DHW Tanks – 40 % Savings by Better Stratification

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Abstract

Thermal stratification of hot water storage is known to have a positive impact on the efficiency of systems for heat supply. An existing method for determining the thermal stratification of combi-storage tanks, based on the second law of thermodynamics, i.e. on the entropy balance of the storage system, was adapted for measuring domestic hot water storage tanks. The updated method uses tapping profiles from EN 16147 to test the stratification efficiency of different sized storage tanks, matching the size of the collector array to the tapping profiles when using solar thermal systems. The resulting test procedure was applied to several tanks and the results were evaluated. It is shown that the efficiency of the storage tanks and their influence on the operation mode of the charging varies considerably. A very well stratified storage tank concept not only achieved higher comfort and created better hygienic conditions, but in combination with a heat pump saved up to 40 % electrical energy compared to another storage tank. These results underline the need for an evaluation of the stratification efficiency of domestic hot water storage tanks, which can be done with the method presented here.

Keywords: Domestic Hot Water, Stratification efficiency, Measurement, Hardware-in-the-Loop, Dynamic operation, Entropy production, Exergetic efficiency

1. Introduction

Stratified water storage tanks are used for storing heat as hot water. When this kind of storage is used in combination with a heat pump, the temperature stratification of the storage is a decisive factor for the overall efficiency and thus for the consumed end energy of the system (Haller et al., 2019).

The temperature stratification of the water storage tanks is automatically established due to gravity and the temperature-dependent density of the water. However, this natural process is counteracted by processes, which can essentially be attributed to three causes:

- Heat conduction and diffusion in the water and in the storage tank internals.
- Plume entrainment caused by buoyant fluid movements.
- Mixing of the inlet jet due to kinetic energy and turbulence during direct storage charging.

Since mixing of fluids with different temperatures (de-stratification) and heat exchanging processes of any kind - and thus all of the effects described above - always result in entropy production (exergy loss), stratification indices can be determined based on the measurement and calculation of entropy and exergy balances.

(Haller et al., 2018) introduced a test method for the evaluation of stratification efficiency of combined storage tanks that store heat for domestic hot water and for space heating in one thermally stratified water storage tank. This method for the determination of stratification efficiency as a key performance indicator is based on the second law of thermodynamics, i.e. on the entropy balance of the storage system. The corresponding laboratory test is based on a 24 hour cycle where realistic and dynamic charging and discharging is applied according to boundary conditions of real weather data of a day in the year. The particular day of the year was chosen because of its representativeness for the effect of stratification on the system performance, such that the annual performance can be derived directly from the performance shown in the 24 h test cycle. The test method was developed in a research project funded by the Swiss Federal Office of Energy entitled "StorEx" (Haller et al., 2015).

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The results showed that the new KPI correlates well with the electricity demand of a heat pump that is charging the storage. While the stratification efficiency was the decisive factor for the energetic performance of the system, i.e. for the electric energy consumption of the heat pump, the determined heat losses of the storage system showed no correlation with the overall energetic performance of the system. However, no information has been available so far for similar kinds of tests for pure DHW storage tanks. Therefore, in this contribution, the existing method is extended to the evaluation of storage tanks for domestic hot water (without space heating), and results are presented for four DHW storage systems.

2. Test Method

2.1 Procedure

The general procedure for measuring the stratification efficiency of combistores is as follows:

- The storage tank is installed on the test bench, including the necessary hydraulics for charging and discharging.
- The test bench simulates and emulates a complete building during the 24-h test cycle, the heat demand of which must be covered. Based on the (real) temperatures measured in the storage tank and the specified test cycle, charging and discharging are started and controlled.
- Based on the measurements, the entropy balance of the storage system is determined.
- The stratification efficiency is determined based on a comparison of the measured entropy production of the storage system with the hypothetical entropy production of a completely mixed reference system.

This general procedure was adopted unchanged for DHW storage tanks. Of course, the emulation of space heating is in this case omitted. Furthermore, a new irradiance profile for the solar thermal collectors was defined and simplifications were made in the emulation of the auxiliary heating.

2.2 Test setup and boundary conditions

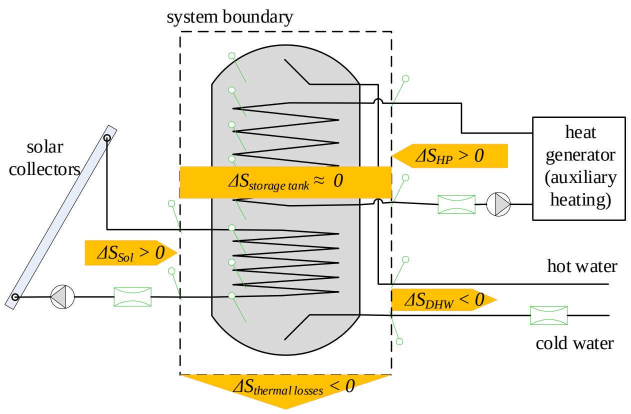

Fig. 1: System boundary for the measurement of stratification efficiency of DHW storage tanks, including the necessary measurement equipment and the entropy terms, which are to be determined from the measured data.

System boundaries

DHW storage systems usually involve heat exchangers for the transfer of heat to the fresh water. These heat exchangers may influence stratification and the heat transfer process as such is a source of entropy generation. Entropy may also be produced in external valves of the overall hydraulics where hot and cold fluids are mixed. That's why a complete storage system is evaluated instead of the bare storage tank.

Fig. 1 shows the system boundaries, using the example of a storage tank with two internal heat exchangers: one each for heat input from solar thermal and from a heat pump.

The parts outside the system boundaries, i.e. the collectors, the heat pump or other heat generators as well as a tapping profile, are simulated and emulated by the test bench.

DHW tapping profile

In the standard EN 16147: 2017 (European Committee for Standardization (CEN), 2017) 24-h load profiles are defined for the DHW demand. The energetic target value of the profiles varies from 0.345 kWh up to 93.52 kWh for one day, the highest target flow rate of a single tapping varies from 2 l/min up to 96 l/min.

The profiles contain the time for the taps, the target flow rate, the usable water temperature and the maximum temperature (minimum water temperature to be reached during a draw-off), whereby the usable temperature lies in the range between 10 °C and 40 °C and the maximum temperature of 55 °C is only defined on a few taps per day. The set point of the incoming cold water is defined as 10 °C.

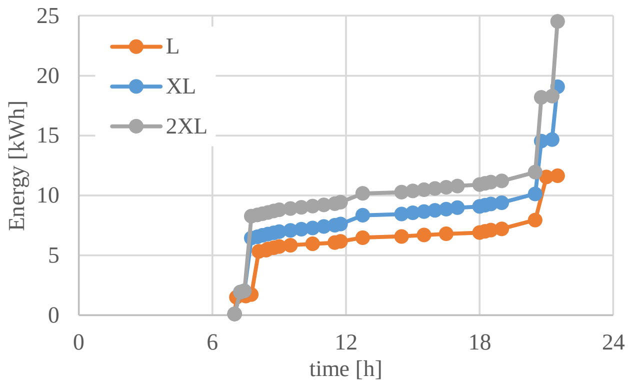

Fig. 2: Cumulated energy from the tapping profiles L, XL and 2XL.

For the measurement of stratification efficiency of DHW tanks, one of the profiles can be selected (depending on the volume/output capacity of the tested tank), with profile L with 11,655 kWh per day as the standard profile. During the measurement, according to the time and the selected tapping profile, the cold water is conditioned to a temperature of 10 °C at the required mass flow. From the difference with the outgoing hot water, the heat output is calculated, which is accumulated to the tapped energy from the moment the usable water temperature is reached. Tapping is stopped when the energy set point is reached.

Heat generator

In contrast to the predefined tapping profile, charging must be flexible. Two different heat sources can be used to charge the tested storage tank: Solar thermal collectors and a heat pump, respectively an auxiliary heater.

Tab. 1: Collector field characteristics.

η_0	a_1 [$Wm^{-1}K^{-1}$]	a_2 [$Wm^{-2}K^{-2}$]	$c_{eff,spec}$ [$Jm^{-2}K^{-1}$]
0.8	3.5	0.015	7000

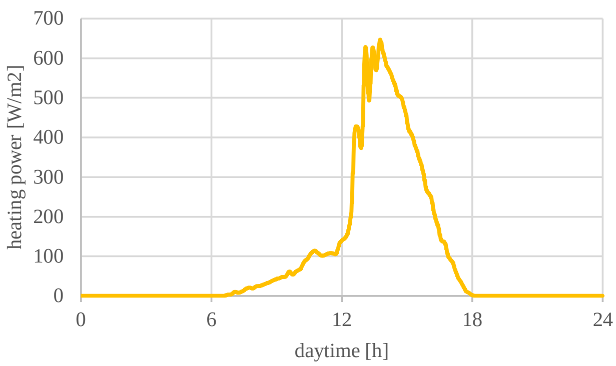

Fig. 3: Generated heat production per m² collector field before subtraction of thermal losses on the test day.

For the simulation and emulation of the solar yield, the standard performance data for a flat plate collector from the IEA SHC Task 32 (Heimrath and Haller, 2007) are used, which were already applied in the StorEx project. The performance characteristics can be seen in Tab. 1. A south orientation with 45° inclination is assumed.

The irradiance on the collector field ($I = I_b + I_d$) was converted to heat produced within the collector ($\dot{Q}_{rad}$) before subtraction of heat losses according to (Eq. 1). The resulting profile for heat generated within the collector according to the irradiance and the incident angle modifiers is presented in Fig. 3.

Tab. 2: Aperture area of the emulated collector array for measurement with different DHW tap profiles.

tapping profile [-]	Aperture area [m^2]
L	5.0
XL	8.2
2XL	10.5

In the simulation model used for the emulation of the solar yield, only the combination of the operating temperature and the power loss to the environment must then be calculated. The difference between the collector temperature and the storage temperature is used as a criterion for operating the collector circuit pump. The size of the collector array depends on the selected DHW tap profile of the test. Tab. 2 shows the aperture area for the different tapping profiles.

$$\dot{Q}_{rad} = A_{sol} \cdot (\eta_0 \cdot K_b \cdot I_b + \eta_0 \cdot K_d \cdot I_d) \quad \textbf{(Eq. 1)}$$

The simulation and emulation of an auxiliary heater is done with reference to a heat pump, meaning with a rather small temperature difference of 5 K. As a simplification, this temperature difference is assumed to be constant over the whole outlet-temperature range, which is limited to maximum 60 °C. The heat pump was chosen as the reference heat generator because, on the one hand, it reacts most sensitively to poor storage stratification and, on the other hand, it poses great challenges to the storage system due to the usually high mass flows at a small delta-T. Moreover, within the framework of Switzerland's energy strategy, heat pumps are considered to play a far more important role in water heating than combustion heat generators.

The temperature positions in the storage tank and on/off hysteresis for the pump are chosen by the manufacturer or commissioner of the storage system.

2.3 Evaluation of test results

Measured values and data acquisition

Temperatures and volume flow rates are determined at the boundary between the tested storage tank and the test rig. The positions are marked in green in Fig. 1. In addition, the temperature of the storage tank is measured with eight contact sensors placed equidistant on the storage wall.

At the limiting boundaries, the following quantities have to be determined based on appropriate measurement devices:

$$\dot{Q} = \dot{m} \cdot [h(\vartheta_{in}) - h(\vartheta_{out})] \qquad \textbf{(Eq. 2)}$$

$$Q = \sum_i \dot{Q}_i \cdot \Delta t \qquad \textbf{(Eq. 3)}$$

$$\dot{S} = \dot{m} \cdot [s(\vartheta_{in}) - s(\vartheta_{out})] \qquad \textbf{(Eq. 4)}$$

$$\Delta S = \sum_i \dot{S}_i \cdot \Delta t \qquad \textbf{(Eq. 5)}$$

Entropy balance

With positive values for heat input into the system and negative values representing heat outputs (or heat losses), the entropy balance, meaning the irreversible entropy production inside the storage system, can be calculated as:

$$\Delta S_{irr,exp} = -\left(\Delta S_{aux} + \Delta S_{sol} + \Delta S_{DHW} + \Delta S_{loss} + \Delta S_{storage\ change}\right) \qquad \textbf{(Eq. 6)}$$

With the measurements and calculations described above at the measuring points at the system boundaries, the following terms can be determined: ΔS_{aux}, ΔS_{sol} and ΔS_{DHW}.

To calculate the entropy change due to the storage losses as well as the storage change, the losses themselves and also the storage temperature must be known. However, this temperature is neither constant over time nor over position. Therefore, the mean value over the height of the storage tank at the beginning and end of the 24-h test sequence $\vartheta_{TES,start}$ and $\vartheta_{TES,end}$ is calculated by means of the contact sensors on the storage tank, as well as the mean value over time $\overline{T_{TES,24h}}$. The calculation of the entropy is then done as:

$$\Delta S_{loss} = \frac{Q_{loss}}{\overline{T_{TES,24h}} + 273.15K} \qquad \textbf{(Eq. 7)}$$

$$\Delta S_{storage\ change} = m_{TES} \cdot \left[s\left(\vartheta_{TES,end}\right) - s\left(\vartheta_{TES,start}\right)\right] \qquad \textbf{(Eq. 8)}$$

Stratification efficiency

From the entropy production measured during the test cycle, the stratification efficiency is determined as a dimensionless quantity. For this purpose, the measured entropy production is set in relation to the entropy production of a completely mixed storage tank:

$$\zeta_{str} = 1 - \frac{\Delta S_{irr,ms}}{\Delta S_{irr,mix}} \qquad \textbf{(Eq. 9)}$$

The entropy production of the completely mixed system depends on assumptions, such as the supply temperature of the heat generator and also on the choice of the tapping profile used. Effectively, the entropy balance is simply the difference between charging and discharging:

$$\Delta S_{irr,mix} = \Delta S_{WW} - S_{60°C} \quad \textbf{(Eq. 10)}$$

For the charging of the completely mixed storage tank, it is assumed that the heat supply always corresponds to the highest temperature required to reach the set temperature of the storage tank, for which 60 °C is used in this case based on the specifications of SIA 385/1:2020. The amount of heat to be supplied corresponds to the load, i.e. the heat demand for hot water. In reality, the thermal losses of the storage tank must also be covered. However, since the entropy losses associated with heat losses are subtracted from the entropy production of the measured system, this entropy change is not shown in the reference system either. Thus applies:

$$S_{60°C} = Q_{DHW}/(273.15K + 60K)$$

The dissipated entropy production via the DHW tapping is calculated with the specific entropy $s(\vartheta)$ between the entering cold water and the required DHW temperature of 50 °C:

$$\Delta S_{WW} = \sum_i m_{WW,i} \cdot [s(T_{KW}) - s(T_{WW})] \quad \textbf{(Eq. 11)}$$

For the test method, tapping profiles from the EN 16147:2017 are used. The entropy production from charging and discharging as well as the balance or the resulting entropy production of the fully mixed storage tank is shown for different tapping profiles in Fig. 4 (left). In the same graphic is also the entropy production of some measured examples shown.

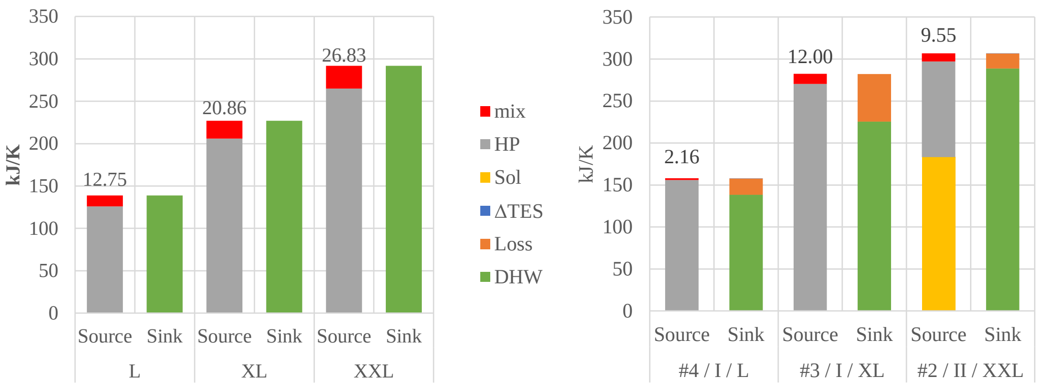

Fig. 4: Reference entropy input and output (left) for different tap profiles and measured examples (right).

Additional indicators

A temperature weighted by power was determined from the temperatures for charging and discharging the storage tanks. The flow temperature of the solar thermal collectors is determined according to (Eq. 12. The determination of the temperatures for the auxiliary heater as well as for the DHW supply is done similarly.

$$\overline{T_{HP,flow}} = \frac{\sum (T_{sol,flow,i} \cdot \dot{Q}_{sol,i})}{\sum \dot{Q}_{sol,i}} \quad \textbf{(Eq. 12)}$$

3. Test Results

3.1 Tested tanks

The test method was applied to four different storage tanks with different concepts and sizes for a variety of boundary conditions such as different mass flow rates on the heat generator. Fig. 5 gives an overview on the schemes and the integration of the tested storage systems.

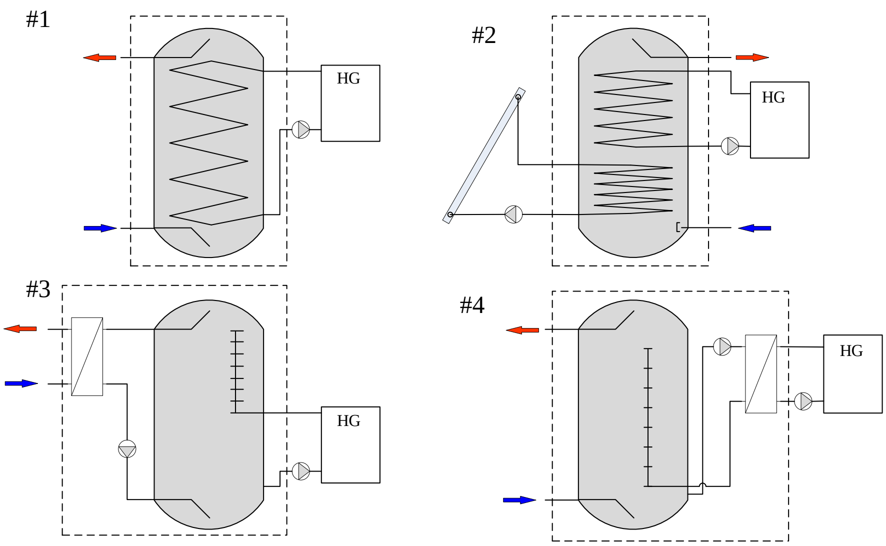

Fig. 5: Schemes and the integration of the tested systems.

Tab. 3: Description of the tested systems and the basic conditions of the measurements.

Tank		#1				#2		#3	#4	
Volume	[l]	200				780		1000	200	
Heat exchangers	DHW	-				-		ext HX	IHX	
	aux	IHX				IHX		-	ext HX	
	sol	-				IHX		-	-	
Test		I	II	III	IV	I	II	I	I	II
Profile		L	L	L	L	XL	XXL	XL	L	L
Solar		No	No	No	No	Yes	Yes	No	No	No
P_{aux}	[kW]	10	10	6	6	10	10	15	6	10
TW[(A)]	[-]	Yes	No	Yes	No	Yes	No	Yes	No	No

[(A)] Time Window for the DHW heating by the aux. heater.

3.2 Temperature curves

Fig. 6 shows the daily trend of a measurement with tank #1. The 200 l tank was charged with a 10 kW heat generator via an internal heat exchanger (IHX). One can see that the temperature on top of the tank drops significantly during the charging process. That’s why the required DHW temperature of 50 °C could not be delivered while a charging process is ongoing.

Fig. 7 shows the course of a measurement of the bivalent storage tank #2. The heat of the collectors and the additional heating was brought into the storage via IHX in each case. At the DHW charges by the additional heating at 6:00 o'clock and at 16:00 o'clock it can be seen that only the upper part of the storage tank is heated up. At the top sensor there is no drop in temperature (a positive finding). The solar heat is brought to the lower part of the storage tank. The heat output of the collectors on the test day is not sufficient to charge the storage tank to the temperature of 50°C required for DHW heating.

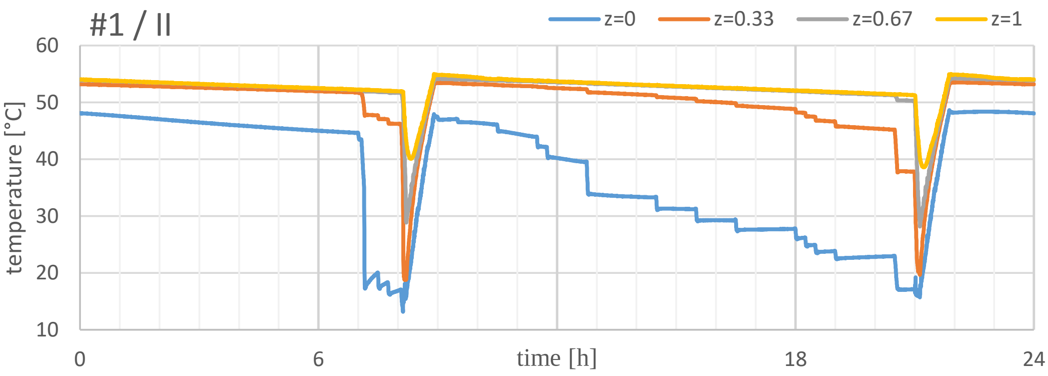

Fig. 6: Course of a measurement of a 200 l tank with IHX.

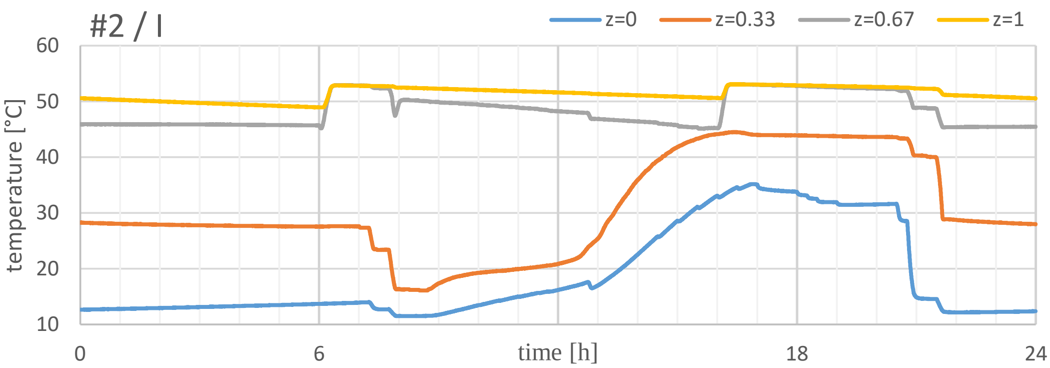

Fig. 7: Course of a measurement of a 780 l bivalent storage tank with two IHX.

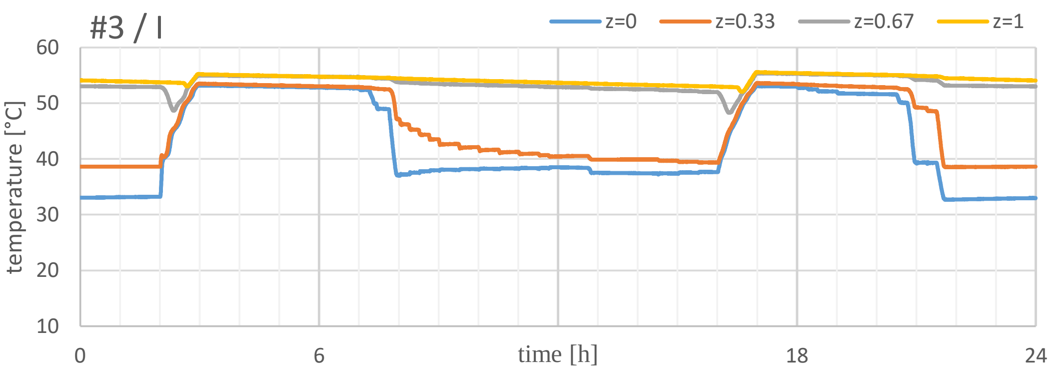

Fig. 8: Course of the measurement of a 1000 l storage tank with external HX for DHW.

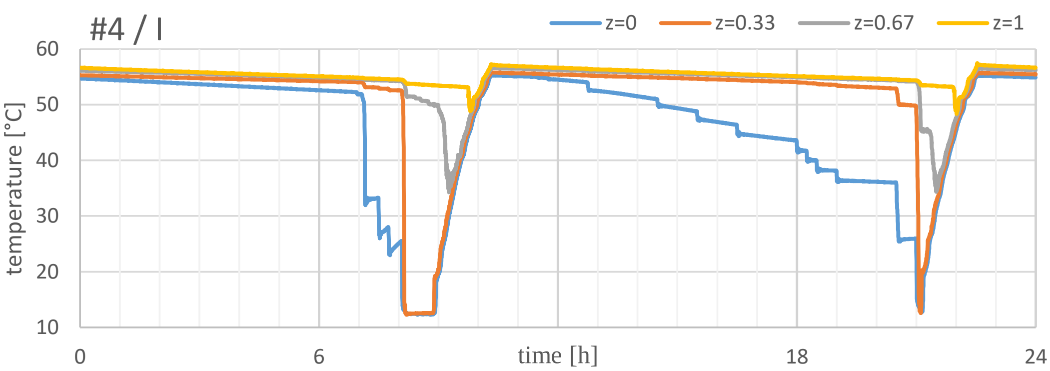

Fig. 9: Course of the measurement of a 200 l DHW tank with external HX for charging via an aux. heater.

Fig. 8 shows the course of a measurement with the 1000 l storage tank #3 that was combined with an external HX for DHW preparation. In opposite to the other measurements, it can be seen that the temperature in the lower part of the storage tank never drops below 30 °C. This has an unfavorable influence on the overall system performance. On a positive note, there is a barely perceptible drop in temperature in the top of the storage tank during charging.

Fig. 9 shows the course of a measurement with the 200 l tank #4 which is combined with an external HX for charging. Here you can see that the recharge is very late. Half of the storage volume is at just about 10°C when a recharge was triggered. During the recharge, the temperature at the top does not drop below 50°C. This means that the required comfort can be guaranteed at all times, despite the rather small storage volume.

The energy-temperature diagram of a typical charging processes for each of the tested tanks can be seen in Fig. 10. For this purpose, the energy from the aux. heating to the storage tank was sorted according to the flow temperature. In addition, the values were normalized for better comparison (in the figure on the right).

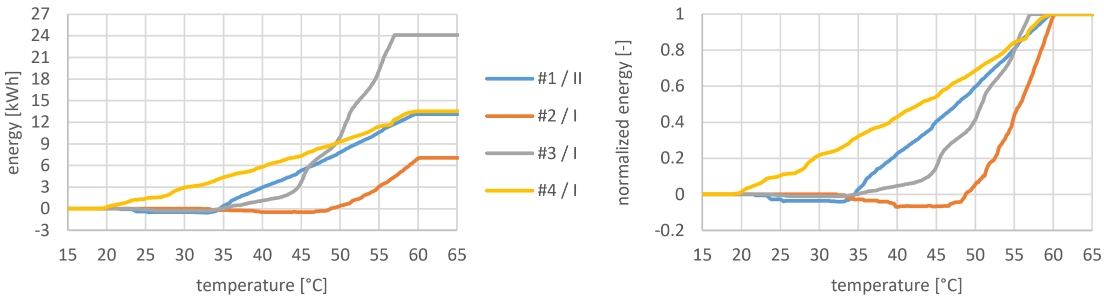

Fig. 10: Energy temperature diagram of the charging process via the aux. heating of one measurement each of the tested tanks. On the left: energy; on the right: normalized values.

The comparison of the charging of storage #1 and #4 at identical load clearly shows that the heat delivery with external HX and vertical stratifier in the storage was consistently at lower temperature level. For tank #1 with IHX, even a negative power of charging can be seen at the beginning of the loading (20 – 35 °C on the x-axis), which occurs due to the mixing processes in the storage. Also for the storage #3 that covers the energy demand for the XL-tap profile shows quite high flow temperatures. Approx. 60% of the energy was provided at over 50°C. The measurement of storage #2 shown was also carried out with the XL-tap profile. Here, however, only a part of the heat was supplied via the aux. heater, because low temperature heat was provided by solar thermal emulation. Hence, the part that was supplied by the aux. heater was in the range between 50°C and 60°C.

3.3 Energy, entropy and exergy

An example for a measured energy balance of a system over the 24-h test cycle is shown in Fig. 11 on the left. The main heat sources are the solar collectors and the auxiliary heater. The change in the net energy content of the storage over the test period is very small. The dominant heat sink is of course the DHW consumption, losses are minor in terms of the total energy turnover. Effects of mixing do not affect the thermal energy balance.

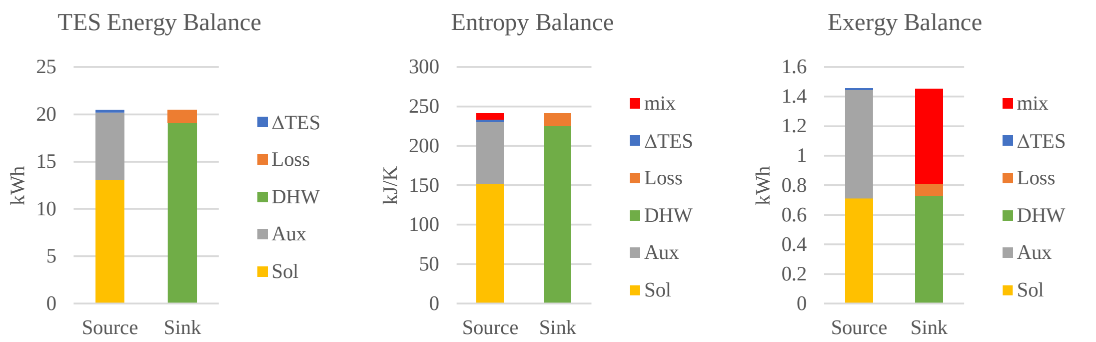

Fig. 11: Energy, Entropy and Exergy balance of the test #2 / I.

The entropy balance shown in the middle part of the same figure. It has a similar distribution, but now the entropy production is visible as a source term. Compared to the total entropy turnover, this value is quite small.

The total entropy production $\Delta S_{irr,exp}$ was 7.86 kJ/K. The entropy production of the reference system $\Delta S_{irr,mix}$ with the same DHW tapping profile is 20.86 kJ/K. The stratification efficiency of the shown example is: $\zeta_{str} = 1 - \frac{7.86\frac{kJ}{K}}{20.86\frac{kJ}{K}} = 0.623\ (62.3\%)$. In the exergy balance that is shown on the right side of the figure, exegetic losses due to mixing are visible on the right side since they are exergy sinks. Quite notably, those internal exergetic losses (reference temperature = 20 °C) are high in comparison with the exergy turnover and by far higher than exergy losses caused by heat losses from the system.

3.4 Stratification efficiency

Fig. 12 shows a summary of the main results of the tested storages.

Storage #1 achieved stratification efficiencies between 52 % and 59 %. The fact that the internal heat exchanger reached almost to the top of the volume of the tank proved to be problematic. It leads to a significant mixing and temperature decrease at the top of the tank at the beginning of charging. As a result, the required DHW setpoint temperature of 50 °C cannot be guaranteed continuously.

Storage tank #2 achieved a stratification efficiency of 62 % in the test with the tapping profile XL. The charging by solar heat via a heat exchanger in the lower part of the storage tank raises the temperature in the storage tank volume, but without guaranteeing the required setpoint temperature for DHW heating. Therefore, the HP has to load the upper part at a high temperature level, reaching the maximum flow temperature of the aux heater.

Storage #3 achieves a stratification efficiency of only 45%. The reason for the low stratification efficiency is assumed to be, on the one hand, high return temperatures of the external HX used for short DHW taps and, on the other hand, circulation effects via the external hydraulics. The losses of the storage tank were with 5 kWh/d or 7.2 W/K more than twice as high as for the other storage tanks with losses in the range of 2.1 to 3.2 W/K.

Storage tank #4 showed very good stratification behavior upon charging by an external heat exchanger in combination with a vertical stratification device in the storage tank. Despite the small volume of 200 l, a consistently very high DHW temperature was ensured with simultaneous low average temperature of the charging. The stratification efficiency in the test with 6 kW and 10 kW HP was 83 % and 85 %, respectively, and thus far exceeded the results of all other tested tank configurations.

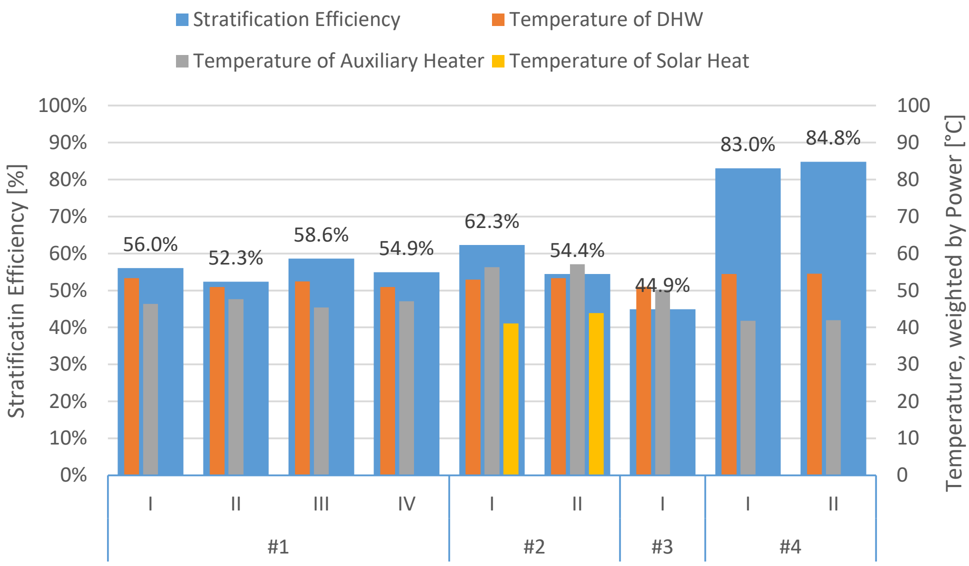

Fig. 12: Results of four different storage tanks with different concepts and sizes for a variety of boundary conditions.

4. Discussion

Based on the average flow temperature of the emulated HP from the individual measurements, weighted by power, an average COP of the heat pump for charging the storage tanks was calculated. Fig. 13 shows the COP of the assumed air-water heat pump as a function of flow temperature for an air inlet temperature of 12.9 °C (mean outdoor temperature of the test sequence). The triangles in the diagram represent the mean flow temperatures of the tested storage systems and their corresponding COPs illustrating the quite significant differences in performance that can be expected from the different storage tank configurations.

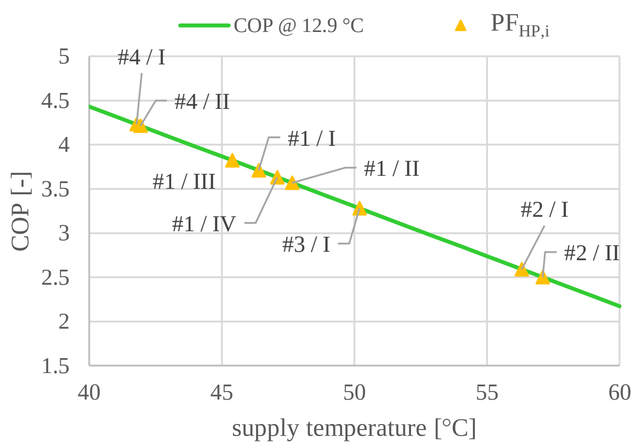

Fig. 13: COP as a function of the flow temperature for different ambient temperatures.

Based on the determined coefficients of performance, a demand for electrical energy for the HP was estimated for each case. The resulting values are presented in Tab. 4 and Fig. 14. With an identical DHW profile, the measurements at storage #1 show a 15% higher electricity demand than the measurements at storage #4, bearing in mind that the delivered DHW temperature of the more efficient storage #4 was up to 3.5 K higher than in storage #1, which in some DHW taps could not reach the required flow temperature of 50 °C. Overall, it is quite remarkable that storage tank #4 delivers the highest hot water temperatures overall, even though the lowest temperatures were measured for the storage tank charging. The system performance factor of 3.7 is also the highest of all the storage tanks, with exception of the solar thermal assisted system. The differences in the system performance factors suggest that the electrical savings of this storage tank compared to storage tank 1 (same tap profile) is 15%, and compared to a storage tank configuration like #3 (that could have profited from a larger tap profile) is as high as 42%, while providing greater comfort.

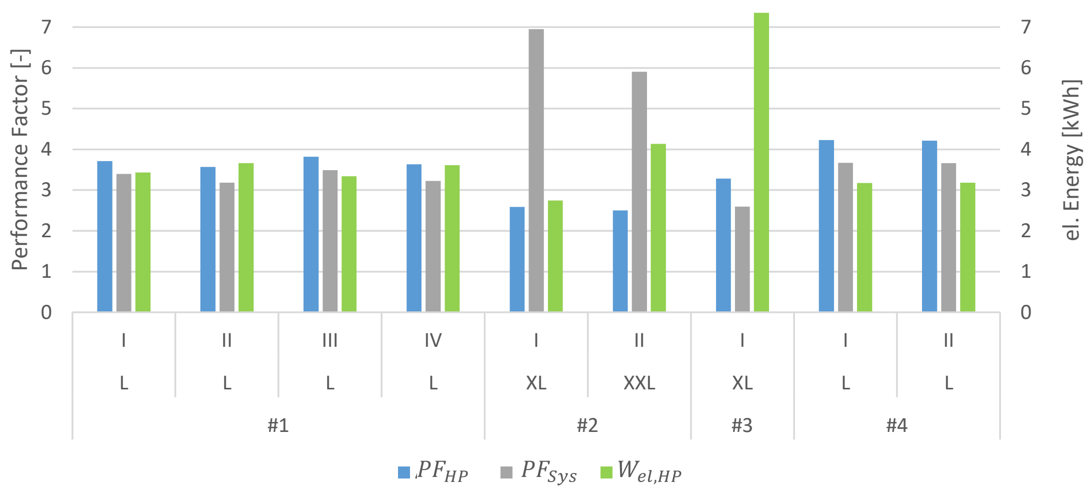

Fig. 14: Estimation of performance factors and el. energy demand.

The estimated electrical energy demand of the HP and the actually delivered DHW were used to determine the coefficient of performance of the storage system. This value, unlike the performance factor of the HP itself, is naturally by far highest for the system with solar thermal assistance. Among the storage systems without solar thermal, the range of results goes from 2.6 (storage system #3) up to 3.7 (storage system #4).

Tab. 4: Estimation of performance factors and electricity demand.

Tank		**#1**				**#2**		**#3**	**#4**	
Test		I	II	III	IV	I	II	I	I	II
ζ_{str}	**[%]**	**56.0**	**52.3**	**58.6**	**54.9**	**62.3**	**54.4**	**44.9**	**83.0**	**84.8**
$\overline{T_{aux,flow}}$	[°C]	46.4	47.6	45.4	47.1	56.3	57.1	50.2	**41.8**	**41.9**
$\overline{T_{DHW}}$	[°C]	53.3	50.9	52.5	51.0	52.9	53.4	51.0	**54.5**	**54.6**
PF_{HP}	[-]	3.7	3.6	3.8	3.6	2.6	2.5	3.3	4.2	4.2
$W_{el,HP}$	[kWh]	3.4	3.7	3.3	3.6	2.7	4.1	7.4	3.2	3.2
Q_{DHW}	[kWh]	11.7	11.7	11.7	11.7	19.1	24.4	19.1	11.7	11.7
$\mathbf{PF_{Sys}}$	**[-]**	**3.4**	**3.2**	**3.5**	**3.2**	**7.0**	**5.9**	**2.6**	**3.7**	**3.7**

5. Conclusions

The method for measuring the stratification efficiency of combi-storage tanks was adapted and simplified for the measurement of hot water tanks with and without solar thermal.

The test method was applied to four different storage tanks with different concepts and sizes. The results of the measurements show clear differences between the individual models and thus reveal weak points in the storage tank configurations that would otherwise remain undetected. The differences between the storages were significantly higher than anticipated by the project team.

Based on the results, it can be assumed that a storage concept with a very good stratification efficiency (for example, storage #4) not only ensures greater comfort and safety (less water in a critical temperature range thanks to better stratification), but also leads to significant savings in final energy at the same time. In combination with a heat pump, this saving can be up to 40% of electrical energy.

The measurements carried out prove the applicability of the method on the basis of a wide range of storage concepts and storage sizes. The assessment of the tested storages on the basis of stratification efficiency is consistent with the simulated electrical (final) energy demand for loading the storage by a heat pump.

Thus, the method for determining stratification efficiency via entropy production can also be applied to hot water storage tanks in the future.

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