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Single Source "Solar Thermal" Heat Pump for Residential Heat Supply: Performance with an Array of Unglazed PVT Collectors

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Abstract

This work deals with the results of experimental measurements of a novel solar heat pump based heating supply concept for residential heat supply. The constructed load case aims at supplying heat for a single-family house with a moderate heat demand of 45 kWh per m² and year. An system design case was measured comprising an array of rear insulated uncovered photovoltaic thermal (PVT) collectors, a brine/water heat pump with deactivated resistance heaters and extended temperature range on the evaporator side, and a two-zone combi storage. The main innovation is to operate the PVT array as the sole heat source for the heat pump and accordingly with extended collector loop temperatures below the freezing point. Performance analysis was carried out on two characteristic cold winter test days. Measurement results show that the present system configuration is able to cover the heat demand comprising domestic hot water and space heating. Analysis of the cold winter test days reveal that insignificant icing occurred on the surface of the collectors. Temperature comfort levels were reached. Improper system design was detected, discussed and several concrete optimization potential measures were identified.

Keywords: solar thermal; heat pump; solar only; single source; photovoltaic thermal collector; PVT

1. Introduction to solar thermal heat pump systems

1.1 System Concepts

In residential heating systems comprising domestic hot water preparation, solar thermal collectors are usually combined with a fossil back up. Replacing the fossil boiler with a heat pump provides the possibility to boost the renewable share of these heating systems. In such a solar and heat pump (SHP) system, solar thermal collectors and the heat pump supply energy for space heating and/or hot water independently of the source(s) of the heat pump. Heating systems with this kind of operation mode were defined as "parallel" within the work of the Task 44/Annex 38 "Solar and Heat Pump" of the Solar Heating and Cooling program of the International Energy Agency (T44A38). In the "serial" configuration the solar thermal collector acts as a source of the heat pump, either exclusively or additionally (Hadorn, 2015). In general, in SHP systems, both, parallel and serial operation modes can be realized within one system (Fig. 1, left). The SHP system investigated in this work is

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a combined parallel/serial solar thermal heat pump system (Fig. 1, right) with an array of unglazed PVT collectors.

1.2 Market barriers

The T44A38 market research conducted in 2011/2012 found 135 solar thermal heat pump system configurations offered by different suppliers and manufacturers (Ruschenburg, Herkel et al., 2013). Results of T44A38 have shown that there are promising systems in terms of energy performance in both, serial and parallel (and regenerative) configurations (Hadorn, 2015). For illustration purposes, Fig. 2 shows the operating temperatures of the main components, solar part and heat pump with either air or ground source, of said SHP configurations ("state of the art"). It can be seen that in (regenerative and) serial SHP configurations, controller and system design rule out operation temperatures below zero degrees in the solar loop.

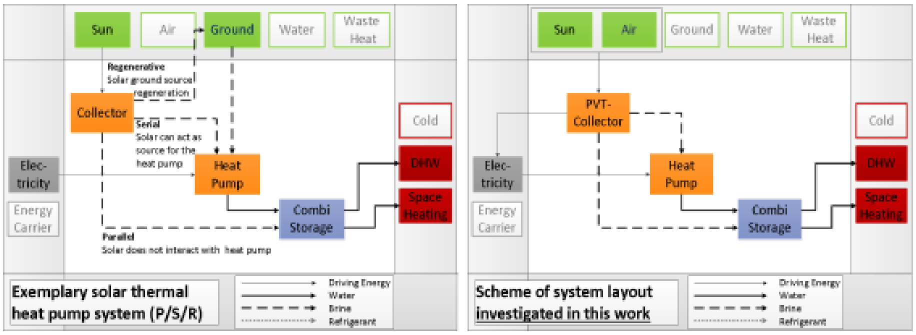

Fig. 1: The left scheme shows an exemplary parallel/serial solar thermal heat pump system according to T44A38 visualization scheme with costly energy sources on the left bar, free energy sources on the top bar, and useful energy sinks on the right bar. The right scheme shows the combined parallel/serial solar thermal heat pump system investigated in this work. Similar to the example on the left, this system has access to two heat sources via the PVT collector array.

Manufacturers may shy away from the risk of formation of ice. On the one hand, the performance of unglazed solar collectors under operating conditions below the frost point was investigated in the past i.e. in (Massmeyer and Posorski, 1982) or more recently in (Bunea, Perers et al., 2015), but studies with operation of entire systems are still rare. Hence, a second heat source is required to cover the remaining (high) heat demand during cold weather periods. A consequence is that the utilization ratio of the solar array is greatly reduced by limiting the operating temperatures of the collector field above the freezing point (i.e. < 2 °C). As a result, the solar fraction of these systems would be in a similarly (low) range as for conventional solar thermal combi systems with fossil back up heater.

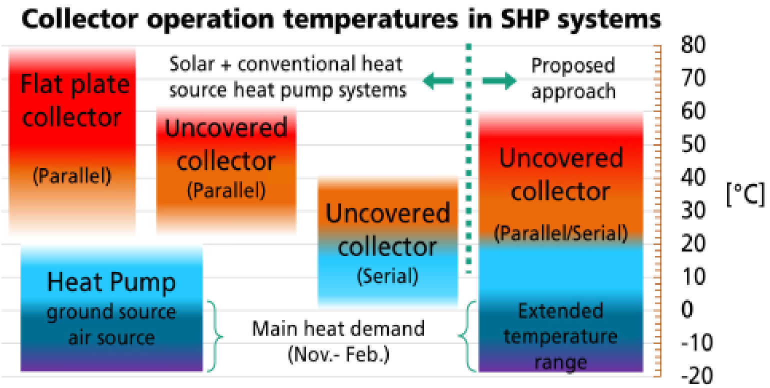

Fig. 2: Most SHP configurations investigated within T44A38 (i.e. market study) do not substitute a conventional heat source. Hence, solar does not cover the main heat load: This design approach limits the solar utilization ratio, thus leading to low fractional energy savings.

Whereas improved system performance can be achieved, the additional investment cost for the solar thermal part (solar collector array, hydraulics and pump group, installation and initial operation, etc.) forms a market barrier: Considering the extra cost of the "optional solar thermal add-on" and its limited saving potential, investors might question its meaningfulness. To date, solar thermal and heat pump systems – if at all – play a niche role in the national heat pump markets. In order to improve the cost efficiency solar thermal heat pump systems, it seems in the nature of the case to substantially increase the utilization ratio of the solar part by extending its temperature operation range below zero degrees Celsius, hence, making the conventional heat pump heat source and cost for its activation obsolete (Fig. 2, "Proposed approach").

2. Single source SHP systems

2.1 Solar thermal and PVT: Previous developments

Only six out of 135 configurations of said T44A38 market study can be called single source SHP systems employing an array of solar collectors as sole source of the heat pump (Table 1). All systems but one use unglazed solar collectors to operate below ambient temperatures. The majority of systems allow for operation below the frost point. All but two systems rely on smaller or larger source side storages, mostly using water as a phase change material, which results in additional cost. None of the systems use PVT collectors. On the one hand, PVT collectors in T44A38 system configurations are only employed as additional low-temperature heat and electricity source for space heating and DHW or to regenerate a heat pump source (ground probe or brine storage). On the other hand, the prospect of a heat pump heating system with a single component PVT collector as heat and electricity generator for the heat pump seems attractive.

Table 1: Overview of collector types, operation temperatures and storage types of all identified single source solar thermal "only" heat pump system configurations of T44A38 market study. Only systems providing space heating and domestic hot water (DHW) are listed. For comparison, the system investigated in this work is shown at the end.

#	Available modes	Collector type	Collector array operated		Storage on heat pump ... side	
			as ambient heat exchanger	below freezing point	source	sink
1	Parallel/ Serial	Flat plate	Presumably not	-	wet sand (large, outdoor)	DHW
2	Serial	Absorber	yes	yes	-	DHW
3	Serial	Absorber	yes	?	ice (large, outdoor)	DHW
4	Serial	Absorber	yes	yes	ice (320 l)	Combi storage
5	Serial	Absorber	yes	yes	-	DHW
6	Serial	Flat plate with integrated fan	yes	yes	ice (320 l)	Combi storage
This work	Parallel/ Serial	PVT	yes	yes	-	Combi storage

While in R&D, the heat supply concept of PVT collectors applied as sole heat source for a heat pump gained increased interest (Zhang, Zhao et al., 2012), especially for DHW applications (Wang, Guo et al., 2017), efficient application in combined space heating and DHW still seems to pose a challenge. To date, the authors are aware of only one single source PVT heat pump system configuration for space heating application which is about to enter the market (Leibfried, 2018).

2.2 Objectives of the study

The objective of this study was therefore to check the principal meaningfulness and furthermore the potential of the single source PVT heat pump as a heat supply concept for residential buildings.

1. Is the array of PVT collectors able to sufficiently supply heat for the heat pump during cold weather periods, including extreme cold and snowfall?
2. To what extent can icing be expected due to the continuous low-temperature operation?

3. Which optimization potential can be derived?
4. How to integrate the electric benefit of PVT into established performance figures?

3. Experimental setup

3.1 Single source PVT heat pump system configuration

A parallel/serial solar thermal heat pump system with an array of PVT collectors, power inverter, combi storage, heat pump, and hydraulics was installed at the system test rig at Fraunhofer ISE in Freiburg, Germany and put into operation in December 2017 (Fig. 3). The main system components are described in further detail in Table 2.

Fig. 3: Investigated single source SHP system with an array of PVT collectors in two rows totaling 20 collectors installed on top of the system test stand at Fraunhofer ISE (left). The photo on the right shows the installed combi storage and the heat pump including a hydraulic unit. The power inverter is installed as well (not shown in the photo).

Table 2: Description of the main components of the measured solar thermal heat pump system.

Component	Subcomponent	Description
Solar PVT array 31,4 m² 5,2 kWp,	Collector type	The array consists of 20 rear-insulated uncovered and therefore infrared and wind-dependent PVT collectors ("2Power HM 1000, 1Power 260 Mono Black"). Rated power is 260Wp (PV), area is 1,57m².
	Electric side	The modules are connected to an SMA inverter, which is connected to the grid. The inverter is powered only by the PV array.
	Thermal side	The array is subdivided into four arrays with five modules each. The hydraulic connectors of the modules are connected according to the Tichelmann system; hence all collectors are flowed through in parallel.
	Control	The parallel mode is set as first priority. Charging is triggered if the temperature of the solar array is above the lower storage tank temperature. Serial mode is triggered by heat pump control settings based on heat demand.
Heat pump P_{rated}: 5,1 kW	Type	Brine-water heat pump with extended temperature operation range on the source side from -20 °C to +30°C.
	Heating rods	Resistance heaters (backup) on source/sink side deactivated during measurements.
	Control	Storage tank temperatures trigger the serial heat pump operation using the solar array. After parallel mode, serial mode is second priority: The heat pump should provide temperatures above 48°C in the DHW section of the combi storage. If this criterion is fulfilled, the space heating section of the combi storage is charged. The required temperature depends on the ambient temperature.
Storage	Type	850 liter combi storage provided with a permeable partition plate to avoid temperature mixing of space heating and domestic hot water zones. Inner outlets in the space heating zone are provided with diffusors (cf. Fig. 4).

3.2 Description of load emulation and measurement equipment

Fig. 4 shows the experimental setup of this work comprising the solar thermal heat pump system (equipment under test), with its main thermal components and hydraulics as well as the interface to the test rig. In total, almost 60 sensors were installed, of which 41 were calibrated for the characterizing measurements, the remaining form part of the equipment under test. Table 3 and Fig. 4 refer to (and show) selected sensors relevant to this work only: Table 3 shows the applied sensor types and accuracies; Fig. 4 shows the denominations of the sensors used in result diagrams (Fig. 5) in chapter 4.

Table 3: Properties of selected sensors used for the measurements and presented in this work.

Sensor	Manufacturer	Type	Accuracy
Temperature (fluid)	TMH	PT 100	0,1 K
Flow Rate	Krohne/Siemens	magnetic inductive (Optiflux 5000/ Sitrans FM MAG 1100)	0,5 %
Solar radiation	Kipp&Zonen	CMP 11	2 %
Relative humidity	B&B Thermo-Technik GmbH	HA-ANA-10V	2 %
Electricity (summer test day only)	EMU	Allrounder 3/75	1 %

Fig. 4 shows the load side on the right side of the dashed line, comprising a space heating loop (radiator) and domestic hot water loop with a freshwater station (heat exchanger and shower head symbol). In both cases, hot water tapping from the combi storage is realized by directly connected hydraulic loops with eccentric worm pumps. Details of the volume flow control specifications are given in Table 4.

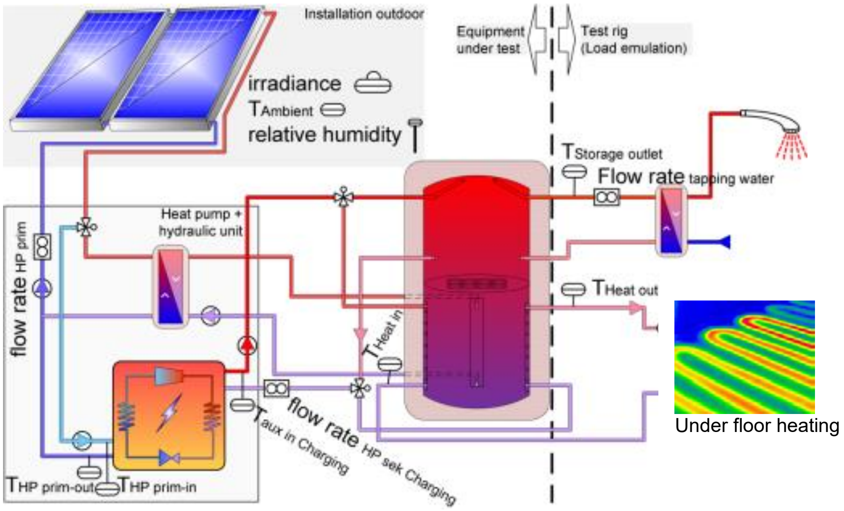

Fig. 4: Experimental setup of this work comprising the solar thermal heat pump system, with its main thermal components and hydraulics, interface to the test rig and sensor positions. The electric side of the solar array is connected to a power inverter (not shown). Sensor denominations given in this illustration are used in Fig. 5 of the next chapter.

Table 4: Set up for the domestic hot water (DHW) and space heating heat loads.

Type	Load description
DHW load:	7,7 kWh at 45 °C carried out in 23 draw offs (shower, small draw offs, dishwashing, bathtub). The load profile is equal for every day and based on the EU tapping cycle “M” with 23 draw-offs, ~5,8kWh (Ecodesign Regulation 814/2013, 2013). The following T44A38 modifications according to (Haller, Dott et al., 2013) were adopted: • Since a daily 3,6 kWh “bathtub” draw off was added to the EU tapping cycle, the energy demand for the other draw-offs was reduced (Total: 7,7kWh instead of 9,4kWh). • Required draw off temperatures were set to 45 °C.

	Deviations from the T44A38: • The bathtub draw-off is every day (not every 7th day) • The dishwasher draw-off is as well at 45 °C (instead of 55 °C) The temperature loss of the freshwater station was assumed 3 K. Hence, the heat pump control was set to provide temperatures of least 48°C in the DHW zone of the combi storage.
Heating load:	65 kWh "Winter peak case" for winter and spring test days based on SFH45 building with underfloor heating (ΔT = 4K) in the climate of Strasbourg. A maximum heating load "winter load case" was defined for all test days except summer. The load is based on T44A38 simulation results for an "SFH 45" building with an underfloor heating in the climate of Strasbourg (45 kWh/m²a) and a heated area of 140m². Based on simulated monthly values given in (Dott, Haller et al., 2013), the mean daily heat demand for the coldest months December and January was obtained (51 kWh/d and 54 kWh/d). Since there are positive (and negative) deviations from these mean values throughout these months, an increased daily space heat load of 65kWh was chosen for said winter load case. The temperature of the space heating zone of the combi storage (hence, indirectly $T_{Heat\ out}$) is controlled dynamically by a heating curve dependent on ambient temperature, as usual in heating systems. The heat load of 65 kWh is achieved as follows: Underfloor heating loops are often operated without night setback because of their inertia; accordingly, the eccentric worm pump operates 24 h with a (constant) volume flow of 10 l/min. Assuming a temperature difference of 4K between $T_{Heat\ out}$ and $T_{Heat\ in}$, ~65kWh is discharged from the combi storage within 24h. $T_{Heat\ in}$ was set dynamically to $T_{Heat\ out}$ - 4 K with a minimum temperature admitted by the thermostat at 20°C.

3.3 Test days

In response to the questions raised in section 2.2 (Objectives of the study), characteristic test days were selected from the measurement period and analyzed accordingly. Refer to the next chapter for test day description and system performance analysis.

4. Results: Performance analysis on test days and optimization potential

4.1 Presentation of measurement results on test days

Result diagram description: Result diagrams of all test days are shown in Fig. 5. The characteristic operating behavior of the system is shown by pairs of diagrams showing the key sensor values of the charging (left) and discharging (right) side of the combi storage. Discharging is carried out via the domestic hot water and/or the space heating loop. Refer to Fig. 4 of the previous chapter for sensor positions. Irradiance and ambient temperature are shown on the left side.

Load side - DHW tapping emulation: 23 draw offs were performed on every test day, with the desired characteristics. Energy quantities as provided by the test rig are presented (Table 5). DHW tapping was carried out as intended: The targeted energy load of 7,7kWh was either met or slightly surpassed presumably due to a temperature level above 48°C in the DHW zone of the combi storage on test days spring and summer.

Load side - space heating load emulation: Due to inappropriate control settings of the thermostat, which in turn lead to temperature differences lower than 4K between $T_{Heat\ out}$ and $T_{Heat\ in}$, the targeted "winter peak case" heat demand of 65 kWh was met only on the second winter test day (03th of March), caused by the control of the test rig. Nevertheless, heat demand is still appropriate: 57.2 kWh is still above the mean heat demand of December (51kWh/d) and January (54kWh/d, cf. Tab 4, "heating load").

Table 5: Measured heat loads for DHW and space heating for given test days. Measurements were carried out in 2018.

Unit [kWh]	1st winter day 28th Feb.	2nd winter day 03th Mar.	Spring day 09th Apr.	Summer day 19th Aug.
DHW: Total	7.6	7.5	8.1	9.1
DHW: Tapping, shower	4.1	4.0	4.6	5.4
DHW: Bathtub	3.6	3.6	3.6	3.7
Space heating: Total	**57.2**	**66.9**	55.7	0.0

General remarks regarding the system behavior – combi storage charge: Charging is carried out either via the heat pump (serial mode) or via the solar plate heat exchanger (parallel mode). In serial mode, the heat pump is charging the DHW and space heating zone of the combi storage alternately (cf. $T_{aux\text{-}in – charging}$). Depending on the heat demand, this occurs continuously, as on the 2nd winter test day, or with interruptions when the combi storage is fully charged (cf. spring test day from ca. 07:30). During these periods, it can be seen that the heat pump control "checks" the source temperature level three times per hour, sending a volume flow of about one minute to the source side pump. On the summer test day, the occurrence of the parallel mode can be observed, starting at about 11:30 (cf. flow rate$_{solar\text{ - }charging}$, $T_{HP\ prim\text{-}in = sol\ out}$ $T_{HP\ prim\text{-}out = sol\ in}$).

General remarks regarding the system behavior – combi storage discharge: Regarding the achieved temperatures of the space heating temperatures, the two winter test days are relevant only (Refer to the following section 4.2). Regarding the achieved DHW temperatures, a general problem can be observed which concerns the heat pump charging behavior: DHW temperature comfort level ($T_{Minimum\ demand}$ = 48°C) is generally met for all draw offs apart from the bathtub where discharge temperatures always drop to about 43 - 44 °C. That this is a storage tank specific issue can be seen on the summer day, where the combi storage is fully charged by the solar array in parallel mode. The heat demand for the entire bathtub is met without any temperature drop. Further malfunctions due to bad control/system layout are discussed in chapter 4.4, Tab 5.

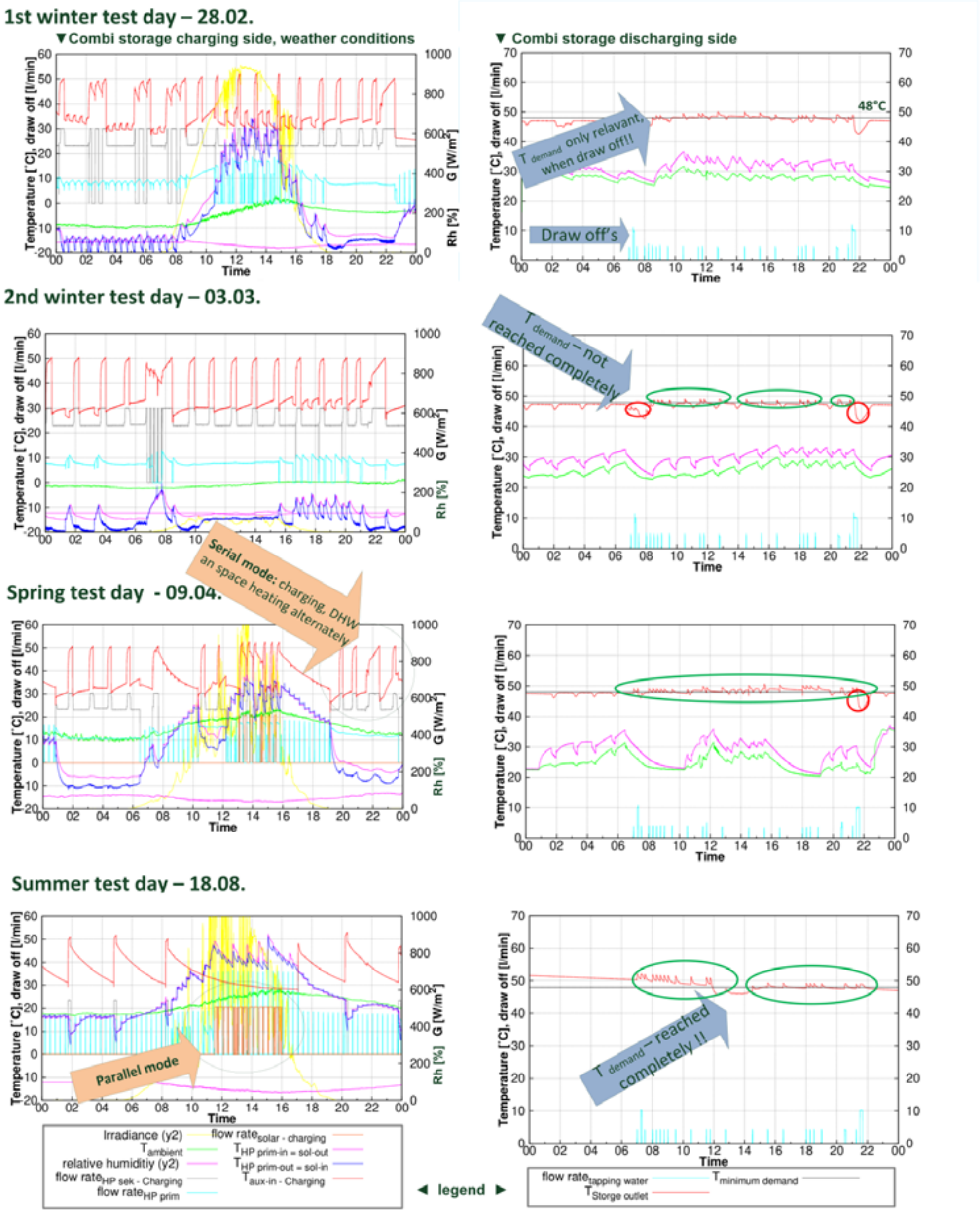

Fig. 5: Diagrams of the following test days: Winter (1st: 28.02.18 and 2nd: 03.03.18), Spring (09.04.18) and Summer (18.08.18). Diagrams on the left show the charging side of the combi storage, diagrams on the right show the discharging side of the combi storage. Legend applies to all diagrams above. Refer to Fig. 4 for positions of the sensors within the system.

4.2 Performance under cold weather conditions

This section deals with the question, whether the array of PVT collectors is able to sufficiently supply heat for the heat pump during cold weather periods, including extreme cold, snowfall, and icing. In response, system performance on two typical winter situations is analyzed.

- 1st winter day (28.02.) Clear sky day/night with ambient temperature ranging from -5 °C to -10 °C. 27th and 28th of February 2018 were the coldest days of the entire winter period 2017/2018 in Freiburg (Fig. 5,6).
- 2nd winter day (03.03.) Cloudy day/night with an ambient temperature between -2°C to 1°C (Fig. 5,7).

Icing: Fig. 6 shows the maximum level of ice formation at 04:53 the early hours of the 1st winter test day. The layer of ice is in places and superficially only (< 1 mm). Similar behavior was observed during measurements of other cold days, as well as in the climate chamber (Schmidt, Schäfer et al., 2018). **"Ice-production" on the module surfaces did not exceed 1cm of thickness (precipitation) on the front side during the entire system measurement phase (Dec. '17 to Aug. '18). No slipping of ice could be detected (tilt angle of 35°).** The comparably low ice formation can be explained due to the low flow of humidity being transported to the surface of the PVT collectors by means of natural convection. In contrast, icing occurs usually in external air units, where humidity is permanently supplied by means of a forced air flow to the compact heat exchanger of the external unit. In consequence, significant amounts of icing on the PVT collector array should only be expected if transport of humidity is assured in large quantities (i.e. precipitation). It could also be observed, that surface ice would melt within a few minutes under direct solar irradiation.

Fig. 6: Maximal observed ice cover on the surface of the solar array in the clear sky night from 27th/28th of February with a thickness approximately below one millimeter (left). On the coldest two days of the winter 2017/2018, the solar array reached operation temperatures above 20 °C from noon to the early afternoon.

The layer of snow on the solar array on the 2nd winter test day (Fig. 7) originates from snowfall on the previous day. It persisted for the entire test day on the solar array due to: low irradiance, the cooled PVT collectors operating as ambient heat exchanger and ambient air temperatures close to the freezing point. As explained below, such a layer of snow still allows for ambient heat exchange with the collector fluid. Measurements on absorbers under comparable conditions (Bunea, Perers et al., 2015) revealed that heat exchange can be increased due to the "naturally" increased collector surface**. The heat pump allows for manual reversing the heat pump cycle to heat the collector with energy by cooling the space heating zone of the combi storage. The mechanism was successfully tested on a layer of snow (+5 cm) on the solar array**: The layer of snow started melting on the side touching the collector array and slipped afterwards to the test rig floor.

Fig. 7: The solar array was covered with a layer of snow with a thickness of ~2 cm on the solar array for the period of 24h of the first winter test day (03.03.). The photos show the first row of the solar array in the morning (07:02) and in the evening (17:41).

System performance on the 1st winter day (cf. Fig. 5): On the second coldest day of the winter period 2017/2018, a space heat demand of 57.2 kWh is covered. Mean temperatures of the space heating loop ($T_{heat\ in}$ + $T_{heat\ out}$)/2 range from 26°C to 33°C. DHW energy demand is met as well; comfort levels are met with exceptions in the morning and in the evening (cf. section 4.2, last paragraph and section 4.5, Table 6, Area "DHW").

Throughout the day, the heat pump is operating most of the time in serial mode (cf. flow rate$_{HP\ Sek\ –\ Charging}$). Due to the cold ambient temperatures, when the evaporator outlet temperature $T_{HP\ prim\text{-}out}$ reaches -20 °C, the heat pump switches to frost protection mode and combi storage charging is interrupted. This occurs frequently from midnight to about 8am in the morning. During this time, it can be seen how space heating loop temperatures decline gradually. Interestingly, during the sunny day, mean evaporator temperatures ($T_{HP\ prim\text{-}in}$ + $T_{HP\ prim\text{-}out}$)/2 increasing along with the sunrise to peak values above 20 °C between 12:00 and 15:00 and decreasing back to low night time values after sunset. Hence, on clear sky days, the single source heat pump can play out its advantage by means of high source temperatures compared to air or ground source heat pumps.

System performance on the 2nd winter day (cf. Fig. 5): On this cloudy winter day, where a layer of snow covers the solar array, a space heat demand of 66.9 kWh is covered. Mean temperatures of the space heating loop ($T_{heat\ in}$ + $T_{heat\ out}$)/2 range from ca. 26°C to 30°C. DHW energy demand is met as well; comfort levels are met with exceptions due to an unknown charging interruption of the heat pump between ca. 7:45 to 8:45 (see also section 4.2, last paragraph and section 4.5, Table 6, Area "DHW").

Likewise, to the first winter day, the heat pump is operating most of the time in serial mode (cf. flow rate$_{HP\ Sek\ –\ Charging}$), charging DHW and space heating zone alternately (cf. $T_{aux\text{-}in\ –\ charging}$). Mean evaporator temperatures ($T_{HP\ prim\text{-}in}$ + $T_{HP\ prim\text{-}out}$)/2 are below -10°C during the whole day reaching minimum values at night time between -15° and -18°C. Apart from two frost protection cycles (at about 01:15 and 03:15), the solar array provides low-temperature energy to the heat pump at all times.

Assessment of system performance under cold weather conditions: Heat demand was met completely. The heat was delivered at the required temperature levels for space heating and with few limitations for domestic hot water.

4.3 Performance indicators used in the project (SPF or AZ)

Measurements in summer, autumn and winter where continued. The test days where repetitive days like mentioned before. The heating demand was adapted to the demand developed in IEA TASK44A38 (Summer: no heating; spring/autumn: 35 kWh/d; winter: 55 kWh/d). The daily DHW draw offs where
8 kWh/d split in the 23 taps based on Erp.

Representative test days where used to determine (daily) performance indicators.

Fig. 8 is based on IEA TASK44A38. The boundaries for the different AZ-values are shown.

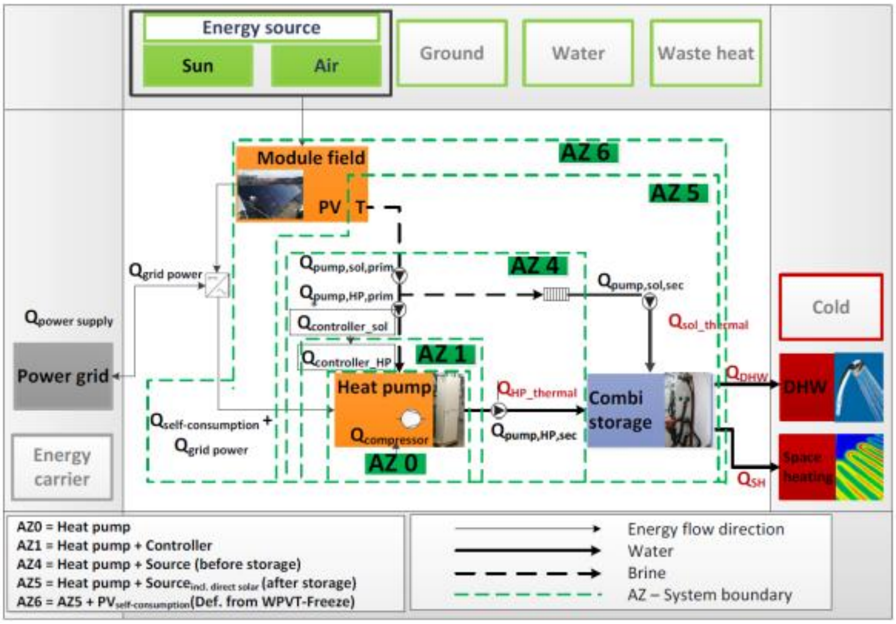

Fig. 8: Boundaries for the performance indicators based on T44A38.

AZ0 – AZ5 are known performance indicators. AZ6 is introduced to show the performance indicator by considering the PV- self consumption.

Day type	Date: / Characteristics of the day	18.08.2018 Summer-sunny	24.08.2018 Summer-cloudy	27.10.2018 Autumn-Rain	06.11.2018 Autumn-cloudy	03.11.2018 Autumn-sunny	24.01.2019 Winter-cold	29.01.2019 Winter - sunny	Seasonal Performance Factor
Load	Domestic hot water [kwh]	9.1	9.1	8.3	8.0	8.4	8.5	8.5	(ISE-internal Extrapolation - Weighting of reference/test days)
	Target value (23 Tapping cycles, ErP)	8	8	8	8	8	8	8	
	Heating load [kwh]	-	-	34.8	33.9	34.2	53.8	51.6	
	Target value (acc. to TASK44)	-	-	35	35	35	54	54	
Seasonal Performance Factor	AZ 0	4.0	3.9	3.5	3.5	3.7	2.9	3.2	3.6
	AZ 1	2.6	2.7	3.3	3.3	3.5	2.8	3.1	3.1
	AZ 4	1.7	2.4	2.7	2.6	2.9	2.3	2.5	2.5
	AZ 5	2.9	2.4	2.4	2.6	2.2	2.0	2.2	2.4
	AZ $6_{\text{incl. PV self-consumption}}$	5.3	3.4	2.5	2.8	2.6	2.1	2.5	3.2
PV	PV - Daily yield [kwh]	19.7	7.9	2.1	1.7	8.6	0.6	6.4	rough first estimation
	PV - Self-consumption [kwh]	1.4	1.1	1.1	1.1	2.8	0.5	3.3	
	PV - Share of self-consumption from daily yield	7%	14%	54%	66%	32%	95%	52%	
	PV - Share (self-consumption) of electricity demand	45%	29%	6%	7%	15%	2%	12%	
Meteo	Irradiation - Daily total [MJ/m²]	18.0	6.7	1.8	0.4	12.5	1.1	9.7	
	T_amb min [°C]	16.7	14.3	5.0	4.8	6.5	-2.8	0.0	
	T_amb max [°C]	28.5	24.2	10.2	13.1	12.5	-1.3	6.0	
	Rain - Daily total [mm]	0	0	14	0	0	0	0	

Fig. 9: Performance indicators, with boundary conditions (daily and a yearly, projected based).

Fig. 9 shows the daily performance indicators with it's boundary conditions. Theses values are based on real measurements. The last column shows the seasonal performance indictors. The base for this are the test days shown in the table, extrapolated to reach a rough first estimation.

4.4 Optimization potential

The realized system has optimization potential. The main points are the following:

- Storage tank design not satisfying. The ErP "bathtub" tapping could also be reached completely by a better use of the buffer volume (readiness volume) for the DHW.
- A controller concept with focus on higher amount of PV self consumption could increase the seasonal performance factors
- PVT modules without back insulation for higher use of latent heat and thermal convection could increase the energy coming from the PVT source

4.5 Key performance indicators according IEA TASK60

Within the IEA TASK60 (PVT systems) Subtask D, key performance indicators are under development. The report of this Subtask D was supposed to be published in January 2020. All formulas with description will be content of this report. A presentation of the Subtask D leader Daniel Zehnhäuser showed a list of KPI's, which are wished "at least". The following chapters show the application of these KPI's, based the measurements on the PVT-heat pump system of project WPVT-Freeze.

4.5.1 Area-specific thermal yield

(listed in the presentation; but no defined calculation method in the Subtask D report)

$$Area\ specific\ yield = \frac{Q\ thermal\ taken\ from\ PVT - field}{Areagross\ (31{,}4\ m^2)} = 312\,\frac{kWh}{m^2 a}$$

Aspects of the PVT-module-field:

- System controlling was not optimized on using solar
- Thermal absorber is glued on the back side of the PV
  -> solar is absorbed by PV and heat conduction to absorber could be optimized
- Back side insulation of the module causes less use of environmental energy
  -> later PVT prototype in the project is without back side insulation

4.5.2 Thermal utilization ratios

$$\omega_{PVT,th}^{gross} = \frac{Q_{PVT}}{\int G_{col}\,dt \cdot A_{PVT}^{gross}} \qquad \text{Equation 3 (Report – Subtask D)}$$

$$Thermal\ utilisation\ ratio = \frac{Q\ thermal\ taken\ from\ PVT - field}{Qsolar\ *\ area\ gross\ (31{,}4\ m^2)} = 0{,}28$$

Aspect:

- $Q_{thermal}$ contains heat taken from sun and environment

4.5.3 Average of the collector mean fluid temperature

$$\overline{\vartheta_m} = \frac{\int_{\substack{collector\ field \\ in\ operation}} dt\, \vartheta_m}{\int_{\substack{collector\ field \\ in\ operation}} dt} = 0.4\ °C\ \ (mean\ value\ /year) \qquad \textbf{Equation 11 (Report – Subtask D)}$$

Aspects:

- Main service time of heat pump during space heating season
- Main heat source of this PVT-application is using environmental energy
  -> low surrounding temperatures during winter

4.5.4 Solar thermal fraction

$$f_{sol,th}^{total} = \frac{Q_{PVT,*}}{Q_{*,primary}+Q_{PVT,secondary}} = 1 \quad \text{(has to be 1)}$$ **Equation 18** **(Report – Subtask D)**

Expl.: Thermal energy PVT / (Energy delivered to heat pump + Energy delivered directly to the storage)

$$f_{sol,th}^{secondary} = \frac{Q_{PVT,secondary}}{Q_{HP,*}+Q_{PVT,secondary}} = 0.1$$ **Equation 19** **(Report – Subtask D)**

Expl.: Thermal energy PVT (direct charge of storage tank) /
(Heat pump energy delivered to storage tank+ Thermal energy PVT, direct charge of storage tank)

Aspects:

- The calculation according to TASK60 seems to focus on multi source systems (see Eq. 18)
- Main heat source of this PVT-application is using environmental energy -> low surrounding temperatures during winter (low value of 0,1 considers only solar)

4.5.5 Performance Factor – heat pump system

$$SPF_{SHP+}^{(total)} = \left[\frac{Q_{HS}}{E_{*,HS}}\right]_{SHP+} = \frac{used\ heat}{electricity\ for\ heat\ pump} = 2.4$$ Equation 28 (Report – Subtask D)

Remark: See also chapter 4.3 (it corresponds to AZ5=2.4)

4.5.6 Performance Factor – heat pump system incl. grid

$$SPF_{SHP+}^{(Grid)} = \left[\frac{Q_{HS}}{E_{Grid,HS}}\right]_{SHP+} = \frac{used\ heat}{electricity\ for\ heat\ pump - PV\ self\ consumption} = 3.2$$

Equation 28 (Report – Subtask D)

Remark: See also chapter 4.3 (it corresponds to AZ6 = 3.2)

4.5.1 Investment cost per square meter

$$i_0^{PVT} = \frac{[I_0]_{PVT}}{A_{gross}^{PVT}} = 378\ €/m^2$$ Equation 48 (Report – Subtask D)

The following costs are considered:

(PVT-modules, inverter, controller, piping and mounting equipment, solar fluid)

4.5.2 Levelized Cost of Heat

$$LCoH = \frac{\left[\sum_{t=1}^{T=25}(I_t+OM_t-RV_t)\cdot(1+r)^{-t}\right]_{HS}}{\sum_{t=1}^{T=25} Q_{HS,t}\cdot(1+r)^{-t}}$$ Equation 44 (Report – Subtask D)

	PVT incl. PV self-consumption **incl. PV costs**	Thermal system (PV-system deducted)
I_0 = Investment costs	31 985 €	21 985 €
C_t= O&M costs	320 €	220 €

T = Service years	25	25
E_t = DHW and heating energy [kwh/a]	8483	8483
AZ6 (SPF) (3.2) or AZ5 (SPF)(2.4)	3.2	2.4
LCoH	28 ct/kWh	25 ct/kWh

Remark: System is not yet optimized, not energetically and not yet on use of PV self-consumption.

Example values for other technologies (based on internet research – whole system costs):

Heat pump deep bore	31 ct / kWh
Heat pump - air heat	24 ct / kWh
Heat pump - surface collector	23 ct / kWh
Oil central heating - calorific value	26 ct / kWh
Oil central heating calorific value	25 ct / kwh
Gas central heating calorific value	18 ct / kWh
Gas central heating calorific value	17 ct / kWh
Firewood central heating	15 ct / kWh
District heating wholesale customer tariff	17 ct / kWh
Central pellet heating	23 ct / kWh
Direct electrical heating	22 ct / kWh

4.5.3 Saved fuel and grid electricity cost

(according to chapter 2.5 of report – Subtask D)

Costs based on calculation, if the whole energy demand would have to be supplied by oil.

Saved **fuel** costs = **9.00 €/a m²** module area

= **282.65 €/a** system

Calculation basis:

saved energy/avoided primary energy	4 948.5	kWh/a	system
	157.6	kWh/a m²	module area
heating oil - calorific value	11.8	kWh/l	
saved fuel	419.4	l/ a	system
	13.4	l/m² a	module area
price for heating oil	67.40	€ /100l	

Saved cost for **electricity** = **47.28 €/a m²** module area

= **1 484.55 €/a** system

(whole energy demand supplied by electricity – 0.30 €/ kwh)

4.5.4 Avoided primary energy depletion

(according to chapter 2.3.5 of report – Subtask D)

Avoided primary energy (based on oil) = **157.6 kwh/a m²** module area

= **4 948.5 kwh/a** system

4.5.5 Avoided global warming impact

(according to chapter 2.4.2 of report – Subtask D)

Avoided primary energy depletion = **44.1 kg CO_2/a m²** module area

= **1 385.6 kg CO_2/a** system

Calculation basis:

saved energy/avoided primary energy	4 948.5 kWh/a system
	157.6 kWh/a m² module area
Example values:	1 litre heating oil ≙ 3.17 kg CO_2
	or 1 kwh ≙ 0.28 kg CO_2

5. Conclusion

Up to date, combined solar thermal heat pump (SHP) systems form a niche in the heat pump markets. The hypothesis is raised that an ineffective integration of the solar collector array forms a market barrier due to increased cost but little-added value, because solar thermal collectors are inactive during the cold periods of a year when heat demand is large. While such inactivity is inevitable in a conventional solar thermal system with a fossil backup heater, the solar array of a SHP system should be active, operating as heat exchanger to ambient during low irradiation conditions and night time. Hence, in order to exploit the full saving potential in SHP systems, it is suggested to substitute other conventional heat pump sources entirely by the solar thermal array. To date, such single source SHP systems typically rely on solar absorbers. The objective of this work was to check the principal meaningfulness and potential of an array of photovoltaic thermal (PVT) collectors in a single source SHP system configuration with a residential heat demand: On the one hand, a PVT collector can provide electricity for the heat pump. On the other hand, the thermal performance is lower compared to a solar thermal absorber.

Results suggest that the single source SHP with PVT collector array is a promising heat supply concept for residential buildings. An extreme case with an array of rear-insulated PVT collectors was investigated. A brine/water heat pump with extended temperature range on the evaporator side and deactivated resistance heaters as well as two-zone combi storage form the other components of the SHP system configuration. The analysis of cold winter test days reveals that insignificant icing occurred on the surface of the collectors and heat demand was met completely. Temperature comfort levels were also met for space heating and with few limitations for domestic hot water delivery. Improper system design was detected and discussed and concrete optimization potential measures identified. The most obvious measure is to replace the solar array with rear non-insulated PVT collectors. A significant temperature increase of source temperatures can be expected. The main aspects are:

- Wider source temperature range down to – 20°C incl. condensation and ice has a big potential
- **PVT Single Source Heat Pump concepts can work!**
- New system concept was designed by project partner PA-ID
- Position of backup heater on the demand side of the fresh water station is recommended
- 2-3 mm max. ice thickness on the front of the PVT-modules during test period
- Key performance indicators for the system where calculated according IEA TASK60 formulas

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