Preliminary Study of a Solar Assisted Heating System

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ABSTRACT

Urgent measures are needed to combat worldwide climate change, including the development of efficient, cost-effective, low carbon, renewable heat sources. The current paper investigates a novel, low cost, solar thermal energy based system, namely a low-emissivity transpired solar collector. This consists of a metal, solar collector plate with a spectrally sensitive surface, and a large number of holes drilled into it, through which ambient air is drawn, into a plenum. The plenum air is then heated by convection from the collector plate, typically increasing its temperature by 15-20 K (27-36°R). The heated air is continually extracted from the plenum and can be used, for example, for space heating or for pre-heating hot water for buildings. This can be achieved either directly by using the solar heated air in ventilation heating systems, or heat exchangers can be used to transfer the heat generated by the solar collector to other air or water circulation systems, thereby combining the solar heat with that from gas, electric heaters or heat pumps. The novel solar heated collector considered in the present study could be used in the form of cladding attached to a vertical building wall. Advantages are that it is: (i) thermally efficient, achieving a significant increase in temperature from a relatively small area, even in winter; (ii) a low/zero carbon renewable heating source; (iii) low cost (in terms of both capital and operating costs), requiring only a low powered fan to force air through the solar collector plate; (iv) readily combined with other heating systems, such as heat pumps.

The current paper investigates the use of the solar collector integrated with other heating systems in a number of configurations, enabling it to meet the heating demand for a range of building types. Its effectiveness in the different configurations was evaluated and compared using models to investigate the various heat transfer steps involved and estimate the heat delivered by each system for a range of applications. In each case, the energy, carbon and cost savings achieved when using the solar collector are compared to those for a conventional (fossil fuel based) heating system. The results from these analyses together with recommendations for further development and future use of the solar collector integrated with other heating systems are reported.

1. INTRODUCTION

The UK government has recently pledged to become the first country within the group of seven largest advanced economies in the world (G7) to end its contribution to climate change, as a new target was set to reach net zero greenhouse gas (GHG) emissions by 2050 (UK Government 2019). This poses a great challenge to the UK's economy, as the country was not on track for achieving its previous climate goal, set by the Climate Change Act (2008), of reducing GHG emissions by 80% by 2050 from 1990 levels (CCC, 2018), and is now pursuing a bolder target of net zero. The UK has managed to cut its GHG emissions significantly over recent years, reducing them by 43% since 1990 (CCC 2018), mainly a result of efforts to decarbonise the power sector, which has seen an uptake in the use of renewable sources to generate electricity and a reduction in the use of coal as an energy source. Much greater efforts from other sectors, such as the heating industry, will be required if the UK is to completely decarbonise

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its economy by 2050, particularly as heating accounts for approximately a third of the UK's carbon emissions and almost half of its energy consumption (BEIS 2018).

At present, very little heating and cooling comes from renewable sources in the UK. The European Environment Agency (2018) reported that the UK has one of the lowest shares of renewable energy used to provide heating and cooling in Europe i.e. only 7%. In London, carbon intensive gas-fired boilers make up 90% building heating systems, representing not only a risk to the climate but also to air quality (GLA, 2018). The Greater London Authority (GLA) has also set a target for London to become zero carbon by 2050, which is aligned with the national goal to tackle climate change. Both the GLA and the CCC have recognised how decarbonising heating is an essential part of their climate strategies, highlighting how heat pumps and renewable energy will both play an important role in future zero carbon and efficient energy systems. This paper investigates a new solar thermal energy technology as a source for a heat pump to provide space heating for buildings. The demand for building space heating arises from the need to maintain internal temperatures at a specified level, above the ambient air temperature, generally to meet requirements for human thermal comfort. In the UK, heat may be needed for up to 8 months of the year, although with peak heat during the winter months (January to March). To meet UK climate change targets there is a need for new, flexible, renewable energy based heating systems. The use of heat pumps is one such technology and can provide high efficiencies e.g. 400%+ with only a low electrical energy input requirement.

Transpired solar collectors (TSCs) enable the addition of solar thermal energy to buildings. They are already widely used around the world for pre-heating ventilation air for buildings and for agricultural drying, although they can also be used indirectly to heat water. However, primarly these collectors are solar air heaters that increase the temperature of outside air when drawn through a solar heated perforated metal sheet. The air temperature rise achieved depends on the environmental conditions e.g. ambient air temperature and solar radiation, as well as the solar collector plate design e.g. distribution of perforations and its surface coating, and the flow rate and face velocity of the air at its surface (Hall et al, 2011).

The coatings used by current generation TSCs are generally high emissivity, which results in high radiation heat losses to the outside environment, however, by using spectrally selective coatings, new, low emissivity coatings have been developed, which maintain the collector's absorptivity, while minimizing re-radiation to the environment. This results in higher collector surface temperatures and higher output air temperatures (Hall and Blower, 2016).

The current study investigates the potential of low-ε TSCs to provide solar enhanced heated air for delivery to buildings via ventilation systems. Although the solar collector can increase air temperatures e.g. by 15-20 K (27-36°R), it is an intermittent heat source, which can only provide heat during the day, and varies with the solar radiation available. For an effective heating system, the solar collector needs to be combined with another heating system which can be used to meet the building heat demand at all times. In the current paper, solar collectors have been used to deliver pre-heated air as a heat source for an air source heat pump, to meet the space heating demand profile of a building. Several configurations of heat pump have been considered e.g. both ambient air and exhaust air heat pumps, and these evaluated in combination with low-ε TSCs, to provide pre-heated air for the heat pump, and compared to the counterfactual case, where the solar collector is not used, to investigate potential benefits.

2. METHOD

The model is made up of 3 main components and these are described below.

2.1 Building model

To evaluate the potential of low-ε TSCs combined with heat pumps for the provision of building space heating, a modelling based approach was used using the commercial software tool Engineering Equation Solver (EES) (Fchart, 2019). A building model was developed to estimate the building space heating demand for the heating system to meet. This consisted of a simple air ventilated building subject to heat losses due the seasonal variation in environmental conditions. This included hourly variation in ambient air temperatures, humidity and solar radiation throughout the year, for a selected region within London, UK, which has been identified as a potential site for a practical case study using the new technology.

Assumptions for the building model included:

(i) Dimensions of 10 x 10 x 10 m (32.8 x 32.8 x 32.8 feet)

(ii) Single internal space i.e. no internal structure was considered

(iii) Ventilation rate of 1 air change per hour

(iv) Building internal i.e. room, set temperatures T_room_set of 24°C (75.2°F) for 11 hours during the day and 22°C (71.6°F) for 13 hours overnight

(v) Fabric heat losses i.e. through the walls, floor, roof, windows and doors, as defined by their overall heat transfer coefficient U and area A values, together with infiltration and ventilation heat losses were compounded into a single heat loss coefficient HLC of 530.4 W/K (1027.8 Btu/hr °F) (see note below)

(vi) The temperature difference ΔT between the set inside room air temperature T_room_set and the outside ambient air temperature T_amb, was the main variable affecting the rate of heat loss from the building i.e. ΔT = T_room_set-T_amb

(vii) The total heat loss for the building, equal to the total building heat demand could then be calculated as: Q_build_heat_demand (W)= HLC x ΔT

Note: The heat loss coefficient (HLC) was calculated from the overall heat transfer coefficients (U values) and surface areas for the building fabric i.e. walls, floors, roof and windows and doors, and the ventilation exhaust heat loss coefficient i.e. mass flow rate of exhaust air x specific heat capacity. The values assumed were: (i) U values in W/m^2K (Btu/hr/ft2/°F): walls 0.18 (0.03172); floor 0.13 (0.0229); roof 0.13 (0.0229); windows and doors 1.4 (0.2467) (UK Building Regulations, 2016); (ii) areas from the building dimensions, assuming windows and doors occupied 40% of the front and back walls only; (iii) ventilation air mass flow rate 0.34 kg/s (2698.5 lb/hr), specific heat capacity 1019 J/kg K (0.2434 Btu/lb °F).

Using the above assumptions, a seasonal building heat demand profile was established.

2.2 TSC model

Transpired solar collectors (TSCs) are constructed from a solar absorbing, perforated, metallic sheet attached to a vertical (south facing, in the northern hemisphere) building wall such as to create a small air gap between the sheet and the outside surface of the building. The sheet is then sealed to the building at each edge to form a plenum i.e. thin box. The outside surface of the sheet absorbs solar radiation heat raising its temperature. A fan is then used to force external ambient air through the thousands of evenly distributed perforations in the absorber surface, and heat is transferred from the sheet to the air by convection. The heated air passes through the plenum to an outlet, from where it can be ducted to a selected application e.g. in the present study, to provide heated ventilation air. There is also a bypass opening to permit ambient air to be used directly for ventilation, when solar heated air is not required.

Within the model, previously described by Hall et al (2011) and Hall and Blower (2016), an absorber plate face velocity of the order of 0.02 m s^{-1} (236.2 ft/hr) was generally found to be of the order needed for effective operation of the TSC. The face velocity obtained depends on the air flow rate and the absorber plate area. Hall and Blower (2016) also reported an improved air temperature rise was obtained with low emissivity coated collector plates compared with high emissivity plates, of 20% for the same absorptivity surface. For the TSC model used in the study a face velocity of 0.022 m/s (259.8 ft/hr) was assumed, an absorptivity of 0.9 and an emissivity of 0.2. The mass flow of air through the TSC was based on the ventilation air flow rate calculated in the building model.

For the model, it was assumed that when the solar heated air temperature exceeded the ventilation air temperature needed to meet the building demand, then the building temperature demand could be met (but not exceeded) by means of a control system to mix the solar heated and ambient air prior to its supply via the ventilation system. When the ambient air temperature exceeded the ventilation air temperature requirement based on the building demand, ambient air was used rather solar heated air, and the building temperature was not controlled i.e. no cooling was included in the simulation. The TSC model used in the present study has been previously described by Hall et al (2011). A schematic of a transpired solar collector and the energy balance at the absorber plate surface are shown in Figures 1 (a) and 1 (b) below.

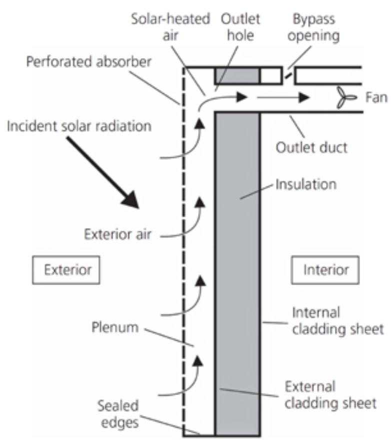


Figure 1 (a) Schematic of transpired solar collector

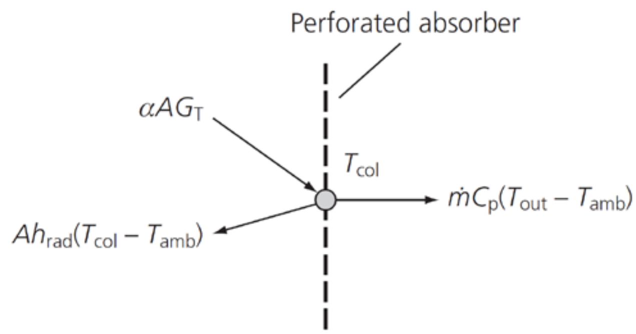


Figure 1 (b) Energy balance at absorber plate surface

In Figure 1 (b), α is the absorptivity of the solar absorber 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}. Cp is the specific heat capacity of the air flowing through the enclosure, and T_{out} is the temperature of the air at the outlet. In fact, the overall energy balance equation, shown below, also includes a term to take account of the convective heat transfer from the absorber plate to the air, namely a heat exchange effectiveness (HEE) coefficient, was calculated using the implementation of Kutscher (1994). The pressure drop through the absorber plate was determined using the Van Decker et al. (2001) scaling factor used to model square pitch perforations. The energy balance for solar absorber plate is given by:

$$\dot{m}C_p HEE\,(T_{col} - T_{amb}) = \alpha AG_T - Ah_{rad}\,(T_{col} - T_{amb}) \qquad (1)$$

Where HEE is defined by:

$$HEE = \frac{(T_{out} - T_{amb})}{(T_{col} - T_{amb})} \qquad (2)$$

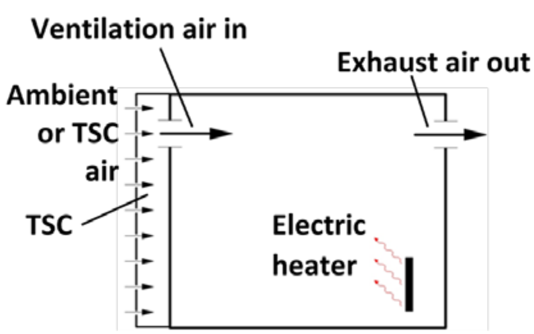
From equation (1) above, it is seen that the main additional input needed was weather data. This was obtained for a location in the Borough of Islington in London, UK, and input to the model as hourly values for ambient air temperature, ground temperature, sky temperature, air pressure, relative humidity, wind speed, global radiation on a south-facing vertical plane and global radiation on a horizontal plane (Meteotest AG, 2019). The TSC model then used the above equations to calculate the hourly variation in solar collector output air temperatures.

2.3 Heat pump model

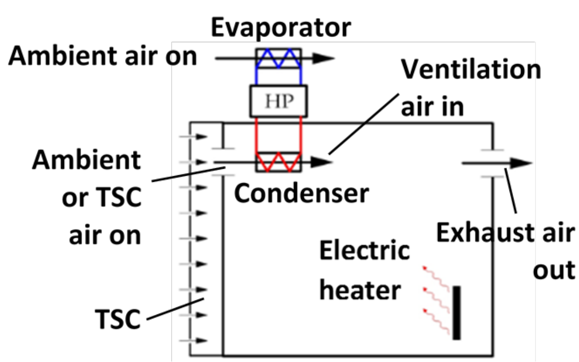
The heat pump model was simulated as a single stage air source heat pump using a series of thermodynamic balance equations to link the input and output parameters and performance of each of its components, as described by Nellis and Klein (2011). Input data for the various components were specified for 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; (ii) the solar collector output air temperatures, which (when greater than the ambient air temperatures), were used either as the condenser air on temperatures, for the heat pump, or mixed with the ambient air and used directly for the ventilation air supply; (iii) the volumetric flow rate for the air over the heat exchangers, which was defined by the ventilation air flow rate required for the building; (iv) the evaporator air on temperature, which for one of the configuration options was ambient air, and for the other was exhaust air from the building (i.e operating as an exhaust air heat pump). The different heat pump systems configurations considered in the present study are described in the next section.

2.4 Options for direct air ventilation based heating systems using TSCs

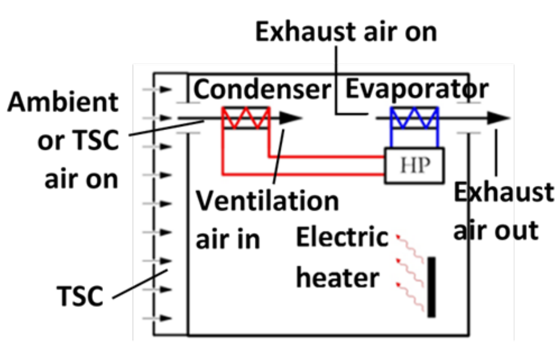
In Figure 2 below, TSC is the transpired solar collector and HP is an air source heat pump. The three heating system configurations were each modelled both with the TSC in operation, and also in bypass mode, where ambient air only was provided.



(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
Figure 2 Building heating system configurations modelled

2.5 Brief description of overall heating system model

Simulation of the overall air ventilation heating system required the use of three different models, namely: (i) a building model, to calculate the building heat demand, and delivery temperature needed for the ventilation air, in order to meet this demand; (ii) a TSC model to calculate the solar heated air temperature i.e. increase above ambient air temperature at the TSC outlet, when solar radiation was available; (iii) a heat pump model, which used output data from the other two models, including the ventilation air delivery temperature needed for the building, and the ambient and solar heated air temperatures from the TSC model. The main output from the model was the electrical input energy needed for the heat pump. The increase in the ventilation air temperature above the room set temperature was limited to 8 K (15°R), so for the settings used in the current model i.e. 24/22°C (75.2/71.6°F), the maximum air ventilation temperature permitted was 30°C (86°F). Where the calculated air delivery temperature (corresponding to the building heat demand) exceeded this value, the heat pump was used to deliver air at 30°C (86°F), and the remaining heat demand met using an electric heater within the building. The total electricity requirement to meet the building demand under each set of conditions was calculated.

3.0 RESULTS FROM MODELLING WORK

Thr building and TSC models were used to generate seasonal (hourly) building heat demand data and seasonal (hourly) TSC output air temperatures respectively, based on the weather data. The outputs from these models are shown in Figures 3 and 4 below.

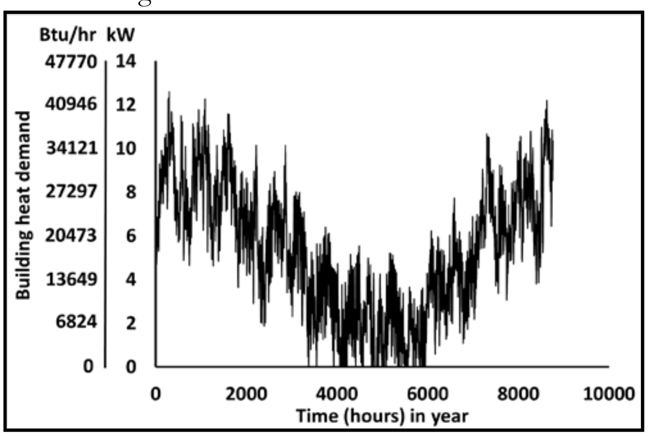


Figure 3 Seasonal variation in building heat demand

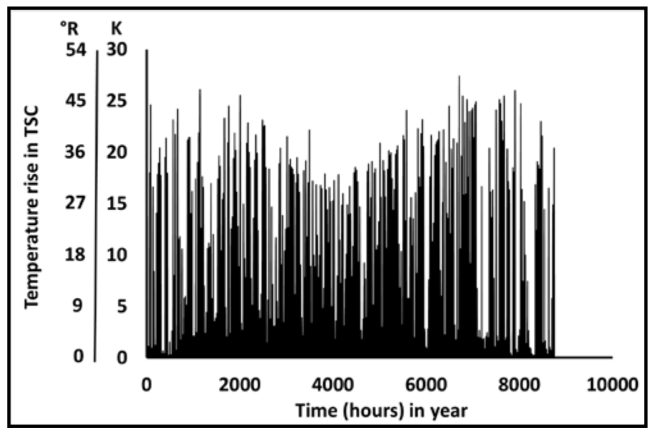


Figure 4 Temperature rise in TSC outlet air

The outputs of building heat demand data and TSC output air temperatures, together with ambient air temperatures were then used by the heat pump model to estimate the electrical energy input to the compressor and fans (and additional top up by the electric heater to fully meet the demand, if required), on an hourly basis throughout the year. The results for two of the configuration options listed in section 2.4, namely option 2 both with and without the TSC applied are shown in Figure 5.

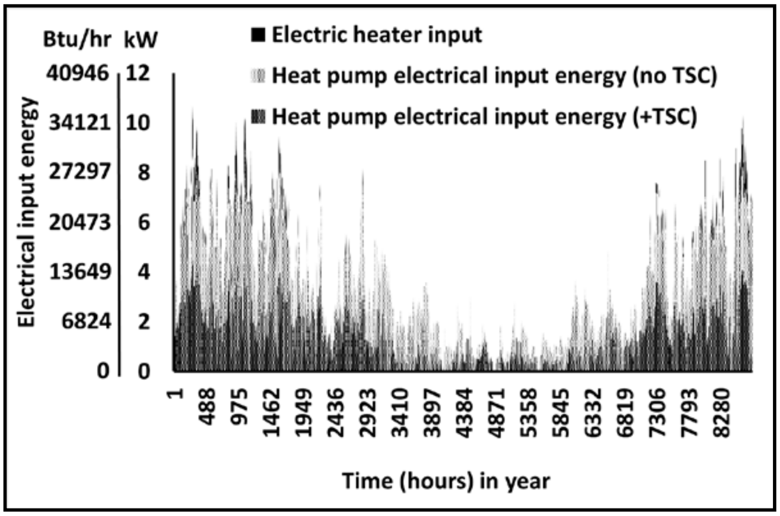


Figure 5 Comparison of electric input required for heat pump (with and without TSC)

The hourly electrical energy input required for each of the 3 options both with and without the TSC air being used were totalled for the whole year, and these together with the annual energy consumption, CO_2e emissions and costs are shown in Table1.

Table 1. Outputs from models

Heating system option	Output	With TSC	Without TSC
Option 1 – Electric heating only, with and without TSC	Total energy use per annum (MWh)(MMBtu)	42.23 (144.10)	48.33 (164.91)
	CO_2e emissions per annum (tonnes)(tons)	10.71 (11.81)	12.34 (13.60)
	Cost per annum (£)($)	£6546.24 ($8191.96)	£7491.54 ($9374.91)
Option 2 – Heat pump using TSC air or ambient air as condenser heat source, and ambient air as evaporator heat source, with electric heater top-up	Total energy use per annum (MWh)(MMBtu)	13.42 (45.79)	15.18 (51.80)
	CO_2e emissions per annum (tonnes)(tons)	3.45 (3.80)	3.88 (4.28)
	Cost per annum (£)($)	£2096.27 ($2623.27)	£2353.65 ($2945.36)
Option 3 – heat pump using TSC air or ambient air as condenser heat source, with exhaust air from room as heat source for evaporator, with electric heater top-up	Total energy use per annum (MWh)(MMBtu)	10.98 (37.47)	12.37 (42.21)
	CO_2e emissions per annum (tonnes)(tons)	2.81 (3.10)	3.16 (3.48)
	Cost per annum (£)($)	£1702.56 ($2130.58)	£1917.70 ($2399.81)

4.0 DISCUSSION

The building heat demand profile shown in Figure 3 runs from January 1st – December 31st for the selected year (2005). It is seen that heat demand is high in the winter months i.e. up to 12.5 kW (42652 Btu/hr), but for a 3-month period in the summer, heat demand is virtually zero. However, throughout the year, the demand varies significantly on an hourly basis, although following a clear overall trend. Figure 4 shows the rise in temperature for the TSC output air, compared with ambient, throughout the year. This shows wide variations in temperature rise throughout the year i.e. between 0 and 25 K (0 and 45°R), as would be expected since the solar radiation will vary substantially from hour to hour. It also indicates that the lowest increase in temperatures was in the summer months when the ambient air temperature is at its highest. Figure 5 shows the data output for electrical energy requirement by the heat pump models for option 2 (+/-TSC), hourly throughout the year. It is seen that there is a significant reduction in heat pump energy input needed where the TSC is used. In both cases, there is very little heat pump energy needed during the summer months, when the heat pump would be switched off. Table 1 shows the total electrical energy input for the year, for the three configuration options both with and without the TSC in operation. There is a clear reduction in energy use of between 11 and 13% where the low-ε TSC is used. CO_2e emissions and costs are also substantially reduced. Comparing the two heat pump configurations (Options 2 and 3), it is seen that Option 3, where the exhaust air from the room is used as the heat source for the evaporator, offers the greatest savings.

5.0 CONCLUSIONS

It is concluded from the model results presented that there is a clear benefit from using low-ε TSCs, with energy, carbon and costs savings of between 11 and 20%, for the three heating system options considered here. Also, it is concluded that low-ε TSCs can combine well with ventilation air and air source heat pump systems with great benefits. Clearly, TSCs will also generated significant increases in output air temperatures during the summer (if a little lower than at low ambient air temperatures), while the building heat demand is low or zero, and the TSC generated heat will not be used at this time, as in the present case. However, if this heat can be used for other applications or stored for future use, further benefits can be realized.

ACKNOWLEDGEMENTS

This research is part of the 'Steel Zero and its Low Carbon Heating Applications' project, which is funded by the HM Government Department of Business, Energy and Industrial Strategy (BEIS) Low Carbon Heating Technology Innovation Fund.

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