Investigation of a Solar Assisted Heating System

Gareth Davies (PhD)[1], John Blower (BSc)[2], Richard Hall (PhD)[2], Graeme Maidment (PhD)[1]
[1]School of Engineering, London South Bank University
[2]Energy Transitions Ltd, Cardiff, UK
daviesg@lsbu.ac.uk

Abstract
Low emissivity transpired solar collectors (low-ε TSCs) consist of metal solar absorber, collector plates, with a spectrally sensitive surface, perforated with holes. Ambient air is drawn through the holes and heated by convection from the solar collector plate, increasing the air temperature by up to 25 K. The heated air can be used for e.g. space heating or pre-heating hot water, in buildings. The performance of low-ε TSCs combined with heat pumps in various configurations, to deliver heat to buildings, has been investigated using computational models and compared with conventional heating systems to determine the potential energy, carbon and cost savings available. The results showed that using low-ε TSCs incorporated into a heat pump based ventilation air system produced annual savings in energy, CO_2e and costs of up to 14%, compared to similar systems where the TSC was not used. The greatest savings were achieved when using an exhaust air heat pump. Recommendations for further development of this novel technology are also presented.

Keywords transpired solar collector; heat pump; low carbon, building heating; energy, carbon and cost savings

1.0 Introduction
The UK Government has recently set a new climate change target of reaching net zero greenhouse gas (GHG) emissions by 2050 [1]. Since 1990, the UK has reduced its carbon emissions by 43%, mainly as a result of carbon reduction measures in the power sector [2], however, significantly more radical solutions and technologies will be needed to meet the new target of zero emissions by 2050. One area of focus will need to be heating, as this accounts for approximately one third of the UK's carbon emissions and about half of its energy consumption [3]. In 2018, the European Environment Agency reported that the UK has one of the lowest shares in Europe for the use of renewable energy for heating and cooling i.e. 7% [4]. There is therefore an urgent need for new, flexible, low or zero carbon, renewable energy systems for use in buildings. Transpired solar collectors (TSCs) are one such technology, which can be used to facilitate the capture of solar thermal energy for use in buildings. TSCs consist of metal plates, with a spectrally sensitive surface, which absorb solar radiation, raising their temperature. The plates are attached to a south facing (in the northern hemisphere), vertical wall of a building, such as to leave a small air gap, and sealed at the edges, to form a plenum i.e. a thin box, or cladding. The collector plate is perforated with many small holes drilled in its surface, through which ambient air is drawn, by a small fan, into the plenum. There, the air is heated by convection from the collector plate, increasing its temperature by up to 25 K. The heated air flow can then be used e.g. for space heating or for pre-heating hot water within the building. The air temperature increase achieved depends on the environmental conditions, the

solar collector plate design e.g. distribution of perforations and surface coating, and the air flow rate and face velocity of the air at its surface [5]. The coatings used by the current generation of TSCs are generally of high emissivity, which although achieving high absorption of solar radiation, are also subject to high radiation heat losses to the outside environment. However, new spectrally selective, low emissivity (low-ε) coatings have been developed, which enable the collector's absorptivity to be maintained, while minimizing re-radiation to the environment. This results in higher collector plate surface temperatures and generates higher output air temperatures [6].

The paper reports the results of an investigation of the potential of low-ε TSCs to provide solar enhanced heated air for delivery to buildings via ventilation systems. The increase in air temperatures obtained from the TSC varies with the solar radiation available i.e. from minute to minute, as well as daily and seasonally, and is only available during the day. Therefore, for an effective heating system, the TSC needs to be combined with another heating system which can be used to meet the building heat demand at other times. In this study, the heated air generated by the TSC has been used as a heat source for an air source heat pump, which is then used to meet the space heating demand profile of a building. A number of configurations for combining the TSC with the heat pump have been evaluated, and compared to the counterfactual case, where the TSC is not used, to investigate its potential benefits. Traditional (high- ε) TSCs have been combined with ventilation air heating systems and evaluated previously e.g. [7], [8], and the present authors have reported the use of low-ε TSCs with ventilation heating systems applied to a domestic house [9]. However, the current study appears to be the first detailed evaluation of such heating systems with low- ε TSCs applied in large e.g. warehouse, buildings.

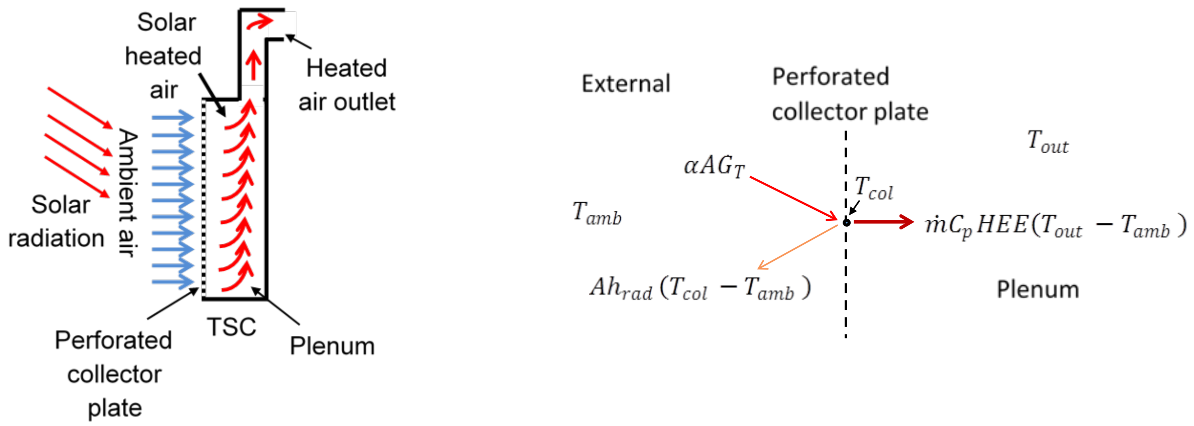
2.0 Methodology

The evaluation of the performance of a number of configurations of TSC-heat pump building heating systems was undertaken using a modelling approach, employing Engineering Equation Solver (EES), a commercial software tool [10]. The performance of the overall heating system was simulated using a series of simultaneous equations to define the relationships between the parameters needed to describe the system. Input data for parameter values e.g. temperatures, fluid flow rates and thermal properties, were provided, and the software used to predict a number of unknown parameter values, such as the total electrical energy input to the system and the efficiency for the heat pump. In fact, to simplify the solution process, the overall heating system was divided into a number of component modules, which were then solved in sequence, in order to evaluate the overall performance of the system. The components included: (i) a TSC model; (ii) a building heat demand model; (iii) a heat pump model; (iv) a control model. The control model was needed to define the order for solving the various component models, and to switch the heat pump on and off, as required, for different input conditions. Details of the various component models are provided below.

2.1 Transpired solar collector (TSC) model

In this study, the heated air generated by the TSC device was ducted to a building, either to provide heated ventilation air directly, or to be used as a heat source for a heat pump. There was also a bypass opening in the TSC to permit ambient air to be used directly for ventilation, when solar heated air was not required. The volumetric air flow rate through the TSC was set equal to the ventilation air flow rate, which was

calculated in the building heat demand model.
In earlier studies [5, 6], it was found that a collector plate face velocity in the range 0.02 – 0.05 m/s was needed for effective operation of the TSC. By using the volumetric air flow rate through the TSC and dividing by the face velocity at the surface of the collector plate (selected to be 0.025 m/s for the current model), the required collector plate area could be calculated. In this study, the collector plate area was estimated to be equal to that of one of the building walls.
In an earlier study [6], it was found that an air temperature increase of 20% could be obtained using a spectrally sensitive low emissivity coating for the collector plate compared with high emissivity plates, for the same absorptivity surface [6]. For the current model, a surface absorptivity of 0.9 and an emissivity of 0.2 were assumed. Further details of the TSC model have been provided previously [5]. A schematic of a transpired solar collector together with the energy balance at the collector plate surface is shown in Figure 1 below.



(a) Schematic of transpired solar collector (TSC)

(b) Energy balance at collector plate surface

Figure 1 Schematic of TSC and energy balance at collector plate

Note: In Figure 1 (b), α is the absorptivity of the solar collector plate surface, A is the area of the plate, G_T is the solar global radiation per unit area, h_{rad} is the linearized radiation heat transfer coefficient, which also incorporates the emissivity of the surface, and T_{col} is the temperature for the solar collector surface. On the inside of the enclosure, m is the mass flow rate of the air, which enters at ambient temperature T_{amb}. C_p is the specific heat capacity of the air flowing through the enclosure, and T_{out} is the temperature of the air at the outlet. HEE is a heat exchange effectiveness coefficient which takes account of the convective heat transfer from the collector plate to the air [11]. The pressure drop through the collector plate was determined using a scaling factor used to model square pitch perforations [12]. The energy balance for the collector plate is given by:

$$\dot{m}C_p HEE(T_{col} - T_{amb}) = \alpha AG_T - Ah_{rad}(T_{col} - T_{amb})$$

Figure 2 Energy balance at collector plate surface

The heat exchange effectiveness coefficient (HEE) is defined by:

$$HEE = \frac{(T_{out} - T_{amb})}{(T_{col} - T_{amb})}$$

Figure 3 Heat exchange effectiveness coefficient for collector plate

In Figure 2, the main (variable) inputs needed were weather data parameters e.g. for air temperatures and solar radiation. These were obtained from a weather database [13], for a location in the London Borough of Islington, which had been selected as a potential trial site for a prototype system. The weather data consisted of hourly values for ambient air temperature, ground temperature, sky temperature, air pressure, relative humidity, wind speed, global radiation on a south-facing vertical plane and global radiation on a horizontal plane. The overall mass air flow rate m through the TSC was determined from the building heat demand model. The TSC model was then used to calculate the hourly variation in the solar collector output air temperatures.

2.2 Building heat demand model

A building model was developed to estimate the building space heating demand to be met by the heating system employed. It consisted of a simple air ventilated building, subject to heat losses through the building fabric due to the difference between the internal set temperature and the seasonal variation in outside environmental conditions, and heat losses from the exhaust ventilation air leaving the building at the internal building set temperature. To maintain the set temperature within the building, it was assumed that the required quantity of heat was added via the ventilation air supply, by raising its temperature, as appropriate. The ventilation air mass flow rate was calculated, based on the internal volume for the building and number of air changes per hour required. The heated ventilation air supply at the appropriate temperature was generated using an air source heat pump, and supplemented by the TSC heated air. When the TSC heated air temperature reached the ventilation air temperature indicated from the building heat demand model, the heat pump could be switched off and the heated ventilation air supplied by the TSC alone.

The building model was based on a large building e.g. warehouse, and heated by a single ventilation air heating system to a common temperature throughout. To calculate the heat demand for the building (at any particular time), the following assumptions were used:

Parameter		**Value**
Building dimensions (m)		44.72 x 44.72 x 20
Volume (m^3)		40,000
Single internal space i.e. no internal structure was considered, for the purposes of the model.		-
Ventilation rate (air changes per hour)		2
Building internal set temperature T_{set} (°C)		22
Fabric heat losses i.e. through the walls, floor, roof, windows and doors, were defined by their overall heat transfer coefficient U and area A values, and these together with the ventilation exhaust heat loss coefficient, were compounded into a single heat loss coefficient HLC (W/K)		30168
Building fabric U values [14] (W/m^2 K)	Walls	0.18
	Floor	0.13
	Roof	0.13
	Windows and doors	1.4
% of front and back walls (only) occupied by windows and doors		40
Ventilation air mass flow rate (kg/s)		26.6
Specific heat capacity (J/kg K)		1019
Temperature difference ΔT between the inside building, air set temperature T_{set} and outside ambient air temperature T_{amb} (K)		T_{set}-T_{amb}
Total heat loss for the building (equal to the total building heat demand) Q_{demand} (W)		HLC x ΔT

Table 1 Building heat demand model parameters used

Note: In Table1, the ventilation exhaust heat loss coefficient was calculated as the product of the mass flow rate of exhaust air and specific heat capacity.
Using the above assumptions, and hourly ambient air temperatures [13], the model was used to determine the seasonal heat demand profile for the building.

2.3 Heat pump model

The heat pump model was simulated as a single stage air source heat pump using a series of thermodynamic balance equations to link the input and output parameters and performance of each of its components [15]. Input data for the various components were specified for the model, and the equations then solved iteratively to predict selected outputs and the overall performance of the heat pump. The model comprised three main component sub-models, namely a single stage compressor, and finned tube air to refrigerant evaporator and condenser heat exchanger models. The condenser heat output capacity (which was matched to the hourly varying building heat demand) was used as input for the heat pump model, which then predicted the compressor swept volume and the electrical energy input to the compressor. It was assumed that a variable speed compressor was used, such that the capacity of the heat pump could be varied in line with the building heat demand, in order for the heat pump to take advantage of the rapidly varying input air temperature conditions provided by the solar collector output. Key inputs for the heat pump model were: (i) the condenser output capacity required; (ii) the condenser output air temperature needed; (iii) the TSC output air temperature; (iv) the volumetric flow rate for the air over the heat exchangers, which was defined by the ventilation air flow rate required for the building; (v) the condenser air on temperature, which for some configurations was ambient air, and for others was TSC heated air (vi) the evaporator air on temperature, which for some configuration

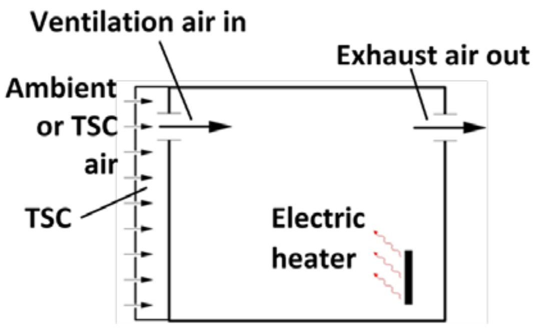
options was ambient air, and for others was TSC heated air, or the exhaust air from the building (i.e. enabling it to operate as an exhaust air heat pump).
Note: When the TSC heated air temperature was greater than the ventilation air temperature needed, while the ambient air temperatures was less than the required ventilation air temperature, it was assumed that the TSC heated air could be mixed with the ambient air to produce the required air temperature, and used directly for the ventilation air supply. In this case the heat pump would be switched off.

2.4 Control model

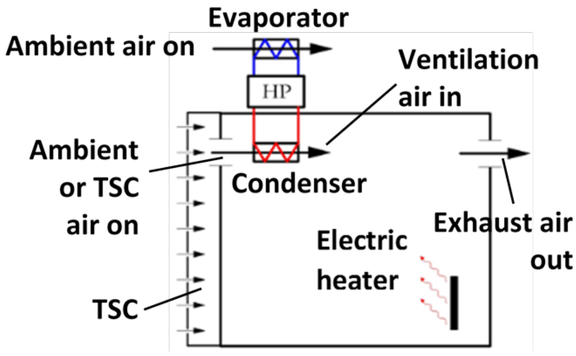
A key requirement for the operation of the heat pump sub-model within the overall TSC-air source heat pump, ventilation air heating system model, was a control sub-model. This was used to determine whether the heat pump was switched on or off at any particular time/set of input conditions, and to read the key parameter inputs for the heat pump model, from the building heat demand and TSC models. It was also used to provide appropriate guess values for iterative solving of the equations defining the heat pump model, in order to facilitate convergence. In addition, it controlled the sequence for execution of each of the sub-models i.e. the building heat demand model, TSC model, and heat pump model.
At any particular time (i.e. time step), the heat required to meet the building heat demand could be derived from several sources, namely: (i) directly from the TSC output air flow; (ii) after upgrading the TSC output air temperature with the heat pump; (iii) directly from the outside ambient air. It was therefore necessary to provide a control algorithm for selection of the most appropriate heat source.
A further consideration was that in some cases, very large temperature differences between the ventilation inlet air and the internal building air temperature were needed to meet the highest building heat demands, and this was undesirable with respect to the thermal comfort level within the building. Therefore, the temperature difference between the ventilation inlet air delivered $T_{delivery}$ and the building air set temperature T_{set} was limited to 8 K [16]. This was also accounted for in the control model. If more heat was needed to meet the building heat demand than could be achieved by the ventilation air, with this limited temperature difference (and since the ventilation air flow rate was fixed), it was assumed that the difference in heat capacity was made up by an electric heater (with 100% efficiency) located within the building. The cost and carbon emissions associated with this additional electricity input needed to meet the building heat demand were also calculated, and added to the totals for energy, carbon and cost inputs for the heating system.

2.5 Ventilation heating system configurations modelled

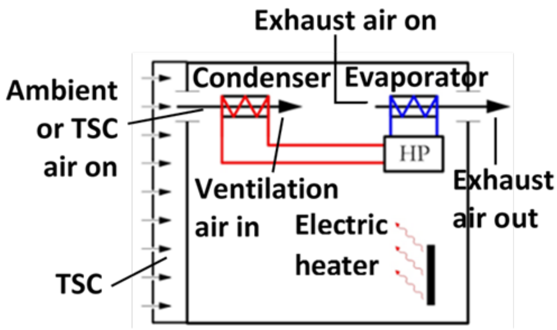
In Figure 4 below, TSC is the transpired solar collector and HP is an air source heat pump. The four heating system configurations shown were each modelled with: (i) the TSC in operation; and (ii) in bypass mode, where ambient air only was provided.



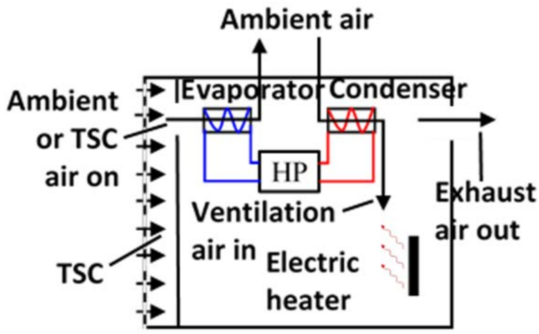
(a) Electric heating of building only



(b) Heat pump with ambient or TSC air onto condenser, ambient air onto evaporator, and electric heater top-up



(c) Heat pump with ambient or TSC output air onto condenser and exhaust air onto evaporator and electric heater top up



(d) Heat pump with ambient air onto condenser, ambient air or TSC output air onto evaporator, and electric heater top up

Figure 4 Ventilation air heating system configurations modelled

For the three configurations using heat pumps shown in Figure 4 above, where either the ambient air or TSC heated air temperatures reached or exceeded the building heat demand ventilation air temperature required, the heat pump was assumed to be switched off within the model.

3.0 Results

The results from the modelling work are presented below.

3.1 TSC model

Figure 5 shows the results for seasonal variation in TSC heated air output temperatures in comparison with ambient temperatures, predicted by the model.

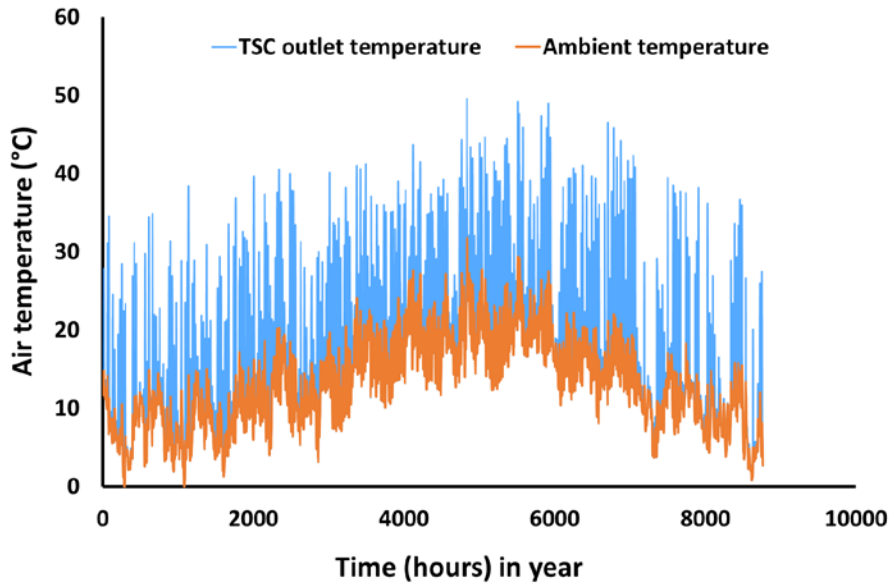


Figure 5 Annual temperature profile for TSC outlet air

It is seen that there was a significant increase in temperature for the TSC outlet air compared to ambient air, of up to 25 K, however, the TSC outlet air temperature increase varied significantly (in the range 0-25K) from hour to hour.

3.2 Building heat demand model

The annual building heat demand profile is presented in Figure 6. It shows the daily variation in heat demand for the (warehouse) building in kWh, over a year. The heat demand values predicted by the model are based on the ventilation air mass flow rate, the set temperature selected for the building, and weather data, particularly outside, ambient air temperatures.

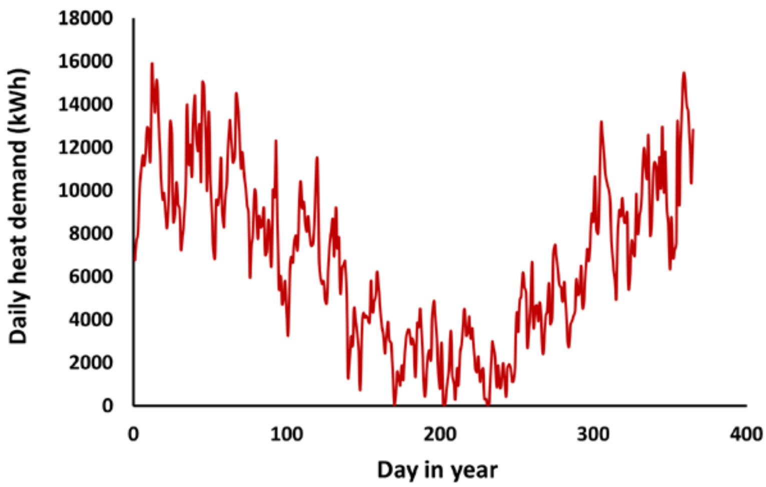


Figure 6 Seasonal variation in building heat demand

In Figure 6, the daily heat demand, follows a sinusoidal trend over the year, with the highest heat demand, up to 16000 kWh per day during the winter months, with the heat demand falling to a minimum level of 0 kWh at times during the summer months, although varying by up to 4000 kWh from day to day, throughout the year. The building heat demand model was also used to predict the temperatures needed for the ventilation air supplied to the building, in order to replace all of the heat losses from the building i.e. to match the heat demand value, on an hourly basis, throughout the year.

3.3 Results for heat pump model and analysis

The outputs from both the TSC model and building heat demand model, together with hourly ambient air temperatures, derived from the weather data, were used as inputs for the heat pump model. The model was then used to determine the total electrical energy needed to supply both the heat pump compressor and heat exchanger fans, together with any additional top up by the electric heater if required, in order to meet the required heat output capacity and ventilation air delivery temperature.

The overall electrical energy used by the ventilation air heating system was then used to calculate the corresponding CO_2e emissions and estimated cost for the electricity, for each of the configurations considered, both with and without the TSC heated air in operation. Assumptions used in calculating the CO_2e emissions and costs were: (i) an electricity carbon factor of 0.2555 kg CO_2e per kWh of electricity used [17]; and (ii) a cost for electricity of £0.155 per kWh of electricity [18].

A summary of the total annual electrical energy input, CO_2e emissions and costs, for each of the heating system configurations, is shown in Table 2.

Configuration	Parameter	Annual total (+TSC)	Annual total (-TSC)	Total annual saving	% annual saving
1. Electric heater only	Electrical energy input (MWh)	2138.5	2481.9	343.4	13.8
	CO_2e emissions (tonnes)	546.2	633.9	87.7	
	Costs	£331,465	£384,693	£53,227	
2. Heat pump heating, TSC or ambient air onto condenser, ambient air only onto evaporator	Electrical energy input (MWh)	540.10	619.93	79.83	12.9
	CO_2e emissions (tonnes)	137.94	158.33	20.39	
	Costs	£83,715	£96,089	£12,374	
3. Heat pump heating, TSC or ambient air onto condenser, building exhaust air onto evaporator	Electrical energy input (MWh)	407.45	472.01	64.56	13.7
	CO_2e emissions (tonnes)	104.06	120.55	16.49	
	Costs	£63,154	£73,162	£10,007	
4. Heat pump heating, ambient air only onto condenser, TSC or ambient air onto evaporator	Electrical energy input (MWh)	560.91	619.93	59.02	9.5
	CO_2e emissions (tonnes)	143.26	158.33	15.07	
	Costs	£86,942	£96,089	£9,147	

Table 2. Electrical energy inputs, CO_2e emissions and costs for a range of air ventilation heating system configurations

It is seen in Table 2 that the best performing heating system configuration i.e. the system with lowest electrical energy use, was configuration 3 i.e. the exhaust air heat pump. This system also demonstrated a further benefit of an additional energy, emissions and cost saving of 13.7 % from using the TSC device.

A seasonal profile for the variation in CO_2e emissions for one of the ventilation air heating system configurations, namely configuration 3, in Table1 (the exhaust air heat pump), is shown in Figure 7, both with and without TSC heated air.

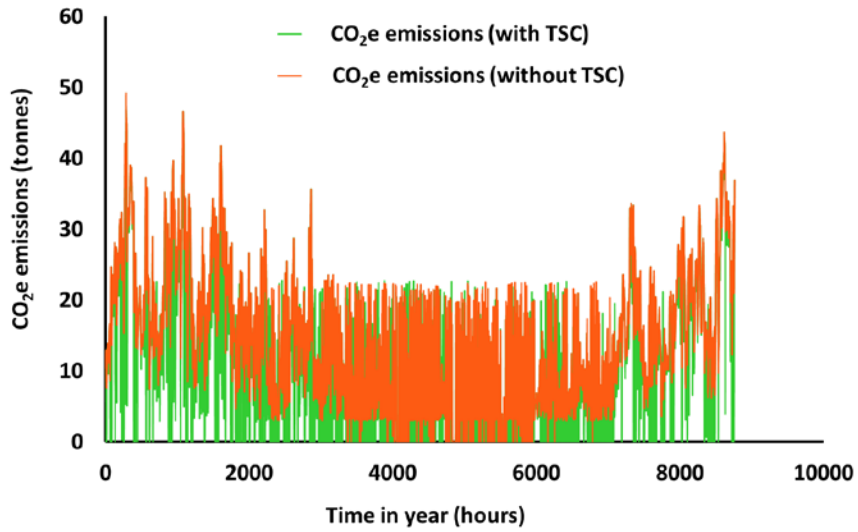


Figure 7 Annual variation in CO_2e emissions for heating system (+/-TSC)

Figure 7 shows that there are significant CO_2e emissions throughout the year, for the selected building and inside air set temperature used, although emissions are higher in the winter months. However, the CO_2e emissions were markedly reduced when using the TSC, particularly during the winter months. The pattern of CO_2e emissions shown in Figure 7 reflects those for electricity input and costs for this heating system configuration.

4.0 Discussion

The TSC outlet seasonal temperature profile shown in Figure 5 follows a similar pattern to that for ambient temperatures, although with increased temperatures of between 0 and 25 K. A similar level of increase in TSC outlet temperatures above ambient is seen throughout the year. There are, however, significant differences in TSC output temperatures from hour to hour, reflecting the variation in solar radiation availability.

The seasonal building heat demand profile in Figure 6 shows a clear annual trend i.e. higher heat demand in winter and much reduced demand during the summer months, although with significant day to day variation. The building heat demand effectively mirrors the seasonal variation in ambient temperatures, and follows a typical seasonal pattern for heat demand in buildings [19]. The building heat demand model was used to calculate the hourly variation in ventilation air temperatures to be supplied to the building in order to meet demand, and these were used as input for the heat pump model.

Table 2 shows the total annual values for electrical energy input, CO_2e emissions and costs for the four heating system configurations evaluated, both with and without the TSC included. In Table 2, the electric heater only system has substantially higher electrical energy input, CO_2e emissions and costs, than for the three heat pump based heating systems However, when using the TSC, a significant annual saving of 13.8% was predicted for the electric heater only system. The best performing system was configuration 3 i.e. the exhaust air heat pump. For this configuration, either the TSC heated air or ambient air was supplied to the condenser, while exhaust air was supplied to the evaporator. A higher energy input for the heat pump (and correspondingly higher CO_2e emissions and costs) was found for configuration 2, which involved either TSC heated air or ambient air supplied to the condenser and ambient air only to the evaporator. In fact, the total annual energy saved when using the TSC compared with ambient air was greater for configuration 2, than for configuration 3, however, the percentage annual saving was reduced to 12.9% for configuration 2. For configuration 4, where ambient air only was supplied to the condenser, and either TSC heated air or ambient air supplied to the evaporator, there was a smaller saving in terms of energy input reduction when using the TSC heated air, compared to ambient air, of 9.5%.

Table 2 indicates that the use of the TSC heated air with the three heat pump configurations was most beneficial, in terms of minimising energy use, CO_2e emissions and costs, when supplied to the condenser to provide ventilation air heating to the building. Also, further reduction in energy input was achieved by using the exhaust ventilation air from the building as the “air on” heat source for the evaporator.

Figure 7 shows the annual CO_2e emissions profile for the best performing heating system i.e. configuration 3. It is seen that the TSC heated air provided lower emissions than for ambient air in the winter, however, the emissions are fairly similar during the summer months. One possible reason for this could be that both the TSC and ambient air temperatures are higher during the summer, and closer to the ventilation air delivery temperature needed, so the heat pump performs similarly in terms of energy input, and corresponding CO_2e emissions.

5.0 Conclusions

This study has investigated the effects of using a low-ε TSC device to generate heated air (compared with using ambient air only) when incorporated into a range of configurations of ventilation air space heating systems for buildings. An electric heating only system and 3 heat pump based heating systems were evaluated in terms of energy input, CO_2e emissions and costs. It is seen from Table 2 that the greatest % annual savings and total annual savings, were achieved when using the TSC with an electric heating only system. However, the electric heating only system required very high energy input, resulting in high CO_2e emissions and costs i.e. by greater than 4 times that for any of the 3 heat pump based heating systems considered, even with the TSC used, so would be the least attractive option of the 4 heating systems considered.

Comparing the use of TSC heated air with ambient air, for the 3 heat pump based heating systems, a similar % annual saving to that for the electric heating only system, of 13.7%, was achieved for the exhaust air heat pump system configuration. In this case, the TSC heated air or ambient air was directed onto the condenser, and the room ventilation exhaust air directed onto the evaporator. The exhaust air heat pump heating system also had the lowest overall energy input, CO_2e emissions and operating costs. The second heat pump heating system configuration consisted of a standard heat pump, with TSC heated air or ambient air directed onto the condenser, and ambient air onto the evaporator, the % annual energy saving when using the TSC was reduced to 12.9%. However, the total annual energy input saving when using the TSC for this heat pump system i.e. 79.83 MWh was greater than that for the exhaust air heat pump system i.e. 64.56 MWh.

For the third heat pump heating system configuration, consisting of a standard heat pump with ambient air was directed onto the condenser and TSC heated air or ambient air directed onto the evaporator, a lower % annual saving in energy, CO_2e and costs, when using the TSC, of 9.5% was obtained. This system was also less efficient overall i.e. with a higher total energy input than for the other two heat pump heating system configurations.

It was concluded that the greatest annual savings were obtained when the TSC heated air was used directly i.e. as the “air on” supply for a heat pump condenser, generating heated ventilation air for the building. In addition, the benefit of the TSC was enhanced when an exhaust air heat pump was employed.
Other potential benefits of using TSCs are that they can act as cladding for buildings, providing extra insulation, and reducing building fabric heat losses.
It is recognised that the application used in the present study i.e. ventilation air heating of a warehouse was focused on a single type of building only, however, previous studies have investigated the use of low-ε TSCs with ventilation air heating systems in domestic houses, and indicated similar % annual savings [9].
In future work, it is planned to investigate the use of low-ε TSCs in other building heating system configurations, additional building types and sizes, and for different environmental conditions e.g. for other countries. The potential of using low-ε TSCs for pre-heating of hot water for buildings, and for combining with storage systems to provide better utilisation of the TSC generated heat, will also be investigated.
Overall, the present study indicates that low-ε TSCs offer significant benefits in providing low cost, low carbon heating for buildings.

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