Techno-economic analysis of an exhaust air heat pump system assisted by unglazed transpired solar collectors in a Swedish residential cluster

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ABSTRACT

Solar heating technologies hold a significant potential to supplement or replace the fossil fuel-driven heating systems in residential and industrial applications. This paper presents a techno-economic study aiming to assess the use of Unglazed Transpired Solar Collectors (UTSC) coupled with an energy system assisted by Exhaust Air Heat Pump (EAHP) in cold climates applied to a residential building cluster. The performance of the system and its components is assessed for different sizes of solar collector field. In addition, a rule-based algorithm is developed to manage the airflow into the UTSC, and a comparative analysis is carried out with conventional flow control. The existing EAHP assisted energy system of a multifamily building cluster in Sweden is modelled by using a simulation software TRNSYS, and the effects of the UTSC integration on the performances of the energy system are evaluated. Results show that the integration of UTSC has a small but positive impact on the overall system performance. Moreover, the developed control based on the variation of the collector airflow rate for UTSC is an effective control strategy to increase the seasonal performance factor of the overall system and to maximize the savings.

1. Introduction

The efforts for the decarbonisation of the global energy supply is primarily focused on increasing the share of renewables in the electricity generation sector by using solar photovoltaic, wind turbines, biomass, etc. As reported by the international energy agency (IEA), however, heat accounts for almost 50% of the final energy consumption and contributes to roughly 40% of the global carbon dioxide emissions (IEA, 2019).

Nonetheless, fossil fuels remain a prominent source for the global heat supply, whereas heat from solar thermal technologies is used to meet only 1.5% of the global heat demand by 2018 (Holeczek, 2014). Solar driven technologies for heating applications hold a significant potential to replace the fossil fuel-driven heating systems in residential and industrial applications. The use of solar thermal collectors for pre-heating of the ventilation air in residential and industrial buildings is studied and demonstrated in IEA SHC Task 14 on “advanced active solar energy systems”, with results well documented in the related deliverables (IEA, 1997). The global installed thermal capacity of solar air heating collectors reached 1.1 GW in 2018 with a total cumulative collector area of 1.6 million m² (Weiss and Spörk-Dür, 2018).

Solar air heating collectors capture the solar radiation using spectrally sensitive absorber surface and transfer the collected heat to a cold air stream using different techniques. The Unglazed Transpired Solar Collector (UTSC) is a type of solar air collector that makes use of a perforated absorber plate with high solar absorptivity coating to absorb the radiation and transfer the heat to the air stream (Hall et al., 2011). UTSC is made of a corrugated metallic sheet with many small holes on the absorber which allows the cold ambient air to pass through them. The cold air exchanges heat with the absorber, and the heated air is delivered to the load via ducting. The collector is usually mounted on the building walls with a gap of around 50–150 mm between the perforated absorber and the wall to form a plenum. The passage of ambient air in the transpired solar collector and the connection ducts causes an additional pressure drop that is usually compensated by a fan. The inlet air from the collector perforations and

Abbreviations: UTSC, Unglazed transpired solar collector; EAHP, Exhaust air heat pump; HP, Heat pump; ASHP, Air source heat pump; COP, Coefficient of performance; SCOP, Seasonal coefficient of performance; SPF, Seasonal performance factor; DHW, Domestic hot water; SH, Space heating; KPI, Key performance indicator.

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plenum volume is withdrawn by a fan unit and is delivered using air ducts, as shown in Fig. 1. In case the temperature of the air is higher than the setpoint, a bypass damper can be used to mix the ambient air at a lower temperature with the solar pre-heated air to reach the setpoint.

1.1. Development and performance of UTSC

The research efforts for the development of UTSC started in the late 1980s when national renewable energy laboratory, USA started investigating the idea of perforated absorbers for low-temperature air heating applications. The efforts were focused to develop a collector for heating of fresh air with high collector efficiency and simplistic design and installation (IEA, 1997). Few prototypes were tested, and the concept was further optimized under the auspices of the solar heating expert group to use these collectors for pre-heating the ventilation air of buildings. A preliminary investigation on a laboratory scale setup of these collectors reported thermal efficiency of up to 84% and a small effect of wind speed on collector efficiency for suction velocities greater than 0.05 m/s (Kutscher et al., 1993). The success resulted in a further intensification of the research efforts aimed to understand the reason for the higher efficiencies by looking into the heat loss mechanisms. Kutscher developed a simple analytical model to calculate the collector heat losses and define the overall collector efficiency equation (Kutscher et al., 1993). A numerical model was further developed to estimate the performance of the collectors under various boundary conditions. The studies concluded that the higher efficiency of the collector is due to lower convection losses. As air flows over the collector surface, the boundary layer approaches an asymptotic thickness, and all the heat in the boundary layer is removed convectively by the suction air and is delivered to the load. Therefore, the low pressure caused by the fan behind the collector surface carries the boundary layer resulting in minimum convection losses and higher efficiency. The model that was developed estimated a steady-state thermal efficiency of around 80% for a collector area of 9 m², global irradiance of 700 W/m², wind speed of 5 m/s, ambient temperature of 15 °C, and airflow rate of 0.05 kg/(m²s).

In the study, it was clearly emphasized that collector design and threshold operating conditions (suction velocities, absorber porosity, and uniform flow) are very relevant factors to achieve higher collector efficiency. An empirical correlation was developed to evaluate the heat exchange effectiveness of UTSCs. The experimental investigation was carried in a wind tunnel and concluded that the effectiveness of the collector is majorly influenced by the suction velocity of the air, the crosswind speed over the collector, and the pitch and diameters of the holes (Kutscher, 1994). A similar study by (Van Decker et al., 2001) identified that solar absorptance, hole pitch, and airflow rate have major influence factor on the heat exchange effectiveness. A drop of 11.5% in effectiveness was observed when the collector pitch was increased from 12 to 24 mm. Other factors such as absorber porosity and hole diameter have small to moderate effects on collector performance. A breakdown of the heat transfer components revealed that almost 62% of the air temperature rise occurs due to the heat exchange with the front surface of the absorber, 28% when air passes through the holes, and 10% on the back surface of the absorber.

The commercialisation of UTSC started in the 1990s when few companies started offering UTSC in various configurations. These products are widely studied, and the performance in the real fields is reported by several researchers (Bandara et al., 2018; Croitoru et al., 2016; Gawlik et al., 2005). Fleck (Fleck et al., 2002) conducted a long-term performance measurement on a 63 m² UTSC installation for Canadian climates. The study intended to understand the effect of local meteorology (especially wind turbulence) on collector performance. Compared to model predicted efficiency, real measurement shows a deviation of up to 60% and it was attributed to the wind turbulence effect. Later on, it was identified that the installation does not fulfill the minimum criteria required to obtained good thermal performance (Kutscher et al., 2003). The suction velocity for the tested system was half of the minimum recommended value, which resulted in sub-optimal pressure drops and made it highly sensitive to local wind variations. These studies further strengthen the importance of well-designed operational conditions for these collectors. The main application for UTSCs remains pre-heating of building ventilation air, but several alternative systems designed were also studied for agricultural and industrial drying processes (Cordeau and Barrington, 2011; Eryener and Akhan, 2016; Hollick, 1999). Currently, most installations are concentrated in the USA, Canada, and U.K and only a limited number of installations are found in the rest of Europe, and no reported installations for Nordic countries.

1.2. Heat pump integration with UTSC

Concerning heat pumps (HP), this technology is nowadays widely used to meet space heating (SH) and domestic hot water (DHW)
demands of the buildings (David et al., 2017; Mohanraj et al., 2018; James et al., 2021). More specifically, air-source heat pumps (ASHP) represent the most commonly used HP technology in Sweden for the residential sector (Poppi et al., 2016). The performance of ASHPs operating in heating mode is negatively affected by a decrease in ambient air temperature, assuming all other relevant parameters (e.g., load temperature, operational capacity) unchanged (Wang et al., 2020). Therefore, the use of preheated ambient air has the potential to increase the coefficient of performance (COP) of the heat pump and reduce its operational electricity consumption in heating mode. In Sweden, exhaust air heat pumps (EAHP), which are a specific typology of ASHP, have lately evolved as an energy efficient option, as they enable to recover the heat from ventilation air and utilize it at the source side of the HP to improve its COP (EHPA, 2009). In a building with EAHP, the ventilation system is designed to extract air from the various building zones and convey the exhaust air volume to a central point through ductwork, so that it can be used by the EAHP as a heat source and then expelled to the outdoor. Sweden is the biggest market for EAHPs in Europe, with more than 17,000 EAHP units sold until 2017, mostly in newly built single-family houses (EHPA, 2009).

Previous projects in Sweden have shown that EAHPs can complement the district heating system in a cost-effective way. EAHP is particularly of interest in buildings that already have centralized heat and hot water distribution and/or exhaust air extraction (Gustafsson et al., 2017). Therefore, EAHP is well-suited option to utilise the exhaust air from building zones, without any requirement of the central air supply system. Other than EAHP, an alternative way to recover heat from ventilation air can be a heat recovery ventilation system to pre-heat the fresh air. However, the retrofitting of a heat recovery system with mechanical ventilation requires a central air supply network and can incur an additional cost. From the energy perspective, if the building envelope is not very air-tight, recovering heat via EAHP is more effective than through ventilation heat recovery, since waste heat can be recovered also from air infiltrations. The increasing popularity of EAHP in Sweden can be proven by looking into the sales growth rate of these HPs, which is more than 20% since 2014. Therefore, it is important to understand the potential utilisation of solar heating collectors with an energy system driven by EAHP.

2. Research gaps and objectives

Previous studies have shown that significant energy savings can be achieved by solar heat pump systems. For example, (Safijahanshahi and Salmanzadeh, 2019) simulated the performance of an HP coupled to a UTSC, where pre-heated air from the solar collector is used as a sole source to increase the brine temperature in HP evaporator.

The results showed that the solar-assisted heat pump could decrease electricity consumption and the CO2 generation up to 10% compared to a system without any air pre-heating. The steady-state analysis focuses on component performance without any modelling of building and hydronics system, the dynamics which can easily affect UTSC, and overall system performance. Similarly, (Persioglou et al., 2017) analysed the performance of a UTSC used as a preheater on the condenser side for an ASHP installed in a residential building in Wales, UK. The results showed that the system contributed to 15% of the heating and cooling demand in one year. (Saini, 2019) analysed the effect of air flow rate on UTSC performance, and its impact on ASHP based energy system for Swedish climates. The results show that the variation in the collector airflow rate can increase the savings up to 60% in a UTSC-ASHP system configuration, and thus airflow control can be used as an effective control strategy to improve the collector and overall system performance.

None of the referred studies, however, addresses the integration of a UTSC with an EAHP energy system. Furthermore, the energy system used in most of these studies consider a centralised ventilation air supply and exploit the UTSC for pre-heating the supply air either directly to the room or in the condenser of the heat pump under steady-state conditions. However, in lack of a centralised air supply system (as in the case with EAHP integrated energy systems) the fresh air inlet is provided by trickle vents installed in the envelope. Therefore, pre-heating with UTSCs can be challenging due to the need for multiple small collector modules, and an intervention that is not likely to be easily replicable or cost-effective (Hall and Blower, 2016). This triggers the need for effective utilisation of UTSC for EAHP system, other than air pre-heating. Besides, the use of UTSCs in Nordic climates is not an issue thoroughly addressed and relevant parameters for system design and operation remain unclear.

To foster the design of innovative solutions that exploit locally available renewable energy sources through technologies already on the market, this paper aims to assess the performance of solar technology as UTSCs coupled with an EAHP system for Swedish climate. This paper analyses an alternative integration scheme for UTSCs, where the pre-heated air is supplied to the source side of an EAHP in addition to the exhaust air recovered from buildings. In the proposed system, warm air from the UTSC is mixed with the exhaust air recovered from buildings and is used to increase the inlet brine temperature of the heat pump using an air-to-brine heat exchanger (HX), with the expectation that higher air volumes and temperature will result in an increased HP capacity and better COP, and thus into a reduction of the electricity consumption. The specific contributions of the study are:

1) Understand the energy gains of UTSC in EAHP assisted energy system through numerical simulation and energy analysis for multifamily building cluster in Sweden;
2) Development and assessment of a rule-based algorithm to manage the airflow into the UTSC, and comparison with the conventional flow controls;
3) Sensitivity analysis to understand the effect of the collector area on the proposed UTSC-EAHP system performance.
4) Economic analysis to understand the profitability of UTSC-EAHP based systems.

To achieve these goals, the EAHP based energy system of an existing multifamily building cluster in Sweden is modelled by using a simulation software TRNSYS and the effects of the UTSCs integration on the performances of the energy system are evaluated through numerical simulations. The paper is structured as follows: Section 3 and 4 illustrate the research methodology and the boundary conditions of this study. Afterward, the numerical results of the energy simulations and the findings of this study are reported and discussed in sections 5, 6, and 7. The paper concludes by highlighting the main results and giving an outlook on future work.

3. Research methodology

This study aims to assess the energy and economic performances of an innovative energy system constituted of a UTSC and EAHP in the context of the Swedish climate. The evaluation is carried out through annual energy simulations performed with dynamic simulation software based on a reference EAHP assisted energy system implemented in a real demonstration case. The impact of UTSC integration is further modelled to quantify the energetic and economic benefits. Hereinafter, a systematic approach is followed to provide the reader with the information needed to understand the results. Fig. 2 shows the workflow used in the course of this study.

Initially, a reference real case energy system is studied and all the relevant information is gathered to develop a computer model. The TRNSYS simulation tool is used to carry out the energy assessment due to its open-source code (University of Wisconsin, 2011). TRNSYS is a component-based program, where the system calculations are solved sequentially based on values of fixed and time-dependent parameters until convergence criteria are met. Moreover, TRNSYS offers a wide
range of components library for solar heating systems and therefore provides higher flexibility compared to other tools such as Polysun or IDA-ICE (Equa, 2019; Velasolaris, 2009). Annual dynamic simulations of the energy system are performed and the results are expressed using relevant key performance indicators (KPI). The climatic, energy system and economic boundary conditions are chosen for the Borlänge region in Sweden. A sensitivity analysis is carried to study the effect of air flow rate control, and collector area on the system performance.

4. System boundaries and description

4.1. Building and load characteristics

This study considers the thermal load and the EAHP energy system of an existing residential building cluster. The building cluster is located in the central Swedish town of Sunnansjö in the municipality of Ludvika with geographical coordinates of 60.2° N and 14.9° E. An aerial view of the building cluster is shown in Fig. 3.

The building cluster was completed in 1973 as part of the “Miljonprogrammet”, a public housing program implemented in Sweden from the Year 1965 to 1974 (Kartikeya Chhaya, 2018). The complex (three buildings) includes 48 apartments over 2 floors and a basement, for a total gross floor area of 4488 m². The façade and the pitched roof have a gross area of 2146 m² and 1750 m² respectively. The external walls of the buildings are insulated and the estimated U-value is equal to 0.33 W/(m²·K). The roof is made of concrete with rock wool insulation and has an effective U-value of 0.21 W/(m²·K). The windows are the typical Swedish window with two panes, clear glazing, and a wooden frame, with an estimated U value of 2.8 W/(m²·K). The general information on the buildings is reported in Table 1.

The total heating load of the buildings cluster is assessed with a numerical model developed in TRNSYS and compared with real measurements within the framework of the Energy Matching project (Energymatching, 2020). The resulting heating load profile from the project is used for this study. The annual thermal energy demand of the building cluster is 527 MWh, and accounts for SH load (424 MWh or 110 kWh/m²) and DHW load (103 MWh or 27 kWh/m²). The frequency distribution of the total heating load is shown in Fig. 4. The peak heating load of the building cluster is 198 kW and occurs only for a few hours in a year. The annual average thermal load of the building cluster is 60 kW. Fig. 5 shows the monthly thermal load split in SH and DHW contributions. As can be seen, the SH demand shows a great seasonal variation, as most of the demand occurs in months with a low ambient temperature. On the contrary, the DHW demand has a small seasonal variation, as the requirement for hot water is consistent over the year.

Table 1

<table>
<thead>
<tr>
<th>General information about the analyzed building cluster.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Year of construction</td>
</tr>
<tr>
<td>Number of floors in each building</td>
</tr>
<tr>
<td>Floor height</td>
</tr>
<tr>
<td>Number of apartments</td>
</tr>
<tr>
<td>Number of inhabitants</td>
</tr>
<tr>
<td>Housing form</td>
</tr>
<tr>
<td>Facade surface gross area of 3 buildings</td>
</tr>
<tr>
<td>South facade surface gross area</td>
</tr>
<tr>
<td>Roof surface gross area of 3 buildings</td>
</tr>
<tr>
<td>Gross floor area of 3 buildings</td>
</tr>
<tr>
<td>Total heated area of 3 buildings</td>
</tr>
</tbody>
</table>
4.2. Reference energy system description

The buildings have a centralised heating system that supplies both SH and DHW energy to all three buildings using a distribution network through culvert. The energy generators are a centralised EAHP and an auxiliary electric boiler. Each building is equipped with one heat recovery unit in the attic, and the circulation loops from all these are connected to a single HP unit in one of the buildings. The fans of the ventilation system operate continuously in the building’s attic, and the air is extracted from the ventilated spaces. This air is then ducted to the air-brine HX units installed in each building, where it exchanges heat with the brine loop, and then expelled to the atmosphere. The heated brine from each building is pumped to the EAHP through a culvert. Fresh air is drawn into the buildings through the envelope with the use of trickle vents installed in each room. The amount of heat that is extracted from the exhaust air is dependent on the ventilation rate, which is generally fixed at the national requirements for hygienic air and corresponds to 0.35 L/m²⋅s in Sweden (roughly 0.5 air changes per hour for normal ceiling heights) (Garman and Ahmad, 2019). In other words, the capacity of the HP is limited by the volume flow rate of exhaust air recovered from the buildings. The use of a back-up system is then required when the heating load overmatches the capacity of the HP.

4.2.1. EAHP

The EAHP considered in this consists of a heat recovery unit (air-brine HX) and an HP unit. In case the installation involves a single building, as popular in Sweden for single family houses, both heat recovery and HP units can be combined offering the advantage of a compact unit. For multi-building installations, however the heat recovery units are detached from HP and are usually placed in a single building in conjunction with an exhaust ventilation system. The HP evaporator is connected via a brine loop to heat recovery units located in each building. As the evaporator is connected via an external loop, there is flexibility in terms of heat sources that can be connected. HP has a variable speed compressor, and the frequency control of the compressor enables to vary the heat output depending on the thermal load in the buildings. In the existing commercial units, this variable speed compressor derives its power from AC, with an AC/DC converter to first create DC power that can then be used to derive variable frequency AC to the compressor. Several heat pumps can be cascaded together to increase the heating capacity up to 500 kW. The EAHP installed in analysed building cluster has a designed capacity of 45 kW with COP of 4.1. 
defined at compressor frequency of 90 Hz, and brine (source) & water (load) temperatures of 0 °C and 35 °C respectively.

4.2.2. System operation

The ventilation air is extracted at room temperature from all three buildings and is ducted to the heat recovery units. These units recover heat from the ventilation air and deliver this heat to the HP evaporator via a brine circuit. The exhaust air is cooled from 21 °C to 0 °C using cooling coils and expelled into the atmosphere. As shown in Fig. 6, two water storages are used to decouple the energy demand and generation, and a centralized electric boiler is used as a back-up energy generator. The distribution system is water-based and reaches the emission terminals in the dwellings through underground culverts from the centralized system. Hot water circulation is used to ensure that there is only a small delay for the delivery of sufficiently hot DHW in the flats. There are six pipes in the culverts:

a) Two pipes for the SH circuit, that is supply and return to/from the emission terminals.

b) Two pipes for the DHW system, that is supply and hot water return circulation pipes.

c) Two pipes for the brine circulation loop to transfer heat from the heat recovery units to the HP.

The HP operates either in SH or DHW mode depending on the DHW tank charging level. The hot water from the HP outlet is directed either to the SH circuit or to a 2.5 m³ capacity storage tank connected to the DHW system. The DHW circuit is designed to supply 55 °C hot water to the user. The maximum allowable heating temperature of EAHP is 65 °C. The hot water is prepared using an external plate heat exchanger unit in combination with a variable speed pump. The design supply/return temperatures of the SH circuit are 55/45 °C respectively and the heat is delivered to the single dwellings through wall-mounted radiator units. The operation mode of the HP is determined based on the charge level of the DHW tank, with the priority being the full charge of the DHW tank. A small storage vessel of 0.35 m³ is connected to the SH circuit and is used to meet the SH loads when the HP operates in DHW mode. The compressor speed of the HP is at maximum when operating in DHW mode whereas it is varied when operating in SH mode so to achieve the designed radiator inlet temperature. If the heat demand of the building exceeds the HP capacity, an electric boiler is used as an auxiliary heat source. The boiler has a capacity of 250 kW and is designed to meet the peak thermal load of the building. A detailed functional description of the system controls is presented in the next sub-sections.

4.2.3. UTSC

The heating system in the analysed building cluster consists of an EAHP. The heat source for EAHP is the recovered ventilation air from the buildings. As the central aim of this paper is to assess the impact of UTSC integration on such energy system, therefore in the analysed reference system, multiple UTSCs are assumed to be installed on the South facade of 2 buildings, for a total collector’s area equal to 208 m². The Energy flow among the various components of the proposed system can be visualised using a square box diagram as shown in Fig. 7. The ambient air exchanges heat with UTSC, and the heated air from solar collectors is mixed with the extracted building ventilation air and fed to the air/brine HX units of the heat pump. The additional heat and air volume from the solar collector’s field lead to a higher heat transfer capacity of the cooling coils, and an increase of the inlet brine temperature and thus to a higher COP of EAHP. EAHP is further used to meet the SH and DHW demand, with an electric boiler as an auxiliary heating source.

4.3. Meteorological parameters

The meteorological data of Borlänge with coordinates of 60.4°N, 15.4°E is obtained from the Meteonorm database (Meteonorm, 2020). Borlänge is the closest available weather data to the location of the project, and data for the year 2014 is used for the analysis. Global horizontal irradiation (GHI) is the amount of solar energy impinging on a horizontal surface in a particular time frame. The annual GHI for the analysed location is 971 kWh/m². The solar collectors are mounted vertically on the South facade and thus it is relevant to understand the monthly variation of the irradiation levels on a tilted vertical surface, shown in Fig. 8 for the South orientation. Due to the higher latitude of the location, from September to March lower Sun altitude angles are observed, resulting in sun beams that have a lower angle on a vertical surface than on a horizontal surface. Therefore, the tilted irradiation is higher by 188 kWh/m² compared to the GHI during these months. The opposite is true from April to August, and the GHI is higher by 195 kWh/m² compared to the irradiation on the tilted surface. On an annual basis, the annual average global irradiation on the tilted surface is 965 kWh/m², only 0.6% lower than GHI.

The monthly average wind speed at 10 m height and ambient temperature for the location is shown in Fig. 9. The annual average wind speed for the location is 3.3 m/s and shows slight variations throughout the year. On the contrary, the ambient temperature is characterized by significant monthly variation and reaches sub-zero temperatures for five months in a year. The annual average ambient temperature of the location is 4 °C.

![Fig. 6. Overview of the analyzed building energy system.](image-url)
5. System modelling and controls

The energy system presented in section 4 is modelled and simulated in TRNSYS 18 with a timestep of 1 min, with post-process calculations in Excel. A detailed description of the system components models and control is provided in this section.

5.1. DHW circuit

The DHW system is modelled based on the energy system installed in the existing building cluster. A 2500 L hot water tank is hydraulically connected to the EAHP without any HX. The stratification in the tank is controlled by specifying the number of nodes in the tank model. The tank is modelled with 10 temperature nodes using TRNSYS Type 158. An external plate heat exchanger is used for DHW preparation. The primary side of this HX is connected to the DHW tank using a variable speed pump. Cold freshwater enters at the secondary side of the HX after each DHW withdrawal, and at the outlet, the DHW is delivered to the distribution system. The DHW is circulated in the various building zones with a total pipe length of 200 m with a design temperature drop of 5 °C to account for the heat losses through the pipe walls. The DHW setpoint temperature is 55 °C, and the flow rate on the primary side of HX is varied so to reach 60 °C at the outlet of the secondary side. A sinusoidal function representing the cold water temperature ($T_{cw}$) is defined in Equation (1) (Werner Weiss, 2003).
5.2. SH circuit

The SH system of the buildings is water-based and uses radiators as emission terminals. A buffer tank with a capacity of 0.35 m$^3$ (Type 158) is installed in the SH loop to act as an interface for seasonal variation of cold water temperature (3.2 ºC), $t$ represent the hour of the year, and $d_{off}$ is the day of the year with maximum cold water temperature (80). The electricity consumption of the pumps is derived from the component models using a nominal power for the flow rate and pressure drop design conditions. The description of the components used in the DHW circuit sub-model is provided in Table 2.

Table 2
System model parameters for DHW sub-circuit.

<table>
<thead>
<tr>
<th>Component</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hot water tank</td>
<td>TRNSYS type 158, Tank capacity 2500 L, Tank height 2.8 m, Heat loss coefficient (Top/edge/bottom) 1.2/1.2/1.6 W/m$^2$K</td>
</tr>
<tr>
<td>Heat exchanger</td>
<td>TRNSYS type 5, Mode Counter-flow type, Overall heat transfer coefficient 19,200 kJ/hr K</td>
</tr>
<tr>
<td>DHW distribution network</td>
<td>Total pipe length 200 m, Pipe internal diameter 0.056 m, Heat loss coefficient 5.3 W/m$^2$K, Designed temperature drop in DHW network 5 ºC</td>
</tr>
</tbody>
</table>

Fig. 9. Monthly Ambient temperature and wind speed variation for project location.

$$T_{aw} = T_{aw} + \Delta T_{AMP} \cdot \sin \left[ \frac{360\left(t + 24\left(273.75 - d_{off}\right)\right)}{8760} \right]$$ (1)

Where, $T_{aw}$ is the annual average cold water temperature for the location (8.5 ºC), $\Delta T_{AMP}$ represents the average amplitude for seasonal variation of cold water temperature (3.2 ºC), $t$ represent the hour of the year, and $d_{off}$ is the day of the year with maximum cold water temperature (80). The temperature drop across the radiators is dependent on the ambient temperature and is governed by a climatic curve. The radiators are simulated using an equation block with a designed supply /return temperature of 60/40 ºC respectively. The climatic curve of the SH circuit that shows the dependence of the radiator supply/return temperatures on the ambient temperature is reported in Fig. 10. The pipe length of the SH distribution is 100 m with an internal diameter of 0.06 m. The temperature drop across the radiators is varied using a variable speed pump (Type 742) with a designed flow rate of 2000 kg/h. The mass flow of the pump is regulated by a PI controller (Type 22) for each timestep of the simulation. A combination of mixture and shunt valves (Type 11) is used to mix the return and the supply water streams and maintain the setpoint temperature.

5.3. EAHP model

The heat pump is available on the market and is modelled using an empirical black-box model based on a quasi-steady state performance map. The performance map uses compressor frequency, brine flow rate and the inlet temperatures of the heat source and the load as independent variables to determine the heat capacity and the electricity consumption of the HP during system operation. The performance map is derived using an extensive set of steady-state measurements performed by the HP manufacturer over a wide range of inlet conditions for a variable-speed compressor module of 16 kW. The performance map also accounts for the electricity consumption connected to dynamic effects such as icing/defrosting of the evaporator. The heat pump heating capacity is determined using a scale function available in Type 1927. This type interpolates the input sample points using a performance map stored in an external file, to obtain the outputs sample points. The list of dependant variables and input points provided to the performance map is shown in Table 3.

The outputs of the model are the thermal capacity and the COP of HP. The frequency of the heat pump compressor is actively controlled based on the operation mode. The heat recovery unit of EHAP is simulated using Type 508, which represents an air-to-brine HX. Exhaust air from the building at room temperature and solar preheated air enter the HX heating the return brine stream from the EAHP. The maximum limit of outlet brine temperature is set to 20 ºC, and thus the cooling coil is modelled so to bypass a fraction of the incoming air and restrict the brine temperature within the limit recommended by the manufacturer.

5.4. UTSC model

The absorbers of UTSCs are perforated metal sheets with very small holes that resemble conventional metal facades. A black-colour UTSC with absorptivity of 0.95 and emissivity of 0.20 is modelled for this
study. A single installation usually comprises many panels that partially overlap to give a continuous appearance along the entire wall. To add structural strength and rigidity, the material is processed through rollers to form corrugations. In the TRNSYS (type 201) used for this study, the corrugations are 39 mm deep and are spaced 183 mm apart. As an exemplification, the geometry of a single UTSC collector panel is shown in Fig. 11. A gap of 70 mm is provided between the collector and the building wall, and no thermal interaction is considered between them. The gap is necessary to allow the heated air to travel up the wall and reach the nearest fan intake through a connection duct. The fan power is designed to compensate for the pressure drop across UTSC, friction pressure drops in the plenum, and ducting. The fan is simulated using type 147 with a specific power consumption of 16 W/m$^2$ of UTSC, designed to compensate for a total pressure drop of 50 Pa.

Dimensions and characteristics of the perforated panel collector can vary depending on the design requirements and needs of each installation. The galvanization protects the steel from rusting and the air movement through the holes dries any moisture that may exist. As the wall is generally vertical, water runs off the wall and the holes are so small that the surface tension prevents most water from entering the holes. The design of the wall and canopy is based on volumetric airflow rate, cost, and appearance. UTSCs are usually vertically installed on the building façade and their productivity can be limited by unfavourable azimuthal exposures or external shading due to nearby buildings or trees. In this study, however, it is assumed that the surrounding area to the building cluster is flat and there is no external shading on the collectors.

An overview of the energy system model developed in TRNSYS is shown below in Table 4.

### 5.5. System operation and controls

#### 5.5.1. Heat pump control

The HP operates either in SH or DHW mode and is governed by a differential control (Type 165) that prioritizes the charging of the DHW tank, as shown in Fig. 12. The HP runs in DHW mode when the water temperature at the bottom of the DHW tank falls below a set point of 60 °C, with a hysteresis of 6 °C. The compressor of the heat pump works at...
full speed in DHW mode to maximise the heat pump capacity and charge the tank as quickly as possible. An external counter-flow heat exchanger (Type 5) and a variable speed water pump are used for hot water preparation. When the DHW tank is fully charged, the HP switches to SH mode. In SH mode, the compressor speed is actively regulated using a proportion integration differentiation (PID) controller (Type 22) to reach the design supply temperature at the radiator inlet. The supply temperature to the radiator inlet is varied depending on the ambient temperature using a climatic curve. The mass flow variation in the radiators is obtained using a PID controller that generates a control signal based on input parameters and a setpoint temperature. The Auxiliary electric boiler is regulated to maintain the temperature setpoint when the thermal load exceeds the HP capacity. The HP is turned off when the DHW tank is fully charged and the SH demand is zero.

5.5.2. UTSC flow control

As already discussed, the UTSC is mounted vertically on the South facades and is integrated with the EAHP so that the warm pre-heated air can be used as the heat source for HP evaporator in addition to the exhaust ventilation air. This paper aims to compare the energy performance of two different control strategies for the management of the airflow into the UTSC. More specifically, the studied rule-based control algorithms guarantee an (a) variable airflow or (b) a fixed airflow behind the collector surface.

With the first control strategy, a variable speed fan is used in conjunction with a PID controller and an equation block to actively change the airflow rate behind the collectors based on several control variables, which may be time-dependent and fixed input parameters. The control strategy is defined as shown in Fig. 13 and is applied to each time step of the simulation. The top control is based on the HP operational status, i.e., the fan remains OFF when the heat pump is not operating. Moreover, an irradiation threshold is defined to start the fan operation and avoid the use of the UTSC when the irradiation level is low. Based on previous experiences and studies, the irradiation threshold is set to 50 W/m² on the collectors plane [24]. Once the fan is ON, the air mass flow rate \( m \) is calculated to maintain the collector air outlet temperature equal/greater than the exhaust air temperatures \( T_{exh} \) from the buildings, as shown in Equation (2).

\[
m = \frac{G_i \cdot \eta}{C_p \cdot (T_{exh} - T_{amb})}
\]

\( \eta \) is the instantaneous efficiency of the solar collectors, \( G_i \) is the global irradiation on the collector plane, \( C_p \) is the specific heat capacity of air and \( T_{amb} \) is the ambient air temperature. The dependence of efficiency on the mass flow rate in Equation (2) is solved using an iterative controller (Type 22). The minimum allowed air flow rate is set to 20 kg/(h·m²), corresponding to a critical suction velocity of 0.02 m/s based on the best practice for collector operation. At minimum airflow rate \( (m_{min}) \), if the outlet air temperature \( T_{aout} \) from the collector is less than the brine return temperature \( T_{bout} \) then the fan is turned off to avoid a reverse heat flow in the heat recovery unit. The maximum airflow rate \( (m_{max}) \) is reached corresponding to a brine outlet temperature limit of 20 °C, selected based on the HP specifications. During periods with high irradiation, the brine flow rate and air flow rate is increased to maximum, so to keep the brine return temperature equal to 20 °C. However, if the brine temperature is higher than 20 °C at a maximum fan speed of 80 kg/(h·m²), and at maximum brine flow rate, then a damper is open in the UTSC duct to bypass a fraction of hot air. The expected
Fig. 13. Flow chart of control strategy 2 based on the variable air flow rate.

- **START**
  - **HP ON?**
    - Yes: Go to Gt > 50 W/m²
    - No: Go to FAN OFF
  - **Gt > 50 W/m²**
    - Yes: Go to FAN ON
    - No: Go to FAN OFF

**Irradiation**
- Calculate air flow rate (m)
  - m < m_{min} → NO
  - m_{opr} = m
  - m_{opr} = m_{min}
  - T_{out} > T_{bystander}
    - Yes: FAN OFF
    - No: FAN ON at air flow of m_{max}

**Ambient and exhaust air temperature**
- Check for 1 minute
  - Yes: FAN OFF
  - No: m_{opr} = m_{max}

**Collector efficiency**
- Bypass ON

- m: calculated air flow rate
- m_{min}: 20 kg/h/m² based on critical suction velocity for collector
- m_{opr}: Operational air flow rate at any given time step
- m_{max}: 80 kg/h/m² based on maximum fan speed
- T_{bystander}: Brine outlet temperature from heat exchanger
- T_{out}: Air outlet temperature from collectors
benefit of a variable flow control strategy is an optimization of the fan power consumption and the temperature output with varying meteorological conditions.

With the second control strategy, a fixed airflow rate of 60 kg/(h·m²) is considered during operation. This airflow management strategy is the one used in most of the real installations, where the master control of an ON/OFF fan is based on the HP operational status and the irradiation conditions. In this case, the airflow is constant, and thus the fan works at a fixed speed.

6. Key performance indicators

There exist multiple approaches to evaluate the performance of solar HP systems. The KPIs defined in this study aim to assess individual component performances (as the solar collectors or the heat pump), and complete system performances. A description of the KPIs used in the discussion is provided in this section.

6.1. Component performance figures

Coefficient of performance (COP): the COP of a heat pump is the ratio of its steady-state heating capacity \( Q_{hp,h} \), and its electricity consumption \( W_{hp} \). The COP is used to evaluate the performances of the HP at a given time, and it does not include the electricity consumption of the circulation pumps in the system. The COP is described as:

\[
COP = \frac{Q_{hp,h}}{W_{hp}} \quad (3)
\]

Seasonal coefficient of performance (SCOP): The SCOP is defined as the integral of the COP of a HP over a certain period based on dynamic operating conditions. The dynamic conditions consider the change in heat source temperature, ambient temperature, load temperature, and part load conditions, etc. The annual SCOP is the annual average COP of the heat pump calculated when it is operational. The SCOP is defined as:

\[
SCOP = \frac{\int_{t=0}^{t=8760} Q_{hp,h}(t)}{W_{hp}(t)} \quad (4)
\]

Collector energy utilization ratio: the energy utilization ratio is the equivalent of the SCOP for a solar collector, as it describes the performances of the component over a period of time, as shown in Equation (5).

\[
\omega_{coll} = \frac{\int_{t=0}^{t=8760} Q_{coll,h}(t)}{A_{coll} G_{coll}(t)} \quad (5)
\]

\( Q_{coll} \) is the annual thermal output of solar collector (kWh), \( G_{coll} \) is the global irradiation on the collector plan (kWh/m²), and \( A_{coll} \) is the gross area of the collector (m²).

6.2. System performance figures

Total electricity use \( (W_{el}) \): this figure accounts for the electricity consumption of the selected system components. For the given system integrated with the solar collector, the total electricity consumption is described in Equation (6),

\[
W_{el} = W_{el,hp} + W_{el,wp} + W_{el,aux} + W_{el,fan} \quad (6)
\]

where \( W_{el,hp} \), \( W_{el,wp} \), \( W_{el,aux} \), and \( W_{el,fan} \) are the electricity consumption of HP, water pumps, auxiliary, and fans respectively. When the solar collector is not integrated, the power consumption of fans to drive airflow in UTSC and relevant pumps are not considered.

Seasonal performance factor (SPF): the SPF is used to express the overall energy efficiency, and provides a comprehensive idea on the operation of the whole system. The SPF is a very relevant KPI to analyse and compare the effects of different control strategies and energy systems. As shown in Equation (7), it is defined as the ratio of the total useful thermal energy supplied to the user \( (Q_{sh} + Q_{DHW}) \) and the total electricity use \( (W_{el}) \).

\[
SPF = \frac{\int_{t=0}^{t=8760} (Q_{sh}(t) + Q_{DHW}(t))}{\int_{t=0}^{t=8760} W_{el}(t)} \quad (7)
\]

6.3. Economic performance figure

Net present value (NPV): NPV is defined as the cumulative profit calculated by subtracting the present values of cash outflows (including initial costs) from the present values of cash inflows over the lifetime of the system. A positive NPV indicates that the projected earnings, generated by a project or investment, exceed the anticipated costs. In general, an investment with a positive NPV will be a profitable one and the higher NPV means higher benefits. This concept is the basis for the NPV decision rule, which dictates that the only investments that should be made are those with positive NPV values.

\[
NPV = \sum_{i=0}^{n} \frac{CF_i}{(1+r)^t} - C_0 \quad (8)
\]

Where, \( CF_i \), \( r \), \( n \), \( t \), \( C_0 \) are the cash flows of a particular year, the discount rate, the number of years, the year of NPV evaluation, and the capital cost respectively.

7. Results

7.1. System performance with variable flow

The reference system analysed in this study is based on a variable airflow control strategy for UTSC. The complete system and component performance is discussed in this section. Fig. 14 shows the annual heat supply to the building cluster for SH and DHW preparation detailed for generation sources. The total heating demand of the building cluster is 527 MWh, of which 51.1% (270 MWh) is met by the HP operating in SH mode \( (Q_{hp,sh}) \) and DHW mode \( (Q_{hp,dhw}) \). The rest of the thermal load is covered by the auxiliary heating system \( (Q_{aux}) \). On an annual basis, the HP covers nearly 96% of the DHW demand (incl. pipe heat losses in the DHW circuit), and rest is covered by the auxiliary system \( (Q_{aux,dhw}) \). The HP generates 160 MWh in SH mode meeting 37% of the SH load, whereas the rest is covered by the auxiliary electrical boiler \( (Q_{aux,sh}) \). During summer, the HP operates primarily in DHW mode, whereas during winter it operates mostly in SH mode operation. The annual heat losses from the SH and DHW tanks are 600 kWh and 4200 kWh respectively.

The total electricity use \( (W_{el}) \) is 368 MWh. The electricity consumption of the various components is shown in Fig. 15. It can be seen that on annual basis, more than 80% (298 MWh) of total electricity is used by the auxiliary electric boiler \( (W_{el,aux}) \), as it must cover a major fraction of the SH demand. The HP covers 51.1% of total heating demand, with total electricity use of 16.8% (62 MWh) by the HP compressor \( (W_{el,hp}) \). Other system components such as fans \( (W_{el,fan}) \), and pumps \( (W_{el,pump}) \) account for about 2.1% (7 MWh) of the total electricity consumption. The annual heating demand is 527 MWh whereas the annual electricity consumption is 368 MWh, resulting in a system SPF of 1.43. The Sankey diagram of the heat flows characterizing the studied system over one year is shown in Fig. 16.

The SCOP of the EAHF is equal to 4.36, and the monthly variation of the average COPs shown in Fig. 17 indicates that the HP operates more efficiently in the period from November to February. This can be explained considering that the prevailing working mode of the HP during wintertime is SH, which has a lower design set temperature compared to the DHW mode, resulting in a better performance of the HP. From July to August, the SH load is significantly lower and thus the DHW mode is the prevailing one. The periods from April to June and from September to October show large variations of the COP, as the HP
is operated almost equally in DHW and SH modes. The horizontal line in each box of the chart represents the median value of the COP and divides the 1st and 3rd quartiles of the data set, whereas cross (x) represents the mean value of COP.

The 208 m² area UTSC field generates a total annual specific thermal output of 36 MWh with collector area-specific output of 176 kWh/m², resulting in an annual utilization factor $\omega_{th}$ of 0.18. The monthly variation of the specific collector output is shown in Fig. 18. It can be seen that the collector thermal output is linearly correlated to the global tilted irradiation, with a higher thermal output observed during summertime when the irradiation and ambient temperature are higher. Similar trends are observed for the monthly utilization factors, with higher values in summer due to higher ambient temperatures and higher flow rates. The increase of ambient temperature decreases the heat losses from the collector surface, which results in a higher collector output. Furthermore, as the GHI, and ambient temperatures during winters is comparatively lower, which further results in lower utilization factors.

7.2. Sensitivity analysis

7.2.1. Effect of flow rate strategy on system performance

It is observed that the use of a fixed airflow strategy to manage the UTSC field leads to worse performances compared to a strategy where the airflow is variable. The improved performance of control strategy two is reflected by an increase of the SCOP of the EAHP and a reduction of the fan electricity consumption. During winter months, as the irradiation and ambient temperature are lower, the control signal from the variable flow strategy reduces the fan speed and thus the airflow rate behind the collector surface. This results in a higher average outlet air temperature from the collectors compared to the fixed flow rate case. Higher source temperatures result into a higher COP of the EAHP. Moreover, reducing the airflow rate from the nominal one allows to decrease the electricity consumption of the fan. During summer months, the irradiation and the ambient temperatures are higher and the fixed flow control strategy cannot control the airflow rate in the UTSC so to limit the brine temperature to 20 °C at the outlet of the HX. In this case, a fraction of the volume of the pre-heated air is expelled through the bypass damper to keep the HP brine temperature within the design limits. This results in an underutilisation of the energy available on the solar collector surface.

However, a variable airflow control strategy can increment the air volume to keep the brine temperature within the design limits and without wasting valuable heat, resulting in better performances of the collector and HP. Fig. 19 compares the COP of the EAHP for both control strategies. The SCOP of the EAHP for variable and fixed flow strategy are 4.36 and 4.25 respectively. A higher relative increase in COP can be seen for months with high irradiation and ambient temperatures. A combined effect of increased EAHP COP, and fan power reduction with variable airflow strategy, leads to a reduction in annual electricity use (W_{el}) of 1.9 MWh. The annual electricity consumption of fan for fixed airflow case is 670 kWh, which is 82% higher than electricity consumption in variable airflow case (368 kWh). The electricity savings due to a higher COP of the EAHP is 1.6 MWh compared to fix flow rate case. The effects of higher COP and reduced fan power consumption are positively reflected in system SPF, which is 0.5% higher for the variable airflow case.

7.2.2. Effect of UTSC integration on the total electricity consumption

The effect of the UTSCs area on the EAHP performance and total electricity consumption is investigated. The collector’s area and the fan capacity are varied using scale factors of 0, 0.5, 1.5 in comparison to the reference system, with a collector’s area of 208 m². The variation of the total electricity consumption is indicated by $\Delta W_{el}$ which represents the difference between the total electricity used of analysed system (at
various scale factors) and that of the reference system (scale factor = 1). The variation in $\Delta W_{el}$ and SPF for various scale factors is shown in Fig. 20. The results show that at smaller scale factors (and thus smaller collector area), the total electricity use is higher, resulting in a positive value of $\Delta W_{el}$. Also, larger collector areas result in a higher SPF, and consumes lesser electricity resulting in a negative value of $\Delta W_{el}$. A system without any solar installation (scale factor equal to 0), that is the case where the only heat source of the EAHP is exhaust air, consumes nearly 1700 kWh of additional electricity compared to the reference system. Therefore, solar collector installation results in electricity savings, and higher system SPF.

Fig. 16. Sankey diagram to represent the heat flows in the energy system.

Fig. 17. Monthly variation in heat pump COP.
7.3. Economic analysis

The profitability of a system integrating UTSCs into an EAHP energy system based on the design discussed above is evaluated. The NPV is used as an economic indicator and is calculated for a zero discount rate and a collector’s lifetime of 30 years. Annual electricity savings equal to 1700 kWh are considered as the energy benefit of the UTSC installation, and an electricity price of 0.207 €/kWh is considered for the NPV calculation. The maintenance cost of the solar collector is neglected, as this might add a source of uncertainty. Fig. 21 shows the NPV for different UTSC investment costs.

Despite the fact that the UTSC integration improves the system SPF, the NPV is negative for the analysed range of specific collector costs. A threshold collector cost is calculated at which the lifetime savings will be equal to the capital investment cost of the collector. The threshold collector cost for positive NPV is 48 €/m², far below the commercial price (300-500 €/m²) for these collectors.

8. Discussion

The results presented in previous sections show that UTSC integration with EAHP system has a positive impact on overall system SPF. However, the impact is not enough to result in any positive cash flow during the collector’s lifetime, despite that several cost components (operation & maintenance, etc.) are not accounted in economic analysis. For the analysed system, the reduction in total electricity consumption is due to the increase in EAHP COP resulting from higher energy input on the evaporator side. However, the impact is quite small, calculated at 1700 kWh for an installed UTSC area of 208 m². The reason for low savings can be attributed to the mismatching characteristics of Swedish meteorological conditions and seasonal variation in building heating demand. For example, in the analysed system configuration the effect of the UTSC on the HP performance is more pronounced from June to August when irradiation and ambient temperatures are high. On the contrary, the heating load is minimal during this period and thus the EAHP operates only for a few hours per day. Therefore, even though the UTSC assisted EAHP operates at high COP, the final effect on annual
electricity savings is low due to limited operational hours, which results in negligible economic savings. During winters, the space heating demand is higher but the low irradiation and ambient temperatures limit the collector outlet air temperatures, and thus the increase of the COP of the HP is relatively small.

The rule-based control based on the variable air flow rate for UTSC has resulted in a positive impact on EAHP COP and system SPF. The calculated reduction in fan power consumption is 370 kWh, and the reduction in EAHP electricity consumption is 1.6 MWh resulting in total electricity savings of 1.9 MWh compared to the fix flow rate case. The developed controls adapt the airflow rate and thus fan power consumption based on inputs from meteorological parameters. During operation in winter months, as the irradiation and ambient temperature are lower, the strategy reduces the fan speed and thus the airflow rate behind the collector surface. This results in a higher average outlet air temperature from the collectors compared to the fix flow rate case. Higher source temperatures result in a higher COP of the EAHP. Moreover, reducing the airflow rate from the nominal one allows to reduce the electricity consumption of the fan. During summer months, the irradiation and the ambient temperatures are higher and the fixed flow control strategy cannot control the air flow rate in the UTSC so to limit the brine temperature to 20 °C at the outlet of the HX. The developed control can be easily adapted to other heat pump systems (air-based/water-based etc) as well, bringing the potential for system performance improvement.

This study approaches the issue considering a single energy system design, a single climate, and a load profile. For the system with similar system boundary conditions, more energy benefits potential may lie in Central and Southern European regions, with favoring load profiles, and meteorological conditions. Moreover, a comparative analysis of the UTSC system without any exhaust air recovery can be of interest to assess the potential of these collectors in a cold climate such as Sweden. In the absence of an exhaust air recovery system, the only source of HP evaporator would be ambient air, which will result in much lower evaporator inlet temperature to HP, and thus lower COP. The integration of UTSC will be more favourable in that case, as air at lower outlet temperature from UTSC can also be utilised to raise evaporator inlet temperature. Another relevant use case for EAHP integration can be

![Fig. 20. SPF and electricity variation at various collector area.](image1)

![Fig. 21. Net present value (NPV) of UTSC for analysed system configuration.](image2)
with UTSC photovoltaic thermal (PVT) collectors. Commercial UTSC system are also available in PVT variants, where PV modules are mounted on top of UTSC, and the airflow helps to recover the heat generated by PV modules, resulting in better performance of PV collectors. PVT configuration can be of interest to analyse, as it can result in higher energy yield and better economic savings.

To increase the rate of diffusion for UTSC collectors without HP system, the biggest applications lie in the pre-heating of ventilation air for residential or industrial applications. However, as the building sector is going through various energy efficiency measures, the integration of heat recovery ventilation or exhaust air heat recovery is becoming popular. Both options are not best suited for UTSC integration due to lower additional benefits from the system perspective. However, buildings with a lack of any kind of heat recovery system seem the best candidate for UTSC integration, with the potential of direct energy savings from pre-heating of ventilation air.

9. Conclusions

The study describes the techno-economic analysis of a UTSC coupled with EAHP in a Swedish multifamily house. The study specifically aims to quantify the potential energy benefits of the UTSC used to pre-heat ambient air as a complementary energy source for an EAHP. A comparative analysis of two control strategies for the management of the airflow into the UTSC is presented, and the effects are evaluated at component and system performance level. The results are further used to investigate the economic feasibility of UTSCs for the given system configuration.

From the analysis of the results applied to a residential building cluster in Sweden, it is observed that:

- The analysed EAHP-UTSC system with a variable air flow rate consumes 368 MWh of electricity to fulfil SH and DHW demand of 527 MWh, resulting in system SPF of 1.43. The component analysis shows that UTSC collector has an annual collector energy utilization factor of 0.18. The SCOP of EAHP is evaluated at 4.36 with a significant monthly variation.
- The integration of UTSCs has a small but positive impact on the overall system performance. A system without any solar collector installation consumes 0.5% additional electricity compared to the reference system due to the relatively lower SCOP of HP. Sensitivity analysis revealed that an increase in the area of the collectors results in lower total electricity consumption and thus a higher SPF.
- The variation of the collector airflow rate is an effective control strategy to increase the SCOP and thus the SPF of the overall system and maximize the savings. A comparative analysis of the effects of the control strategies suggest that the optimization and variation of the collector airflow rate resulted in a 2.5% increase in the SCOP of the EAHP and a 0.5% increase in the system SPF.
- The economic analysis reveals that even though the UTSC integration improves the system SPF, the NPV is negative for the analysed range of specific collector costs. The threshold collector cost for positive NPV is 48 €/m², much lower than the current cost of these collectors in the market.

UTSC are often marketed for their easy mounting, lack of hydronic system, and longer lifetime. This results in a reduction of a major cost in the installation of systems. However, despite lower costs and a higher collector lifetime, the unfavourable meteorological conditions and the large seasonal variations of the heating demand in the Swedish climate seem to be a limiting factor for the economic profitability of these collectors. This can be a barrier to the wide adoption of the technology for the proposed system configuration. Despite, this study has analysed the added value of UTSC for EAHP based system in a residential building cluster and can be used as a stepping stone towards the identification of additional research opportunities for such systems.

CRediT authorship contribution statement

Puneet Saini: Conceptualization, Methodology, Validation. Bonato Paolo: Conceptualization, Methodology, Writing - review & editing. Frank Fiedler: Conceptualization, Project administration, Supervision. Joakim Widen: Project administration, Supervision. Xingxing Zhang: Conceptualization, Project administration, Supervision.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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