Title: A comprehensive investigation of using mutual air and water heating in multi-functional *DX-SAMHP* for moderate cold climate.

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Abstract

Solar energy assisted heat pump systems (*SAHP*) have been used in this application. *SAHP* system with solar collectors and the heat pump are combined into one unit so as to convey the solar energy to the refrigerant. The solar collector is used as the evaporator, where the refrigerant is directly vaporized by solar energy input. Due to the complicated technical issues associated with a combined system that provides air for space heating and domestic hot water, most of the previous studies have concentrated on water heater heat pump mechanism. The current work is aimed at examining the use of a new multi-functional heat pump (*DX-SAMHP)*, air for space heating mutually with solar for domestic hot water without employing an auxiliary heater. Comprehensive experimental and analytical studies in the first of its kind have been performed on the new system. The novel system with ternary panels and the thermal performance of the collector has been examined in this study. Results indicate that the *DX-SAMHP* using solar inner and outer panels for space and water heating is a promising substitute for the existing *DX-SAHP* water heater. Compared to the conventional solar-assisted *SAHP* heat pump systems, the coefficient performance of the new design doubles that of the conventional *DX-SAHP* systems.

Keywords

Heat pump in low temperature, Solar energy system, Refrigeration cycle**,** Heating system application.

28 Nomenclature

A_{coll}	area of the solar collector, (m^2)
A_{tank}	total heat transfer area of the wall of the water tank, (m)
$COP_{\text{h-sys (t)}}$	heat pump coefficient of performance for air (-)
$COP_{\text{h-sys, av}}$	coefficient of performance average of the system, air (-)
$COP_{\text{h,w-sys (t)}}$	heat pump coefficient of performance for air and water (-)
$COP_{\text{effective}}$	coefficient of performance effectiveness of the system
Cold-air-in	inlet air temperature to the system, (°C)
C_{p_air}	specific heat capacity of air ($kJ{\cdot}kg^{-1}°C^{-1}$)
C_{pw}	specific heat capacity of water ($kJ{\cdot}kg^{-1}°C^{-1}$)
D	refrigerant tube diameter, mm
D_i	inside refrigerant tube diameter, (m)
dt	variable time, per second, (t)
F'	collector efficiency factor
Hot-air-out	hot air gained by the condenser, (°C)
hf_i	heat transfer coefficient, (-)
h_{refout}	refrigerant/Collector outlet enthalpy, ($kJ{\cdot}kg^{-1}$)
h_{refin}	refrigerant/ Collector inlet enthalpy, ($kJ{\cdot}kg^{-1}$)
h_w	wind heat transfer coefficient, ($W{\cdot}m^{-2}°C$)
$W_{\dot{m}}$	total uncertainty in the mass flow rate of the water, (%)
$\dot{V}$	volume flow rate, ($l{\cdot}s^{-1}$)
I_{coll}	the intensity of the solar radiation, ($W{\cdot}m^{-2}$)
$\text{Inlet}_{\text{air, av}}$	average of outdoor air temperature,(°C)
M_W	water mass in the water tank
N_u	Nusselt number (-)
P_r	Prandtl number (-)
Plate-inlet	fluid enters the collector, (°C)
Plate-out	fluid exits the collector, (°C)
$Q_{\text{condenser air}}$	condenser heat gained experimentally, (kW)
Q_W	heat gained at the *DHWT*, (kW)
$Q_h\ (t)$	heat exchange rate in the plate condenser, (kW)
$Q_{\text{evaporator}}$	evaporator heat gained experimentally, (kW)
Q_{solar}	collector's heat gained theoretically, (kW)
$Q_w\ (t)$	condenser gained heat delivered by a heat pump (water), (kW)
R_e	Reynolds number (-)
T_a	ambient temperature, (°C)
T_2	outlet collector's temperature, (°C)
T_3	discharge compressor temperature, (°C)
T_{room}	room temperature, (°C)

U_L	Collector's heat losses, ($W{\cdot}m^{-2}$°C)
$U_{L,t}$	water condenser heat losses, ($W{\cdot}m^{-2}$ °C)
V	compressor's displacement, ($m^3{\cdot}h^{-1}$)
W_{comp}	compressor's work, (W)
$W(t)$	system input power, (W)
Abbreviations	
DHWT	water temperature at domestic hot water tank cylinder
HR	room heating
LPM	litre per-minute
L	litre
SPF	Seasonal performance factors
s	time, per second
Greek symbols	
$\dot{m}$	mass flow rate,
$\dot{\eta}_r$	refrigerant mass flow rate, ($kg \cdot s^{-1}$)
$\dot{\eta}_{coll\_ref}$	refrigerant mass flow rate, ($kg \cdot s^{-1}$)
η_{coll_ref}	collector efficiency, (%)
η_{comp}	compressor's efficiency, (%)
$\check{T}$	mean refrigerant temperature in the collector/evaporator, (°C)
ρ	fluid density (kg m^{-3})
f_{comp}	compressor's frequency (Hz)
α	collector's adsorption rate (-)
τ	operating period of duration, per-second (s)
Subscripts	
after-comp	fluid temperature after being compressed
after-cond	fluid temperature after being condensed
hot-rif	hot refrigerant at heat exchanger plate
hot-water	hot water at heat exchanger plate
h-sys	heating space system
h-sys,av	heating space average
refout	refrigerant collector outlet
refin	refrigerant collector inlet
coll	collector's
air-ave	an average of out-door air temperatures
solar	enthalpy calculation via modelling
ref	refrigerant
comp	compressor
tank	water tank
room	heated room area
av	average
a	ambient temperature
w	water
r	refrigerant/fluid
(t)	time, per second

1 Introduction

Solar energy systems and heat pumps are one of the promising means of decreasing the consumption of fossil fuel [1]. A heat pump is also a promising means of reducing the consumption of energy resources [2]. The idea of improving the conventional heat pump was suggested to enhance the system performance. A number of research groups have investigated powering heat pumps with solar energy and the idea of a combination of heat pump and solar energy has been proposed and developed around the world, which is called, the solar-assisted heat pump (SAHP) system [3-5]. The SAHP can be divided further into two types: the direct-expansion type (DX-SAHP) and indirect-expansion type (IX-SAHP).. Thereafter, a great deal of research concentrating on numerical and experimental studies of both SAHP systems was implemented as early as in the 1970s [6]. Furthermore, theoretical and experimental SAHP studies were also performed later in the 1990s [7]. Comakli et al. [8] have designed a solar heat pump for residential heating using energy storage system. Another study with IX-SAHP type was found suitable for regions abundant in solar insolation, while DX-SAHP coefficient of performance (COP) of 4.0 was found more economic impact [9]. The DX-SAHP system directly integrates Reverse-Rankine refrigeration device with the solar collector was firstly considered by Sporn and Ambrose [10]. Huang and Chyng first proposed the design of an integral-type DX-SAHP that integrates the heat pump, solar collector, to come up with a unitary system that is easy to install [22]. Kuang et al. [11] have performed experimental and analytical studies on DX-SAHP system as applied in Shanghai. In this study, the effects of various parameters under variable compressor speed were investigated.

A further developed DX-SAHP system which was able to supply multi-functional low heating costs to domestic buildings, including space cooling during the summer, space heating during the winter, and hot water supply is named direct-expansion solar-assisted heat pump water heater (DX-SAHPWH) for year-round was investigated in [12] and [13]. Since the solar collector serves as an evaporator while the refrigerant absorbs the solar incident energy (and/or ambient air energy), and the energy discarded by the condenser contributes to water heating. Provided that, the system of solar collectors can provide energy in steady state condition for overall performance analysis [14]. The overall COP s of the system is affected considerably by the load demands and changes in climatic conditions especially for low-temperature water heating applications [15]. It was also reported that a considerable amount of theoretical and experimental studies have been conducted on DX-SAHP system especially for the water heater [16]. Chaturvedi and Shen [17] investigated a bare collector/evaporator for water heating applications. Li et al. [18] introduced and applied experimentally a methodology for design optimization in two direct expansion solar-assisted water heater systems. Ito et al. [7] studied theoretically and experimentally the structure parameters on the DX-SAHP. In this study, the results showed that the system reached COP of 3.3 under various weather conditions for space heating. They optimized the collector/evaporator structure parameters such as plate materials, thickness and collector area. Mohamed [19] developed a new concept of ternary collectors used to enhance the direct-expansion solar-assisted multifunctional heat pump (DX-SAMHP) for air-space heating. The result showed that the system had a significant improvement compared to the conventional heat pump. A multi-fold-functional system in cold climate region is essential year-round and high utilization rate makes the option economically attractive [1, 20]. Since space heating and domestic hot water (DHW), in winter for a household is indispensable. In addition, the coefficient of performance of the DX-SAHP system would increase over that of the air-source heat pump system alone [21]. DX-SAHP can utilize heat from solar radiation and ambient air simultaneously [22] and can also operate using surroundings domestic/industrial exhausted heat. Even in the absence of solar insolation, this may utilize for space heating and water heating applications [23]. Recently the authors conducted a study for water and air space heating to evaluate the heating performance of multi-functional DX-SAMHP [24]. The experiment agreed with the view of the fact that the system can be a good alternative to existing DX-SAHP. The study also indicated that the formation of frost in an air source heat pump for the fan-driven condenser is inevitable. Thermal refractory performance investigation is performed on collectors using a thermal camera.

On the other hand, space heating and domestic hot water production are deemed to be key applications in this sector:53 percent and 16 percent respectively [25, 26]. However, the heat pump is widely known as heat recovery system capable of increasing the temperature of recovered heat. Hence, enable the wasted heat to be elevated to the levels that are more useful.

Although, and besides the aforementioned advantages the use of heat pump for combined space heating and water heating, particularly in the *DX-SAMHP* option is not popular. The studies, however, are scarcity in numbers and they all have their limitations with respect to cold regions [27]. Most of the previous studies focused on the developments under typical working conditions, in particular, water heater system, while other systems that under low temperature heating conditions was little until the last three consecutive years [28].

The incentive of this study is to develop a new design arose from utilizing a wasted surrounding heat energy as the main source. The objective of this new design is to decrease the temperature difference across the collector/evaporator and condenser using SAHP energy system, thereby enhancing the heating capacity at lower ambient temperatures. By providing air heating rather than heating the space through hot water, this technique would drastically reduce the water tank required area, and shrink the use of primary energy sources.

This study demonstrates theoretically and experimentally that the *DX-SAMHP* could work satisfactorily with a very high efficiency at a low temperature and in absence of solar irradiance.

2 Schematic design, fabrication and description of the proposed system

A multi-mode-functional *DX-SAMHP* system for domestic applications is shown in Fig. 1 and 2. It mainly consists of a ternary unique coated aluminium flat-plates, a variable speed hermetic compressor, centrifugal fan-coil units (air-cooled condenser), a water-to-refrigerant heat exchanger, a water circulating pump and piping, a domestic hot water tank (DHWT) with an immersed condensing coil loop (water-cooled condenser), thermostatic expansion valve, electronic control and electrical valves. Supplementary components are added to facilitate cycle running such as centrifugal fan, expansion vessel, water circulating pump, and solenoid valve.

Fig.1: Details of the schematic developed *DX-SAMHP* system

Fig.2: Operations principles of *DX-SAMHP* system

The developed design is comprised primarily of collectors. The fabricated unglazed solar absorber evaporator is formed by integrating bare ternary soft aluminium solar-collectors, which are connected in series form (Fig.2). This unglazed solar flat-plate collector is used as a heat source acting as an evaporator for the refrigerant R407C. The piping network design is inlaid between the three aluminium plates. The lumped collectors formed by over-pressurizing the network so that the fluid serpentine circuit constructs within the fin. The aluminium plates are then retained by bonding and rolling them together. The plates thicknesses are 2 mm, while the inner diameter of the tube is 13 mm with the total piping length of 25 m. The heat pump composed of a refrigerant cycle, air circulation, and water cycle (Fig.1 and 2). As above-mentioned, the collectors comprise of two flat plates, which are externally placed and integrated into the structure of a house roof, whereas another plate is internally mounted in the house loft space to absorb domestic wasted heat (Fig.2). The heat pump primarily of a rotary hermetic-type compressor is utilised in the system with power rated of 800 W at a frequency (50 Hz), with the compressor volumetric displacement of 7.8 L/s (Fig.3). The objective of this concept is to decrease the temperature difference across the collector/evaporator and condenser, in which the wasted energy stored in the surrounding air, solar radiation and ambient air could be used. Thereby enhancing the heating capacity of the system at lower ambient temperatures.

Fig.3: *DX-SAMHP* system components

The variation of the compressor's speed was obtained via variable frequency drive, to avert the compatibility between its variable loads and in order to reach the steady capacity of the compressor (Fig.3, 1). A refrigerant receiver (Fig.3, 2), and the accumulator is embedded in the system to facilitate in controlling the refrigerant distribution, while the external pressure equalizer with thermal expansion valve to control the degree of superheat at the compressor inlet by controlling the pressure. It also liquidises the refrigerant's flow to evaporators (Fig.3,3). The heat pump has two heat rejection modes, one of which is made of copper tube and aluminium fins (Fig.3, 4); whereby centrifugal fan can deplete energy to contribute to space heating load. The other is copper tube as coil closed loop immersed in a cubage of 200 L, a fully insulated domestic hot water tank (Figure.3, 7), and water-to-refrigerant plate heat exchanger (Fig.3, 5). The water-to-refrigerant plate heat exchanger is linked to the DHWT through circulating piping pump, expansion vessel (Fig.3, 6-8) and water flow rate controller (Fig.3, 9) in order to maintain the hot water demand. The energy rejected by the condenser (copper tube and aluminium fins plate) is used for the purpose of exchanging the heat between the refrigerant and air inlet source to contribute to space heating.

In general, the multi-mode-functional *DX-SAHP* system can offer three fundamental operating modes; Space heating-only-mode, water heating-only-mode and it can also produce DHW and space heating modes. In the present experiment, the switching between those modes is by means of valve position and on–off controls. There is a two-way solenoid valve (Fig.3, 13), the one-way non-reversing valve on the refrigerant pipes at the locations shown in Fig.3, 14. A controller box is employed to determine and govern the operations running and modes of the system, which is supplied with digital cabinet output to organise the compressor's frequency (Fig.3, 12).

3 Experimental setup and data acquisition

The experiments were carried out twice, earlier in winter 2016 and the latter one in winter 2017, emerged between laboratory and actual weather conditions based on British Standard Institute/International Organisation for Standardization *BSI/ISO 13612-1:2014* for testing and performance standard. The system is proposed operating at night time hours and early morning and remains to idle for the rest of the day. Temperatures were measured with constantan thermocouples (Table.1). All thermocouples were calibrated in a high accuracy thermostatic bath using standard platinum resistance thermometer. A solar Pyranometer was used to maintain consistent fallen solar radiation to cover the entire surface area of the plates equally (Table.1). Electronic power meters were utilised to evaluate the compressor's energy consumptions, and the power of the whole system including, a booster water pump, electrical valves and fan consecutively (Fig.3, 10), (Table.1, 2).
The indoor temperature, ambient temperature, the collector's (surface, inlet/outlet) temperatures, and evaporation temperature, alongside with relative humidity were measured in real weather conditions to an accuracy of ± 0.1°C and 3 percent respectively (Table.1, 2). A solar simulator was configured indoor to simulate incident solar insolation on the surface of the collectors. The evaporator/collector surface temperatures were measured by 'K' type thermocouples on indoor/outdoor collectors (Table.1). These surfaces temperatures were determined at points near of the six corners, in middle between the 7th and 9th tubes, and 15 cm from the upper and lower of the evaporator/collector plates. As the spectrum distribution fulfils the European standard, and the instability and heterogeneity of the solar simulator were under 4 percent, therefore the simulator efficiently can simulate the solar radiation. In addition, the flexible luminous area is 2.22 m^2 were situated in parallel to collector's surface area. Pyranometer with a sensitivity of 60 to 100 $\mu V \cdot w^{-1} \cdot m^{-2}$ (*SP lite2 compares to ISO 9060*), was mounted on the collector to read the instantaneous range of adjustable solar radiation on the collector between 0 to 200 $W \cdot m^{-2}$ (Table.1).

Pressures were gauged using two pressure sensors located at different points (outlet-collector, outlet-compressor, and outlet condenser) to measure the suction and discharge lines the influence of the operating parameters of the system performance with accuracy ± 1 percent (Fig.3,), (Table.1). The refrigerant mass flow rate was measured in order to evaluate heat pump using R407C in charge and discharge sides by a clamp on pipe ultrasonic flowmeter with an accuracy of ± 0.5 percent (Table.1). The switching between modes is by means of an on-off control valve. Three rates of water were also fixed to determine the water cycle mass flow rates with the uncertainty of ± 5 percent as shown in Fig.3, 9. In addition, the power input was measured precisely using digital power metre with accuracy ± 1 percent in order to calculate the consumption of the electrical and electronic devices (Centrifugal fan, motor pump and controller) (Fig.3, 10,11,12 and 13), (Table.1). During the experiment, major operating parameters were measured or calculated numerically and experimentally in actual time. The experimental results were later compared with the analytical model. Tests were performed fairly between ± 1°C to ± 8°C of the outdoor temperatures. Several solar irradiances varied from 0 to 200 W·m^{-2} were applied in order to examine the normal operation of the system. and compare it to the heating performance results of other conditions. Each experiment was clearly measured under quasi-static condition.

Table.1: The specifications of instruments measuring tools and accuracy

Table.2: System parameters for thermal analysis

4 Thermodynamics analysis, and numerical modelling

When the solar irradiance falls on the solar collector, the main part of the radiation(I), strikes to the aluminium absorber plate electroplated with black paint selective coating. A part of the solar radiation is absorbed by the working fluid ($Q_{evaporate}$) and the remained part is dissipated through top and bottom of the absorber plate to the surrounding. In other words, to compute the outlet temperature and efficiency of the solar collector, first the heat losses to surrounding should be calculated. The rate of useful energy extracted by the collector experimentally ($Q_{evaporate}$) expressed as the rate of extraction under steady-state condition is proportional to the rate of useful energy absorbed by the collector. The relation between the absorbed heat and heat losses are as follows:

$$Q_{evaporate} = A_{coll} * \left[\tau \propto I - U_L (T_p - T_a)\right] \qquad (1)$$

$$= F' A_{coll} \left[\tau \propto I - U_L (\check{T} - T_a)\right] \qquad (2)$$

Where:

Ac: solar collector's surface area, (m²)

∝: the collector's absorption rate

I: the intensity of the solar radiation in (W·m^{-2})

U_L: the collector's overall loss coefficients, (W·m^{-2})

T_p: the collector/evaporator temperature, (°C)

T_a: ambient temperature, (°C)

F': solar collector’s efficiency factor that is dependent on tube-and-sheet

$\check{T}$: mean refrigerant temperature in the collector/evaporator, (°C)

T_{evp}: the refrigerant temperature at the collector/evaporator

$$\mathrm{U_L} = \mathrm{U_{top}} + \mathrm{U_b} + \mathrm{U_e} = h_c + h_r \qquad (3)$$

Where $\mathrm{U_{top}}$ the heat loss coefficient from the top/plate, $\mathrm{U_b}$ is the heat loss coefficient from the bottom/plate and $\mathrm{U_e}$ is the heat loss coefficient from the edges of the collector. To calculate $\mathrm{U_L}$, the following equations can be used [29, 30]:

$$h_c = 2.8 + 3.0v \qquad (4)$$

$$h_r = \varepsilon\sigma\left(T_p^2 + T_a^2\right) * (T_p + T_a) \qquad (5)$$

Where h_c , the convection coefficient due to the wind and h_r, the heat transfer coefficient due to radiation, and v the velocity of air. Whilst, ε is the emissivity of the collector (0.9), and σ is the Stefan-Boltzmann constant (5.7x10^{-8} W·m^{-2}·C^{-4}). The heat losses from the edges are out of scope in this paper.

An initial value of $\mathrm{T_p}$ is measured experimentally, and the quantities of $\mathrm{U_L}$ and $\mathrm{Q_{Solar}}$ are calculated. $\mathrm{Q_{Solar}}$ can be computed through modelling via refrigerant enthalpy chart:

Where; $\mathrm{Q_{evaporator}}$ via laboratory experiment $\simeq$ $\mathrm{Q_{Solar}}$ via modelling

For experiment $$Q_{evaporator} = \dot{m}_r * (h_{refout} - h_{refin}) \qquad (6)$$

For modelling $$Q_{Solar,ref} = \dot{m}_r * (h_2 - h_1) \qquad (7)$$

Where; $\mathrm{h_{refout}}$ is Refrigerant/Collector outlet enthalpy $\mathrm{h_{refin}}$ is Refrigerant/ Collector inlet enthalpy, and $\dot{m}_r$ Collector mass flow rate ($kg.\mathrm{s}^{-1}$) calculated from;

$$\dot{m}_r = \mathrm{Q} * \rho \qquad (8)$$

Where the volumetric flow rate in ($\mathrm{m^3 s^{-1}}$) is computed from;

$$\mathrm{Q} = \frac{\pi}{4} D_i^2 * \mathrm{V} \qquad (9)$$

Where;

D_i: inner tube diameter of risers (mm)

ρ: Fluid density ($kg.\mathrm{m}^{-3}$)

V: The mean velocity of the fluid (m s^{-1})

For this work, it is not necessary to develop a completely new analysis of the tube-sheet relation situation, Hottel-Whillier-Blis have developed the collector efficiency factor F' [31] [29], for the tube-sheet relation in the formula as follows:

$$F' = \frac{1}{W\left[\frac{1}{[D+(W-D).F]}+\frac{W.U_L}{\pi D_i h f_i}\right]} \quad (10)$$

Where:

W: Pitch between the serpentine tubes of the collector

D: Outer tube diameter of risers

D_i: Inner tube diameter of risers

F: Fin efficiency factor of the collector plate calculated by;

$$F = \frac{\tanh[m(W-D)/2]}{m(W-D)/2} \quad (11)$$

4.1 The evaporator/collector section

The size of (inner/outer) evaporator/collector area is carefully chosen, as cold regions require a large collector to match moderately the high heating load. The collector model is used to determine the value of the outlet collector's temperature T_2 for given values of ambient temperature T_a andI_{coll}, refrigerant properties h_1 and h_4 and the heat pump parameters (specific volume v_1 and the displacement volume rate (VD) are given in Table.2. The net energy absorbed by the fluid circulating via the collector/evaporator equals to the incident solar radiation minus the heat loss from the collector/evaporator, and accordingly, the outlet fluid temperature T_2 can be calculated through the steadystate energy balance on the collector/evaporator in modelling part, expressed by:

$$\frac{VD}{\nu_1}(h_2 - h_4) = F'A_{coll} * [I_{coll}(\tau \propto) - U_L(T_2 - T_a)] \quad (12)$$

From the above equation, one can solve for T_2 as:

For modelling part $$T_2 = T_a + \frac{I_{coll.(\tau\alpha)}}{U_L} - \left(\frac{VD}{\nu_1}\right) * \frac{(h_2-h_1)}{U_L.F'.A_c} \quad (13)$$

For a given location (Nottingham, UK), the values of the collector/evaporator temperature at given time of the day is measured experimentally and compared to those assumed values of enthalpies at state points 2 and 1 which are modelled and computed from the polynomial fit for the refrigerant properties. New results of T_2 according to circulated processes are compared with the previously measured experimental data and modelled values of T_2. The findings of these parameters are incorporated over a given month. The thermal energy produced by the heat pump Q_H (W) is calculated from the following equation:

$$Q_{H_{ref}} = \dot{m}_{coll\_ref}(h_3 - h_4) \qquad (14)$$

$$Q_{H\_air} = \dot{m}_{air} * C_{p\_air}(T_{out} - T_{in}) \qquad (15)$$

Where $\dot{m}_{coll\_ref}$ and $\dot{m}_{air}$ are the collector's refrigerant and condenser's rejected air mass flow rate respectively, whereas C_{p_air} is the air specific heat capacity. Consequently, the efficiency of the solar-collector η_{coll_ref} can be defined both experimentally and in model part, as expressed below:

$$\dot{m}_{coll\_ref} = \frac{V_D}{v_1} \quad (kg\,.\,s^{-1}) \qquad (16)$$

$$\dot{m}_{air} = \rho * V * A \quad (kg\,.\,s^{-1}) \qquad (17)$$

Where ρ is the density of flowing air ($kg.m^{-3}$), V is the airflow velocity ($m\,s^{-1}$) and A the flow area (m^2).

The instantaneous efficiency of the evaporator/collector is expressed as follows:

For experiment part
$$\eta_{coll\_ref} = \frac{Q_{evaporator}}{I_{coll} * A_{coll}} \qquad (18)$$

4.2 The compressor section

As mentioned earlier, the compression of the refrigerant vapour is assumed to be a polytropic process, the compressor work W_{comp} or compressor power $W_{power} = \dot{m}_{refrigerant} * W_{comp}$ for a given pressure ratio P_2/P_1 is determined for the modelling part from expression [32]:

For modelling part
$$W_{comp} = \frac{P_1 v_1}{\eta_{comp}}\left(\frac{k}{k-1}\right)\left[\left[\frac{P_2}{P_1}\right]^{\frac{k-1}{k}} - 1\right] = \dot{m}_r(h_3 - h_2) \qquad (19)$$

Where [k] the ratio of specific heat and for R407C.It is has a value of 1.14. The assumption of ideal gas behaviour during the compression process appeared to be reasonably acceptable since the compressor's work computed under such assumption is slightly overestimated in comparison with the compressor's work calculated instantly through the experiment work

4.3 The condenser section

In this study, two condensers are employed: air-cooled condenser with an area of 14.17 m² and water-cooled condenser with an area about 3.22 m². The air-cooled is representative of the heating finned coil for the hot-air heating system. Whereas, the water-cooled is to heat the DHWT when the output of the system satisfies the space heating requirements. The air velocities were measured so as to further obtain the air flow rate at the outlet of condensation fan and the heating capacity. The operating performance of a heat pump is related to the temperature that is the difference between the temperature of the heat source and the output temperature of the heat pump. Therefore, the $COP_{\mathrm{heat\ pump}}$ determined solely by the condensation temperature and the temperature lift (condensation-evaporation temperature). The actual heating capacity is obtained from equation (20) by measuring the air temperature difference between inlet and outlet of and the air mass flow rate at outlet orifice.

$$Q_{condenser_{air}} = \dot{m}_{r_{air}} * c_{p_{air}}(T_{out.condeser} - T_{in.condenser}) \quad (20)$$

4.4 Thermostatic expansion valve section

An orifice thermodynamic expansion valve is modelled, through which the refrigerant is extended from condensing to evaporating pressures. Liquid mass flow rate passing through it can be formulated as follow:

$$\dot{m}_r = \mathrm{C_v A_0} \sqrt{2_{\rho\mathrm{i,I}} \Delta \mathrm{P}} \quad (21)$$

Where $\mathrm{C_v}$ is liquid flow coefficient, which depends on the valve opening degree, and when the valve is fully open, it can reach its maximum value; $\mathrm{C_v}$ is evaluated by [33][32] as follow;

$$\mathrm{C}_v = 0.02005\sqrt{\rho\mathrm{i,I}} + 0.634\mathrm{v_0} \quad (22)$$

While $\mathrm{A_0}$ is the minimum flow area across the orifice. Whereas, $\mathrm{v_0}$ is the refrigerant specific volume at the outlet valve. $\rho\mathrm{i,I}$ is the density of the liquid that can be calculated at the inlet valve. $\Delta\mathrm{P}$ is the variation of the pressure across the orifice valve.

For the process of an isenthalpic in the expansion device, the following equation is achieved:

$$\mathrm{h_4 = h_1} \quad (23)$$

Where; $\mathrm{h_4}$, and $\mathrm{h_1}$ are the refrigerant’s specific enthalpies at valve inlet and outlet.

Condenser/Domestic hot water storage tank.

The serpentine copper tube is used for hot water tank condenser, which is immersed in the DHWT (submerged heat transfer coil). Similarly, to collector/evaporator, the condenser copper tube can also be divided into sections equal to enthalpy difference. Supposing that the DHWT is non-stratified, the energy balance can be obtained with the immersed condenser as follow:

$$Q_w = M_w C_{pw} \frac{dt_w}{d_\tau} = \dot{m}_r(h_3 - h_4) - U_{L,t} A_t (T_{w,out} - T_{w,in}) \quad (24)$$

Where Q_w is the condenser's gained heat, which is released and transferred heat rate into DHWT by the condenser. M_w is the water mass in the water tank, C_{pw} is defined as the water specific heat, while $T_{w,in}$ and $T_{w,out}$ are the water temperature inlet and outlet respectively, τ is the time, and h_3 , h_4 are the refrigerant's specific enthalpies at the condenser's inlet and outlet respectively. The total heat loss coefficient of the tank is $U_{L,t}$ and A_t is the total heat transfer area of the wall of the water tank.

4.5 Refrigeration compression cycle and system design

The optimal size selection is important to design HPs because oversized systems operate with lower efficiencies, would lead to excessive running cost. For system planning, components must be designed to interact optimally in order to ensure reliable operation and high performance level. Therefore, to achieve constant quasi-steady heat pump condition by means of constant temperature and condensation, the work is sufficient enough to meet the minimum requirement for the heat pump, hence allowing it to operate reversibly and to determine the COP of the heat pump. In the current study analytical investigation were considered in order to examine the thermodynamic performance of the cycle along with experimental running tests to validate the mathematical model of the developed system.

As mentioned in schematic design, the system consists of ternary flat panel's acts as an evaporator, two condensers, one heat exchanger and domestic hot water tank with an immersed condensing coil loop. The working fluid evaporates at a temperature T_{EV} by extracting heat from surrounding and available solar irradiation. It is then compressed and gives up its latent heat for two directions as it condenses at higher temperature T_{CO} at the first condenser for space heating and another is immersed inside a domestic water tank for heating water purpose. Whilst an external heat exchanger plate's is used to extract heat from the fluid to water. The condensed liquid is then expanded adiabatically and irreversibly throatily through an expansion valve and is throttled into evaporator/plates to complete the cycle.

The heat pump cycle is the reverse of the power cycle and can be illustrated with reference to R407C pressure enthalpy diagram shown in Fig.4. This heat pump system is designed to operate at discharge line pressures (P_H) between 17 to 19 bar, and suction line (P_L) is 3-4 bar (Fig.4). The compression ratio (CR) of the corresponding pressures in the condensers and evaporator (P_{CO}/P_{EV}) is about 5.1 bar. The highest obtainable condensing temperatures are relatively modest up to 40°C in ideal conditions as seen in Fig.4.

Fig.4: *DX-SAMHP* refrigeration cycle [Enthalpy diagram (kJ/kg)]

The cycle in Fig. 4 shows that the working fluid at S2 is in the form of saturated vapour at an evaporation temperature (T_{EV}) of -3.98°C. It is isentropically compressed to point D1 in the superheated vapour region. The superheat (HD1-HD2) is then removed and it is isothermally condensed from saturated vapour at D2 to saturated liquid at point D3 at a condensing temperature (T_{CO}) between 36-40°C. From D3 it is isenthalpically expanded to a mixture of liquid and vapour at point S1= -3°C from which it is isothermally evaporated at a temperature of -3.98°C to point S2. Theoretically, the coefficient of performance of a heat pump can be determined as (COP) = (HD1- HD3)/ (HD1-HS2) ≈ 3.83 where H is the enthalpy per unit mass.

5 Operation modes and procedures

5.1 Space-heating-only mode

This mode can be used to provide the room with space heating during the cold season when heating of the room air is essential. At this mode, the refrigerant-filled solar collector array on the expected roof acts as an evaporator, while the finned coil tube heat exchanger works as a condenser. The refrigerant vapour from the compressor enters the heat exchanger directly. The centrifugal fan was used to dissipate the heat to the room, and meanwhile, the water-to-refrigerant heat exchanger is switched off. As a part of the control strategy, the heat pump operates only from 20:00 pm to 06:00 am, because of targeting lower outdoor temperatures to reach stabilise planned experimental conditions; ascertain that the system would relatively be in the same conditions for all tests (high-pressure side, low-pressure side, and ambient temperature), so that a comparison can be evaluated. On the other hand, the water circulation pump was stopped and the water loop closed.

Using the space-heating-only mode, the function of producing hot water is ineffective, and the system COP for space heating at any time (t) is defined as [12];

$$COP_{h-\mathrm{sys}}(t) = \frac{\mathrm{Q_h}(t)}{\mathrm{W}\ (t)} \qquad (25)$$

Where: $\mathrm{Q_h}$(t)is the heat exchange rate in the plate condenser, and W (t) is the system power input. If W (t) is stated as the power input for the compressor, the rate from equation (25) is so-called the heat pump COP_h which mainly reliant on the difference in temperature between evaporating and condensing processes. τ is the duration of the operating period which much dependent on the compressor capacity. The $COP_{\mathrm{h-sys,av}}$ average is therefore, can be determined as:

$$COP_{h-sys,av} = \frac{\int_0^{\tau} Q_h\,(t)\,dt}{\int_0^{\tau} W\,(t)\,dt} \qquad (26)$$

5.2 Space and water-heating mode

This mode is used for both hot water and space heating production. At this mode, the two-way valve is positioned after the compressor serves as two fluid lines to feed both heat exchangers through bypassing the refrigerant flow. The water in the hot domestic tank (DHWT) is heated up to 60°C through water-to-refrigerant plate heat exchanger. The immersed condenser coil-tube dissipates heat to the water tank. Meanwhile, the booster water pump on the loop is powered-on. Whereas rejected energy by the finned-coil tube condenser contributes to space heating. In this operation, the solenoid valve shuts the water loop side down once the water temperature in the tank exceeds the load temperature (65°C). This makes the refrigeration cycle operate in a steady state condition except for the storage tank.

The COP for space and water-heating mode is defined as [12];

$$COP_{h,\mathrm{w-sys}}(t) = \frac{\mathrm{Q_h}(t) + \mathrm{Q_w}\ (t)}{W\ (t)} \tag{27}$$

And

$$COP_{\mathrm{h,w-sys,av}} = \frac{\int_0^{\tau} \mathrm{Q_h(t)\,dt}\ +\ \int_0^{\tau} \mathrm{Q_w}\ (t)\mathrm{dt}}{\int_0^{\tau} W\ (t)\mathrm{dt}} \tag{28}$$

Where; $\mathrm{Q_h}(t)$ is the heat exchange rate at the air heat exchanger, and $\mathrm{Q_h}(t)$ is the heat exchanger rate at the water storage tank (Condenser).

5.3 Water-heating-only mode

At this mode, hot water production is used only to heat up the water lingeringly at the DHWT. The condenser–air fluid line supplier is shut-off. In this case, the compressor serves as one fluid-line-way to feed refrigerant flow only to the water-to-refrigerant heat exchanger. The immersed coil dismisses heat to the domestic storage tank. The compressor is shut down once the water temperature in the tank exceeds the load temperature (65°C). The working temperature of the submerged condenser rises and the efficiency of the system declines accordingly. Meanwhile, the system COP is sensitively varying with the change in water tank temperature due to the delay of heat transfer processes. In order to investigate the *DX-SAHP* water heating capability, evidently, the temperatures of water storage are tested and recorded during the testing period. The COP for water-heating mode is defined as;

$$COP_{w-\mathrm{sys}}(t) = \frac{\mathrm{Q_w}(t)}{\mathrm{W}\ (t)} \tag{29}$$

$$COP_{\mathrm{w}-sys,av} = \frac{\int_0^{\tau} Q_w\ (t)\ dt}{\int_0^{\tau} W\ (t)\ dt} \tag{30}$$

6 Results and discussion

A series of consecutive five months during winter 2016 and winter 2017, repetitive experiments were conducted for three fundamental operating modes during the cold season under typical Nottingham indoor/outdoor weather conditions. The simulation model allowed the parametric study of important variables and their identification. For the purpose of validation, a comparison between simulated and experimental results was made. It is important to mention that, due to unstable climatic conditions and the numbers of tests, experiments were performed for one hour for each case. Solar insolation of 0, 57, 100 and 200 $W{\cdot}m^{-2}$ with 1, 2 and 3 LPM was applied. The wind speed factor is not considering in this study because it has no great influence on such system performance [21]. The effect of relative humidity as varies between 26 to 40 percent on the *DX-SAMHP* performance is not also significant, since the relative humidity values are reasonably low. Variant COPs with correlation with different solar irradiance and several flow water rates have been taken into account and their characteristics studied. Experimental data are then compared with the analytical results. The *DX-SAMHP* system operated for sixty minutes for each case. The impact of various system parameters on the response on the water temperature variation in the heat storage tank, indoor air temperature of the building, electrical power consumptions and heating capacities of the system were investigated.

6.1 Space-heating-only mode

In order to investigate the performance of air space-heating capability of the developed *DX-SAMHP* system, temperatures of outdoor air, (indoor, ambient) temperatures, and the system components has also been examined and recorded during the testing period without any heating apparatus. Experimental data listed in Table.3, which reveals that the averaged indoor temperatures ($T_{room,\ av}$) varied from 18.35 to 20.1°C. The outdoor ambient temperature was changing between -1 to 5°C, and values of the heat pump COP-sys, av were 2.8-3.86 and 3.87-3.93 in all cases respectively. It observes that in Fig.5 the refrigerant leaves the evaporator/collector in a low temperature and wet condition, these results in uniform temperature distribution and maximizes the evaporator/collector capacity. The surface temperatures of the evaporator/collector also indicate a unique refrigerant circulation between the collectors and on each plate. Tests result illustrates that the refrigerant temperature at negative region (0, -1 and -2) was cooling the collector were reduced to yields solar energy and indoor/outdoor ambient energy as seen in Fig.5.

Table.3: Performance of *DX-SAMHP* space-heating-only-mode

Fig.5: Performance of *DX-SAMHP* space-heating-only-mode

As it can be seen from Table.3, the heated air production in all cases was remained in steady state condition with a very little fluctuation due to cold air differences. Thereby the room and the system heated up with the increase in solar radiation as a low-grade energy source (Fig.6). It is clearly noticed that the effect of solar energy as radiation increased the system hot air reached its peak value 26.6°C within specified testing time. This behaviour improves the evaporator/collector efficiency, and the system performance was enhanced. However, the condenser reached maximum on 195 W·m^{-2} and followed by a decrease at 200 W·m^{2}. It is worthy to mention that in this case, the fluid was in the superheated vapour state at the exit of the collector which is considerably decreased the mass flow rates of the refrigerant, thereby less heat gained and less fluid condensation was taking out from the $Q_{condenser,\ air}$ (air-cooled condenser). It is spotted that in the presence of solar energy the compressor work was relatively reduced, the mass flow rate of refrigerant decreased and the COP-sys, av evidently increased (Fig.7). This because the increase in solar intensity, is leading to rapid increase in evaporating temperature causes a reduction in the heat pumping process. Consequently, the lower operational cost was achieved, and at inadequate solar insolation, the compressor electrical power consumption at a variable speed relatively fluctuated. The reason for this occurrence is that the compressor was overcome the mismatch between the collector and the compressor capacity to maintain the load. In such situation, the compressor relative stability performance enabled a higher COP-sys, av.

Fig.6: The influence of solar radiation on *DX-SAMHP* system

Fig.7: Average COP of the *DX-SAMHP* system at different solar intensities

It is interesting to note that, with or without solar energy, the system is capable to operate and provide sufficient space heating at satisfactory temperature level (Table.3). It would be operated either absorbing of the two or both (ambient/indoor) hot air simultaneously or heated by the heat pump instead. Ambient air energy was mainly transferred to the collector through natural convection and thus can still gain adequate heat although there was no solar energy. In this case, the total energy consumption was 0.91 kWh with an averaged COP-sys, av of 2.8. The averaged value of the solar irradiance input ratio was ± 0.0 suggesting that thermal energy from indoor and ambient air accounted for 100 percent of the total energy obtained. Hence, under this condition, the COP-sys, av was higher than the traditional *DX-SAHP* system.

Fig.8 presents the performance characteristic of *DX-SMHAP* space heating-only-mode. Showing the system temperatures versus the coefficient of performance for various solar radiation. The basis of this presentation is to illustrate the influences of parameters variation on the performance characteristic.

Fig.8: The influences of parameters on the DX-SAMHP performance characteristic

It is true that the COP-sys, av can be determined by temperature differences between evaporation and condensation processes. Thus, experimental data obtained had to condense and evaporating outlet temperatures of approximately 45°C, -1°C respectively. The thermal energy delivered to heat the space ($Qcond_{air, av}$) was raised steadily from 321.16, 3139.1, 3182.5 and 3191.1 W corresponding to 0, 57, 100 and 200 W/m² solar radiation in order to supply stable temperature at heating room averaged (HR, av) about 24°C in all cases (Table.3). Huang and Chyng [22] derived that in their study in solar source heat pump, and COP was ranged of 2.5-3.7. Huang and Wang [12] stated in multi-functional heat pump *DX-SAMHP* system study, that the COP was 2.1 in cloudy days for space-only mode and 3.5 in clear sunny days in winter. It can be concluded based on obtained values in the present study that the COP was fairly much better to those aforementioned by 12 %, 7.84 % and 11.43 % consecutively and in total by 33.3 %

6.2 Space-and-water heating mode

As solar radiation, ambient/indoor temperature, relative humidity and water flow rate are varying over a wide range of experimental operations. It would be useful to study their effects on the present developed *DX-SAMHP* system. As shown in Figs.9, the heat capacity increases by the evaporation temperature rise (T_{evp})with the growth of solar radiation (I_{coll}) and water flow rates. This principally because of two explanations; firstly, the increase of (I_{coll}) assists to obtain a higher evaporating temperature of the refrigerant, decreasing the evaporator/collector efficiency (η_{coll_ref}), and consequently resulting in a higher $COP_{h,\mathrm{w-sys}}$ (Fig.10). Secondly, for a given relative humidity and ambient air temperature (T_a), the higher I_{coll} allows the temperature of the collector/evaporator plate (T_p) to rise, which translates to change of temperature difference between T_a and T_p. When T_a is higher than T_p, collector/evaporator could attain useful energy gain from the surroundings (Fig.13, 1.a, b). This is proven that the η_{coll_ref} could exceed 100 % with zero or lower I_{coll} intensity. A higher T_p enables temperature difference between T_p and T_a to reduce, even though T_a is lower than T_p, and thus, η_{coll_ref} declines due to the change of heat transfer between the ambient air and collector/evaporator plate.

Fig. 9: *DX-SAMHP* system behaviour under different parameters

Fig. 10: The total $COP_{\mathrm{h,\,w\text{-}sys}}$ of the *DX-SAMHP* system

On the other hand, Fig.13.4 (b), 6(b) and 11(b) show that the performance of the system improves slightly with raising ambient air (T_a) or indoor temperature (T_{room}). It can be attributed to the fact that, the increasing of T_a drops the heat loss from the collector/evaporator and increases the refrigerant temperature in the collector/evaporator, which permits a higher

$COP_{h,\mathrm{w-sys}}$ as well as $\eta_{\mathrm{coll\_ref}}$. The evaporator more often works at a temperature lower than ambient temperature, with averaged values of -3, -2.6, 0.5 and 1.8°C respectively (Fig.13. (A's)). From Fig.13, it can be clearly seen that the effect of the Increment of solar irradiance on the energy consumption of the system. The reason is that when the solar irradiation increases the evaporator temperature increases causing evaporating pressure rises as well as the refrigeration mass flow rate. This causes the surge of energy consumption of the system. The values are 934 W, 1000 W, 1050 W, and 1086 W at the end of the experiments. It is obvious that the evaporator absorbs energy from both ambient and solar irradiations. This enhances the evaporating heat exchange rate of the system, as shown in Fig.13, (A's). Accordingly, this leads to the growth of condensing heat exchange rate. The specific values are 1474 W, 1560.9 W, 1630.3 W, and 2254.6 W corresponding to solar irradiances of 0, 57, 100, 200 $W{\cdot}m^{-2}$ and water flow rates 1, 2 and 3 LPM at the end of the experiments (Fig.9). Hence, the air temperature and water outlet in the DHWT versus time were graphically depicted in Fig.13.

Fig.13 (A's), also demonstrates the refrigerant temperatures of the collector/evaporator inlet, outlet and at the heat exchanger inlet.

Fig. 11.(1-12),(a, b): The influences of operating parameters on the system performance

Fig.14 shows the percentages of energy obtained via the two condensers given by compressor. The percentage values of both condensers were varying from 23 to 70 percent depending on solar intensity and water flow rates.

As shown in Fig.15, the coefficient performance of air and water ($COP_{h,\mathrm{w-sys}}$) elucidated separately in each test. The $COP_{h,\mathrm{w-sys}}$ increase owing to increasing of solar insolation. From a comparison point of view, Bi et al. [34] reported that in cold season, the heat pump solar energy source COP was found to be 2.73. Yumrutas and Kaska [35] demonstrated that for an experimental solar source heat pump space heating system with energy storage tank: the COP values is about 2.5 for a lower source temperature and is up to 3.5 for a higher supply temperature.

It should be noted that here, even though both the condensing hate exchange rate and energy consumption increases with solar irradiance, the boost of the condensing heat gain rate is larger. Subsequently, the coefficient performance average (COP-sys, av) of the system also rises with solar irradiance, with the specific values as 2.7, 2.9, 3.0, and 3.5 as shown in Fig.10 at the end of experiments. When solar irradiance improves from zero to 200 $W{\cdot}m^{-2}$, the heating capacity of the *DX-SAMHP* system is enhanced by 52.94 %, and the total $COP_{h,w-sys}$ increases by 41.3 %. In contrast, when water flow rate enhances from 1 to 3 LPM, the heating capacity also improves by 29.9 % and the $COP_{h,\mathrm{w-sys}}$ surges by 16.65 %. It should be mentioned that the heating air capacity gained is higher than the water heating capacity due to the priority of the design. Thus, when the heating air capacity percentage increases with the solar irradiance, the water heating capacity decreases. But it is also observed that with the increase of water flow rate the heating air capacity decreases (Fig.14,15). In return, the water heating capacity raises with the increase of solar irradiance, and also increases with the rise of water flow rate because of the rate of exchange heat in the water-to-refrigerant heat exchanger. The percentage differences values between heating water and heating air are plus or minus 30 percent to 70 per cent respectively (Fig.14). Therefore, the COP_{air} is greater than COP_{water} under the tested conditions (Fig.15).

Fig. 12: *The system heating capacity percentages for both air and water condensers* Fig. 13: Details of the coefficient performance of the developed *DX-SAMHP* system

6.3 Water heating only mode

The simulation model results in Figs.14.1, 2, 3 and 4 are taken at water-heating-only mode. In this case, the DHWT total volume is 200 (litres) with final temperature selected to be 65°C. It can be observed that the expected operation time τ is 1 hour on a typical night-time weather in winter. It is necessary to note that, here, *DX-SAMHP* operation time τ depends essentially on the compressor capacity. A larger compressor capacity permits a larger mass flow rate of the fluid, which leads to a higher compressor work and a better heating energy, consequently resulting in a less τ and lower $COPw\text{-}sys,av$. In these assessments, the system performance is governed generally by the change of the incident of solar insolation and indoor/outdoor air temperature. The hot water was heated in the DHWT by submerged water-cooled condenser with plates heat exchanger. Therefore, the system can operate in an appropriate efficiency for water heating and there is no need of water heater apparatus. The water heated from initial temperature 20°C to a final temperature about 65°C. Once the DHWT outer temperature reached 65°C, the refrigerant flows valve to water-side cycle was closed and the circulating water pump remained in the mix the stratified hot water in the *DHWT*. According to space and water heating experimental results, the equilibrium water temperature was 3°C less than actually measured one. The simulated values of the $COP_{w-sys,av}$ have the same behaviour with 4 percent average relative error. Thus, the lower the outdoor air temperature, solar intensity and water mass flow rate was, the longer the time was needed to reach final water temperature. This indicates that the water heating capacity of the system increases as the solar radiation and water flow rates surges.

In the first case, it took 60 min for heating 200 litres of water from 20°C to about 27, 33 and 40°C, under zero $W{\cdot}m^{-2}$ solar intensity with water flow rate at 1, 2 and 3 LPM respectively and fixed ambient temperature at 5°C (Fig.14.1). In the second case, the Fig.14.2 represents the values of 29, 36, and 45°C for water tank temperatures with 57 $W{\cdot}m^{-2}$ irradiance, and an increase of heating rate by 25 % is achieved. The third case reveals that with 100 $W{\cdot}m^{-2}$ solar insolation, and specific water heating values of 33, 43, and 55°C system performance was enhanced by percent 28 % as shown in Fig.14.3. the final case shows that under 200 $W{\cdot}m^{-2}$ irradiation the system delivers much better values with 35, 52, and 65°C, and an increase of heating rate by 25 % within 1 hour corresponding to specific water flow rates (Fig.14.4). The improvements between the three cases are 7.4, 13.8 and 6.1 % respectively. From a comparison point of view, Kuang, Y. and R. Wang **[12]** reported that the system COP in the multi-functional heat pump is between 2.1-3.5 for a water-heating-only mode in spring. **[22]**derived that in their study in solar source heat pump, and COP was ranged of 2.5-3.7. It is worthy to mention that here, in the present study the $COP_{w-sys,av}$ was always above 4.4 for all cases.

Fig. 14: Water-heating-only mode performance under (a) 57 W.m-² and (b) 200 W.m-² under different water flow rates

6.4 Validation of the mathematical model and comparison

To validate the model of the system, a *DX-SAMHP* unit was developed. The size and dimensions of evaporator/collector of the unified unit and main parameters are included in Table.2. The compressor of the unit was a frequency hermetic rotary compressor, type Mitsubishi RE165VA with frequency can be adjusted spontaneously by internal frequency converter from 15 to 110 Hz. The main parameters of the compressor are presented in Table.2 in detail. The condensers of the unit were fabricated as a finned coil (copper tube and aluminium fin) and immersed coil tube (Table.2). The expansion valve and added water-to-refrigerant heat exchanger of the unit was *SWEP*, and *Danfoss* (TZ2+1#) type respectively, whose specifications are shown in Table.2. To get a typical experimental environment for the unit, the unit was mounted in a similar attic house place as portrayed in Fig.2, which was used to attain a real-time temperature and relative humidity. The layout of measuring spots is exposed in Fig.1. All the data for every spot were recorded in real-time by a data taker DeLogger4V4R2 (Table.1).

The comparison between predicted values of the proposed design and experimental results is important to validate the performance of the new system. Therefore, numerical modelling was executed to predict the performance of the developed *DX-SHMAP* system. In this study, the system was assumed operating only at lowest temperature hours on the night and early morning, when the evaporating temperature was at least 5°C less than ambient temperature. The evaporator pressure was first presumed, and modulation looped through evaporator/collector, compressor, condensers and heat exchanger by iteration, until a convergence solution was reached. Enthalpy and entropy states of the leaving refrigerant were precisely assigned. In case of calculation differences were not within the tolerance limit, then leaving refrigerant was amended and the same process was repeated. The *DX-SAMHP* components were partitioned into loop sections as mentioned in section four, and then the performance of each loop elements was determined (Fig.17). The system modellation of the air and water heat gain percentages are illustrated as shown in Fig.18. Fig.19 demonstrates the values of calculated coefficient performance of the proposed developed system. Fig.20 compares the predicted values of *DX-SAMHP* system and heating capacity of experimental data. The results, elucidate that the computed results agree noticeably with the experimental results. That indicates excellent agreement between present data and numerical data. Figures

21-23 are plotted to show the variation of modelled air heating performance and experimental results. It also selected to display for reflecting the time-dependent ($\boldsymbol{\tau}$), and performance of the *DX-SAMHP* system. It is found that the modellation values are slightly higher than experimental results with average deviations in the range of ± 4 %. It is also noticed that in one case the experimental data is a little bit higher than modellation ones (Fig.21). This is because the experimental results of t_a are transient measured ones. The rest of air section comparisons and findings are seen to be quite aligned as shown in Fig.22 and 23. In contrast, the water in the DHWT was firstly heated from different start points in relation to water tank state temperature up to about 31, 24.5,26 and 39°C. This is corresponding to 0, 57, 100, and 200 W·m^{-2} respectively under 1LPM water flow rate (Fig.24). Followed by 26, 31, 33, and 43°C under 2LPM and 35, 36.7, 33.5 and 48°C under 3LPM consecutively as seen in Fig.25 and Fig.26. It is noted that at higher solar intensity and water flow rate the system is more likely affected by losing heat to surrounding environment due to a higher conduction heat transfer. As also observed that, the experimental water temperature deviates no more than 2°C from predicted values after one hour of operations. It is also elucidated that, in some cases, experimental results for the heating time ($\boldsymbol{\tau}$), the heat gain at the immersed condenser Q_w and $COP_{\mathrm{h,w-sys,av}}$ are a little bit less than the predicted values but they still have the same trend. In conclusion, that indicates a very good agreement between predicted values and experimental results.

Fig. 15: Heat capacity of the *DX-SAHP*

Fig. 16: Comparison between numerical and experimental air value of the system

Fig. 17: Comparison between numerical and experimental water heating values of the system.

7 Conclusions

A new concept of a multi-functional *DX-SAHP* system for space heating and DHW provision was fabricated at the laboratory as aforementioned. This is can be considered the first of its kind as it provides air space heating in contrast to previous studies that allocates the hot water for space heating. The use of solar and excess heat; is one of the developed system advantages without any auxiliary apparatus. The present study employed experimental and modelling examinations to simulate the thermal performance of *DX-SAHP*. It can offer multi-fold-functions for residential uses in order to provide two fundamental operating modes: domestic hot water and space heating in winter period. To study the performance of multi-functional *DX-SAHP* under different conditions, experiments were carried out with a varied solar intensity. The present study was tested under real time conditions of indoor/outdoor temperatures and experimental environment of -1 to 8°C in Nottingham city. Incident solar insolation simulator is used and solar irradiations are 0 W·m^{-2}, 57 W·m^{-2}, 100 W·m^{-2} and 200 W·m^{-2} with different water flow rates; 1 LPM, 2 LPM and 3 LPM during wintertime 2017. The tests and simulations of the newly developed system have been done under the lowest winter 2017 ambient temperature. Experimental results show that the system operating in space-heating-only mode can produce adequate space heating during winter. In this case, the COP-

sys, av system average is up to 2.8 indicating a good agreement with theoretical model findings with a maximum deviation less than 8 %.
On the other hand, according to space-and-water heating mode results, when the water temperatures in the condenser tank increase with time, the condensing temperature also increases. However, by modifying the system by adding water-to-refrigerant heat exchanger, the performance was enhanced by cooling the liquid before it passed through a thermostatic expansion valve, and corresponding $COP_{h,w-sys,av}$ and collector efficiency values remain in steady state condition. Average values of $COP_{h,w-sys,av}$ ranged from 2.8 up to 3.4 and solar collector efficiency varied between 40 % and 75 %. Water temperatures at the condenser tank were varying between 43°C and 55°C within two hours in correlation with available heat energy. For water-heating-only mode, the multi-functional *DX-SAHP* system could provide 200 litres of hot water, with final temperature up to 65°C. Results also indicate that the performance of the system was influenced significantly by collector area, load temperature, ambient temperature and solar irradiation. It is worth to mention that, variations of both ambient/indoor air temperatures and solar irradiation was leading to a large fluctuation in the thermal load imposed on the collector/evaporator of the multi-functional *DX-SAHP*. However, the system can reach adequate performance and mutually improve without the presences of solar energy or even auxiliary traditional sources. The energy consumption rises slightly with the increment of relative humidity. The reason is that when the relative humidity rises, the condensing latent heat increases the evaporating pressure, resulting in rising of energy consumption.

In addition, it can be seen that the inner collector plate improves the collector's efficiency by increasing the portion of collected heat via absorbing the surrounding deplete heat at loft area, and frost accumulation would become less serious. Solar energy collection raises the collector temperature, leading to lower temperature lift in the heat pumping process. It is fact that the energy input of the system slightly increases because of solar irradiations, thus, the refrigerant specific volume decreases and mass flow rate increases. This, in turn, was translated into an increase in both energy consumption and heating capacity of the system.

One observes that owing to the increase of heating capacity is more significant than that of energy consumption. Thus, $COP_{h,w-sys,av}$ of space-and-water, the heating mode was improved from 2.8 to 3.4 and the heating performance of the system improved accordingly. Therefore, solar irradiance can considerably reduce frosting formation of multi-functional *DX-SAHP* and benefit the performance of the whole system. The developed multi-functional *DX-SAHP* system could guarantee an appropriate operation under very low temperature and relatively low running cost. The privileged collector solution in cold climates consists of inner and outer plates makes it promising to satisfy the entirety of demands (space heating and *DHWT*) during winter. Comparing to a traditional *DX-SAHP* and *DX-SAHP* water heater, with or without sufficient solar irradiation available, the use of developed *DX-SAHP* was obviously more advantageous. This design as the present study reveals energy conserving solutions compared to existing; conventional and electrical resistance heaters systems. Not only was able to provide water and space heating efficiently for residential heating but also, helped to solve the incompetence of not being able to operate properly when solar energy absence. It can be concluded that for domestic space and water heating purposes in cold region the system is decidedly best-suited applications in the 50-70°C load range.

For future studies, a larger scroll variable compressor capacity with high speed can be replaced to keep abreast of enhancing the system performance, in order to overcome the mismatch between the collector, variable load and compressor constant capacity in the developed multi-functional *DX-SAHP* system.

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