Build-up and performance test of a novel solar thermal roof for heat pump operation

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ABSTRACT
Global increase in energy demand and fossil fuel prices loaded ever-increasing pressure on identifying and implementing new means to utilise clean and efficient energy resources. Due to the environmental benefits, technical and economic possibilities of Solar-Assisted Heat Pump Systems, there has been a growing interest for such hybrid systems with a variety of system configurations for various climates. International Energy Agency Task 44 of the Solar Heating and Cooling Programme has recently started working on finding methods to most effectively use solar heat pump systems for residential use. In the present study, a novel solar thermal roof collector was developed by primarily exploiting components and techniques widely available on the market and coupled with a commercial heat pump unit. The proposed indirect series Solar-assisted Heat Pump system was experimentally tested and system performance was investigated. Yet, the analysis based on indoor and outdoor testing predominantly focuses on the solar thermal roof collector. A detailed thermal model was developed to describe the system operation. Also, a computer model was set up by using Engineering Equation Solver to carry out the numerical computations of the governing equations. Analyses show that the difference in water temperature could reach up to 18°C while maximum thermal efficiency found to be 26%. Data processing of the series covering the test period represents that Coefficient Performance of the heat pump (COPHP) and overall system (COPSYS) averages were attained as COPHP = 3.01 and COPSYS = 2.29, respectively. An economic analysis points a minimum payback period of about three years for the system.

Nomenclature

| Variable | Definition |
|----------|------------|
| A_eff | Effective area, m² |
| C | Specific heat |
| COP | Coefficient of performance |
| D_i | inner diameter of heat exchanger tube, m |
| D_o | Outer diameter of heat exchanger tube, m |
| e | Per cent deviation |
| F | Collector efficiency factor |
| G_r | Grashof number |
| g | Gravitational acceleration, m/s² |
| h | Heat transfer coefficient, W/m²K |
| H | Enthalpy, kJ/kg |
| I | Incident solar radiation, W/m² |
| K | Thermal conductivity, W/m.K |
| m | variable defined to solve differential equations |
| m | Mean flow rate, m/s |
| Nu | Nusselt number |
| Q | Energy, W |
| T | Temperature, °C |
| U | Thermal transfer coefficient, W/m²K |
| W | Distance between tubes, m |
| W | Compressor power, W |
| V | Wind speed, m/s |

Greek symbols

| Symbol | Meaning |
|--------|---------|
| α | Absorption |
| δ | thickness, m |
| ε | Emissivity |
| η | efficiency, % |
| λ | thermal conductivity, W/m.K |
| Σ | Total |
| ρ | density, kg/m³ |
| θ | Stefan–Boltzmann constant |
| T | Transmittance |

Subscripts

| Subscript | Meaning |
|-----------|---------|
| a | ambient air |
| abs | absorber surface |
| c | cover layer |
| cp | electrical energy in compressor |
| e | collector side |
| c | thermal energy in condenser |
| cv | convective |
| e | evaporator |
| e | thermal energy from evaporator |
| f | fluid |

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1. Introduction

Excessive increase in energy demand, escalating fossil fuel prices, global warming and climate changes have brought a new dimension to seek for sustainable energy resources with a minimum impact on the environment. Solar energy – as a clean, sustainable and environment-friendly energy source – can play an important role in the use of energy resources (Amin and Hawlader 2013). Significant utilisation of solar energy takes place especially in regions where relatively low temperature (\(< 100^\circ\text{C}\)) is obtained, that is, drying in agricultural processes (Chua et al. 2002).

Solar collectors for domestic hot water production and space heating have been extensively used, especially where the solar radiation is abundant. However, conventional solar thermal collectors have some practical drawbacks, which include susceptibility to damage during transportation, which could lead to large financial losses and material waste; expensive to make and to install; not aesthetically appealing, nor easily integrated into roofs or facades; fragile means they could easily be vandalised (Buker, Mempouo, and Riffat 2013). Integration of solar systems is essential, and some recent investigations have been closely focused on this (Kalogirou 2013). Integrating unique polyethylene heat exchanger loop into a novel roof construction to form a ‘Sandwich’ thermal roof collector could be an efficient alternative to overcome those drawbacks of conventional solar thermal collectors. ‘Sandwich’ Thermal Roofs, as solar absorbers, would be used to maximise the solar heat energy absorption. It is wise to say that roofs have great solar thermal potential due to their large size, being free from shadowing and orientation limitations (Stasinopoulos 2002).

For heat pumps, a fundamental factor of great importance for a cost-effective operation is the availability of a cheap, reliable heat source for the evaporator. The grade of energy can be improved by coupling solar energy with heat pump and the combination can represent interesting solutions for various domestic applications. It was shown that the evaporator–collector employed in such a system can absorb both solar thermal and ambient energy (Georgiev 2008). Also, a series of experiments were conducted with R 22 and 744 to identify proper refrigerant for such a system (Choi, Kang, and Cho 2014).

Solar-assisted heat pump systems (SAHP) to produce hot water have been widely investigated in both theoretical and experimental studies. To name a few, a considerable number of studies, both theoretical and experimental, have been conducted on the exergetic performance evaluation of heat pumps systems with various heat sources (Pons 2012) such as ground heat, solar heat and heat recovery (Beccali, Finocchiaro, and Nocke 2012), heat pump system having photovoltaic PV/T evaporator (Liu et al. 2009), solar-assisted heat pump with an energy storage (Karacavus and Can 2008), on a direct-expansion solar-assisted heat pump water heating system (Kuang, Sumathy, and Wang 2003), heat pump systems for district heating (Zhen et al. 2007) and a direct-expansion solar-assisted heat pump with integral storage tank for domestic water heating under zero solar radiation conditions (Fernández-Seara et al. 2012). The measurements have been undertaken to discover the building space heating capabilities of the SAHP (Bakirci and Yuksel 2011; Yang, Wang, and Zhu 2011). Dynamic simulation and parametric optimisation of a solar-assisted heating and cooling system have been carried out (Calise, Dentice d’Accadia, and Vanoli 2010). Experimental performance analysis of a solar-assisted ground-source heat pump for green house heating (Choi, Park, and Kang 2014) and the development and performance analysis of absorption heat pump system driven by solar energy (Sözen, Altıparmak, and Usta 2002) have been analysed. Solar-assisted heat pump applications and performance assessments for agricultural and marine products drying have been reviewed (Daghgh et al. 2010), mushroom drying concept has been analysed (Şevik et al. 2013) and the performance of a multi-functional solar integrated heat pump air conditioner, water heater and dryer has been studied (Hawlader and Shaochun 2008). A techno-economic evaluation of a solar-assisted combined heat pump has also been made in three different locations (notably Ankara, Denver and Bochum) to indicate the economic feasibility of the system (Schimpf and Span 2015). A grid-tied PV system to power a heat pump water heater has been experimentally studied in a hotel in South Africa, with the objective to reduce the energy cost and obtain competitive feed-in tariff rates (Sichilalu and Xia 2015). A direct expansion SAHP and long-term performance monitoring and long-term life-cycle-cost (LCC)-based techno-economic analysis of the energy conversion potential and economic viability of the system has been evaluated and meaningful results have been obtained (Chaturvedi, Gagrain, and Abdel-Salam 2014). An economic optimisation of evaporator and air collector area of a solar-assisted heat pump drying system has been presented by comparing the cost flows recurring throughout the lifetime of the solar and conventional alternative systems, and sufficient amount of savings during the life cycle with a minimum payback period of four years has been obtained (Rahman, Saidur, and Hawlader 2013).

Combination of the heat pump and solar energy is a mutually beneficial way of enhancing the coefficient of performance of a heat pump and solar collector efficiency. The heat pump Coefficient of Performance (COP) can be elevated to a great extent on the temperature of the evaporator. The solar collector loop enables us to boost the heat source temperature of the heat pump, thereby improving the annual and seasonal performance...
of the heat pump. A COP value of about 6.38 was experimentally obtained in a SAHP domestic heating system with an evacuated tubular collector (Çağlar and Yamali 2012); the COP value attained in a SAHP water heating system using unglazed flat-plate collectors (Georgiev 2008) was 9. The average COP of heat pump system for agricultural drying was found to be 7 in tropics (Choi, Park, and Kang 2014). Zhang et al. (2014) developed and tested a novel solar PV/loop-heat-pipe HP system for space heating or DHW. The solar heat was exchanged through a flat-plate heat exchanger acting as the condenser of the heat-pipe loop and the evaporator of the heat pump cycle. The average values of COP for both thermal and PV/T were found to be 5.51 and 8.71 under the prevailing climatic conditions of Shanghai, China. The result proved that the proposed system has a potential of achieving enhanced solar thermal efficiencies around 1.5–4 times more than conventional counterparts. In another study, Shilin and Fei (2010) proposed and experimentally studied an indirect-expansion SAHP radiant floor space heating system in Beijing, China. The configuration of the system varied as solar alone, solar with water-source heat pump and water source heat pump alone, depending on the solar collector’s outlet temperature. All those studies listed above and many others prove that combination of solar energy with heat pump system can yield high performances and contribute to more sustainable environment.

Exergy analysis is a primary thermodynamic performance evaluation of energy systems in addressing the impact of energy resource utilisation on the environment. It is an effective technique to determine the actual magnitude of wastes and losses for more efficient use of energy resources via applying conservation of mass and energy principles along with the second law of thermodynamics (Pons 2012).

1.1. IEA: SHC Programme Task 44

The objective of International Energy Agency (IEA)’s Solar Heating and Cooling (SHC) programme Task 44 was to optimise combined solar and heat pump systems for single house applications and to provide a common definition for the performance of such systems (Hadorn 2012). This project was performed along with the IEA Heat Pump programme denoted as Annex 38. The task mainly focused on small-scale heating and domestic hot water systems, electrically driven heat pumps, etc. Task 44 also involved the assessment of solar and heat pump systems under Subtask B that aims to set common definition of performance metrics and assessment criteria as well (D’Antoni and Sparber 2011). Loose et al. (2011) performed field tests of various combined SAHP systems with different heat sources and presented results, as part of IEA, SHC Programme Task 44. The system employed collectors and geothermal heat pump with borehole heat exchangers to provide DHW and space-heating to a new structure with radiant floor heating system. The collectors fed the storage directly when sufficient solar radiation was available. Otherwise, low-grade energy from the collectors would be used in the heat pump for space heating. Upon reaching energy storage capacity in the tank, the heat pump operation was inactivated; the collectors would feed the borehole heat exchanger. So, the heat pump would draw the stored energy from the boreholes after summer time. This configuration was monitored for three years and promising results were obtained. The solar regeneration of ground boreholes ensured that the HP system could run with a high rate of seasonal performance factor in a long term. In another study, conducted as part of IEA, SHC programme Task 44 Haller and Frank (2011) introduced a mathematical relation model by using TRNSYS in order to determine if combined solar collector-evaporator of a dual (solar-air) source heat pump configuration was more advantageous than utilising solar heat directly. The dual configuration examined in this study can switch its operation between serial and parallel mode. Indirect utilisation of solar heat was shown to be more beneficial in terms of system performance factor when the solar radiation is under a certain level, which depends on solar collector and heat pump characteristics. According to simulation results, using uncovered solar collectors was more advantageous in series mode and also, increasing the runtime of the collectors could enhance the performance of the series configuration.

1.2. Objective

The objective of this study was to investigate a roof integrated ‘Sandwich’ solar thermal roof collector that properly blends into surrounding, thus avoiding ‘add on’ appearance and having a dual function (heat absorption and roofing). Even though a low-cost ‘Sandwich’ roof may produce lower fluid temperatures than water-based glazed collectors, it is also able to absorb heat from the ambient. Another objective is to address the inherent technical pitfalls and practical limitations of conventional solar thermal collectors by bringing an inexpensive, maintenance-free and easily adaptable solution. Although substantial studies have been performed on the combination of solar modules with heat pumps, integration of such a novel roof construction with heat pump has yet to be considered. The present study will both theoretically and experimentally evaluate the thermal performance of the ‘Sandwich’ roof unit with heat pump according to performance assessment criterions set by IEA: SHC Task 44.

2. Description of the ‘Sandwich’ roof unit and its combined operation with heat pump

The system consists of a Thermal ‘Sandwich’ Roof collector that can act as both roof element and low-grade heat source for the heat pump. The roof module contains several layers, namely the outer surface is a black-painted aluminium solar radiation absorber, polyethylene heat exchanger loop tightly screwed to the black aluminium surface for enhanced heat transfer and injected insulation foam back layer against heat loss. Furthermore, thermally conducting adhesion was used to improve the heat transfer between the aluminium cover and poly heat exchanger pipes. By this means, all layers are integrated into the same module to form a ‘Sandwich’ Thermal Roof collector.

In the roofing structure, steel frames form a roof truss which weighs and fits the modules to build up a well-integrated roof. As for the plumbing of the system, supply and return pipes are placed underneath the roof truss and joined using flexible coupled connectors with valves to provide a leak-free connection. In doing so, the supply pipe feeds cold water/glycol mixture to polyethylene heat exchanger, and the return pipe transports the
hot water/glycol mixture from heat exchanger. To some extent, the sandwich module and roof support are well fixed together with the screws to form a properly functional solar thermal roof. Figure 1 is the illustration of layers of ‘Sandwich’ thermal roof collector.

In operation, the ‘Sandwich’ module receives a substantial amount of solar insolation falling upon the black-coated solar absorbing surface as a means of thermal energy, while dispersing remainder to the surrounding. The black-painted aluminium top cover enhances the solar heat absorption. As the second law of thermodynamics states that when an object is at a different temperature from its surroundings, heat flows so that the body and surroundings reach the same temperature, at which they are in thermal equilibrium. Such a spontaneous heat transfer occurs from high temperature to low temperature. Thus, setting the surface at a lower temperature than the surrounding by circulating cold water/glycol mixture will cause that heat flow, thermodynamically, moves towards roof collector. The absorbed heat, then, will be conveyed through the circulating water/glycol mixture across the heat exchanger. Consequently, hot water, thus enhanced solar thermal efficiency would be expected. Moreover, fixing the tubes and risers of polyethylene heat exchanger tightly to the black aluminium surface by means of screws would potentially lead to good contact between the absorber and heat exchanger tubes, thus having relatively low temperature difference between the module surface and heat carrying fluid. The heated fluid (water/glycol mixture) obtained out of this operation can be utilised for heat pump operation.

The system also comprises 3 kW thermal output commercially available heat pump package designed for space heating (through underfloor or radiator) and domestic hot water for buildings. The Indirect Solar-assisted HP system includes a solar thermal collector, evaporator, compressor, condenser and an expansion valve. In operation, the ‘Sandwich’ structure roof unit supplies free, low-grade solar thermal energy to the heat pump evaporator. The solar heat is exchanged through a coil heat exchanger acting as the condenser of the collector loop. Within the secondary sealed side of the evaporator heat exchanger, there is heat transfer fluid, also called refrigerant. When the water/anti-freeze mixture enters the evaporator, the solar thermal energy absorbed by the novel roof is transferred into the refrigerant, which begins to boil and becomes a gas. The refrigerant physically does not mix with the water/anti-freeze mixture. They are separated by a sandwich-like heat exchanger which enables the heat transfer. The gas is then introduced to the compressor, in which the refrigerant gas pressure increases, which rises gas temperature as well. The high-temperature refrigerant gas then moves along a second heat exchanger, also called the condenser, which has same characteristics as set of heat transfer plates. The condenser supplies

Figure 1. Cross-sectional view of the Novel ‘Sandwich’ thermal roof collector.

Figure 2. Schematic of the combined system.
Table 1. Technical parameters.

| Component                   | Parameter Description                                      | Value                  |
|-----------------------------|------------------------------------------------------------|------------------------|
| Sandwich Roof               | Exposed roof area ($A_{eff}$)                             | 1.92 m$^2$             |
|                             | Dimensions ($H \times W \times D$)                        | 1.6 m $\times$ 1.2 m $\times$ 0.02 m |
| Aluminium                   | Thickness                                                 | 5 mm                   |
|                             | Thermal conductivity                                      | 450 W/m$^2$ K at 25°C  |
| Polyethylene heat exchanger | The internal diameters of tubes ($D_i$)                   | 0.0027 m               |
|                             | The external diameters of tubes ($D_o$)                   | 0.0043 m               |
|                             | Distance between tubes ($)                                 | 0.001 m                |
|                             | Max temperature allowed in poly HE ($T_{max}$)            | 60°C                   |
|                             | Max pressure allowed in poly HE ($P_{max}$)               | 1 MPa                  |
| Two part polyurethane liquid foam | Expanding size                                 | 25 times of its original volume |
|                             | Time to begin reaction when mixed (%)                     | 20–25 s                |
|                             | Time to reach max size (%)                                 | 150 s                  |
| Heat pump                   | Nominal thermal output ($)                                 | 3 kW                   |
|                             | Process medium (refrigerant) ($)                          | R-134a                 |
|                             | Max inlet temperature ($)                                  | 15°C                   |
|                             | Min outlet temperature ($)                                 | −5°C                   |
|                             | Max flow temperature ($)                                   | 65°C                   |
| Circulating pump            | Dimensions ($H \times W \times D$)                       | 0.53 $\times$ 0.475 $\times$ 0.37 m |
|                             | Dry weight                                                | 50 kg                  |
| Hot water storage           | Capacity                                                  | 55 l                   |
|                             | Max working head                                           | 10 m                   |
|                             | Standing heat loss                                         | 1.5 kW/h per 24 h      |
| Solar simulator             | Tilt angle                                                | 15°                    |
|                             | Max lighting                                               | 400 W                  |

2.1. Prototype design

A prototype of the combination of solar thermal roof and heat pump system was assembled and tested both in the laboratories of Institute of Sustainable Energy Technology at the University of Nottingham and under the prevailing climatic conditions of Nottingham, UK, which is geographically located at 52.57°N and 1.09°W. The key innovation in the proposed system is the utilisation of the solar thermal energy as a renewable source with relatively efficient and low-cost components by a thermodynamically effective design. Initial design parameters are given in Table 2. Figure 4 shows the schematic diagram of the roof system.

As absorber, 5-mm-thick aluminium cover was coated with high-temperature silicone spray in order to enhance solar thermal absorption and avoid moisture condensation and freezing on the collector. As heat extraction component, polyethylene heat exchanger loop was tightly fixed to the black aluminium surface via screws to have a good touch to the absorber for enhanced heat transfer. As roof finishing, insulation foam was injected as back layer against heat loss.

The heat exchanger loop is made of polyethylene thermoplastic substance and its physical structure makes it first-of-its-kind. The pipes and fittings were incorporated via welding with soldering iron. The flow channels have 2.7 mm of internal and 4.3 mm of external diameters, and the distance between the flow channels is 10 mm. Technical parameters of the heat exchanger can be found in Table 1. In order to maintain the heat pump operation under the climatic conditions where temperatures are below freezing point, the anti-freeze concentration within the novel roof arrays is kept 20% for the protection level of −10°C. The concentration level can be detected with a refractometer.

The heat pump unit was connected to a 55-l storage tank with a circulation pump and a flow meter regulator to modulate the mass flow rate of hot water. The storage tank was thoroughly insulated with foam insulation material in order to reduce the heat losses. The water inside the storage tank was heated by employing a helical copper tubular heat exchanger. An expansion vessel was also installed in between the heat pump and storage tank coupling to reduce the likely damages due to excessive increase in pressure, and also the fluid pressure in the solar thermal roof unit is monitored by a pressure gauge, as shown in Figure 2.
Figure 3. Experimental design – novel solar thermal roof with solar simulator, heat pump, water storage.

The circulating pumps employed in close loop solar roof systems must overcome the head pressures. Also, the pump should be compatible with solar applications in terms of its size, speed, efficiency, and hydraulics and power consumption. Technical details of the circulation pump are provided in Table 1.

An array of 30 halogen floodlights was employed in order to simulate the artificial solar sunlight. The solar sunlight simulator was evenly fixed on an aluminium frame installed above and in parallel to the ‘Sandwich’ roof, as illustrated in Figure 2. The array was formed as three sub arrays and was connected to the grid through a three-phase transformer, which allows the level of the radiation flux to be gradually regulated. The maximum electrical power to be consumed by each floodlight is 400 W. A Kipp and Zonen pyranometer with sensitivity of $17.99 \times 10^{-6} \text{V/W/m}^2$ was mounted on the vertical surface to measure the radiation flux on the surface of the novel roof. An anemometer was also installed for outdoor test. K-type thermocouples were inserted to investigate the temperature variation across system. All related data were recorded on the data logger, DT500; direct connection between data logger and PC was provided to store data and export on Excel spreadsheet (Figure 3).

3. Mathematical modelling and EES model set-up

Energy transfer arose within the proposed system has two major processes, including conversion of solar radiation into thermal energy by ‘Sandwich’ roof unit and upgrading the evaporation heat into higher grade energy through the heat pump. These two processes are interconnected and eventually reach a balance under the steady-state operation. A numerical model which is well suited to the physical structure of the proposed system was adapted to simulate the thermal performance of the novel system by using EES (Engineering Equation Solver) Software. Several assumptions have been considered during the first and second law analyses of the system as follows:

(a) The overall performance of the system is at a quasi-steady-state condition with negligible potential and kinetic energy effects and there is no chemical reactions taken for all processes.
(b) All surfaces of layers cover uniform temperature and temperature gradient around tubes is negligible.
(c) There is forced flow through heat exchanger tubes and the pressure drop is negligible due to the short length of pipelines.
(d) Due to advanced thermal resistance features, heat loss through the insulation and edge is negligible.

The method and mathematical equations presented in this study are not necessarily the most accurate available, but they are widely applied, easy to use and adequate for most of the design computation. Heat transfer profile of the roof unit and P-h chart of the refrigeration cycle are shown in Figures 4 and 5, respectively.

3.1. Energy analysis of ‘Sandwich’ Roof unit

Initially incoming Solar Energy is defined as

$$Q_{\text{abs}} = \alpha_{\text{abs}} \cdot A_{\text{eff}} \cdot I.$$  \hspace{1cm} (1)

Heat loss is mainly through the top due to the good back insulation features of ‘Sandwich’ Roof unit. So the heat loss can be expressed as

$$Q_l = h_c \cdot (T_{\text{abs}} - T_a) + \varepsilon_{\text{abs}} \cdot \sigma \cdot (T_{\text{abs}}^4 - T_{\text{sky}}^4).$$  \hspace{1cm} (2)

Combining above equations provides a revised expression of the useful heat energy ($Q_t$):

$$Q_t = Q_{\text{abs}} - Q_l.$$  \hspace{1cm} (3)

The natural convection heat transfer coefficient ($h_c$) from roof to the ambient is taken into account for wind velocities lower
than or equal to 6 m/s (V) by (Sharles and Charlesworth 1998);

\[ h_c = 6.5 + 3.3 \cdot V. \] (4)

The sky temperature can be obtained by Swinbank equation as a function of the ambient temperature by

\[ T_{sky} = 0.037536 \cdot T_a^{1.5} + 0.32 T_a. \] (5)

Radiation heat transfer is defined as

\[ h_{rd} = \varepsilon_c \cdot \sigma \cdot (T_{abs}^4 - T_{sky}^4). \] (6)

The heat flow goes through conduction across the absorber (black aluminium), the walls of heat exchange tubes and eventually conveyed to the flowing water/glycol mixture. The heat gain of the water/glycol mixture is as equal as the useful heat energy as

\[ Q_t = A_{eff} \cdot U_t \cdot (T_{abs} - T_w). \] (7)

Overall thermal resistance between absorber and refrigerant is

\[ U_t = (U_{abs,heo}^{-1} + U_{heo,hein}^{-1} + U_{hein,w}^{-1})^{-1}. \] (8)

Heat transfer coefficient \( U_{abs,heo} \) from absorber aluminium layer to outer wall of heat exchanger can be obtained by

\[ U_{abs,heo} = \frac{\lambda_{abs}}{\delta_{abs}}. \] (9)

The temperature at the outer wall of the polyethylene heat exchanger becomes

\[ T_{heo} = T_{abs} - \frac{Q_t}{A_{eff} \cdot U_{abs,heo}}. \] (10)

Heat transfer coefficient \( U_{heo,hein} \) from outer wall to interior wall of heat exchanger is defined as (Zhao et al. 2011)

\[ U_{heo,hein} = \frac{2 \cdot \pi \cdot \lambda_{he}}{\ln \left[ \frac{D_o}{D_i} \right]}. \] (11)

The average temperature at the inner wall of polyethylene heat exchanger is given as (Zhao et al. 2011)

\[ T_{hein} = T_{heo} - \frac{Q_t}{A_{eff} \cdot U_{heo,hein}} \] (12)

Heat transfer coefficient \( U_{hein,r} \) from interior wall of heat exchanger to water can be obtained as follows;

\[ U_{hein,r} = \frac{Nu_r \cdot \lambda_r}{D_i}, \] (13)

where Nusselt number is expressed as (Duffie and Beckman 1980)

\[ Nu_r = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4}. \] (14)
The Reynolds number for flows in pipe or tube is generally defined as:

\[ Re = \frac{Q \cdot D_{he}}{\nu \cdot A_p} \]  (15)

In order to compare the theoretical results with the experimental outcomes, the root mean square per cent deviation (\(e\)) and the coefficient of correlation (\(r\)) have been examined by using the following expressions:

\[ e = \sqrt{\frac{\Sigma(e_i)^2}{N}} \]  (16)
\[ r = \frac{n(\Sigma X \times Y) - (\Sigma X) \times (\Sigma Y)}{\sqrt{n(\Sigma X^2) - (\Sigma X)^2} \times \sqrt{n(\Sigma Y^2) - (\Sigma Y)^2}}. \]  (17)

Stating the expressions used by

\[ m^2 = \frac{U_1}{K_\delta}. \]  (18)

Then, the fin efficiency factor (\(F\)) can be found by

\[ F = \tanh \left( \frac{mW - D_o}{2} \right). \]  (19)

The collector efficiency factor (\(F'\)) becomes

\[ F' = \frac{1}{\frac{W_U_1}{D_oU_{abs,pl}} + \frac{W_U_2}{D_o} + \frac{W_U_3}{D_o + (W - D_o)F}}. \]  (20)

The heat removal factor (\(F_R\)) is given by

\[ F_R = \frac{mC_f}{A_{eff}U_fF'} \left[ 1 - \exp \left( - \frac{A_{eff}U_fF'}{mC_f} \right) \right]. \]  (21)

Energy efficiency is usually defined as the ratio of total energy output to the total energy input as

\[ \eta = \frac{E_{out}}{E_{in}}. \]  (22)

### 3.2. Energy analysis of heat pump unit

The water from the Novel solar thermal roof transfers heat to the evaporator in the heat pump. There it releases heat to liquid refrigerant passing separately across the evaporator, allowing the environmentally friendly refrigerant R-134a to vapourise. The water then returns to the solar thermal collector to capture heat again in a continuous cycle. The preheated refrigerant vapour goes through the compressor, where it is compressed to high pressure and upgraded to a much higher temperature. High-temperature vapour, then, passes through the condenser, where it is surrounded by water from the heating system and releases heat to the water circulating via channels. Consequently, vapour becomes saturated liquid with a same pressure. The temperature and pressure of the refrigerant is further reduced by flowing through the expansion device, from where it turns back to the evaporator to repeat the cycle (Zhao et al. 2011).

The heat received by the evaporator (\(Q_e\)) is given by the following equation:

\[ Q_e = m_t \cdot (H_1 - H_4). \]  (23)

The heat introduced to the condenser (\(Q_c\)) is given by

\[ Q_c = m_t \cdot (H_2 - H_1). \]  (24)

The electrical power required to drive the compressor (\(Q_{cp,e}\)) is given by

\[ Q_{cp,e} = m_t \cdot (H_2 - H_1). \]  (25)

Coefficient of performance of the heat pump (\(COP_{HP}\)) is given by

\[ COP_{HP} = \frac{Q_c}{W_{cp}}. \]  (26)

Coefficient of performance of the overall system (\(COP_{SYS}\)) is the ratio of condenser load to the total power consumption of the compressor and other two circulation pumps and it is given by (Atmaca and Kocak 2014)

\[ COP_{SYS} = \frac{Q_c}{W_{cp} + W_{cs} + W_{ls}}. \]  (27)

### 3.3. Uncertainty analysis

The experimental uncertainties were determined by applying Gauss propagation law. The result \(R\) is calculated as a function of the independent variables \(x_1, x_2, x_3, \ldots, x_n\) and \(w_1, w_2, w_3, \ldots, w_n\) represents the uncertainties in the independent variables. Then, uncertainty \(R\) is expressed as (Buker, Mempouo, and Riffat 2014)

\[ w_R = \left[ \left( \frac{\partial R}{\partial x_1} \right)^2 + \left( \frac{\partial R}{\partial x_2} \right)^2 + \ldots + \left( \frac{\partial R}{\partial x_n} \right)^2 \right]^{1/2}. \]  (28)

The independent parameters measured in the experiments are solar radiation, temperature of solar thermal roof, temperature of outlet water, ambient temperature, flow rates and power consumption of the heat pump. Experiments were conducted by using following instruments: a Kipp and Zonen CM11 pyranometer with 1% accuracy, K-type thermocouples with the maximum deviation of ±0.3°C, liquid flow indicators with the accuracy of ±2% and a watt meter with ±2% accuracy.

It is obtained from the Equations (1)–(3), (6), (10), and (13) that the \(\eta_t\) is the function of several variables, each subject to uncertainty:

\[ \eta_t = f(T_w, T_{abs}, T_a, \dot{m}, \dot{I}, W_{cp}), \]  (29)

where subscripts \(T_w, T_{abs}, T_a, \dot{m}, \dot{I}\) and \(W_{cp}\) stand for temperature of outlet water, temperature of collector surface, ambient temperature, mass flow rate, solar radiation and power consumed by compressor, respectively.
Total uncertainty for overall system efficiency can be expressed as

$$w_R = \left[ \left( \frac{\partial \eta}{\partial T_w} w_{T_w} \right)^2 + \left( \frac{\partial \eta}{\partial T_{abs}} w_{T_{abs}} \right)^2 + \left( \frac{\partial \eta}{\partial T_a} w_{T_a} \right)^2 + \left( \frac{\partial \eta}{\partial \dot{m}} w_{\dot{m}} \right)^2 + \left( \frac{\partial \eta}{\partial I} w_{I} \right)^2 + \left( \frac{\partial \eta}{\partial \dot{W}/\Sigma} w_{\dot{W}/\Sigma} \right)^2 \right]^{1/2}$$

(30)

Total uncertainty rate affecting the efficiency of the proposed system was computed by using Equations (28)–(30). The estimation implies that total uncertainty in calculation of the efficiency is found to be 4.08%.

4. Results and analysis

The theoretical analysis and experimental investigation were performed to evaluate the performance of the solar sandwich roof along with a heat pump unit. In analysis, the effects of the various operating parameters (including thermal and overall efficiencies) on the performance of the envisaged system were also investigated. The developed mathematical model to describe the system operation was programmed by using EES. In this software, energy and mass balance equations and the boundary conditions are solved simultaneously. In order to validate the mathematical model, the theoretical and experimental results were compared and shown that there is rather a good compliance between them. The series of outdoor testing was performed according to standard testing procedure EN 12,975 (Standard and BS EN 2006) and the results belonging to testing period was presented. From the experimental data, the temperature of the water outlet from the innovative solar thermal collector, the mean values of the COP of the heat pump and

**Table 3.** Measured parameters and experimental results in average.

| Parameter                                              | Value       | Unit   |
|--------------------------------------------------------|-------------|--------|
| Solar radiation                                       | 701.3       | W/m²   |
| Condensing temperature (space heating only mode)       | 35          | °C     |
| Evaporating temperature                               | 8           | °C     |
| Temperature of potable water                          | 12          | °C     |
| Temperature of water in energy storage tank            | 29.87       | °C     |
| Water-glycol flow rate in solar collector              | 0.074       | kg/s   |
| Water flow rate in load side                          | 0.074       | kg/s   |
| Refrigerant flow rate in heat pump                     | 0.106       | kg/s   |
| Average outdoor air temperature                       | 27          | °C     |
| Power input to the heat pump                          | 0.8         | kW     |
| Power input to the refrigerant water pump              | 0.125       | kW     |
| Useful heat received from the solar thermal collector  | 2.9         | kW     |
| Instantaneous collector efficiency                     | 0.209       | –      |
| Heating COP of the heat pump                          | 2.904       | –      |
| Heating COP of the overall pump                        | 2.29        | –      |

![Figure 6](image-url) Performance of the novel roof under 400 W/m² solar intensity and various flow rates.
of the overall system and efficiency ratios were deduced and presented in Table 3.

The novel solar thermal roof unit presented in Figure 1 was experimentally investigated to test the system performance under various solar intensities employing the solar simulator to control the incoming solar radiation. The reason that Figures 6–8 provided is to demonstrate the collector performance under various solar intensities and the effect of different flow rates on the temperature of flowing water/glycol mixture. The graphs show the dynamic heat transfer on change in modular roof’s performance before reaching a steady state. Figure 6(a) presents the performance of the ‘Sandwich’ thermal collector under 400 W/m² and 0.074 kg/s water flow rates. The surface temperature ranges between 20.5°C and 21.6°C with a flowing water temperature in the range of 9.7°C and 11.3°C and a maximum temperature increase of around 2°C. Figure 6(b) presents performance of the ‘Sandwich’ thermal collector under 400 W/m² and 0.049 kg/s water flow rates. The surface temperature increases from 21.4°C to 23.8°C, while water temperature varies between 11°C and 12.3°C under 0.049 kg/s flow rate. Figure 6(c) presents performance of the ‘Sandwich’ thermal collector under 400 W/m² and 0.025 kg/s water flow rates. The change in surface temperature is noted from 23.7°C to 31.7°C and corresponding change in water temperature is between 12.4°C and 16.1°C. One should note that maximum temperature increase is achieved while flow rate is 0.025 kg/s and the least is noted as the flow rate is 0.074 kg/s.

Figure 7(a)–(c) represents the performance characteristics of the ‘Novel’ roof under 600 W/m² and water flow rates of 0.025, 0.049 and 0.074 kg/s, respectively. The surface temperatures increase from 23.4°C to 42.4°C, and 49.2°C to 40.7°C, 48.2°C and 50.6°C and water temperatures are from 17.8°C, 22°C and 23.9°C to 21.4°C, 23.4°C and 25.9°C while corresponding flow rates are 0.074, 0.049 and 0.025 kg/s, respectively. Again, the maximum temperature rise both in surface and water temperature is attained at the lowest flow rate of 0.025 kg/s and the minimum is at the maximum flow rate of 0.074 kg/s on average.

Figure 8(a)–(c) presents the associated performance findings of the proposed solar thermal roof under 800 W/m² solar insolation and 0.074, 0.049 and 0.025 kg/s water flow rates, respectively. In Figure 8(a), the performance of the ‘Sandwich’ thermal collector under 800 W/m² and 0.074 kg/s water flow rate is shown. The surface temperature ranges between 55.6°C and 59.9°C with a flowing water temperature in the range of 28°C and 30.2°C and a maximum temperature increase of around 2°C. In Figure 8(b), the performance of the ‘Sandwich’ thermal collector under 800 W/m² and 0.049 kg/s water flow rate is shown. The surface temperature ranges between 62.5°C and 65.8°C with
a flowing water temperature in the range of 30.7°C and 34.3°C and a maximum temperature increase is more than 4°C. Again, the maximum temperature rise both in surface and water temperature is attained at the lowest flow rate of 0.025 kg/s and the minimum is at the maximum flow rate of 0.074 kg/s on average.

Figure 9 presents the efficiency change of the ‘Novel’ solar thermal collector under various solar radiation and water flow rates. It can be seen from the graph that although higher flow rate has the greater efficiency ratios for low solar insolation settings while slower flow rate such as 0.025 kg/s has the superior efficiency ratios for the higher solar radiation levels. As incoming solar radiation to the roof system increases, the efficiency level also increases with the slower flow rates.

Figure 10 presents the collector efficiency, from the outdoor test, as a function of the flowing fluid, air temperature and solar radiation on the surface. The steady state had been reached before data were taken. The thermal efficiency is inversely proportional to the increasing \((T_w - T_a)/I\) ratio. Any momentarily change in climatic conditions could have various impacts over the thermal efficiency of the roof unit as follows; when the air temperature is relatively low, this may lead to more heat dissipation from hot surface to the ambient, thus causing to lower thermal efficiencies. Higher solar radiation contributes to enhanced heat transfer from black aluminium surface towards the polyethylene heat exchanger layer and this will eventually conduce to greater thermal efficiencies of the Novel roof. Also, high temperature of flowing water through the heat exchanger results in low efficiencies due to reduced heat transfer across the poly heat exchanger and confined structure of the ‘sandwich’ roof prevents high heat dissipation towards environment.
4.1. Outdoor testing

4.1.1. Solar roof
Figure 11 presents the environmental conditions on 23rd of June that outdoor test has been performed, outer surface and water temperature variations depending on the external conditions. The wind speed was measured over the course of the external test to find out the heat flow rate through natural convection between the blackened aluminium surface and the surroundings. Average solar radiation ($I$), ambient air temperature ($T_a$) and wind speed were measured as 701.3 W/m², 27.3°C and 1.09 m/s, respectively. It is clearly shown in Figure 11 that the surface temperature was expectedly higher than the circulating water temperature. Around 9°C increase in water temperature was noted while the surface temperature was in the range between 22.6°C and 44.6°C. One may conclude that conceiving the changes in temperature tendencies and increase in both surface and water temperature, there seems to be a fair agreement with the results obtained.

The design parameters in Tables 1–2 and climatic data have been utilised to assess the water and surface temperatures in Figures 11–13. The root mean square per cent deviation and correlation coefficient obtained by Equations (16) and (17) have also been evaluated and presented in the same figures. Figure 12 presents the comparison between experimental and theoretical values of the surface temperature. The results indicate that predicted and experimental results are in a good agreement as the correlation coefficient and root mean square values are attained as 0.88% and 4.55%, respectively. Figure 13 also presents the comparison between the theoretical and experimental results regarding the water temperature increase as a result of circulating within the heat exchanger being heat extraction component of the novel roof. Data gathered from the analysis show that the results obtained from the numerical investigation are in good compliance with the results of the experimental study. The correlation coefficient and root mean square values for water temperature comparison are attained as 0.92% and 3.03%. So, the estimated and experimental results for surface and water temperatures were found to be closely correlated.

4.1.2. Heat pump
Figure 14 presents the experimental data plotted regarding the COP, power consumption of the heat pump and heat gain in condenser changing with respect to time. The water temperature at the condenser was set constant at 35°C for space heating purposes. The energy gain at the condenser was found to be in the range of 1.9 kW to 3.43 kW while the COP is shown to fluctuate between 2.4 and 3.1. Meanwhile, the average power used
to operate the heat pump and to pump the water through the solar collector during the testing period was 0.7 kW.

Figure 15 presents the temperature variation in the water storage tank vs. time of the day. The temperature difference observed between the top and bottom of the storage tank is up to 4°C. However, the temperature deviation tends to decrease gradually.

Figure 16 presents the COP of the heat pump vs. COP of the overall system in the test day. The COP of the heat pumps is found to be higher than that of the overall system COP. The heat pump COP ranges between 2.4 and 3.1 while system COP is from 1.7 to 2.5, respectively. Daily average values of the measurements taken from 8 am to 5 pm are provided in Table 3.

4.2. Economic studies

Economic parameters for the proposed system are provided in Table 4. In addition, Figure 17 illustrates the variation of payback period as a function of collector area for various inflation rates.

Figure 17. Variation of payback period as a function of collector area for various inflation rates.

Table 4. Economic parameters.

| Parameter                  | Value  |
|----------------------------|--------|
| Fuel escalation rate       | 6%     |
| Discount rate              | 8.75%  |
| General inflation rate     | 2.8%   |
| Interest rate              | 8.2%   |
| Fuel cost                  | $0.012/MJ |
| Life cycle                 | 20 years |
| Down payment               | 10%    |

The increase in collector area due to a rise in collector-area-dependent cost. In the economic analysis, the least payback period at around three years was achieved corresponding to a collector area of about 5 m².

5. Conclusion

A Novel solar thermal roof collector coupled with a heat pump unit was investigated by simulation and experimental methods. The performance test results and energy evaluation analysis were presented. The developed numerical models were validated by the experimental findings collected on the testing
period. Also, the impacts of environmental conditions and operating parameters were examined by setting the system for space heating mode. The concluding remarks are outlined as follows.

The selection of the heat pump unit was mainly done because of its good compliance with the operating conditions, economic viability, its ability to handle varying evaporating temperatures adequately and environmental impacts, etc. Although initial costs of the heat pump systems are usually high, their low operating, maintenance and LCCs and long life cycle expectancy attract considerable interest as a potential alternative to the most conventional systems.

The validation of the developed thermal model was carried out by comparing the theoretical results with the actual findings during experiments. As the comparison indicates, thermal model developed for this prototype shows a good agreement with the experimental results obtained.

Overall results prove that the water flowing through tubes and risers of the heat exchanger layer can successfully convey the solar thermal energy and have a cooling effect on the roof surface. The initial tests conducted at the laboratory environment show that the solar roof collector yields the maximum of 31.7°C surface temperature and 16.1°C water outlet temperature at 400 W/m² solar radiation, 50.6°C surface temperature and 25.9°C water outlet temperature at 600 W/m² solar radiation and lastly 65.9°C surface temperature and 34.6°C water outlet temperature at 800 W/m² solar radiation, respectively. Although the flow rate of 0.025 kg/s seems nominal at first instance, capturing heat slowly from the surface leads to overheating of the surface. This, however, may result in exceeding the maximum temperature threshold for the heat exchanger and may be harmful to the risers and pipes. To minimise this conflicting trend, the tests were initiated with a water flow rate of 0.074 kg/s for cooler surface, and then flow rates were gradually reduced until 0.025 kg/s throughout the test season. Hence, the optimum flow rate was assumed as 0.049 kg/s for outdoor test of the system. As for the outdoor performance test conducted, the temperature of water could increase up to 30°C. Thus, temperature increase of the water circulating through the innovative roof module could reach up to 10°C when the liquid within the system is at around 20°C. The efficiency of the solar thermal collector varies between 0.24 and 0.11, respectively. Experimentally obtained results present that the proposed solar thermal collector can act as an efficient heat extraction component for solar source heat pump systems. One can conclude that the possible reasons why it took so long before reaching a steady state may be the effect of turbulent flow inside the tubes, relatively poor heat transfer rate of the materials used and insulation.

The COPs of the heat pump and overall system are calculated in terms of experimental data obtained from the outdoor test. Overall COP values of the system increase with the ambient air temperature and solar radiation and reach a maximum at 3.2 at an average of 2.98 over the course of testing for space heating mode. The maximum heat transfer rate experimentally attained in the condenser is 3.43 kW. This indirect SAHP system can assure long-term operation under various climatic conditions and relatively low operation costs during wintertime. From the economic analysis of the system, a minimum payback period of around three years was achieved. However, its advantages need to be further proved by more experimental and theoretical studies in the future.

Although a number of studies have been conducted on the combination of solar collectors with heat pumps, integration of such a novel roof construction, which was not known before in literature, with heat pump has not been considered yet. The performance investigation of the system reveals that substantial amount of energy can be saved by utilising solar thermal energy when implementing such a novel system. This new system may also provide a simple yet effective alternative for regions suffering from energy scarcity in the residential and commercial areas. SAHP also provide enormous environmental benefits as they do not exhaust natural resources, and mostly do not allow any kind of air emissions or waste products. Therefore, these systems may also contribute to mitigation of the environmental pollution and provide a remedial action especially in the areas where air emissions arise to dangerous levels (e.g. China) (Loose et al. 2011).

Disclosure statement

No potential conflict of interest was reported by the authors.

Funding

The authors gratefully acknowledge the financial support by the Institute of Sustainable Energy Technology, University of Nottingham.

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