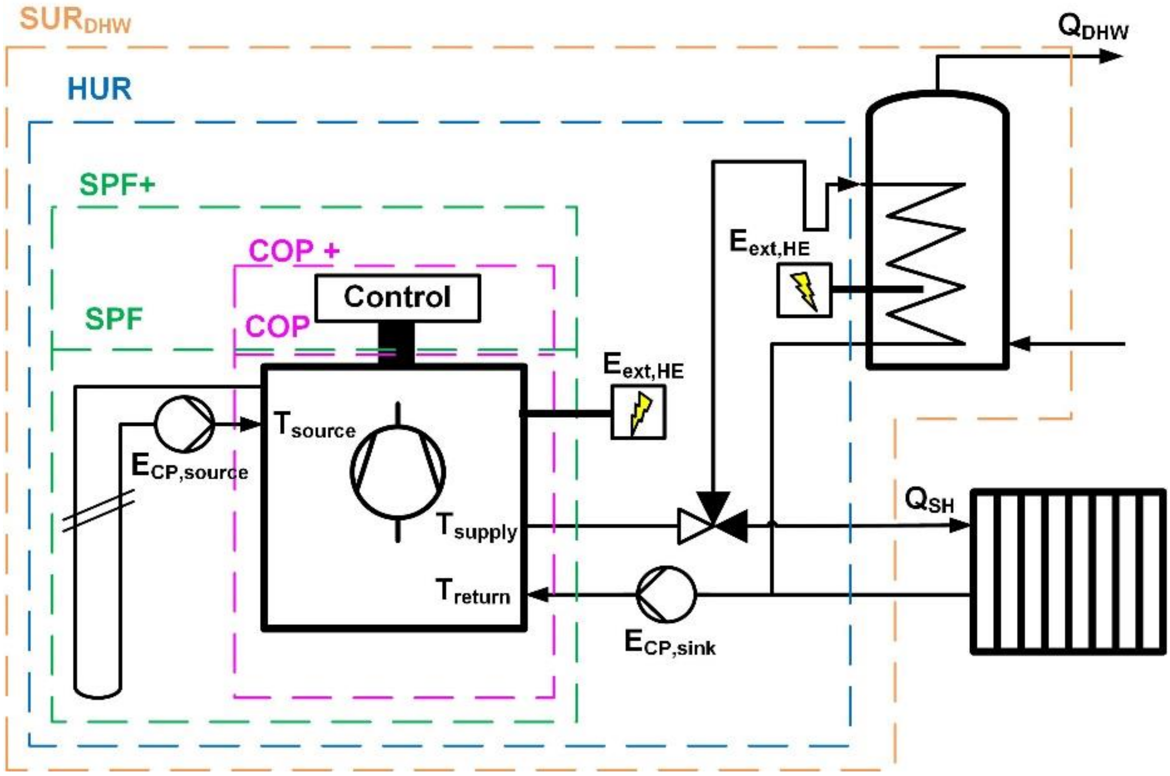

Fig 1. System boundaries and key performance indicators (SUR, HUR, SPF, COP) of a brine-to-water heat pump with a direct heating circuit and domestic hot water heating with a storage tank (Definitions based on EnergieSchweiz [2], [10]).

The total electrical energy requirements for the entire HP system, commonly referred to as electricity consumption, are summarized in the value E_{tot}. Notably, only the performance indicators SPF^{+}, SPF^{+}_{SH}, SPF^{+}_{DHW}, and SUR^{+}_{DHW} do not include standby losses since the electrical energies are only considered during active compressor operation. On average, the share of electrical standby losses is 2 % to 3 % of the total annual electrical energy demand [9].

The specific DHW heat requirement (HR_{DHW}) (Eq. 4) and the total heat requirement (THR) (Eq. 5) refer to the energy reference area of the building (ERA) for better comparability of different building sizes. For the evaluation of the specific heating and electrical energy demand (Q_{HD} and E_{HD}) (Eq. 6 and Eq. 7), the required charging energy (Q_{SH} and E_{SH}) is related to the ERA.

The heating degree days (HGT) reflect the weather influence of a period (month or heating season) and/or location. A heating limit temperature of 12 °C and an indoor target temperature of 20 °C are used for new buildings ($HGT_{20,12}$). For renovated buildings, the heating limit is usually set at 16 °C ($HGT_{20,16}$). Heating degree days are only counted if the average daily temperature is lower than the heating limit [9]. Finally, the HGT results from the difference between the average daily temperature and 20 °C.

To sum up, the following formulas and parameters are used for data evaluation of the field measurements:

$$SPF^{+} = \frac{Q_{SH}+Q_{DHW}}{E_{tot}-E_{CP,Sink}-E_{ext,HE}} \quad (1) \qquad HUR = \frac{Q_{SH}+Q_{DHW}}{E_{tot}} \quad (2) \qquad SUR_{DHW} = \frac{Q_{DHW}}{E_{DHW}+E_{ext,HE}} \quad (3)$$

$$HR_{DHW} = \frac{Q_{DHW}}{ERA} \quad (4) \qquad THR = \frac{Q_{SH}+Q_{DHW}}{ERA} \quad (5) \qquad Q_{HD} = \frac{Q_{SH}}{ERA} \quad (6)$$

$$E_{HD} = \frac{E_{SH}}{ERA} \quad (7) \qquad ER_{DHW} = \frac{E_{DHW}}{ERA} \quad (8)$$

with
SPF Seasonal performance factor [-]
HUR Heat utilization ratio [-]
SUR_{DHW} System utilization ratio of domestic hot water [-]
Q_{SH} Thermal energy requirement for space heating (SH) [kWh]

Q_{DHW}	Thermal energy requirement for DHW [kWh]
E_{tot}	Electrical energy input of the entire HP system [kWh]
$E_{CP,sink}$	Electrical energy of the circulating pump at the heat sink [kWh]
$E_{ext,HE}$	Electrical energy of the external heating elements [kWh]
E_{SH}	Electrical energy of the HP in SH operation [kWh]
E_{DHW}	Electrical energy of the HP in DHW operation [kWh]
E_{HD}	Specific electrical energy demand in heating operation[kWhel/m²]
Q_{HD}	Specific heating demand [kWh/m²]
HR_{DHW}	Specific DHW heat requirement [kWh/m²]
ER_{DHW}	Specific electrical energy requirement in DHW operation [kWh/m²]
THR	Total specific heat requirement (SH and DHW) [kWh/m²]
ERA	Energy reference area of the building [m²]
$HGT_{20,12}$ or $HGT_{20,16}$	Heating degree days [°C] with a heating limit of 12 °C (new house) or 16°C (renovation) and room temperature of 20 °C

3. Results

3.1. Effects of control optimization measures of a HP in a renovated building

This section presents some control optimization measures in a renovated single-family house from 1975 with inverter-regulated AWHP (Object No. 24 in the field study) [9]. Based on the evaluation of the heating season HS 2020/21 in the field monitoring study, some optimization measures could be implemented for the 2021/22 heating season. Data evaluation enables a direct performance comparison of the two heating seasons. The effects of control optimization on the HP system performance were as follows:

- **Speed reduction of the heat sink circulation pump:** The speed was reduced by approx. 10 % resulting in a decrease in the average pump power from 30.4 to 17.3 W. The running time decreased by 4 %, and the annual energy consumption of the pump reduced by 45 % (from 78.1 to 42.7 kWh/a).
- **Speed reduction of the compressor:** The compressor's minimum speed was reduced, lowering the minimum compressor power consumption from 4 to 2 kW. This increased the compressor's control range, and the compressor's annual operating hours increased by 2.5 % to 2,374 hours, although the total specific heat requirement (THR) decreased by 9 % to 66.3 MWh.
- **Reduction of the maximum heating capacity in the DHW mode:** The maximum thermal load in the DHW mode has been reduced from 8 to 6 kW to allow lower supply temperatures during storage tank charging and thus improve efficiency. Figure 2 (A) shows that the average supply temperature in DHW mode ($T_{supply,DHW}$) decreased from 51.4 °C in the heating season HS 2020/21 to 49.0 °C in HS 2021/22. At the same time, the specific DHW heat requirement (HR_{DHW}) decreased to 87 %, while the demand for electrical energy for DHW heating (ER_{DHW}) decreased to 61 % compared to the previous year (Figure 2, B).
- **Reduction of the heating curve and temperature rise in the buffer tank:** The temperature rise in the buffer tank was reduced in HS 2021/22, and thus the supply temperature in heating mode. Figure 2 (C) shows that the adjusted heating curve in HS 2021/22 is approx. 7 K lower than in HS 2020/21. The mean value of the supply temperature in heating mode ($T_{supply,SH}$) reduced from 45.0 °C to 37.8 °C (Figure 2, A). The heat source temperatures remained constant. There was no loss of comfort identified.

Figure 2 (B) compares the changes in the heating degree days and the thermal and electrical energy requirements. The heating degree days ($HGT_{20,16}$) decreased by 7 %, and the specific heating demand (Q_{HD}) by 8 %. However, the specific electrical energy demand in heating mode (E_{HD}) decreased by 31 %, which is attributed to higher efficiency (Figure 2, D) resulting from the mentioned improvements. The efficiencies increased in the heating and DHW operating modes. The SPF^{+}, HUR, and SPF_{SH}^{+} parameters increased by about 37 % (e.g., from 2.2 to 3.1), while the SPF_{DHW}^{+} and HUR_{DHW} increased by 42 %, and SUR_{DHW}^{+} even doubled (from 0.6 to 1.2). In DHW operation, the thermal energy demand (HR_{DHW}) decreased by 13 %, and electrical energy demand (ER_{DHW}) by 39 %.

Overall, the example of Object No. 24 shows that relatively simple control optimization measures can significantly impact the performance of an AWHP system.

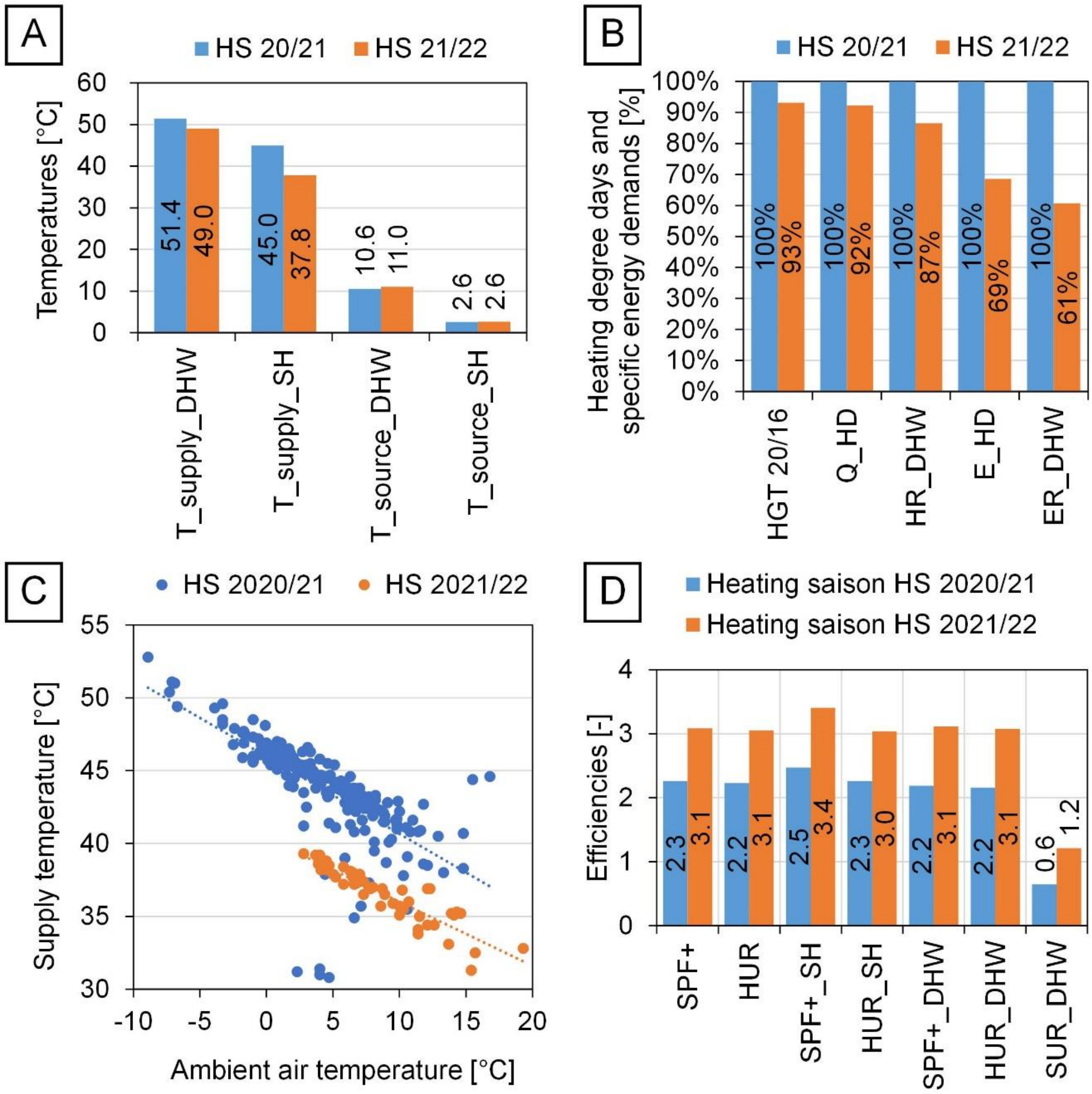

Fig. 2. Comparison of two heating seasons, HS 2020/21 and HS 2021/22, in a renovated single-family house from 1975 (Object No. 24 equipped with an inverter-regulated AWHP [9]) after control optimization on the HP. (A) Change of the supply and source temperatures in heating and DHW mode, (B) Comparison of the heating degree days and the specific thermal and electrical demands in heating and DHW mode (thermal demand decreased by 8 %, electrical demand decreased by 31 %), (C) Adjusted heating curve in HS 2021/22 (approx. 7 K lower supply temperature) compared to HS 2020/21 after a reduction of the temperature rise in the buffer tank, (D) Comparison of the efficiencies in heating, DHW, and combined operating modes for the heating seasons HS 2020/21 and HS 2021/22 (Data source: [9]).

3.2. Shifting the DHW charging operation to time windows with high outdoor air temperatures for AWHPs

As a next example, Figure 3 (A and B) compares the DHW charging characteristics of two renovated single-family houses (Objects No. 11 and No. 15) equipped with AWHPs. The graphs show hourly averaged data from 365 days (9/1/2021 to 9/1/2022) for several parameters over the daytime (00:00 to 24:00).

In Object No. 11 (Figure 3, A), the DHW operation often occurs in the morning between 7:00 to 8:00, accounting for about 48 % of the thermal energy. This time window corresponded to an annual average outdoor air temperature of 10.0 °C, thus the time with the lowest outdoor air temperature. Conversely, the highest outdoor air temperature was 16.6 °C and occurred between 15:00 and 16:00. Therefore, shifting the DHW mode operation to times with higher outdoor air temperatures would significantly increase the overall efficiency of the AWHP system if it does not result in a loss of comfort due to too low tapping temperatures.

As can be seen, in the time window from 11:00 to 12:00, the electrical consumption of the compressor was almost zero due to active power blocking by the local grid operator. Such blocking times are based on the load profiles of grid utilization and prevent controllable devices like HPs from consuming electricity at peak loads. Thus, blocking time detection can help optimize PV self-consumption or source temperature maximization.

Figure 3 (B) shows an equal representation for Object No. 15 (renovated single-family house with inverter-driven AWHP). Here, the DHW charging operation by the AWHP was concentrated at midday. 88 % of the DHW was generated between 12:00 and 15:00. This means that most of the DHW charging occurred when the outdoor air temperature (or the heat source temperature of the AWHP) was close to its daily maximum. In the selected time frame from 12:00 to 15:00, the average outdoor temperature was 15.7 °C. Ideally, the DHW charging operation would have to occur even two hours later (i.e., from 14:00 to 17:00) to reach the highest possible heat source temperature of 17.2 °C. In this context, the summertime changeover must also be considered. A one-hour shift results from the change of the clock to summertime. Another time shift results from later daily maximum temperatures in summer.

To sum up, the analysis of the HP operation over the day of Objects No. 11 and No. 15 showed that the DHW heating demand is virtually identical (273 W thermal power on average). However, Object No. 11. required more than twice the electrical compressor power (693 W vs. 299 W on average) and heating power (1’997 W vs. 999 W) due to operation in the early morning at the lowest outdoor air temperatures. Therefore, the timing of DHW charging after noon, between 13:00 and 15:00, is crucial for the high efficiency of the AWHP system and can be set in the HP controller settings. As a rule of thumb, a 10 °C higher source temperature results in 25 % higher HP efficiency.

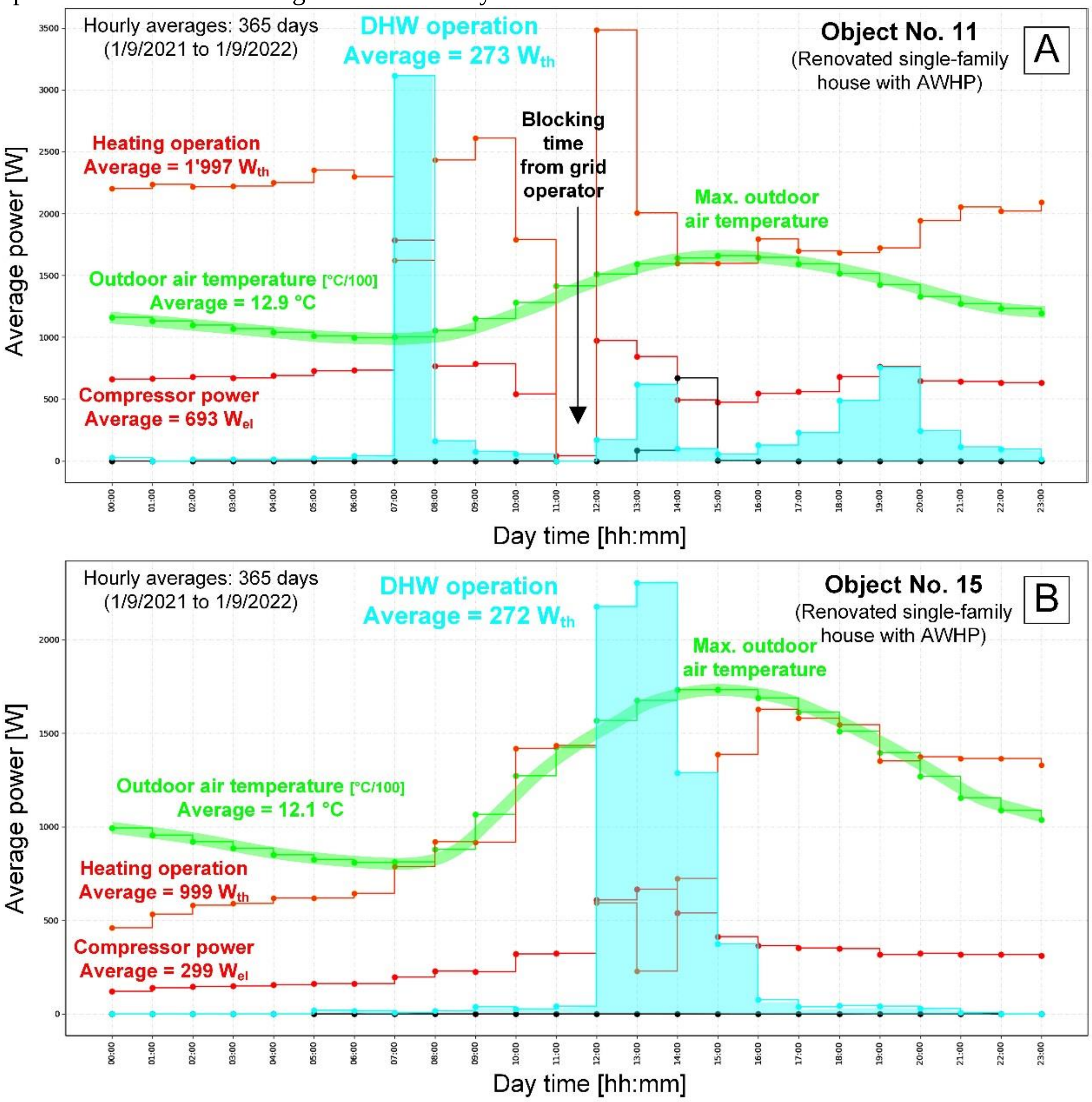

Fig. 3. (A) DHW operation at Object No. 11 occurs predominantly between 7:00 and 8:00 when the outdoor air temperature is low. The highest outdoor air temperatures are measured between 15:00 and 16:00. Between 11:00 and 12:00, blocking times of the electrical grid operator are active. (B) DHW operation at Object No. 15 is shifted to high outdoor air temperatures between 12:00 to 16:00. The operating window could even be moved to 14:00 to 17:00 to profit from the highest outdoor air temperature of 17.2 °C.

3.3. Comparison of fixed speed (on/off) and variable speed (inverter-driven) HPs

Inverter-driven HPs can modulate the compressor speed and thus dynamically adapt the HP performance to the required heating or DHW demand within certain limits. In contrast, the compressor speed is constant in conventional on/off HPs. In the following, some results of inverter-driven and on/off HPs are compared, and topics such as performance, sizing, running time (e.g., the compressor starts, cycling), and standby losses are discussed [8], [9].

Figure 4 shows the seasonal performance factor of 8 variable-speed and 4 fixed-speed AWHPs in heating mode (SPF_{SH}) as a function of the temperature lift between source and supply temperature (outdoor air temperature range between 0 and 10 °C accounts for >70 % of the annual heat demand). Each point represents a daily average. The Carnot efficiency of 30 % and 50 % is also plotted for orientation. Likewise, power function trendlines are added for each compressor group.

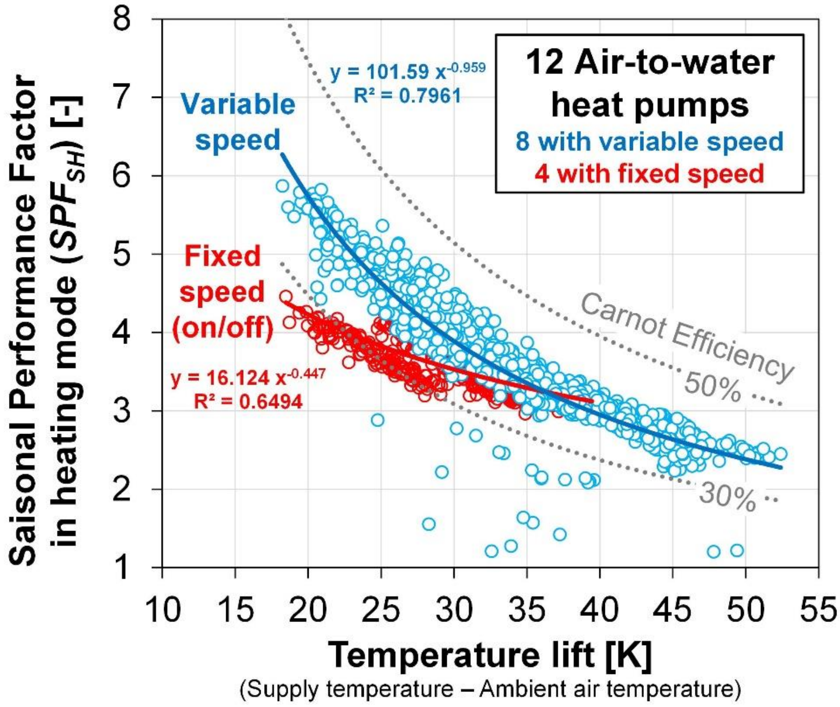

Fig. 4. Comparison of the seasonal performance factor of variable speed and fixed speed (on/off) AWHPs in heating mode as a function of the temperature lift between supply temperature and ambient air temperature as the heat source (ranging from 0 to 10 °C). Each point is a daily average (Data source: [8], [9]).

The evaluation shows that the Carnot efficiency of the variable-speed AWHPs was about 40 % over the entire temperature range. At a temperature lift of 25 K, the examined AWHPs with inverter (SPF_{SH} 4.67, variable speed) were on average 22 % more efficient than those with fixed speed (SPF_{SH} 3.85) [8]. Both compressor types achieved similar Carnot efficiencies for larger temperature lifts of about 40 K. However, the smaller the temperature lift, the lower the efficiency of the on/off HPs. At small temperature lifts, the Carnot efficiency also decreased below 30 %. A major reason is that fixed-speed HPs switch on and off much more frequently than variable-speed AWHPs. In addition, the temperature differences inside the heat exchangers of the HP are smaller with a variable speed compressor in part-load operation. Furthermore, the type of compressor used can also influence efficiency. For example, variable-speed rotary piston compressors usually work more efficiently at small temperature lifts than scroll compressors with fixed speeds [11]. In conclusion, variable speed AWHPs are especially suited for new residential buildings with low supply temperatures, e.g., 30 °C to 35 °C for floor heating.

However, the measured data for the heating season HS 2021/22 also revealed that the potential of inverter-driven HPs is not always fully exploited. Figure 5 shows that not all variable speed HPs have a significantly longer running time per start (red dots on the secondary axis). On average, the evaluated AWHPs with inverters had about 4 times more running time per start than on/off units (2 hours per start vs. 0.5 hours per start). For BWHPs, the average running time per start of inverter HPs was even 9 times higher than for on/off units (6.5 hours per start vs. 0.7 hours per start).

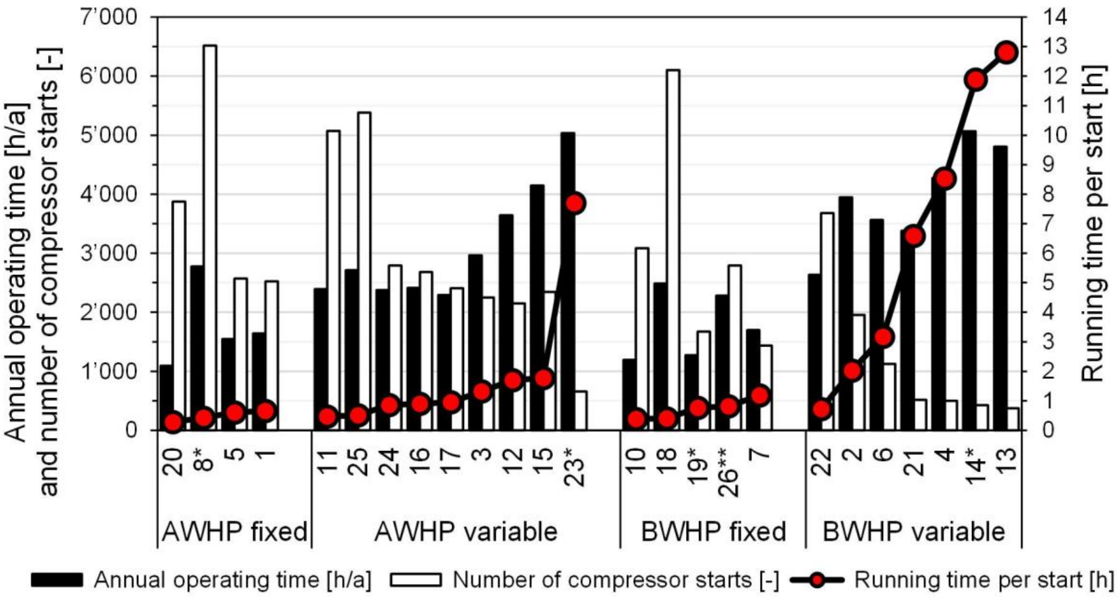

Fig. 5. Annual operating time, number of compressor starts, and running time per start of different fixed speed and variable speed AWHPs and BWHPs in HS 2021/2022, *Objects with separate DHWHP. **2 HPs with 4 parallel compressors (Data source: [9]).

The annual operating times differed considerably in some cases. For example, on average, speed-controlled inverter AWHPs had 1.7 times the operating times of on/off units. The difference was even greater for BWHPs. Here, the inverter machines had, on average, even 2.2 times longer operating hours.

For both AWHPs and BWHPs studied, one inverter system (Objects No. 23 and 14) ran for over 5,000 operating hours in the HS 2021/22. The average running time of the corresponding AWHP was 7.7 hours per start, and that of the BWHP was 11.9 hours per start. Even with optimal design, inverter-driven BWHPs achieved a higher average running time as the power control range depends essentially on the heat source temperature, which varied considerably more for AWHPs than for BWHPs.

In conclusion, the following general recommendations can be made for inverter-driven HPs:

- **Design:** Inverter-driven HPs should be well-matched to the heat demand of the building. Oversizing leads to a limitation of the control range due to the minimum HP capacity. Then, continuous operation of AWHPs is only possible at relatively low outdoor temperatures to deliver the minimum capacity to the building. If the heating demand is lower, an inverter-driven HP must also switch to cycle operation. Therefore, the advantages over an on/off HP are no longer given in these time ranges.
- **Start-up:** Good and solid commissioning is especially important for inverter-driven HPs. Software parameters such as the heating curve, heating limit, legionella activation, etc., must be determined and correctly set.
- **HP manufacturer:** There is potential to increase the efficiency of inverter-driven HPs. Some inverter-driven HPs have high standby losses in combination with long downtimes. In DHW operating mode, attention should be paid to low compressor speeds, as these lead to lower temperature differences in the hot water heat exchanger and thus increase the COP. In many cases, DHW generation is performed at a constant high speed because the main focus is likely to be on a short charging time and less on efficiency.

In addition to significant differences in operating behavior and efficiency, there were also revealing differences in standby losses between fixed-speed and variable-speed HPs. In general, standby losses are the energy that occurs during a HP standstill, i.e., when the compressor, including the auxiliary heater, is not in operation. The main causes of standby losses are the control system, an oil sump heater, and, in the case of inverter-driven HPs, the inverter. In the field monitoring study, these standby losses vary considerably depending on the object.

Figure 6 shows the average standby powers and the resulting annual standby losses grouped by 8 on/off and 16 speed-controlled HPs examined in the field monitoring study. The highest standby losses were found in some inverter-driven HPs. However, others with low standby losses indicate that high standby losses in inverter-driven HPs were not system related. The average standby power for the measured on/off HPs was 12 W, while the mean value of the inverter-driven HPs was 27 W and, thus, 2.3 times higher. On average, the five HPs with the highest standby losses (all with speed-controlled compressors) consumed 49 W.

Depending on the object, the electrical standby losses ranged between 0.5 % and 10 % of the total annual electrical demand. Across all HPs, the standby losses accounted for 2 % of the total annual electrical energy demand. For some inverter-driven HPs, there is still a relatively high potential for optimization, as a comparison with the "best" systems shows.

Qualitatively, the annual standby losses (secondary axis of Figure 6) showed a similar picture as the average standby power. However, the standby energy depends on the respective running or standby time, which is influenced by many factors, including the HP design (e.g., dimensioning of the heating capacity), the hydraulic integration, parametrization, and the actual user behavior (e.g., selected room temperature or DHW demand). For example, this effect can be seen in the comparison of Objects No. 2 and No. 24, whose average standby power was almost identical. However, the annual standby losses of No. 24 were about 29 % higher than those of No. 2. Thus, lower standby power does not necessarily mean lower standby losses [8], [9].

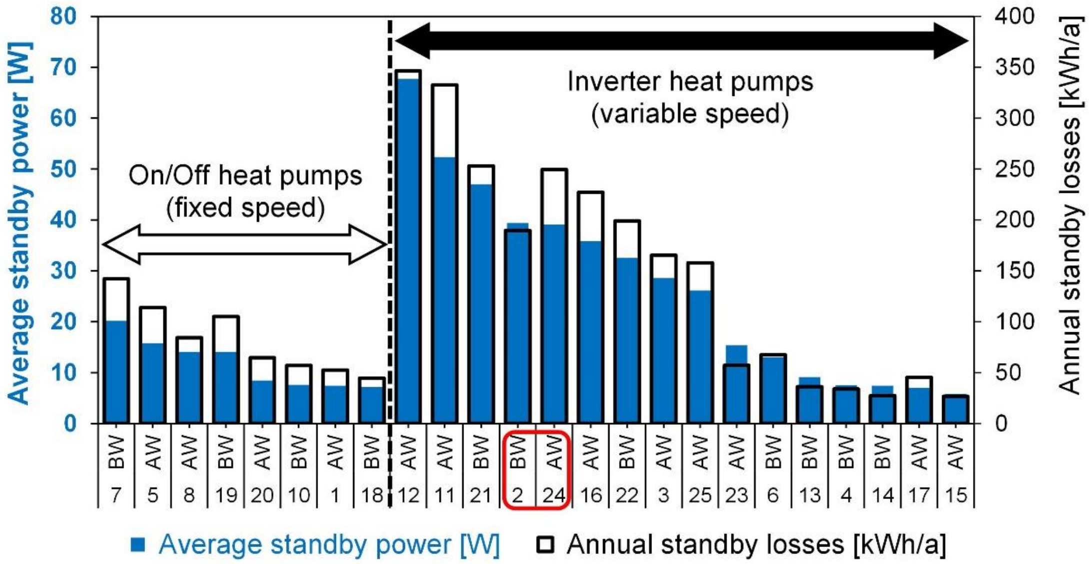

Fig. 6. Average standby power and annual standby losses of 8 fixed-speed on/off HPs and 16 speed-controlled inverter-driven HPs examined in the field study, sorted by decreasing standby power (Data source: [8], [9]).

In summary, comparing variable speed HPs with conventional on/off HPs shows a clear difference in efficiency and control strategies. Furthermore, standstill losses were much higher with an inverter because the HP control and frequency converter were always running. On the other hand, fixed-speed systems started up more frequently than variable-speed systems. The average running time of variable-speed compressors was more than twice that of fixed-speed systems. This indicates that many variable-speed compressors run at a more optimal operating point than units without capacity control, even though they tend to be oversized and often not optimally parameterized.

3.4. Seasonal performance factor for AWHPs and BWHPs by 2050

Table 1 summarizes the average *SPFs* of AWHPs and BWHPs in heating (*SH*), *DHW*, and combined (*SH*+*DHW*) mode based on 2018 field measurement data and provides an outlook for future development until 2050 for an average scenario with 60 % Carnot efficiency [12]. For the forecast, the considered efficiency strongly depends on the supply temperature of the building category (30 to 35 °C assumed for a new building, 40 to 45 °C for renovation, and 50 to 55 °C for old buildings). AWHPs currently achieve a measured SPF_{SH+DHW} of 3.5 in new single-family houses, while BWHPs achieve an SPF_{SH+DHW} of 4.9. Assuming the average-case scenario, SPF_{SH+DHW} values of 6.3 and 7.9 appear reachable by 2050, corresponding to a significant efficiency increase compared to today [12]. However, a prerequisite is that the economic and political framework conditions are set in such a way that further development of HP technology by the manufacturers takes place.

The difference in efficiency between AWHPs and BWHPs is most evident in the low supply temperature range, where the higher heat source temperatures have relatively more influence. In contrast, the difference in efficiency is smaller for old buildings because much running time of the AWHPs is at high outdoor temperatures. Nevertheless, field measurements show that BWHPs with higher supply temperatures perform better than those in the low-temperature range. Furthermore, BWHPs benefit from a stable geothermal heat source and have an average source temperature (7.9 °C) that is about 4 K higher than AWHPs (3.8 °C) [9].

Table 1. Seasonal performance factor (SPF) of AWHPs and BWHPs in heating (SH), hot water (DHW), and combined mode (SH+DHW) in 2018 and for an average future scenario in 2050 with 60 % Carnot efficiency (Data source: [12]).

Heat pump type	2018			2050 (Carnot efficiency 60 %)		
	SPF_{SH}	SPF_{DHW}	SPF_{SH+DHW}	SPF_{SH}	SPF_{DHW}	SPF_{SH+DHW}
New building: 35 to 30 °C (supply temperature at design point)						
AWHP	3.7	2.8	**3.5**	**6.5**	**4.9**	**6.3**
BWHP	5.7	3.2	**4.9**	**8.4**	**4.7**	**7.9**
Renovation: 45 to 40 °C						
AWHP	3.3	2.8	3.1	**5.2**	**4.9**	**5.1**
BWHP	5.0	3.2	4.6	**6.6**	**4.7**	**6.0**
Old building: 55 to 50 °C						
AWHP	2.9	2.8	2.8	**4.6**	**4.9**	**4.5**
BWHP	4.4	3.2	4.3	**4.9**	**4.7**	**4.8**

4. Conclusions

AWHPs with variable speed compressors are well suited as a heating system for the conditions in Switzerland, especially for new residential buildings with low supply temperatures (e.g., floor heating). Due to the low heat demand in new buildings, AWHPs are usually the most economical and frequently chosen HP type. On the other hand, BWHPs are preferred in renovations due to the higher supply temperatures and heating capacities. However, correct dimensioning, integration, and parameterization are key to the efficient and long-lasting operation of all heat pump types.

Regarding efficiency, the analyzed AWHPs with variable speed compressors perform better on average than fixed speed models at a temperature lift below 40 K. At around 25 K temperature lift, the advantage of inverter-driven AWHPs is approximately 22 % in the SPF. However, in standby mode, inverter-driven HPs revealed considerable standby losses, which, combined with the low compressor running time, can significantly decrease system efficiency. The annual standby losses can vary between 25 kWh/a and 350 kWh/a depending on the system (factor of 14!).

Although good HP efficiencies are already achieved in the field, these can be increased by simple tricks in the controller settings. The greatest potential for optimization was identified on the HP control side. Recommendations are:

- Optimization of the supply temperature (i.e., settings of the heating curve and heating limit, 1 °C lower supply temperature corresponds roughly to 2.5 % efficiency gain)
- Optimizations in the control of the heat sink pump and the compressor speed
- Timing of DHW operation for AWHPs in the early afternoon (e.g., 13:00 to 15:00), as outdoor temperatures are higher than in the early morning hours (activation of summer function if available).

Assuming an average scenario with 60 % Carnot efficiency, SPF values (heating and DHW) of 6.3 for AWHPs and 7.9 for BWHPs seem achievable by 2050 if the economic and political framework conditions are set in a direction favorable for highly efficient HPs. What is needed in the medium term are better control systems for HPs that simplify installation and HPs that optimize themselves according to the heat demands of the building and the customer needs.

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Nomenclature

AWHP	Air-to-water heat pump	
BWHP	Brine-to-water heat pump (geothermal heat pump system with vertical boreholes)	
COP	Coefficient of performance	[-]
DHW	Domestic hot water	
DHWHP	Domestic hot water heat pump	
E_{Co}	Electrical energy of the heat pump compressor	[kWh]
$E_{CP,Sink}$	Electrical energy of the circulating pump at the heat sink	[kWh]
E_{DHW}	Electrical energy of the heat pump in DHW operation	[kWh]
$E_{ext,HE}$	Electrical energy of the external heating elements	[kWh]
EnergieSchweiz	Federal authority on behalf of the Swiss Federal Office of Energy (SFOE)	
ERA	Energy reference area of the building	[m²]
E_{tot}	Electrical energy input of the entire heat pump system	[kWh]
FAWA	Field analysis of heat pump installations	
GWHP	Groundwater/water heat pump	
$HGT_{20,12}$	Heating degree days for 20 °C, 12 °C (heating limit) (new building)	[°C]
$HGT_{20,16}$	Heating degree days for 20 °C, 16 °C (heating limit) (renovation)	[°C]
HP	Heat pump	
HS	Heating season	
HUR	Heat utilization ratio according to the definition of SFOE	[-]
IES	Institute for Energy Systems (IES) at OST, Campus Buchs	
OST	Eastern Switzerland University of Applied Sciences	
Q_{DHW}	Thermal energy requirement for DHW	[kWh]
Q_{HD}	Specific heating demand	[kWh/m²]
Q_{SH}	Thermal energy requirement for space heating (SH)	[kWh]
SH	Space heating	
SPF	Seasonal performance factor according to EnergieSchweiz	[-]
SUR_{DHW}	System utilization ratio according to SFOE	[-]
$T_{source,SH}$	Average source inlet temperature in SH mode of the HP	[°C]
$T_{source, DHW}$	Average source inlet temperature in DHW mode of the HP	[°C]
$T_{supply,SH}$	Average supply outlet temperature in SH mode of the HP	[°C]
$T_{supply, DHW}$	Average supply outlet temperature in DHW mode of the HP	[°C]
THR	Total specific heat requirement (SH and DHW)	[kWh/m²]
WPZ	Heat pump test center (in Buchs SG, CH) (in German: Wärmepumpen Test Zentrum)	

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