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Efficiency and Hygiene in DHW Systems for MFH: Evaluating Heat Exchanger Arrangements and Charging Strategies with R290 Heat Pumps

Robert Haberl, Vera Gütle, Michel Y. Haller

OST-SPF, Oberseestr. 10, 8640 Rapperswil, Switzerland

Abstract

This study examines domestic hot water (DHW) preparation in multi-family dwellings using an R290 brine-water heat pump under realistic dynamic conditions, focusing on energy efficiency and hygiene compliance. Three system concepts were tested: Potable water storage tank with external heat exchanger (two charging strategies), buffer water tank with instantaneous flow water heater (IFWH) (two variants) and buffer water tank with internal heat exchanger for discharging. Charging strategy strongly influences efficiency. Multi-pass charging achieved the lowest heat pump temperature and highest performance factor, while single-pass charging required higher temperatures and reduced efficiency. Systems using buffer tanks with instantaneous flow heaters or internal heat exchangers for discharging showed lower performance due to higher storage temperatures in the lower part of the tanks and DHW circulation effects.

Swiss hygiene standards (≥55 °C in circulation return, ≥50 °C at taps) were met in all systems. Histogram analysis confirmed that potable water storage tanks offer strong Legionella protection because most water remains hot and incoming cold water remains below the critical range between 25 and 40 °C due to good stratification, while storage tanks with internal heat exchangers for discharging may have localized zones near this range. Buffer systems do not store potable water, so risk is confined to the circulation loop.

In conclusion, multi-pass external heat exchanger charging offers the highest efficiency, while temperature analysis does not show long periods of time with temperatures within the hygienically critical range and temperatures in the upper part are always sufficiently high. Optimized control strategies, inlet velocities lower than 0.1 m/s and DHW circulation management are essential to enhance stratification and to minimize electricity use of the heat pump.

For systems with ISWH, it has been shown that a separate heat exchanger for circulation may improve performance and safety, though economic feasibility must be considered.

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Keywords: heat pump; domestic hot water; potable water hot; efficiency; hygiene; multi-pass charging; single-pass charging

1. Introduction

The urgent need to decarbonize heat supply systems has positioned heat pumps (HPs) as a cornerstone technology for both space heating and domestic hot water (DHW) production. In 2000, HPs accounted for just 4.1% of primary heat generation in buildings in Switzerland. By 2021, this figure had risen to 17.0 %, and by 2024 it had already reached 23.0 %. The picture is similar for DHW preparation in particular. Here, a share of 16.8 % was achieved for heat pumps, with a strong upward trend [1]. The share of space heating in total energy consumption in private households was 64.5 % in 2020, down 17.6 % since 2000. The share of DHW preparation was 15.5 % in 2020, up 6.6 % since 2000 [2]. In modern, highly insulated buildings, DHW can account for 30–50 % of total heat demand, making its efficient provision increasingly critical. This challenge is amplified in multi-family dwellings (MFH) with circulation systems, where continuous flow and elevated storage temperatures lead to additional energy losses of up to 15–20 % compared to single-family systems.

Meeting hygiene requirements, particularly for Legionella prevention, imposes strict temperature constraints (⩾55 °C in circulation return, ⩾50 °C at taps) [3,4], which conflict with the low-temperature optimization of conventional HPs. Previous field studies report system performance factors between 1.8 and 2.2 for DHW applications, significantly below theoretical COP values [5,6]. Laboratory investigations confirm that elevated sink temperatures and circulation losses are primary drivers of this gap [7].

A wide range of system configurations exists for integrating HPs into DHW heating. One common approach involves using water directly as the storage medium, in which case heat is transferred from the HP to the storage via heat exchangers (HX), which may be integrated into the storage (IHX) or external of the storage. External heat exchangers allow for various charging strategies, including single-pass or multi-pass charging. In contrast, when buffer water (non-potable heating water) is used as the storage medium, heat is transferred from the HP to the storage without an additional heat exchanger, but instead heat must be transferred from the storage to the potable water via heat exchangers. Also in this case, these heat exchangers may be internal (IHX) or external of the storage. External heat exchangers for discharging can be configured as instantaneous flow water heater (IFWH) and may also include separate heat exchangers for DHW circulation loops.

Despite the diversity of available system architectures, the impact of these configurations on the overall efficiency of HPs remains insufficiently understood [8,9]. A systematic investigation is needed to assess how different DHW concepts influence the thermodynamic performance of HPs, particularly under real-world operating conditions. The present study addresses this gap within the framework of the EffPlusWW project. The project aims to evaluate the efficiency implications of various DHW system designs and identify optimization potentials for future implementations.

2. Methods/Approach

2.1. Test Setup

The measurements were carried out as realistically as possible in dynamic operation. For this purpose, an 18 kW, inverter-controlled brine-water HP (refrigerant: R290) was installed on the test bench and combined with various storage tanks one after the other. The brine-water HP was selected for several reasons. MFHs often require high heating capacities and stable performance under varying loads, which brine-water systems can provide more reliably than air-water HPs, particularly in the Swiss climate where winter temperatures of the ambient frequently fall below freezing. Air-water HPs are less suitable for such conditions and face additional restrictions in densely populated urban areas due to noise emissions and space limitations. R290 (propane) was chosen as the refrigerant because of its excellent thermodynamic properties, enabling high supply temperatures necessary to meet stringent hygiene requirements for Legionella prevention. In addition, R290 has a very low global warming potential (GWP), making it an environmentally responsible choice. To ensure operational safety when using a flammable refrigerant, the test bench was equipped with a dedicated ventilation system and appropriate leak detection measures. These precautions comply with current safety standards for hydrocarbon refrigerants in HP applications in Switzerland.

Fig. 1 shows a picture of a system installed on the test bench on the left and a schematic representation with system boundaries marked on the right. The hydraulics were adapted to the storage tank and the arrangement of the heat exchangers in combination with the respective charging or discharging concept in each case.

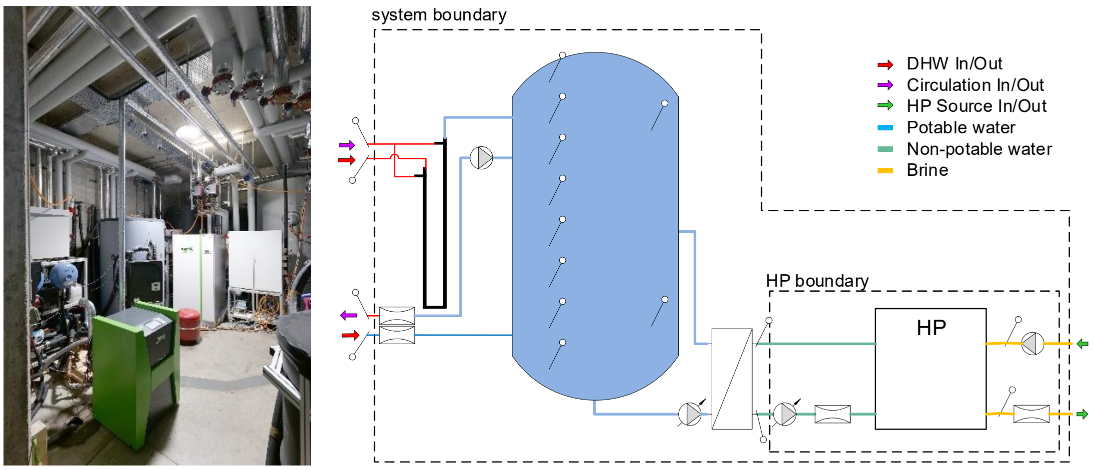


Figure 1: Left: Picture of one of the tested systems on the test bench. Right: Measuring concept and hydraulic scheme of system #1b. Everything inside the “system boundary” is installed as physical component on the test bench. The parts outside of the boundary are simulated and emulated.

The test bench emulates the environment of the system under test: geothermal probes as a heat source and a DHW draw-off profile with conditioning of the cold-water temperature and flow rate. A riser pipe, representing the pipe through which the water would be transported from the technical room to the consumer, was installed, emulating volume and heat losses of a real installation in the field, the thermal losses of the circulation pipe back to the tank were also emulated by the test bench.

A reference building was used to define the DHW demand, the capacity of the HP (suitable to meet also the building's space heating requirements) and to design the circulation pipes. The building is an apartment building with 6 apartments [10]. The total heat demand for DHW preparation was designed in accordance with the Swiss standard SIA 385/2, to do a proper dimensioning of the volume of the storage tanks. The heat demand is 70 kWh/d, consisting of the actual DHW demand, the discharge losses, the circulation losses and the thermal losses of the storage tank. The DHW demand at tap and the discharge losses of the single-feed pipes were combined and used as a target value for the tapping test profile.

The 24-hour tapping profile was generated using the dhwCalc software [11], post-processed and implemented into the test bench. Figure 2 shows the cumulative energy demand and the mass flow of the individual taps.

The boundary conditions for circulation were also defined in line with the tapping profile, considering specifications from the SIA 385/2 [12] regarding discharge times, etc., on the one hand, and field data on circulation losses [13] on the other. The circulation losses emulated by the test bench amount to 500 W or 12 kWh/d, which is in between the value resulting from calculations and measured values from real applications.

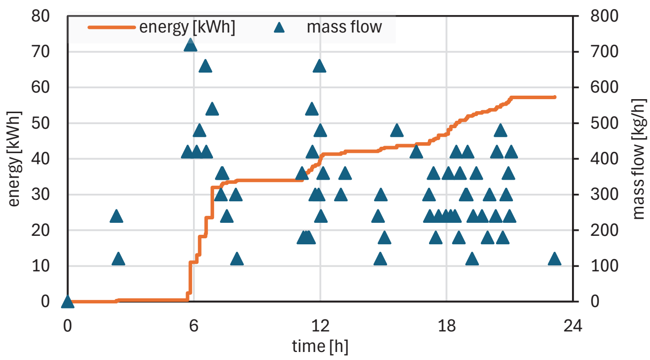


Figure 2: 24-h DHW profile.

A test cycle runs for 24 hours. The test cycle was repeated until there was only a negligible change of the energy stored in the storage tank between the beginning and end of a 24-hour cycle. To determine this change in the status of the storage, the average storage temperature at the beginning and end of the 24-hour cycle was determined using 8 to 9 contact sensors that were attached to the wall of the storage tank, below the insulation.

For all tests hygiene requirements specified in the Swiss standard SIA 385/1 are met. According to this standard, the temperature in the PWH (potable water hot) as well as of PWH-C (potable water hot circulation) must be at least 55 °C (return line) and may only fall below this level for a short time during charging of the storage or when large amounts of PWH are drawn. If a system did not meet this requirement, the test was repeated and the set temperature in the storage tank was increased until the condition was met. As shown in Figure 1, the plant was equipped with various measuring points that allow for a comprehensive qualitative and quantitative assessment.

The variables that are important for determining the heat transfer from one point to another (e.g. HP to storage or DHW tapping energy) are recorded using immersed temperature sensors in the flow and return pipes (inlet and outlet respectively) and a mass flow meter. These values are used to calculate the heating power ($\dot{Q}$) from the enthalpy difference between inlet and outlet ($h(\vartheta_{in/out})$) every second, which in turn is integrated or summed over time to form the thermal energy (Q):

$$\dot{Q} = \dot{m} \cdot \left[h(\vartheta_{in,i}) - h(\vartheta_{out,i})\right] \tag{1}$$

$$Q = \sum \dot{Q} \cdot \Delta t \tag{2}$$

The electrical power was measured using contacted electronical power measuring devices, with the pumps being recorded individually wherever possible.

The performance factor of the system (PF_{system}) provides information about the efficiency of the DHW system. It is calculated from the ratio of the useful energy Q_{DHW} delivered at the tap and the electrical energy required by the HP ($W_{el,HP}$). Here, $W_{el,HP}$ includes the auxiliary energy required for the primary and secondary sink pumps of the HP, the source pump, and the control system. The performance factor of the heat pump itself (PF_{HP}) is based on the same value for electricity demand in combination with the heat that is supplied from the heat pump to the storage.

$$PF_{system} = \frac{Q_{DHW}}{W_{el,HP}} \tag{3}$$

$$PF_{HP} = \frac{Q_{HP}}{W_{el,HP}} \tag{4}$$

Furthermore, weighted average temperatures were computed using power as a weighting factor, as demonstrated using the example of the flow temperature of a HP:

$$\overline{T_{HP,fl}} = \frac{\Sigma(T_{HP,fl,i} * \dot{Q}_{HP,i})}{\Sigma\, \dot{Q}_{HP,i}} \tag{5}$$

The above-mentioned contact sensors on the storage tank wall are used to visualize the processes in the storage tank and to determine the energy status at the beginning and end of a test day. In addition, zones in the storage tank were defined in accordance with Figure 2 for further evaluation of the measurement data.

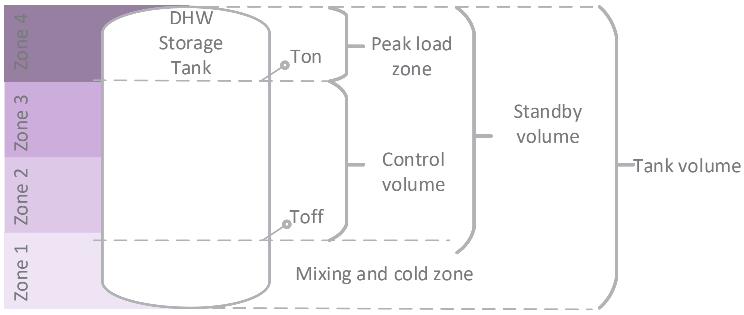


Figure 3: Zones in the hot water tank. On the left, a division into uniform zones for evaluation; on the right, a functional description of the main zones.

2.2. Tested Systems

Inlets of the storage tanks were designed in a way that the velocity of water entering the storage was always below 0.1 m/s at the point of release into the bulk volume of the storage in order to avoid inlet jet mixing and achieve the best-possible thermal stratification. For this purpose, diffusors and devices for deviating flows and calming down turbulences were used inside the storage tank.

2.2.1. Overview

This article presents the results of five different measurements on three different system types. The systems tested are shown below. Buffer water tanks with internal HX for charging were considered as less relevant in MFHs applications and were therefor excluded in this study.

2.2.2. System #1: Potable water storage tank with external heat exchanger for charging

Measurements are shown for two different charging concepts: single-pass charging with throttle valve and multi-pass charging.

#1a: With single-pass charging, the HP's flow is regulated directly to the target temperature. The heated water enters directly into the peak coverage volume of the storage tank. Since the modulation range of a conventional pump is not large enough to achieve the necessary range in the volume flow, a throttle valve was used on the secondary side of the heat exchanger to reduce the mass flow to very low values from start.

#1b: In multi-pass charging, charging takes place with a constant temperature difference of 5-10 K between the supply and return of the primary and secondary side of the heat exchanger in combination with a large volume flow rate. The storage tank inlet is located below the peak coverage volume in the control volume.

2.2.3. System #2: Buffer water tank with external HX for discharging

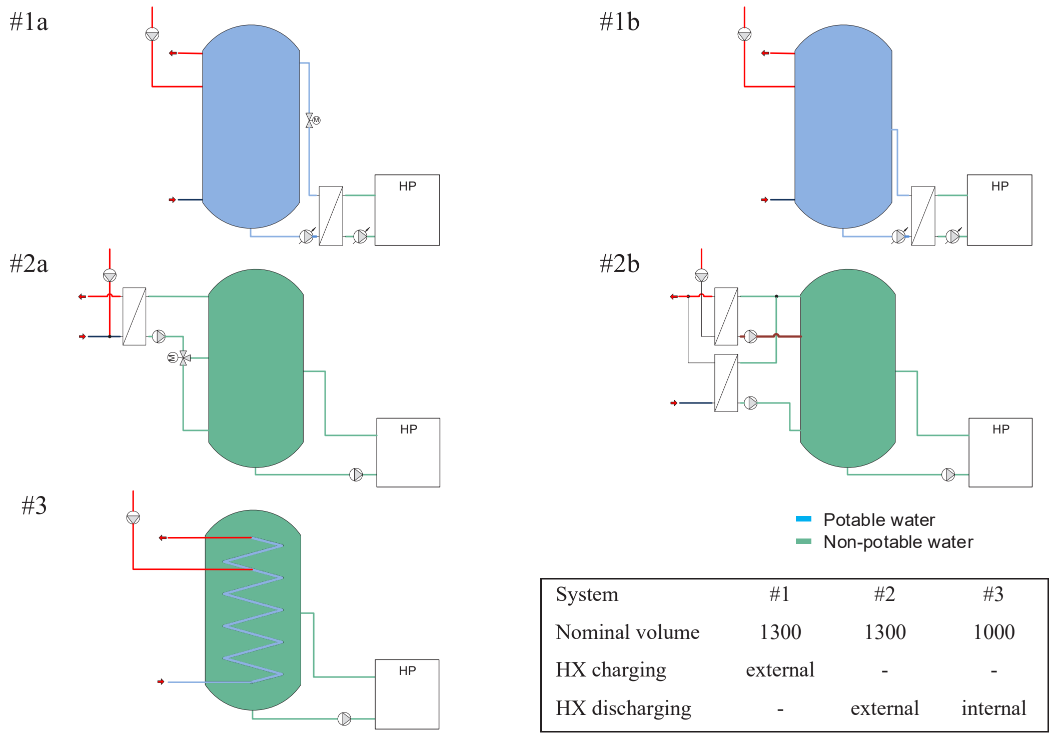
Two different concepts of buffer tanks in combination with IFWH were tested:

In system #2a, a single IFWH is used to heat both PWH and PWH-C. The boundary conditions for the two operating modes differ significantly: For PWH, a large secondary-side temperature difference from inlet to outlet must be maintained while ensuring a high volume flow rate. In contrast, heating PWH-C operates with both a low flow rate and a small temperature difference. Consequently, the return temperature on the primary side of the IFWH is considerably higher during circulation than during DHW consumption. To address this, a switching valve is installed in the return line, directing the cooler return flow to the bottom of the storage tank during DHW consumption.

System #2b is also an instantaneous water heating concept with buffer tank. However, in this system, two different heat exchangers are used for heating water during tapping and when only PWH-C is to be heated, respectively. For the heat exchanger that serves tapping, the primary side return temperature is usually very low, close to potable water cold (PWC), and therefore the primary return flow of this heat exchanger is directed into the bottom of the storage tank. The primary return temperature of the heat exchanger that only covers circulation losses on the other hand is expected to be higher than the 55 °C of the PWH-C return line and is therefore directed into the storage tank just below the peak coverage volume.

2.2.4. System #3: Buffer water tank with IHX for discharging

This storage tank contains mainly buffer water and is charged directly via the HP. PWH is heated via an internal heat exchanger, which is designed as a spiral pipe and heats the water using the flow-through principle. An additional connection to the upper part of the spiral pipe in the heat storage tank is used to return the circulation (PWH-C) to the storage tank.



System	#1	#2	#3
Nominal volume	1300	1300	1000
HX charging	external	-	-
HX discharging	-	external	internal

Figure 4: Sketches of the hydraulic integration of the storage concepts examined. System #1: Potable water storage tank with external heat exchanger for charging; System #2: Buffer water tank with external HX for discharging; System #3: Buffer water tank with IHX for discharging. Drawn in blue: Potable water, drawn in green: Non-potable water.

3. Results

The measurement results are summarized in table 1. The coefficients of performance of the HP and of the entire system are shown in Figure 5. The graph also shows the average flow and return temperatures of the HP and the average return temperature of the circulation. The highest coefficients of performance were achieved in test #1b where the average return temperature of the HP was the lowest. The average flow temperature of the HP was significantly lower in this test than in all other measurements.

The difference between measurements #1a and #1b lies in the charging strategy, with identical components. Multi-pass charging with external HX delivers significantly better results than single-pass charging. The measurements in systems #2 and #3 were also carried out with multi-pass charging but show lower performance factors due to the discharging concepts that do not lead to equally low temperatures in the storage.

Figure 7 shows the energy-temperature diagrams of the measured variants. In energy-temperature diagrams, the thermal energy is plotted over temperature. The heat supplied by the HP is shown with supply, return and mean temperatures. Here “mean temperature” is referring to the mean temperature between supply and return.

Table 1. Results of the measurements.

Test		#1a	#1b	#2a	#2b	#3
HP: DHW set-temp.	[°C]	62	62	65	63	65
HP: Hysteresis	[K]	6	6	6	6	6
IFWH: set temp. PWH / PWH-C	[°C]	-	-	60/60	58/58	-
PWH	[kWh]	57.0	57.3	57.3	57.3	57.3
PWH-C	[kWh]	12.2	12.0	11.5	11.5	12.7
Heat losses storage + hydraulics	[kWh]	15.7	13.5	11.9	8.7	8.5
Heat supply	[kWh]	85.3	83.0	80.8	77.1	78.4
El. Energy HP	[kWh]	26.3	21.4	28.4	24.2	27.6
El. Energy hydr. Pumps	[kWh]	2.0	2.6	2.0	1.9	2.0
Operating time HP	[hh:mm]	5:28	4:16	5:13	4:48	5:28
Mean storage temperature	[°C]	53.6	56.2	60.4	58.6	59.5
Mean return temp. PWH-C	[°C]	58.1	56.7	56.4	55.1	59.6
Mean flow temp. HP (weighted)	[°C]	63.1	48.1	60.1	55.1	58.8
Mean temp. HP (weighted)	[°C]	54.9	45.9	57.8	52.6	56.6
PF_{HP}	[-]	3.0	3.5	2.7	2.9	2.6
PF_{system}	[-]	2.0	2.4	1.9	2.1	1.9

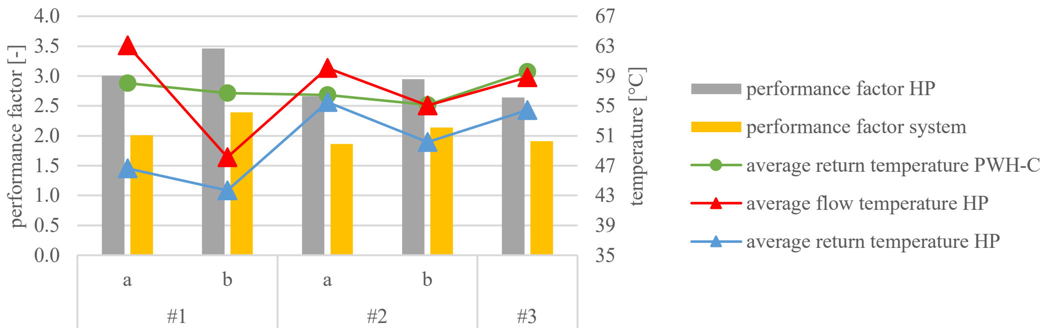


Figure 5: Performance Factors and power weighted temperatures of the 24-h test sequences for each system.

Since the efficiency of the HP is highest when there is a small difference between the source and the sink and hence also between the evaporation and condensation temperatures of the working fluid, it is advantageous to keep the temperatures of the heat supply as low as possible.

Figure 6 shows the temperature profile of the storage tanks during the 24-hour test cycle.

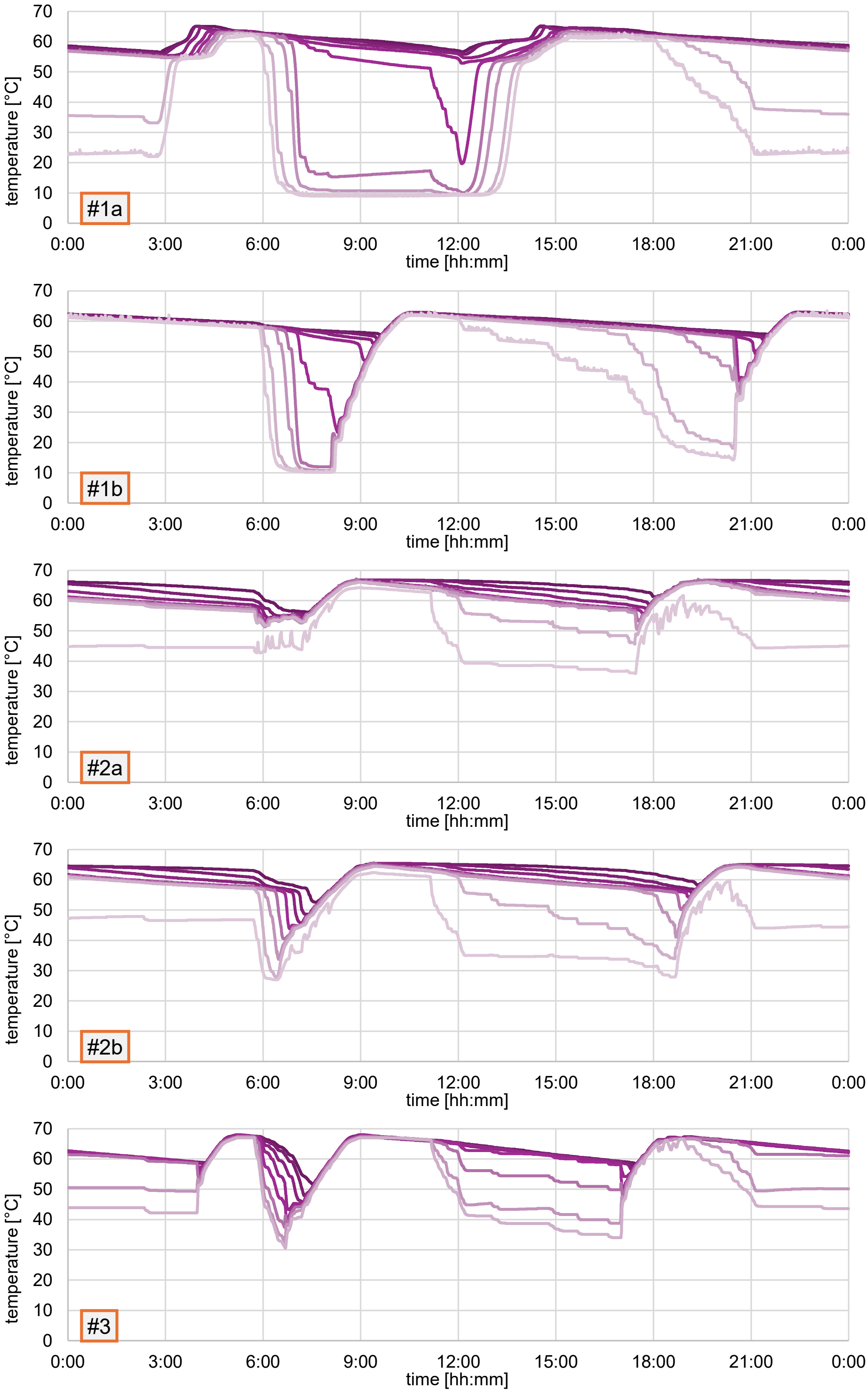


Figure 6: DHW storage tank temperature over the 24 h test cycle, measured with 8-9 contact sensors along the storage tank wall.

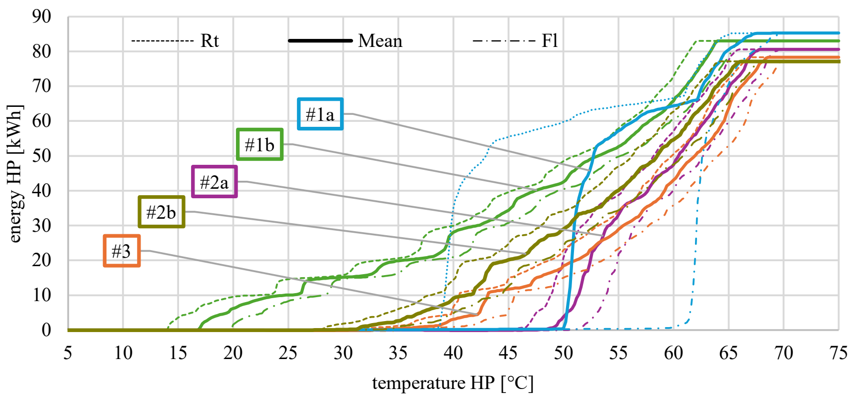


Figure 7: Energy-temperature diagram of the measurements. The thermal energy is sorted by the temperature at which it was delivered. The solid line represents the thermal energy sorted by the mean temperature of the heat supply via HP, the dotted line the energy sorted by the return temperature (Rt) and the dash-dotted line by the flow temperature (Fl).

4. Discussion

4.1. Charging of the storage tank

As described in section 2.2.2, in single-pass charging the HP supply temperature is regulated to the required target temperature from the start of charging, whereas in multi-pass charging, the temperature difference between the supply and return remains constantly low. Figure 8 shows the temperature curve for a single-pass and for a multi-pass charging process. The primary side temperatures and mass flow rate of the HP as well as the temperatures at the top and bottom of the storage are shown.

Under ideal single-pass charging conditions, water is entering the heat pump with a low temperature (return) and is then heated by the HP to a supply temperature that is high enough for the secondary side outlet temperature of the heat exchanger to meet the desired DHW temperature. This regime is maintained over most of the duration of the charging and is characterized by a large temperature differential between the return and the flow. As described above, this requires a very low secondary mass flow rate over the storage. The measured data shows that this ideal is not achieved. The primary-side (HP) return temperature of the single-pass charging with throttle control was above 40°C for most of the charging process. The average temperature of the HP, weighted by power, was 54.9 °C.

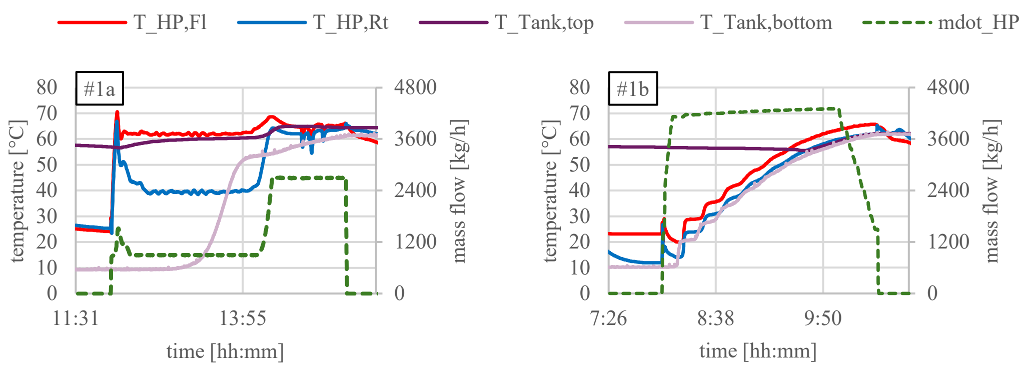


Figure 8: Temperature curves for one charging process each from measurements of system #1a and #1b. The primary side (before the HX) flow (Fl) and return (Rt) temperatures and mass flow (mdot) of the HP and the temperatures at the top and bottom of the storage are shown.

With multi-pass charging, a significantly lower average temperature of 45.9 °C was achieved for the HP supply and return. The low temperature in the storage tank (10 °C at the start of the charging process) was transferred efficiently to the HP via the heat exchanger over the first "pass" of the multi-pass charging, leading to an inlet temperature of the heat pump between 14 and 17 °C (Figure 6). The storage tank temperature at the top shows that the temperature remained consistently high throughout the entire charging process. The challenge of multi-pass charging does not lie in the control of the refrigerant circuit or the inlet and outlet temperatures of the water to be heated using variable flow rates, throttles or mixing valves. Instead, it lies in preventing inlet jet mixing of water that has been heated with a small delta-T and therefore enters the storage tank with a high flow rate. It is crucial that the high momentum of the incoming flow is considered and mitigated to a flow velocity of < 0.1 m/s by corresponding design of inlets.

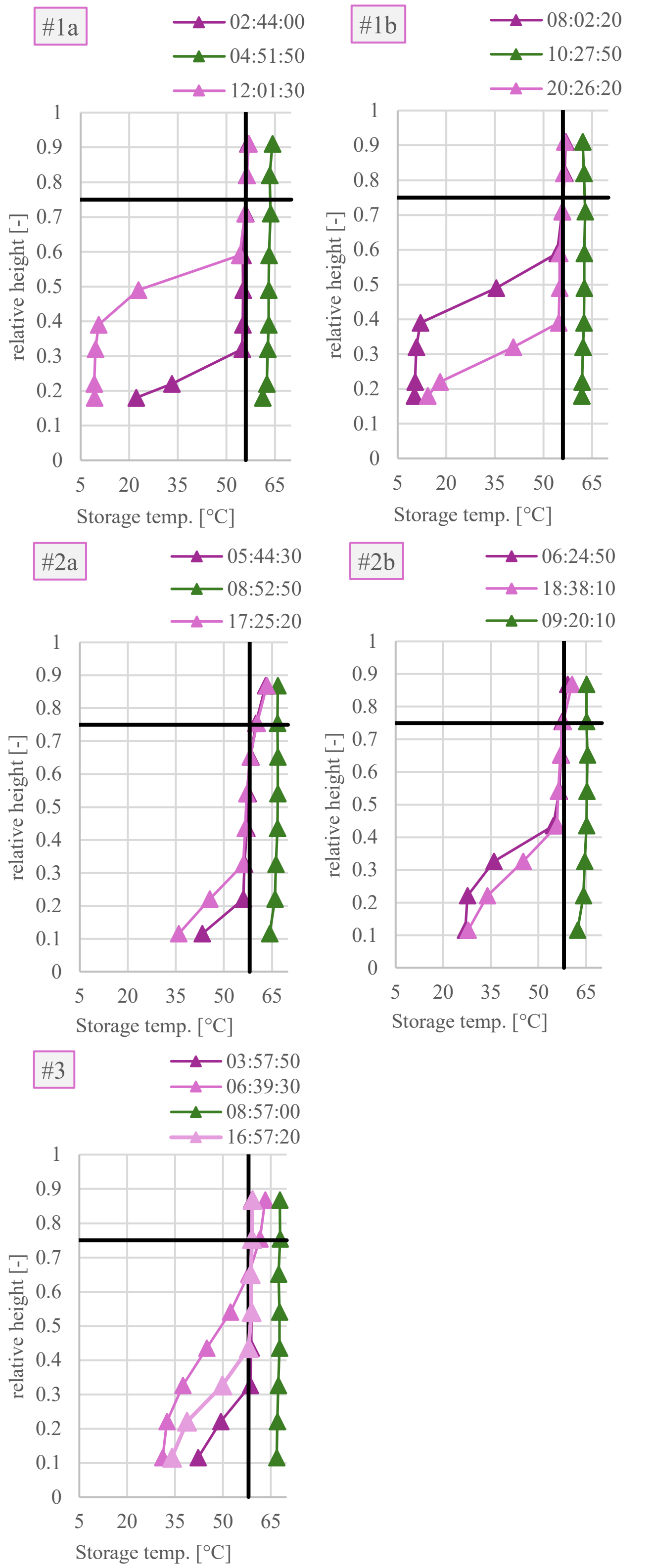


Figure 9: Storage temperature at the points in time when recharging was triggered (purple), as well as at the end of one of the charging cycles (green).

This task can be effectively managed by redirecting and expanding the flow cross-section at the entry point to the storage volume, with a sufficiently long stabilization section, as shown in projects StorEx [14] and BigStrat [15,16], as well as the guideline on the SPF homepage). These recommendations have been followed in the tested storage tanks.

Measurements #2a, #2b, and #3 with buffer tanks were only performed with a multi-pass charging concept. However, unlike the measurements with the potable water storage tank, the temperatures in the lower part of the storage tank were significantly higher in all three cases.

4.2. Discharging of the storage tank

4.2.1. Potable water hot

Hot water is drawn directly from the storage tank in system #1, i.e. without a heat exchanger. Cold water inlet is stratified into the bottom of the storage tank. If the stratification of the cold water is implemented with the aid of flow calming, a piston flow is created throughout the entire storage tank with a sharp separation between the cold water at the bottom and the hot water above it. In tests on system #1, a sharp thermocline formed with hardly any mixing zone. The storage tank had a high exegetic efficiency (based on a target temperature of at least 50 °C at the point of water draw-off). The presence of the cold-water reservoir in the storage tank below is a prerequisite for efficient operation of the HP.

In systems #2 and #3, domestic hot water was heated using heat exchangers that transfer heat from a buffer storage to the DHW in a flow-through principle. When heating with IFWH, the primary side temperature depends on the flow control. For the IFWH to be able to control temperature efficiently, the temperature in the storage tank must be higher than the setpoint on the secondary side of the HX. Otherwise, the frequency of the primary side pump of the IFWH may – depending on PI(D) control settings – increase to its maximum. In measurement #2b, the storage tank temperatures indicate that the return temperature must be lower than 28 °C. The same IFWH was used in measurement #2a, However, the temperatures in the mixing and cold zone of the storage were significantly higher here (compare Figure 6). The reason for the higher temperatures in the lower part of the storage tank in test #2a compared to #2b is discussed in the next section.

The PWH in System #3 is heated using an IHX. During phases without hot water consumption, the water in the IHX is at the same temperature as the surrounding non-potable water. When hot water is drawn, this water is expelled and DHW is heated based on flow-through principle. The more the lower part of the storage tank has cooled down, the more heat must be transferred to the PWH in the upper part of the storage tank, thus forcing cooling of the storage at the top. To ensure hygiene requirements with this system, recharging must therefore take place earlier than in the other systems.

4.2.2. Impact of potable water hot circulation

In all systems tested, an influence of PWH-C on system behaviour was observed. The "mechanism" that influences the recharge sequence varies depending on the system concept. Figure 9 shows the storage tank temperature at the points in time at which charging was triggered and at the end of one of the charging cycles.

During discharging of system #1, cold water is layered at the bottom and hot water is drawn off at the top. Recharging starts when the upwards moving thermocline reaches the position of the upper temperature sensor. Therefore, temperature in the top of the tank should remain high during discharge. Cooling of the upper volume occurs both through heat loss to the environment and through PWH-C. In the best case, the return flow of the circulation is directed below the peak coverage volume into the storage tank. It initially descends into the control volume because just after charging its temperature is lower than the temperature of the volume between the two sensors. This effect is observed even when the temperature difference between the circulation return flow and the tank water at the inlet position is negligible.

As shown in Figure 9, two different charging events happen within 24 h in both variants of system #1. Only one of the two charging events was triggered at a time when the lower part of the storage tank had cooled down. With time, the upper storage tank volume cools down homogeneously in each case. A distinct thermocline formed by the circulation return flow cannot be observed due to the small temperature difference between the fluid entering the tank and the temperature in the tank. The circulation was operated with a temperature difference of 3 K between supply and return. At least once a day, this cooling due to the circulation leads to premature charging of the storage tank when the lower part is not cold yet. Since the average temperature of the storage and hence also the supply and return temperatures of the HP are significantly higher in this case, this charging is also quite less efficient. Further charging takes place at a point in time at which the spatial distance between the “cold water thermocline” and the switch-on sensor is at least 15 % of the storage tank volume. The storage tank temperatures measured at the sensors with the relative height of 0.59 and of 0.71 are almost identical. Therefore, the pronounced cooling in the upper part of the storage tank caused an early recharging, which was not caused by the DHW tapping.

In System #2, circulation also has a significant influence on the overall system. Chapter 2.2.3 described that there are different primary return temperatures for the IFWH, depending on whether it is operating for PWH or PWH-C. The measurements show that the use of a separate heat exchanger for PWH-C results in significantly lower storage temperatures and, as a result, higher efficiency of the storage charging by the HP. Under the selected boundary conditions in the test, this solution would not be the ideal economic solution, as the efficiency gains, and thus the reduction of operating costs, do not outweigh the additional investment for such a small system with only six apartments. For larger systems with cascaded IFWH, the solution would make sense not only from an energy perspective, but also from an economic perspective.

4.3. Hygiene

When discussing hygiene in DHW systems in MFHs, the main concern are usually Legionella bacteria. Legionellae are facultative pathogenic bacteria that can cause Legionnaires' disease under certain conditions. Infection occurs via the lungs through the inhalation of aerosols or fine water droplets. According to current knowledge, drinking of water that is contaminated with Legionella is not generally harmful to health. The optimal temperature range for the growth of Legionella is between 25 °C and 45 °C. Above this range, reproduction is inhibited. At around 50 °C, the bacteria begin to die off, and at temperatures of 60 °C and above, Legionella bacteria are usually reliably killed.

No drinking water is stored in systems #2a and #2b. Compliance with the maximum permissible return temperature of the PWH-C (55 °C) and a temperature of at least 50 °C at the tap are in-line with current recommendations of Swiss standards SIA 385/1 to prevent the growth of Legionella bacteria in the warm part of the pipe system. In systems #1a and #1b, drinking water is stored, while in system #3, the drinking water is in the integrated heat exchanger. In these systems, the water can theoretically remain in the critical temperature range over longer time periods.

Figure 10 shows a histogram of the storage temperatures in zones 1 to 4 (compare Figure 3) from measurements #1a, #1b, and #3. Hygiene requirements set by Swiss standards were met in all systems, but the risk profile differs. The potable water storage tank (System #1) provides a strong safety barrier against Legionella because the stored potable water is maintained at high temperatures throughout most of the volume, effectively preventing bacterial growth.

In the storage tank with IHX for discharging, the lower zone remains in the critical temperature range for the longest time. In addition, only a few liters of water need to be expelled from inside the heat exchanger before the water from the lower part of the heat exchanger rises to the top. This means that the time that this water remains at high temperature after being heated up is significantly shorter compared to system concepts #1, thus leaving only limited time for killing eventual bacteria loads from the lower part.

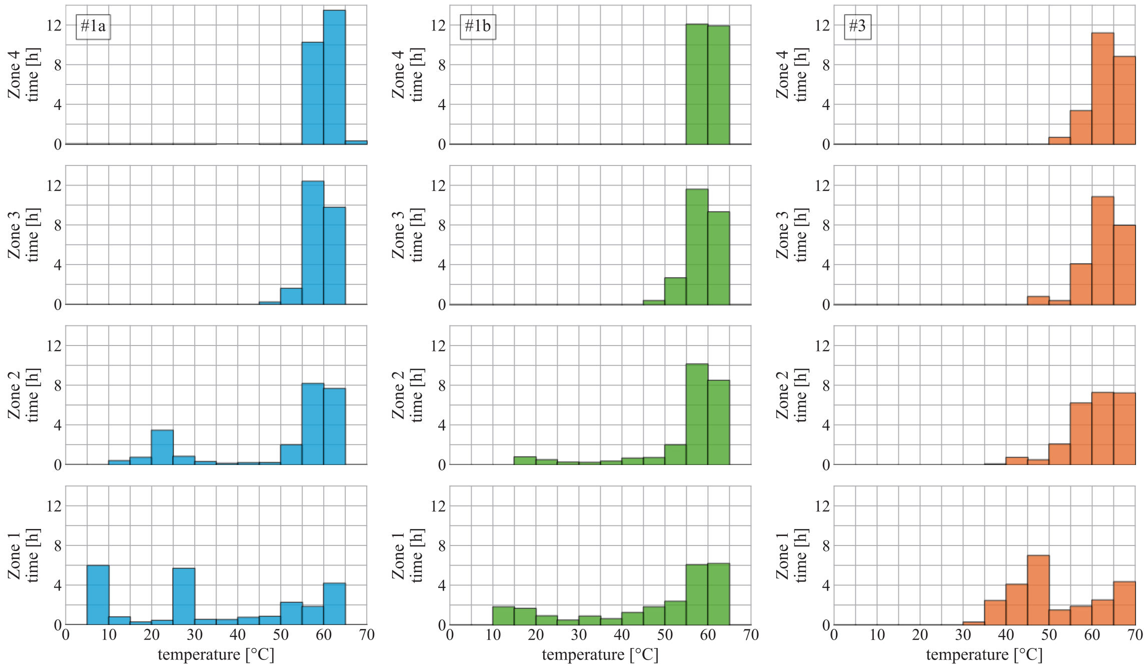


Figure 10: Histogram of the temperature in storage zones 1 to 4 of measurements #1a, #1b, and #3 (left to right).

5. Conclusions

This study investigates DHW preparation in MFHs using an R290 brine-water HP under realistic conditions, focusing on energy efficiency and Legionella prevention. Three concepts were tested: a potable water tank with external heat exchanger (two charging strategies), a buffer tank with instantaneous flow heater (two variants), and a buffer tank with internal heat exchanger for discharging, respectively DHW preparation.

Multi-pass charging clearly outperforms single-pass charging in terms of HP efficiency, as it reduces average supply temperatures of the heat pump and improves the coefficient of performance. System design and heat exchanger configuration strongly affect both energy performance and hygiene compliance. Potable water storage with multi-pass external heat exchanger charging offers a good balance between efficiency and safety. DHW circulation loops, however, cause premature recharging in all tested systems and thus efficiency losses, making optimized return positioning and good control strategies essential.

Multi-pass charging (System #1b) achieved a mean HP temperature (average of flow and return) of 45.9 °C and the highest performance factor, while in single-pass charging (System #1a) the condenser operated at 54.9 °C on average, and thus with lower efficiency.

Systems using instantaneous heaters or IHX for DHW preparation showed lower performance factors due to higher storage temperatures (>55 °C) and the effects of the circulation return flow.

Swiss hygiene standards ($\geqslant$55 °C in circulation return, $\geqslant$50 °C at taps) were met in all cases. The measurements indicate that potable water tanks offer better Legionella protection, though tanks with IHX showed occasional exposure to the critical 25–45 °C range.

For systems with ISWH, it has been shown that a separate heat exchanger for circulation may improve performance and safety, though economic feasibility must be considered.

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