Quasi-Dynamic Testing of Air-Brine-Collectors and Numerical Simulations of a Cold District Heating Network

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Abstract

So-called air-brine-collectors increasingly available on the market are mostly uncovered absorbers without thermal insulation. Hence, their thermal performance is significantly affected by ambient conditions such as wind speed, sky and air temperature. In order to model the thermal performance of air-brine-collectors, it is necessary to know how the above-mentioned ambient conditions affect their thermal performance.

For this purpose, the thermal performance of an air-brine-collector available on the market was determined in two variants according to ISO 9806 by means of outdoor testing using the quasi-dynamic test method. The characteristic values resulting from the measurements are then used within an annual system simulation of a typical cold district heating network, also called fifth generation district heating and cooling (5GDHC) network, in the simulation environment TRNSYS. Whereas the air-brine-collectors in combination with an ice store act as heat sources for decentralized heat pumps in the heating and cooling system.

Keywords: air-brine-collector, quasi-dynamic test method, heat pumps, cold district heating, 5GDHC, TRNSYS

1. Introduction

Since the 1970s, research and development has been carried out worldwide on various types of covered and uncovered photovoltaic thermal (PVT) collectors. In addition, heat pumps have been understood for a few years as one of the most promising technologies for future decarbonized energy supply. Therefore, for the last 10 years about, a trend has been developing towards the increased investigation of PVT collectors in system combination with heat pumps (Kumar et al., 2015) (Al-Waeli et al., 2016). Thus, the research, development and commercialization of the so-called air-brine-collectors, which serve as the heat source of the heat pump, has also been intensified. Air-brine-collectors are designed both with and without photovoltaic modules as a cover. The air-brine-collectors with photovoltaic modules as a cover also belong to the PVT collectors.

Air-brine-collectors are heat exchangers that extract heat from the ambient air and also convert solar radiation into usable heat. Thus they also function as solar collectors and are ideally suited as heat sources for heat pumps. Especially in heating and cooling systems based on cold district heating networks, also called fifth generation district heating and cooling (5GDHC) networks, but also in single- or multi-family buildings, air-brine-collectors as a heat source for a heat pump are an important and very cost-effective alternative to conventional air source heat pumps (Giovannetti et al., 2018).

The air-brine-collectors increasingly available on the market are mostly without transparent cover, i.e., uncovered and without thermal insulation, which means that their thermal performance is significantly affected by ambient conditions such as wind speed, sky and air temperature (Giovannetti et al., 2018). In order to characterize the thermal behavior of air-brine-collectors it is necessary to know how the above-mentioned ambient conditions affect their thermal performance (Leibfried et al., 2019).

For this purpose, the thermal performance of an air-brine-collector available on the German market was determined in two variants at the Institute for Building Energetics, Thermotechnology and Energy Storage of the University of Stuttgart (IGTE) following ISO 9806 (ISO, 2013) using the quasi-dynamic outdoor test method. Fig. 1 shows the two variants investigated. The uncovered variant 1 (one) is shown on the left and the covered variant 2 (two) on the right. The covered variant represents the case where photovoltaic modules are attached above the air-brine-collector.

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Fig. 1: Illustration of the two variants investigated, left: uncovered (variant 1), right: covered (variant 2)

The characteristic values for the numerical description of the thermal behavior of the air-brine-collectors determined by means of measurements according to ISO 9806 are subsequently used within annual system simulations of a typical 5GDHC network in the simulation environment TRNSYS. Thereby, it was investigated how the thermal performance of the air-brine-collectors influences the performance of the entire heating and cooling system.

The procedure for determining the thermal performance of the air-brine-collectors, the boundary conditions and results of the measurements as well as those of the annual system simulations of the two sun-air-collector variants as heat sources for a cold district heating system are presented in detail in the following.

2. Measurements and characteristic values

The measurements to determine the characteristic values were performed in accordance to ISO 9806. Deviating from the specifications of the standard, measurements without irradiation (nighttime measurement), measurements with an average fluid temperature below the ambient air temperature, and measurements with negative thermal performance, i.e. heat dissipation, were also performed.

The measurements without irradiation and at a mean fluid temperature below the ambient air temperature represent typical operating conditions of air-brine-collectors. The measurements with negative thermal performance are used to determine a set of characteristic values that is also valid within the simulations, if the air-brine-collectors are used for recooling purposes.

Equations (eq. 1) and (eq. 2) show the modeling of the thermal collector performance according to ISO 9806.

$$\frac{\dot{Q}}{A_G} = \eta_{0,b} K_b(\theta_L, \theta_T) G_b + \eta_{0,b} K_d G_d - c_6 u G - c_1(\vartheta_m - \vartheta_a) - c_2(\vartheta_m - \vartheta_a)^2$$
$$- c_3 u(\vartheta_m - \vartheta_a) + c_4(E_L - \sigma T_a^4) - c_5 \frac{d\vartheta_m}{dt} \qquad \text{(eq. 1)}$$

$$K_b(\theta_L, \theta_T) = K_b(\theta_L, 0) \cdot K_b(0, \theta_T) \qquad \text{(eq. 2)}$$

With

A_G	m²	Gross area of collector
c_1	W m^{-2} K^{-1}	Heat transfer coefficient at $(\vartheta_m - \vartheta_a) = 0$
c_2	W m^{-2} K^{-2}	Temperature dependent heat transfer coefficient
c_3	J m^{-3} K^{-1}	Wind speed dependent heat transfer coefficient
c_4	-	Factor for calculating the radiant heat losses in dependency of the sky temperature

c_5	kJ m^{-2} K^{-1}	Effective specific heat capacity of the collector
c_6	s m^{-1}	Coefficient for calculating the wind dependence of the conversion factor
E_L	W m^{-2}	Longwave irradiance (λ > 3 µm)
$\eta_{0,b}$	-	Conversion factor based on beam irradiance
G_b	W m^{-2}	Beam irradiance
G_d	W m^{-2}	Diffuse irradiance
G	W m^{-2}	Total irradiance ($G_b + G_d$)
$K_b(\theta_L,\theta_T)$	-	Incidence angle modifier (IAM) for beam irradiance
K_d	-	Incidence angle modifier (IAM) for diffuse irradiance
$\dot{Q}$	W	Thermal performance extracted from collector
t	s	Time
T_a	K	Ambient air temperature
u	m s^{-1}	Wind speed
ϑ_a	°C	Ambient air temperature
ϑ_m	°C	Mean temperature of heat transfer fluid
θ	°	Angle of incidence of beam irradiance G_b
σ	W m^{-2} K^{-4}	Stefan-Boltzmann constant σ = 5.67 x 10^{-8} W m^{-2} K^{-4}

Tab. 1 shows the determined characteristic values and Tab. 2 and Tab. 3 the incidence angle modifiers of the two tested air-brine-collector variants. For the investigated north-south orientation of the longitudinal axis of the air-brine-collector variants, the incidence angle modifiers for beam irradiance in the longitudinal direction are used to characterize the influence of the annual season and the associated position of the sun on the thermal performance of the air-brine-collector variants. Since the measurements were performed in only two out of twelve months, the longitudinal incidence angle modifiers were calculated analytically from the geometry. The gross area of the air-brine-collector in both cases is 4 m².

Tab. 1: Characteristic values of the two air-brine-collector variants

Characteristic Value	Unit	Variant 1	Variant 2
$\eta_{0,b}$	-	0.75	0.71
c_1	W m^{-2} K^{-1}	47.30	33.50
c_2	W m^{-2} K^{-2}	0.00	0.00
c_3	J m^{-3} K^{-1}	20.20	32.40
c_4	-	0.75	1.00
c_5	kJ m^{-2} K^{-1}	77.49	91.61
c_6	s m^{-1}	0.09	0.04
K_d	-	1.01	0.95

The conversion factors $\eta_{0,b}$ determined for the beam irradiance G_b appear surprisingly high at a first glance. However, if the fact that it is a volumetric absorber and thus the radiation-absorbing area is a multiple of the reference area (gross area) is taken into account, the conversion factor for variant 1 is realistic. Equation (eq. 1) represents for such air-brine-collectors, consisting of volumetric absorbers with opaque cover, not the physical but only the mathematical description of the thermal performance. Thus, by inserting an almost freely selectable combination of incidence angle modifiers and a related conversion factor into equation (eq. 1), a curve fitting is performed to fit the measured thermal performances of the air-brine-collectors at different ambient conditions including different angles of incidents. The conversion factor of variant 2 is therefore not directly comparable with conversion factors of other air-brine-collectors. The very high incidence angle modifiers for beam irradiance in the transverse direction for variant 2 result from the very low incidence angle modifiers in the longitudinal direction due to the cover.

Tab. 2: Incidence angle modifiers for beam irradiance (variant 1)

Angle of Incidence θ in °	**0**	**10**	**20**	**30**	**40**	**50**	**60**	**70**	**80**	**90**
$K_b(\theta_L,0)$ in -	1.00	0.99	0.97	0.94	0.92	0.88	0.8	0.66	0.35	0.00
$K_b(0,\theta_T)$ in -	1.00	1.04	1.08	1.11	1.15	1.37	1.42	1.74	2.47	0.00

Tab. 3: Incidence angle modifiers for beam irradiance (variant 2)

Angle of Incidence θ in °	**0**	**10**	**20**	**30**	**40**	**50**	**60**	**70**	**80**	**90**
$K_b(\theta_L,0)$ in -	0.00	0.00	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.00
$K_b(0,\theta_T)$ in -	0.00	11.00	23.90	28.30	34.50	44.40	83.40	100.00	219.00	0.00

The differences in the heat transfer coefficients c_1 and the wind-dependent heat transfer coefficients c_3 of the two variants can be explained as follows: Due to the cover of variant 2, the upward convection currents are obstructed, resulting in a lower value for c_1. On the other hand, the cover creates a certain tunnel effect which increases the wind speed inside the volumetric absorber compared to the surroundings, resulting in a higher value for c_3 at the same time.

The thermal collector performance is described analytically according to equation (eq. 1), in particular as a function of the temperature difference between the mean fluid temperature and the ambient air temperature. The temperature dependent heat transfer coefficient c_2 is used to account for the nonlinear increase in thermal losses, especially for high temperature differences or high mean fluid temperatures. Due to the low stagnation temperature of approx. 20 K above ambient air temperature air-brine-collectors are operated within a narrow range of temperature difference to the ambient air temperature. For these temperature differences, the behavior of the thermal losses can be described as linear to the temperature differences. Thus, c_2 is set to zero for the air-brine-collectors investigated.

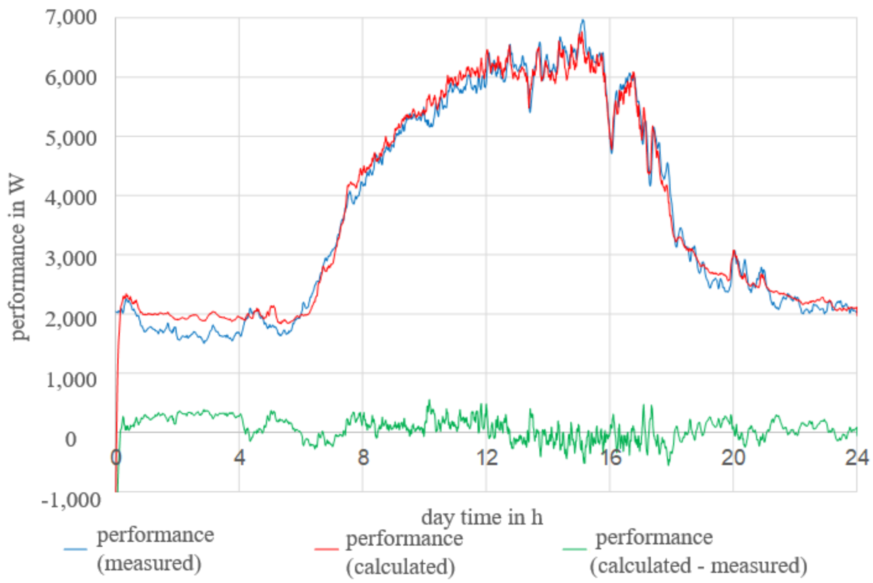

Fig. 2: Measured, calculated and the difference between calculated and measured thermal performance extracted from the air-brine-collector of variant 1 on Aug. 21st, 2020

Fig. 2 shows an example of the measured thermal performance extracted from the air-brine-collector of variant 1 together with the thermal performance calculated using the characteristic values as well as the difference between the calculated and the measured thermal performance for a day used for the evaluation (Aug. 21st, 2020). Fig. 3 shows the irradiance and Fig. 4 shows the wind speed as well as the difference between mean fluid temperature and ambient air temperature for variant 1 on Aug. 21st, 2020.

From Fig. 2, it can be seen that the measured thermal performance extracted from the air-brine-collector can be replicated with good accuracy using the determined characteristic values. However, it should also be mentioned that the heat losses of both variants depend not only on the wind speed, but also on the wind direction, which was not recorded during the measurements and is not taken into account by ISO 9806.

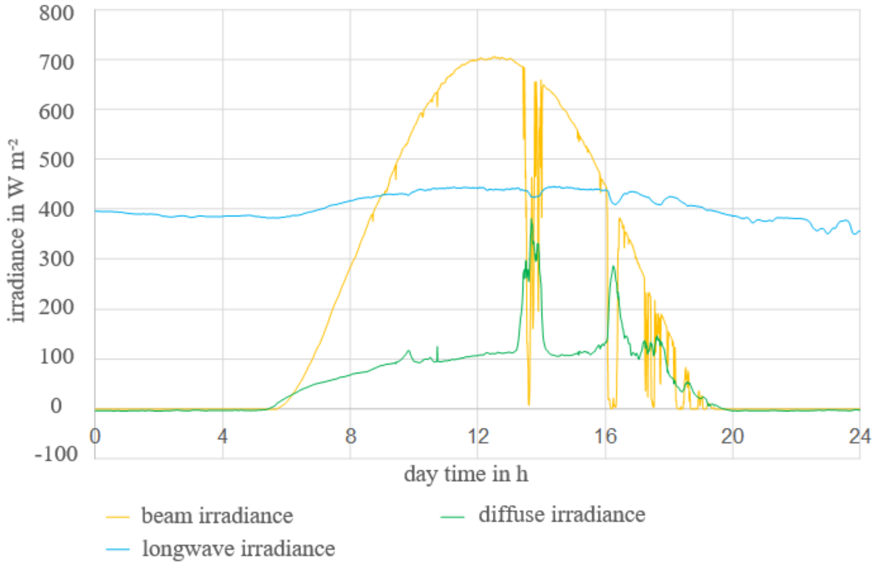

Fig. 3: Irradiance on Aug. 21st, 2020

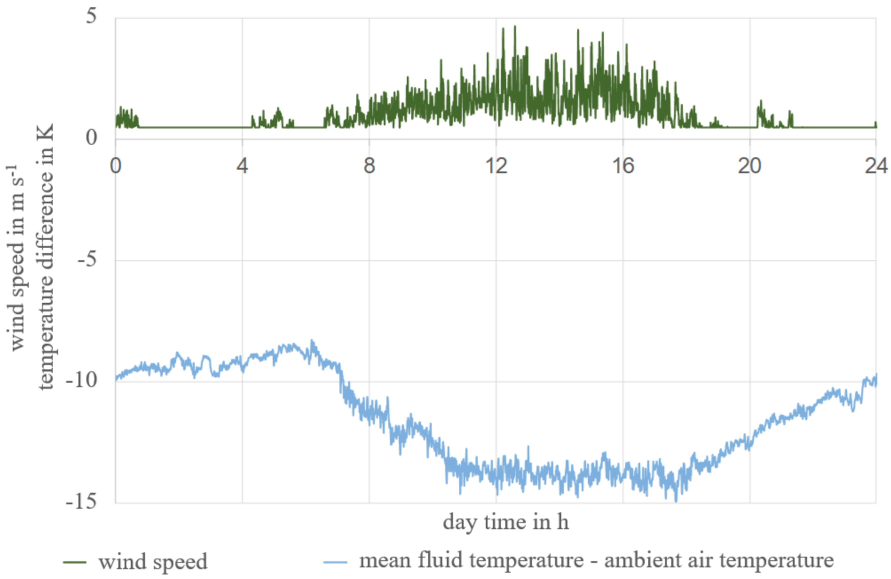

Fig. 4: Wind speed and temperature difference between mean fluid temperature and ambient air temperature for variant 1 on Aug. 21st, 2020

3. System simulations

Using the determined characteristic values, dynamic system simulations with TRNSYS were performed to investigate how the two variants of air-brine-collectors behave in a heating and cooling system and to what extent the two variants differ.

The influence of the two variants on the heating and cooling system is characterized by a comparison of the annual environmental and solar thermal useful heat supplied by the respective air-brine-collectors. The environmental and solar useful heat supplied by the air-brine-collectors depend on the operation of the heating and cooling system, in addition to the characteristic values, the incidence angle modifiers and the ambient conditions. Therefore, the boundary conditions and the control strategy of the heating and cooling system are described in the following.

3.1 Boundary conditions

Use case and location

A typical application for the investigated air-brine-collectors is their use as heat source for heat pumps in 5GDHC networks or so-called anergy networks. Therefore, the heating and cooling system of a new building district in Ludwigsburg, a city in the south of Germany near Stuttgart, is modeled. Within the framework of the project 'Development of integrated solar supply concepts for climate-neutral buildings for the "city of the future" (Sol4City)' the described 5GDHC system is to be investigated, analyzed and optimized.

The new building district comprises nine multi-family buildings and a kindergarten with three residential units above. The buildings are equipped with unheated basements. In total, the district with 107 residential units has a heated usable floor space of 8,567 m². For the design of the system components and the 5GDHC network, a heating demand of 507 MWh a^{-1} and a cooling demand of 166 MWh a^{-1} were assumed.

For the system simulations, a weather data set of the German weather service (DWD) was used for a current average test reference year for the Ludwigsburg site. The annual global radiation sum amounts to 1,048 kWh m^{-2}. The mean ambient air temperature is 10.4 °C and the mean wind speed is 2.9 m s^{-1}.

Heating and cooling concept

The heating and cooling supply of the district is based on a 5GDHC network, which is operated with temperatures between - 10 °C and + 20 °C. A central ice store with a water volume of 770 m³ and a central collector field consisting of air-brine-collectors with a total gross area of 137 m² serve as heat sources and sinks for the decentralized brine-to-water heat pump of each building. The total rated heating capacity of the total of ten heat pumps is 428 kW_{th} at the standard rating conditions according to DIN EN 14511-2 (DIN, 2019). Whereby the standard rating conditions correspond to a brine temperature at the evaporator inlet of 0 °C and a water temperature at the condenser outlet of 35 °C and will be referred to as operating point B0/W35 in the following. The total electrical power consumption of the heat pumps at this operating point is 93 kW_{el}. Each heat pump is hydraulically decoupled from the consumer circuit by a buffer store with a volume of 1.5 m³ each. The cylindrical ice store has two separate heat exchangers - regeneration and discharge heat exchanger - through which a water-glycol mixture is circulated. The storage medium is water.

The heat supply within the buildings is provided by underfloor heating with a constant supply temperature of 35 °C and an assumed constant return temperature of 28 °C. The domestic hot water is generated exclusively by means of electric instantaneous water heaters and is therefore not taken into account in the system simulation.

Additional heat exchangers are also used on a decentralized building-by-building basis to perform building cooling via the 5GDHC network. In the cooling period, the underfloor heating of the buildings serves as the heat source of these decentralized heat exchangers. The buildings are cooled with a constant supply temperature of 17 °C on the consumer side and a corresponding assumed return temperature of 20 °C. In the passive cooling mode, the so-called natural cooling (NC) mode, the ice store represents the heat sink of the 5GDHC network. For the possibility of an active cooling, a second additional heat exchanger, the so-called residual heat exchanger, is installed in the building no.10. An active cooling (AC) mode is achieved by the heat pump in building no. 10 using the air-brine-collectors as heat sink on the condenser side. The cooling energy thus generated on the evaporator side is fed from there into the 5GDHC network. Buildings no. 1 to no. 9 do not have this second additional heat exchanger, but are supplied with additional cooling energy via the 5GDHC network using the decentralized building-by-building heat exchangers described above. The active cooling mode is used exclusively when the natural cooling mode is insufficient.

For simplification, not all possible operating modes are represented in the system simulations described here. Only the heat supply modes and the natural cooling mode of the new building district in Ludwigsburg are considered. The simplified hydraulic structure of the modeled system as well as the initial values for the annual system simulations can be seen in Fig. 5.

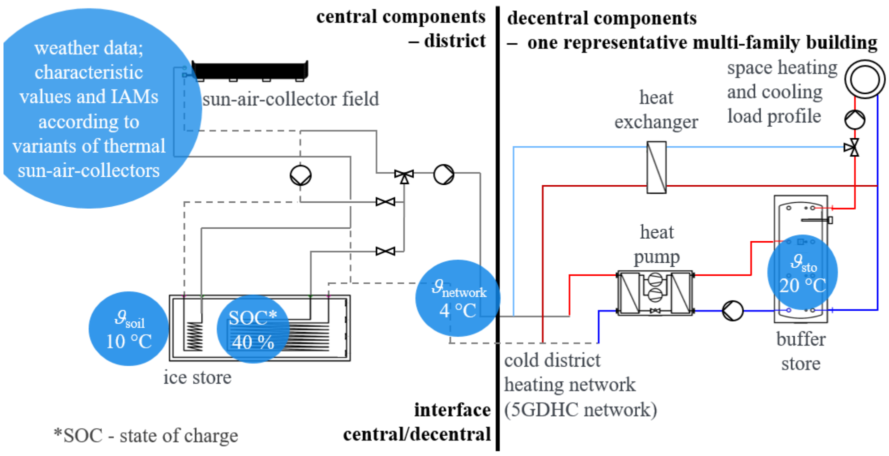

Fig. 5: Simplified hydraulic scheme of the heating and cooling system with initial values for the annual system simulations

Assumptions
To further simplify the simulation model, only one of the ten buildings is modeled. For this purpose, a representative space heating and cooling load profile of one characteristic building of the district is used.

Due to the difference in thermal loads of the building no. 10 with a kindergarten, a non-residential building, and of the residential buildings no. 1 to no. 9, dynamic building simulations in TRNSYS of both one of building no. 10 and one of building no. 1 were performed. Based on target room air temperatures of 20 °C in winter and 26 °C in summer, the hourly heating and cooling capacities that must be supplied to the respective building in order to achieve the target room air temperature were determined. These hourly heating and cooling capacities represent the respective space heating and cooling load profile. To create the representative space heating and cooling load profile, the load profile of building no. 10 with kindergarten was summed with that of nine residential buildings and then divided by the factor of ten for the number of buildings of the district. The factor of ten was chosen because the new building district in Ludwigsburg consists of ten buildings, each with a heating center. In this way, the dynamic thermal behavior of the components decentralized in the respective heating center, such as buffer stores, heat pumps and heat exchangers, can also be investigated. At the "central/decentral" interface (see Fig. 5), the mass flow rate is converted accordingly. Which means that the mass flow rate from the network to the individual building is divided by the factor of ten and from the individual building back to the network is multiplied by the factor of ten. The supply and return temperatures are not changed or converted as all buildings are connected in parallel. Decentralized components are designed for the representative space heating and cooling load profile, centralized components for the entire district.

Simultaneity effects are neglected, since only the heat supply for space heating and not that for domestic hot water is accounted for. Thus, it is assumed that the heat load peaks of the individual buildings always occur at the same time. The heat losses or gains of the pipes of the 5GDHC network are neglected. This can be justified due to an existing thermal insulation and the low network temperatures, respectively the small temperature differences between the fluid in the pipes and the surrounding soil. For the ice store, the heat exchange with the soil is taken into account and for the buffer stores heat losses to the surrounding air are considered.

For both the heating and cooling supply, a variable consumer-side mass flow rate results from the definition of the target temperatures at the consumers, in order to be able to meet the representative space heating and cooling load profile that changes over time. On the network side, a constant mass flow rate is assumed in accordance with the planning. In the 5GDHC network, a water-glycol mixture (33 %) is also assumed as the heat transfer medium. In the consumer circuit, the heat transfer medium is water.

Furthermore, in order to avoid condensate formation, a minimum inlet temperature of 15 °C is aimed for on the network side of the building-by-building heat exchangers during the cooling period. Since the network temperature in natural cooling mode (NC mode) corresponds to the outlet temperature of the ice store, the required heat exchangers inlet temperature in this modeling is often only achieved by mixing the outlet temperature of the ice store with the building-by-building heat exchangers outlet temperature on the network side. If the ice store outlet temperature is too high, the building-by-building heat exchangers inlet temperature is also too high. For this reason, it is assumed that the cooling demand can only be met up to an average ice store temperature of 16 °C. For simplicity, the described possibility of active cooling mode via the residual heat exchanger is not considered at this point.

During the heating period, a controller ensures that the temperature in the decentralized buffer stores is maintained between 40 °C and 46 °C in the upper 60 % of the store. If the buffer store temperature falls below this point, the heat pumps are switched on. If the evaporator temperature exceeds - 10 °C, the set heating supply temperature of 35 °C on the consumer side is reached. If the evaporator temperature falls below - 10 °C and the buffer store temperature falls below the target heating supply temperature of 35 °C, the heating demand can no longer be met in the simulation. In the real heating and cooling system, electrical heating elements are then used as emergency heaters.

The TRNSYS Type 832 is used to model both variants of the air-brine-collectors, covered and uncovered. This Type uses the collector model also used for the quasi-dynamic outdoor measurements according to ISO 9806 (ISO, 2013).

System control

A total of seven operating modes will be available in the new building district in Ludwigsburg. For the sake of simplicity, however, only four of them are modeled here: discharging mode, absorber-direct mode, regeneration mode and natural cooling mode.

It is assumed, that the air-brine-collectors are available for the heat pumps as a heat source in the operating mode **absorber-direct mode** between November 15th and March 15th of each year, when the collector outlet temperature is above - 4 °C. If this temperature is below, the ice store serves as the heat source to the heat pumps during this period until the collector outlet temperature is either above the ice store outlet temperature or the ice store is already consisting to 90 % of ice. During this period of the year, the absorber-direct mode is the prioritized operating mode of the system simulations under consideration. This is to ensure that the heat pumps are supplied with the highest possible evaporator inlet temperature and thus operate as efficient as possible.

Furthermore, the ice store may also be regenerated during this period. For this purpose, heat is transferred from the air-brine-collectors to the ice store when the collector outlet temperature is above the average ice store temperature and the ice store is at least consisting of 50 % ice and no absorber-direct mode is required. This operating mode is named **regeneration mode.**

From March 16th of each year until the start of the cooling period on April 30th the air-brine-collectors are only available as a heat source if no more heat can be extracted from the ice store. This is the case when it is iced up to a maximum of 90 %. During this period, **discharging mode** is prioritized and no regeneration mode may take place. The reason for this is a desired targeted maximum icing of the ice store before the start of the cooling period, in order to provide as much cooling energy as possible by the ice store in the summer months. After the end of the cooling period on September 30th until November 14th the ice store is also initially preferred as a heat source in order to additionally utilize the sensible heat of the completely liquid storage medium of the ice store, which can reach up to 16 °C.

During cooling in the summer months, the extracted space heat of the district is first dissipated into the 5GDHC network via the building-by-building heat exchangers. For building cooling with the **natural cooling mode**, the cold district network is then cooled by using the ice store as a heat sink. As a result, the storage medium melts. Once the ice store has reached a completely liquid state and a temperature of 16 °C, the active cooling mode should theoretically take place. However, as already explained, this mode of operation has not yet been implemented in the simulation. Fig. 6 shows the simplified system control described here schematically in relation to the representative hourly space heating and cooling load profile used.

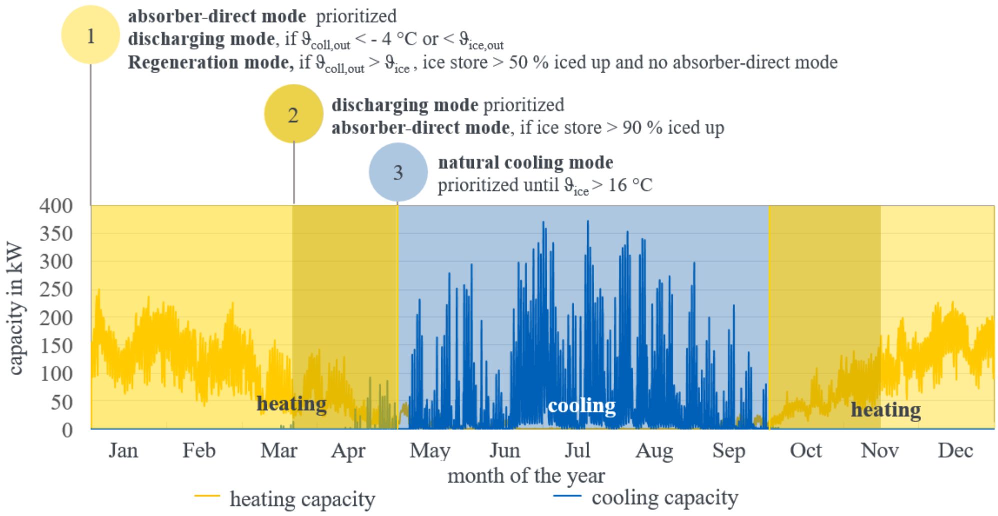

Fig. 6: Schematic representation of the operating modes of the system control with reference to the representative space heating and cooling load profile

The system control is based on so-called "hard switchover points" between heating and cooling operation, i.e. from a fixed date onwards, the system switches between heating and cooling operation. Between May 1st and September 30th, only cooling is possible. During the rest of the year, only heating is possible. It is assumed that heating demands that occur in the cooling period and cooling energy demands that occur in the heating period are negligible. These are therefore not covered by the heating and cooling system. The switching points can vary from year to year depending on the weather data.

3.2 Results

In order to achieve a full coverage of the heating and cooling demand of the district, the cooling demand must be met within the specified summer period and the heating demand must be met within the specified winter period due to the hard switching points. The composition of the monthly supply of thermal energy to meet the monthly heating and cooling demands of the described heating and cooling system and the corresponding monthly heating and cooling demands of the district are shown in Fig. 7 and Fig. 8. The heat losses of the buffer stores are included in the heating demand of the district. The simulations of the heating and cooling system were carried out for variant 1 and variant 2 of the air-brine-collectors (see section 2).

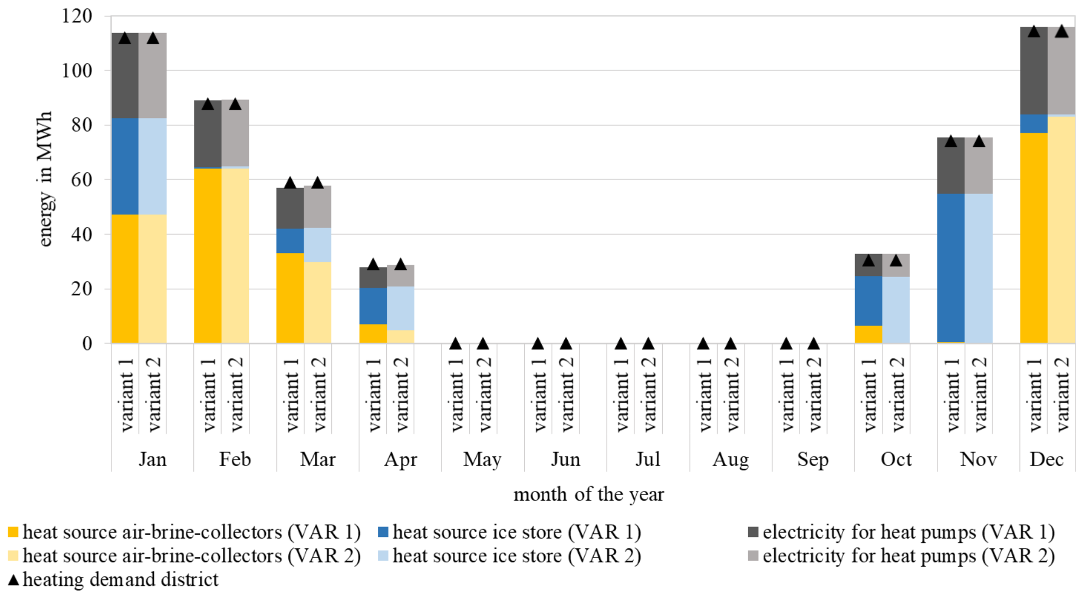

Fig. 7: Monthly heating demand and composition of the heating supply for variants 1 and 2 of the air-brine-collectors

Fig. 7 shows that the heating demand of the district can be completely covered by the previously described heating and cooling system with an assumed initial icing of the ice store of 40 % and the described system control for both variants of the air-brine-collectors. The annual operation of the 5GDHC system does not mean that the ice store must be 40 % iced again at the end of December. Thus, for both variants of the air-brine collector with the implemented simplified control, a level of icing of 90 % is reached at the end of December. In order to cover the heating demand, the control system would have to be adjusted accordingly in real operation. At low ambient air temperatures and low solar irradiation, the heating and cooling system relies on the heat of the ice store, especially in the month of January, even though the absorber-direct mode is actually prioritized. In February, on the other hand, the collector outlet temperature is predominantly either above a temperature of - 4 °C or above the outlet temperature of the ice store, so that in the majority of time the absorber-direct mode, which is prioritized in this month, can be used. Contrary to the prioritization of the discharging mode in April (cf. Fig. 6), the absorber-direct mode is required in April, as the ice store is already at least consisting of 80 % ice at the beginning of April. This high ice content may be caused by the intensive use of the discharging mode in January. The conditions for the regeneration mode are not reached at any time of the simulations, which is why the ice store is not regenerated by the air-brine-collectors.

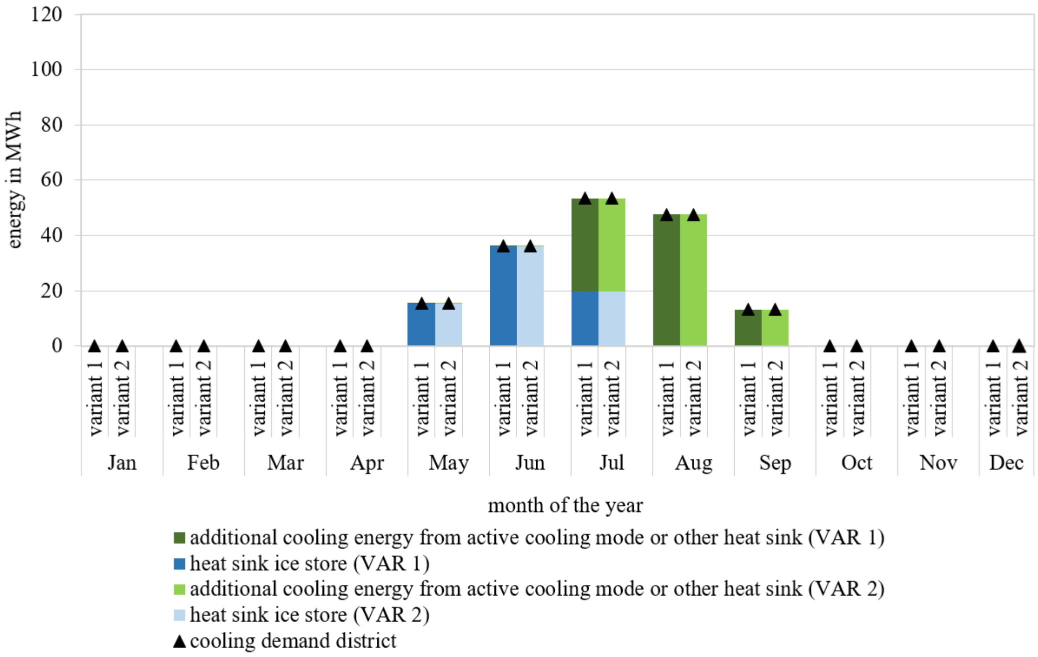

Fig. 8: Monthly cooling demand and composition of cooling supply for variants 1 and 2 of the air-brine-collectors

From Fig. 8, it can be estimated that approx. 43 % of the annual cooling demand for both variants can be met by the natural cooling mode. Thus, for the months of May and June, the ice store provides sufficient thermal energy for cooling the buildings completely. In July, 37 % of the cooling demand can still be covered by the natural cooling mode. In the remaining cooling period, an additional active cooling mode is required, since the ice store is completely liquid from mid-July onwards and has reached an average temperature of 16 °C. As a result, heat is available again from mid-July to cover the heating demand with the discharging mode, but this heat is not required until October 1st. Thus, the ice store remains completely liquid until October 1st and cannot be further used for building cooling.

With variant 2 of the air-brine-collectors 45 % of the annual heating demand is covered by the air-brine-collectors as the heat source of the heat pumps. The ice store supplies 28 % of the annual heating demand. 27 % of the annual heating demand is generated by the electrical energy required for the operation of the heat pumps. It is noticeable that the environmental and solar thermal heat source equipped with variant 1 of the air-brine-collectors, compared to that equipped with variant 2, covers the annual heating demand by 1.2 percentage points more, which means that the ice store heat source is used somewhat less.

If the annual environmental and solar thermal useful heat supplied by variant 2 of the air-brine-collectors is used as a reference, the annual environmental and solar thermal useful heat supplied by variant 1 is only 2.6 % (relative) higher. The deviation in the individual months of the heating period can be either positive or negative, as shown in Fig. 9. Overall, however, it can be concluded from the simulation results that in the considered application, assumptions and simplifications within the given model and measurement uncertainties, there is no significant difference between the two investigated variants of the air-brine-collectors.

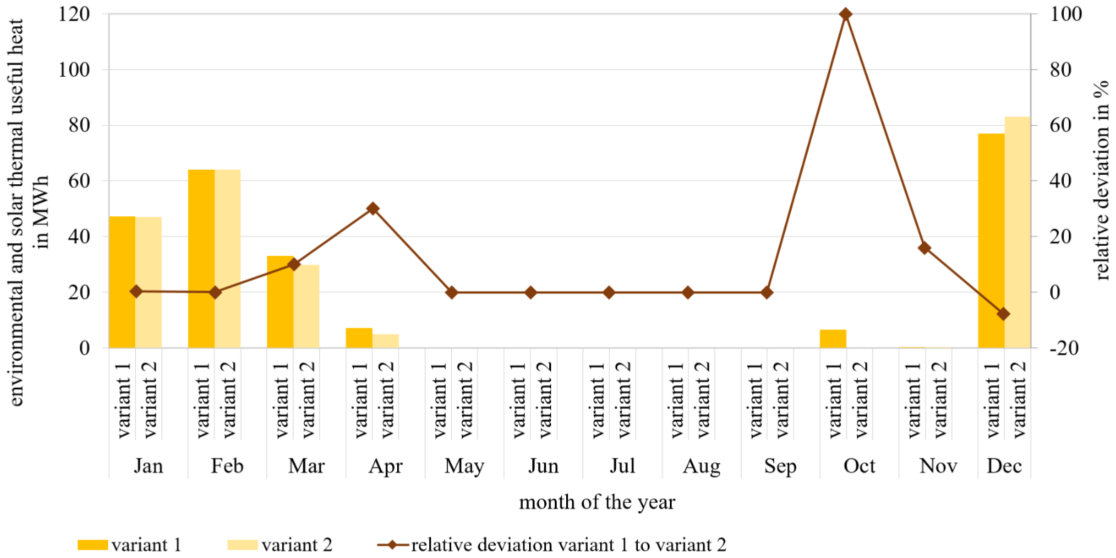

Fig. 9: Monthly environmental and solar thermal useful heat of the two air-brine-collector variants and associated relative deviation

Since the regeneration mode does not occur in the system simulations carried out, the environmental and solar thermal useful heat of the air-brine-collectors shown in Fig. 9 is exclusively the heat used at the evaporator of the heat pumps, which is provided by the air-brine-collectors in the absorber-direct mode. So in this concept, the environmental and solar thermal useful heat of the air-brine-collectors is at such a low temperature level that it cannot be used directly to heat the district.

4. Conclusion and outlook

According to ISO 9806 the characteristic values of two variants of air-brine-collectors were determined by means of outdoor testing using the quasi-dynamic test method. Due to measurements with an average fluid temperature below the ambient air temperature, the two sets of characteristic values are suitable for typical operating conditions of air-brine-collectors in a heating and cooling system combined with an ice store. The resulting characteristic values from the measurements were used for the numerical description of the thermal behavior of the air-brine-collectors within annual system simulations of a heating and cooling system.

The results of the system simulations carried out so far show that for the application studied, almost half of the required heating demand can be provided by the air-brine-collectors as heat source for the evaporator of the heat pumps. The difference in the annual environmental and solar thermal useful heat supplied between the sun-air-collectors of variant 1, i.e. without a cover, and that of variant 2, i.e. with a cover, is less than three percent (relative). The differences of the two variants of the air-brine-collectors related to their thermal performance and thus their influence on the heating and cooling system under consideration are within the given model and measurement uncertainty and thus negligible. This applies especially in this application of the air-brine-collectors in heating and cooling systems based on cold district heating networks, also called fifth generation district heating and cooling (5GDHC) networks, with an ice store and the resulting utilization period of the air-brine-collectors.

With regard to the simulation results it should be noted, that the characteristic values and incidence angle modifiers were determined for a single air-brine-collector under outdoor conditions. Placing multiple air-brine-collectors side by side can further affect both, convective and solar gains. Therefore, in further considerations, system simulations should be performed taking into account the arrangement as a field. The influence of the arrangement of multiple air-brine-collectors as a collector field on the characteristic values required for the system simulations must be investigated in more detail. It is still unclear whether a measurement of a collector field is necessary to model the effects of field arrangement or whether the effects can be described analytically in connection with a measurement of a single air-brine-collector. Accordingly, it remains to be analyzed whether new measurements are necessary for the modeling of an air-brine-collector field, which would result in new characteristic values, or whether a correction of the characteristic values presented here is possible on the basis of an analytical description of the effects of field arrangement. Further, it can be assumed that the effects of a field arrangement influence the convective and solar gains for the air-brine-collectors with a cover, such as variant 2, more than the convective and solar gains for variant 1 without a cover.

Furthermore, it can be expected that the dimensioning of the central components, i.e. the volume of the ice store as well as the number of air-brine-collectors respectively the total resulting gross area, has a significantly larger influence on the composition of the heating and cooling supply than the difference between the two investigated variants of the air-brine-collectors. Within the framework of the aforementioned project "Sol4City", this assumption will be investigated by means of further system simulations.

Additionally, heating and cooling supply concepts like the one described, but without ice store, represent another typical application area of the investigated air-brine-collectors and hence will be investigated as well.

The influence of other system control strategies on the environmental and solar thermal useful heat in such a system will also be investigated in further system simulations.

The aim of the measurements and system simulations carried out is to derive optimization potentials and improved control strategies for both, the solar heating and cooling system and the individual components, in order to realize the most energy-efficient and cost-effective heating and cooling supply possible.

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