THERMAL PERFORMANCE OF A SOLAR-ASSISTED HEAT PUMP WITH A DOUBLE PASS SOLAR AIR COLLECTOR UNDER CLIMATE CONDITIONS OF IRAQ

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Abstract

The prime aim of the current research is to investigate the thermal performance of the solar-assisted heat pump (SAHP) under the Iraqi climate experimentally and theoretically. In the winter season, the ambient air temperature reduces which causes a reduction in the coefficient of performance (COP) of heat pumps. By utilizing the thermal energy of solar to raise the heat transfer rate of the evaporator, compressor work diminishes and thus the COP of heat pump rises. The experimental setup of SAHP is performed by joining of a solar air heater and an air-to-air heat pump. In this arrangement, the inlet of the air evaporator has been preheated by a solar air heater. The mathematical model based on energy balance was evolved and the performance of this system has been studied over a cold season of Baghdad city (placed in the middle of Iraq). The consequences revealed that the presence of porous media in the lower channel of the absorber plate providing a great surface area for convective heat transfer, therefore, the variation of air temperature and thermal efficiency of the solar air heater are raised. The average thermal efficiency for models (II, III, IV, and V) over the model I (Conventional) are (16.6%, 21.2%, 26.2%, and 30.3%) respectively at mass flow rate 0.02172 kg/s.

Keywords: Air source Heat Pump, Solar Assisted Heat Pump, Power Consumption

I. Introduction

Approximately 1.74 x 10¹⁴ kW (174 PW) of solar radiation reaches the Earth's outer atmosphere. Of this solar energy, roughly thirty percent is reflected into space and the remaining seventy percent is absorbed by the oceans, landmass, and clouds in the Earth’s atmosphere. In fact, the Earth’s atmosphere absorbs enough solar radiation in just over an hour which equals the total amount of energy that the entire world used in 2007 (Smil, 2008). This shows the potential for solar energy and
how it can be used as an environmentally friendly alternative to reduce the reliance on fossil fuel sources.

For the Heat Pump (HP), an elementary factor of huge significance for its efficacious application is the readiness of an inexpensive, dependable heat source for the evaporator-preferably one at comparatively high temperature. The coefficients of performance (COP) of HP systems rely on numerous factors, like the temperature of low-energy source, the temperature of delivered useful heat, the working medium used, the characteristics of components of HP systems, etc. Based on above, the temperature of the evaporator stands for a key factor (Kara et al, 2008).

There is a large quantity of heat available in the ambient air. By means of air-source heat pumps, this extremely distributed low-grade energy is feasibly extracted. Air source heat pumps effectively take heat from air. Throughout winter, heat is taken from the external air and throughout summer, heat is taken away from the inside air. Air source heat pumps have been generally employed in new buildings and retrofits. As compared with other heat pump categories, installation expenses have been relatively low. Nevertheless, air source heat pumps can produce several noise problems and the devices can be problematic to hide (Brumbaugh, 2004).

Air source heat pumps have been proportional to warmer climates. In colder climates, air-source heat pumps can work well, but the rate of COP decreases as the outside temperature dropscausing less economic operation. This takes place throughout the winter time, as the heating load is at biggest. Consequently, an auxiliary heating system has been necessary in colder climates. Furthermore, air-source heat pumps in colder climates can have several energy mistreatment as a result of the defrost cycle (Dincer, 2010).

A combined system, termed as a Solar Assisted Heat Pump (SAHP), can be employed for solving the drawbacks of system operating independently. The benefit for the heat pump cycle in coupling it with a solar thermal collector, is an intensification in evaporator temperature over either air-source or ground-source heat pumps. This intensification in temperature causes an enhanced heat pump coefficient of performance (COP). Based on the solar collector approach, using heat pump lessens the fluid temperature returning to the collector near or below ambient temperature. This lower temperature raises the collector efficiency, and tolerates for substantial heat gains with low cost unglazed solar absorber panels, even under marginal conditions (Chaturvedi and Shen, 1982), (Morrison, 1994), (Freeman and Harrison, 1997). The combined system tolerates for efficient operation over a broaderassortment of seasons and weather conditions, and for more hours in all over the day.

Numerous structures of SAHPs were examined and applied in the past. Most of preceding studies have dedicated their use in space heating uses involving both “parallel” and “series” configurations.

The parallel solar heat pump integrates solar collectors besides a heat pump with a supplementary source (ambient air, ground, …etc.). They work autonomously from each other and deliver thermal energy to the storage tank dependent on solar in solation and heat necessity. The benefit of this structure is that COP hp doesn’t rely
on the collector temperature; its drawback is the requirement of a supplementary heat source as in a ground collector.

In the series of solar heat pump system, solar energy has been supplied to the heat pump evaporator through solar thermal collectors. The collectors have been the only heat source for the heat pump that delivers thermal energy to the heating system. Series configured systems can be further sub-categorized as “direct” or “indirect” types. In a ‘direct” (or direct-expansion, DX) system, heat pump refrigerant is straightforwardly spread by a solar collector that acts as the system’s evaporator. Solar energy absorbed in the collector/evaporator has conveyed to the load through the heat pump’s condenser.

Masoud and Misaph [10] investigated theoretically the performance of a DX-SAHP used for domestic water heating in the range of 20-45 °C. That domestic water heating system consists of a solar collector that is a flat plate containing a three-square meter individual cover, water tank and tubes immersed in the tank as a condenser. The consequence of solar radiation and compressor speed on the COP were investigated. The analysis of the collected data showed that COP of such HPs is better than those of conventional. Also, the COP is increased with larger area collector and the lower compressor speed. System COP has been 6.37 and 8.39 at solar irradiance values of (450 and 950) W/m². Vladimr and et al. [11] studied experimentally and analytically the performance of a SAHP of evaporator consist of two flat plate solar collector. The working fluid was R134 a. The effect of solar radiation and the compressor speed on the performance of the HP was investigated. From through this study that found that increasing solar radiation leads to increase coefficient of performance (COP). On the other hand, increasing the speed of the compressor leads to reduce coefficient of performance (COP). Also, this study found that increasing the temperature of evaporating temperature leads increase coefficient of performance (COP). Kush [12] experimentally investigated the performance of heat pump used for space heating. It was noted that the HP using water sources such as wells, natural surface water had not exploited all the thermodynamic available COP and using solar collector could increase energy of the source. The experiments were conducted on SAHPs in series configuration with R-12 as a refrigerant. The obtained results revealed the increasing of source temperature above 10 °C improves the COP. Chaturvedi et al. [13] investigated experimentally the performance of DX-SAHPs which used in domestic water heating. That domestic water heating system consists of a solar collector that is a bare flat plate. The utilized compressor was of a variable speed in order to conform between the compressor and the collector for different ambient conditions. The result from experiments showed that decreasing the compressor speed lead to improve the COP. Yumrutas and Kaska [14] investigated experimentally the performance of SAHP used for heating. This heating system consists of a solar collector that is a flat plate with heat pump device and the heating system ends with a water tank. The working fluid is water for the collector and R-12 for HP. The data were collected in February. The COP reached 2.5 at the lower storage tank temperature and 3.5 for the higher storage temperature. Hawlader et al. [15] through this study, theoretical and practical investigations the performance of a DX-SAHP working on refrigerant R-134a. In order to find the compatibility in the work of the radiation of the collector and the compressor device was used flat plate system where the results of this paper...
indicates that this system is noticeably affected by a number of factors, the most important factors surface area of the radiation collection system as well as the speed factor of the speed of the compressor device and finally the amount of radiation falling on the surface area of the collector radiation. On the other hand, the results revealed that the performance of coefficient (COP) was influenced by the temperature of the water stored in the evaporator tank. The coefficient of performance (COP) values range from 4 to 9 when the temperature of the water stored in the evaporator tank was range from 30 °C to 50 °C. Gorozabel Chata et al. [16] this research paper investigated the performance of a DX-SAHP thermal systems by experimenting with two different types of radiation collection refrigerants system. The two types used in this study are a bare radiation collector and a glass cover radiation collector. On the other hand, three types of refrigerant gas were used R-12, R-22 and R-134a as individually gases. also, a mixture of gases was used R410a, R407c and R404a. The results of the study indicated that the highest values of the coefficient of performance (COP) observed in the use of individual gases were for the refrigerants gases R-12, R-22 and R-134a while the results of mixed refrigerants gases were the highest value for cop was for gas R410a followed by gas R407c and then gas R404a. Ahmet and Cemil [17] from through this study, both theoretical and practical aspects of a SHAPE performance were examined from used the empty tube collector. The results of this research paper showed that the highest values for the performance of COP factor should reach 6.38. Li et.al, [18] Through this paper the DX-SAHP performance was achieved with the use of some factors, for example, the speed of rotation of the compressor was steady in addition to the use of plate made of aluminum material for the collector radiation and for the water evaporator and finally was used a water submerged coil device for the condenser. The results of this research paper showed that the value of daytime performance of coefficient was higher than the performance of coefficient in rainy night where the values were 6.61 and 3.11 respectively. Also, the results showed that the Seasonal COP and collector efficiency values of this system can also reach 5.25 and 1.08 respectively. Kadhum [19] investigated experimentally and theoretically the heated effectiveness of the IX-SAHP under Iraqi climate. To achieve this, a heat pump of 1 hp rating capacity was connected to an evacuated tube solar collector. TRNSYS software was implemented for the theoretical analysis, from through this paper, the study showed that the factors that affect the efficiency of the performance of the heating system include the temperature of the evaporator and condensing device in addition to the surface area exposed to sunlight for the radiation collector. The results of this research showed that the efficiency of the surface area exposed to sunlight to the radiation collector was increasing by increasing the temperature of the room, in addition to the amount of radiation falling on it. Therefore, this leads to an increase in the heat transfer rate of the evaporator which leads to the raising of the efficiency of COP values from 2.2 to 2.39 when the ambient temperature increases from 9.9 degrees Celsius to 14.9 degrees Celsius with an increase in the amount of solar radiation from 268 W/m² to 689 W/m².

The key point of this paper is to lessen electrical consumption and \( \text{CO}_2 \) emission of SAHP as compared with conservative HP. With the aim of achieving this
goalmouth, a calculated model based on energy principle method has established for both solar air heater and air source heat pump.

II. Modeling of Solar Air Collector

II.i. Energy Balance on the Solar Air Collector

The overall performance of solar air collectors can be characterized by using energy balance. The energy distribution of the solar air collector is calculated by dividing the total heat gained to the total heat loss. The energy balance on the solar air collector can be clarified as follows:

\[ Q_i = Q_u + Q_{\text{cond}} + Q_{\text{conv}} + Q_{\text{rad}}. \]  

Where:
- \( Q_i \): Total incident solar energy rate (W).
- \( Q_u \): Useful energy rate from collector (W).
- \( Q_{\text{conv}} \): Convective losses rate from collector (W).
- \( Q_{\text{cond}} \): Conduction losses rate from collector (W).
- \( Q_{\text{rad}} \): Radiation losses rate from collector (W).

The total heat losses from glass cover and absorber plate surface are represented by \( Q_l \) as shown:

\[ Q_l = Q_{\text{cond}} + Q_{\text{conv}} + Q_{\text{rad}}. \]

II.ii. The Absorbed Energy

There are several factors that affect the amount of absorbed energy such as:

The relation is used to calculate absorbed energy from the following equation (Duffie and Beckman, 2013):

\[ Q_{\text{abs}} = F_t \times I_t \times A \]  

Where, \( F_t \) is the effective absorption-transmittance factor which is determined from the equation bellow, and \( I_t \) is the total incident solar radiation:

\[ F_t = (\alpha_p \tau_F) \times F_{\text{sh}} \times F_d \]  

where;
- \( F_d \): Dust factor.
- \( F_{\text{sh}} \): Shading factor.
- \( (\alpha_p \tau_F) \) is given by as follows:
\[ (\alpha_p \tau_g)_{e} = 1.02 \times \alpha_p \tau_g \] (5)

\( \tau_g \): glass cover transparency.
\( \alpha_p \): Absorptance of absorber plate.

**Energy Loss**

Heat loss takes place from all parts of the solar air collector. The absorber plate loses the largest part of the heat losses which takes place from the top part of the collector. Also, the losses occur from the sides and bottom of the collector, the amount energy lost is determined by:

\[ Q_l = U_l \times A_c \times (T_p - T_a) \] (6)

Where:
\( A_c \) = Solar collector surface area (m²).
\( T_a \) = Ambient temperature (K).
\( T_p \) = Average temperature of absorber plate (K).
\( U_l \) = Overall heat loss coefficient (W/m². K).

The overall heat loss coefficient involves the heat losses that occur from all the external surfaces of the solar air collector as following:

\[ U_l = U_e + U_b + U_t \] (7)

Where:
\( U_t \) = Heat losses coefficient from top surface (W/m². K).
\( U_e \) = Heat losses coefficient from edges surface (W/m². K).
\( U_b \) = Heat losses coefficient from back surface (W/m². K).

The edge loss coefficient can be expressed as follows (Grag 1987):

\[ U_e = \frac{K_e \times A_e}{X_e \times \lambda_c} \] (8)

\( A_e \): Collector edge area (m²).

From the top of absorber plate the heat loss coefficient to the environment is estimated as following, (Duffie and Beckman, 2013).

\[ U_t = \left[ \frac{1}{h_{cfg}+h_{rgg}} + \frac{1}{h_{cga}+h_{rgs}} \right]^{-1} \] (9)

The back-loss coefficient of the conventional solar air collector (single-flow) as shown in Figure (3.1) is defined by:
The back-loss coefficient of the dual channel (double-flow) solar collector as shown in Figure (3 a, b) can be calculated from the following (El-Sebaii, et. al. 2011):

\[
U_b = \frac{K_b}{X_b} \quad (10)
\]

\[K_b: \text{Thermal conductivity of insulating material.} \]

\[X_b: \text{Thickness of insulating material.} \]

The heat transfer radiation coefficient between the absorber plate lower surface and the bottom plate \((h_{rpb})\) can be expressed as follows:

\[
h_{rpg} = \frac{\sigma(T_p + T_b)(T_p^2 + T_b^2)}{\frac{1}{\varepsilon_p + \varepsilon_b} - 1} \quad (12)
\]

The heat transfer radiation coefficient between the absorber plate upper surface and the lower surface of glass cover \((h_{rpg})\) is determined by:

\[
h_{rpg} = \frac{\sigma(T_p + T_g)(T_p^2 + T_g^2)}{\frac{1}{\varepsilon_p + \varepsilon_g} - 1} \quad (13)
\]

\[T_g = \text{Average temperature of glass covers (K).} \]

\[T_b = \text{Average temperature of back plate (K).} \]

\[\varepsilon_b = \text{Emission of back plate.} \]

\[\varepsilon_g = \text{Emission of glass cover.} \]

\[\varepsilon_p = \text{Emission of absorber plate.} \]

The heat transfer radiation coefficient between the glass cover upper surface and the sky \((h_{rgs})\) is determined by:

\[
h_{rgs} = \varepsilon_g \sigma(T_g + T_s)(T_g^2 + T_s^2) \quad (14)
\]

Where sky temperature \((T_s)\) is taken same to ambient temperature.

The convective heat transfer coefficient \((h_w)\) of glass cover depends on wind speed \((v_w)\), (Duffie and Beckman, 2013):

\[
h_w = 5.7 + 3.8 v_w \quad (15)
\]
The empirical correlation for the solar flat plate collector with smooth absorber plate to define the Nusselt number for the turbulent flow (Re > 6000) at flows in rectangular duct exposed to a constant heat flux from side and insulation from another side as follows, (Ong, 1995):

\[
\text{Nu} = 0.036 \text{Re}^{0.8} \text{Pr}^{1/3} (D_h/L)^{0.055}
\]  

(17)

\[
D_h = \frac{2W \times t}{W + t}
\]  

(18)

Where \((w,t)\) is width and thickness and \((D_h)\) is the hydraulic diameter respectively of the test region of the collector.

2.5 The Useful Energy

The amount of useful energy earning by the solar air collector is defined as follows (Struckmann, 2008):

\[
Q_u = Q_1 - Q_1 = I_t (\alpha_p t_g) A_c - U_1 A_c (T_p - T_a)
\]  

(19)

As well as, the useful energy gain can be obtained from fluid passing in the solar collector and carried the heat amount, as follows:

\[
Q_u = m_c p (T_o - T_i)
\]  

(20)

\(T_i\): Inlet temperature of collector.

\(T_o\): Outlet temperature of collector.

2.6 Factor of Heat Removal

The factor of heat removal can be expressed as the amount of effective useful heat energy (actual heat transfer) to the greatest possible useful energy. The greatest possible useful energy takes place when the temperature of absorber plate is at the inlet fluid temperature; heat losses to the surroundings are then at a minimum as follows (Duffie and Beckman, 2013):

\[
F_R = F'' F'
\]  

(21)

Where the collector efficiency factor \(F'\) and the collector flow factor \(F''\) are estimated as following equations:

\[
F'' = \frac{mc_p}{A U_1 F'} \left[ 1 - \exp \left( \frac{A U_1 F'}{mc_p} \right) \right]
\]  

(22)

\[
F' = \left[ 1 + \frac{U_1}{h_p h^+ (1 + \frac{1}{h^+ 1})^{-1}} \right]^{-1}
\]  

(23)
The heat removal factor ($F_R$) is given as (Struckmann, 2008):

$$F_R = \frac{\dot{m} c_p (T_0 - T_i)}{A_c [I_t (\alpha_p T_R) - U_l (T_i - T_a)]}$$

(24)

Subsequently, the effective useful energy can be expressed by multiplying the heat removal factor of the collector by the maximum possible of useful energy. This permit the reformulation of equation (2.19):

$$Q_u = F_R A_c [I_t (\alpha_p T_R) - U_l (T_i - T_a)]$$

(25)

### Thermal Efficiency

The solar collector performance is represented by energy efficiency. The solar air collector potential was estimated by a concept of energy efficiency in the process of energy conversion. Additionally, it gives a hint of heat energy absorbed by the solar air collector. It expresses a ratio of actual useful heat energy to incident solar radiation on the area of solar collector, from the following equation:

$$\eta = \frac{Q_u}{A_c I_t}$$

(26)

Therefore;

$$\eta = \frac{F R A_c [I_t (\alpha_p T_R) - U_l (T_i - T_a)]}{A_c I_t}$$

(27)

### III. Modeling of Heat Pump

The main components of a Heat Pump are a compressor, an expansion valve, an evaporator and condenser.

In general, the predominant criteria to assess the Heat Pump effectiveness are the coefficient of performance. The coefficient of performance for a heating mode can be described as the ratio of a load of the condenser ($Q_{\text{con}}$) to the total input work throughout the cycle. Besides, the entire input work is the summation the work of compressor ($W_{\text{comp}}$) and also the fan work for condenser and the evaporator ($W_{\text{fan,con}}, W_{\text{fan,eva}}$). Hence, the COP equation can be expressed as:

$$\text{COP} = \frac{Q_{\text{con}}}{W_{\text{comp}} + W_{\text{fan,con}} + W_{\text{fan,eva}}}$$

(34)

In this study, a steady-state condition will be considered when Heat Pump is analyzed. Furthermore, the mathematical model consists of four loops; three of them are concerned with the parts of Heat Pump (included compressor, condenser, and evaporator). The loops were independently modeled using equations of energy conservation and including the relationship of empirical and quasi-empirical. In addition, the main loop is the fourth loop. Many
assumptions were used to analyze the mathematical modeling of the proposed SAHP system:

- The pipes which are connecting between different parts can be considered adiabatic.
- The process is considered as an isenthalpic expansion.
- Polytropic compression process the refrigerant passes it with a constant polytropic index (n).
- The heat losses related to the compressor are ignored.
- The changes in the kinetic and potential energy are negligible.
- Frictional losses in the condenser and the evaporator are assumed to be insignificant.
- Water does not condensate on the external surfaces of the evaporator.

III.i. Compressor Work

The compressor is a major part of a Heat Pump, which consumes the biggest portion of power. The goal of modeling a compressor of a heat pump is to obtain shaft speed, refrigerant outlet temperature, and required compressor work utilizing many known parameters like inlet temperature, the refrigerant mass flow rate, outlet and inlet pressure. Work of the compressor can be calculated as for as:

\[ W_{\text{comp}} = \dot{m}_r (h_{c,\text{out}} - h_{c,\text{in}}) \]  
(28)

III.ii Capillary Tube

The capillary tube is utilized to reduce the temperature and pressure of the refrigerant fluid from a high temperature and pressure states to a low temperature and pressure states. Also, the expansion valve is utilized to control the flow rate of refrigerant to regulate the level of superheating and to guarantee total condensation of the refrigerant fluid in the evaporator. The process of occurring in the thermostatic expansion valve is assumed an isenthalpic expansion process as there is no work or heat output or input, given as:

\[ h_{r,i} = h_{r,o} \]  
(29)

III.iii Evaporator Heat gain

The refrigerant receives the latent heat from the load and exits from the evaporator (heat exchanger) as a dry gas. Then analysis of the evaporator is almost the same as that of the condenser.

\[ Q_{\text{evap.}} = \dot{m}_r (h_{r,i} - h_{r,o}) \]  
(30)

Where \( Q_{\text{evap.}} \) represents the heat absorbed in (kW) and \( (h) \) is the outlet enthalpy at the evaporator in (kJ/kg).
The heat absorbed by the air is given as:

\[ Q_{\text{evap}} = \dot{m}_a C_{p,a} (T_{a,o} - T_{a,i}) \]  

(31)

III.iv. Condenser Heat Rejected

Air-cooled condenser consists of uniformly spaced parallel plates fins and a range of tubes that pass perpendicularly through the plates. It’s dissipates the heat absorbed from the internal space to the surrounding. The dissipated heat in the condenser from the equation as follow:

\[ Q_{\text{cond}} = \dot{m}_r (h_{r,i} - h_{r,o}) \]  

(32)

The heat rejected by the air is given as:

\[ Q_{\text{cond}} = \dot{m}_a C_{p,a} (T_{a,o} - T_{a,i}) \]  

(33)

IV. Solar Assistant Heat Pump concept

The following correlation was utilized after deriving the mathematical models to execute the Solar Assistant Heat Pump Concept for each part of the combined system (Heat Pump and solar air collector).

The evaporator inlet air temperature equals to the collector outlet air temperature:

\[ T_{(c,\text{air,out})} = T_{(\text{eva,air,in})} \]  

(35)

V. Experimental Setup

The experimental setup consists mainly of three parts, the solar collector, the heat pump and the test cabin. Auxiliary components were used to connect the main parts. The auxiliary components are a flexible duct and evaporator hood. The instrumentation used for the measurements are solar intensity meter, power consumption meter and temperature sensors. The experimental setup is shown in Figure 1.
Two solar collectors with single glass cover were fabricated according as reported in ASHRAE recommendations. SAHs were constructed with porous media longitudinal. The solar collector consists of a single glass sheet,

Figure (1): Photograph of the experimental test rigs.

Figure (2): Schematic diagram for the test rig

V.i Solar Air Collector

Two solar collectors with single glass cover were fabricated according as reported in ASHRAE recommendations. SAHs were constructed with porous media longitudinal. The solar collector consists of a single glass sheet,
Strips of porous media are fixed on the upper and lower face of the absorber plate. Aluminum chips are packed in form of strips. The cross-section area of the strip $(5 \times 7) \text{ cm}^2$ and the length of the strip line is the same as the absorber plate and arranged uniformly in longitudinally rows in the flow directions as shown in (Fig. 3, a). A Schematic diagram for the porous media installed inside the SAH was shown in (Fig. 3, b) The absorber plate surfaces were painted with matte black paint for absorbing maximum portion of the sun’s radiation.

The experiments were implemented during January and February 2019. The test rigs were operated under the weather condition of Baghdad ($33^\circ$N, $44^\circ$E). The solar air heater is assembled on an inclined frame having an angle of $43^\circ$ and during all experiments is oriented to the south to receive the maximum solar radiation. For each collector three thermocouples are fixed on the absorber plate at an equal distance. Two thermocouples are used to measure the inlet and outlet air. The ambient temperature is measured by a thermocouple shielded from the direct solar radiation. The air velocity at the inlet is measured by vane meter.
V.ii. The Heat Pump

It consists of a rotary compressor, an expansion valve, a condenser and an evaporator.

The evaporator used in experiments consists of finned copper tube of (8.5) in diameter. The evaporator absorbs the heat from the hot air supplied from the solar collector. The Condenser specification are tabulated in Table (3)

The Condenser consists from finned tubes. It supplies absorbed heat in the evaporator, to the cabin. The sensible heated is reject at first then the latent heat.

**Figure (4.a):** Schematic diagram for porous media arrangements

**Figure (4.b):** Solar air collector with porous media arrangements(I), (II), (III), (IV) and (V) used in the present study.
Table 1: The Condenser specification

| Parameters                  | Specifications | Parameters                  | Specifications |
|-----------------------------|----------------|-----------------------------|----------------|
| Height                      | 485 mm         | Fin thickness               | 0.125 mm       |
| Width                       | 600 mm         | Fin pitch                   | 1.5 mm         |
| Length in air flow direction| 33 mm          | Vertical distant between tubes | 12.5 mm      |
| Outside diameter            | 8.5 mm         | Number of rows              | 22             |
| Inside diameter             | 8.075 mm       | Number of fins              | 428            |
| Tube wall thickness         | 0.425 mm       | Refrigerant                 | R22            |

The expansion valve used is capillary tube. The function of this valve is to reduce the pressure of the refrigerant. The expansion process is achieved by the pressure drop caused by the flow of refrigerant from large diameter tube to a small diameter tube.

An air conditioner compressor is the component in the system that raises the temperature and pressure of the vapor refrigerant that leaves the evaporator coil. The rotary compressor is used for most ac split.

Table 2: The Compressor specification

| Model                  | Displacement cm³/rev | Cooling Capacity W | Power W | Capacitor | Dis. Pipe LD mm | Suction Pipe LD mm |
|------------------------|----------------------|--------------------|---------|-----------|------------------|---------------------|
| PH215G1C-4DZ1          | 21.3                 | 3645               | 12429   | 1215      | 35/370           | 8.2                 | 9.8                 |

V.iii. The Test cabin

The cabin, used as a space to be conditioned, is made of sandwich panels of thickness 10cm. The dimensions of the cabin are (1.5 length × 1.8 width × 2 height) m³. The condenser is fixed inside the cabin. Two thermo couples are fixed inside the cabin to measure the space interior temperature. Thermocouple is fixed in front of condenser to measure the temperature of air from the condenser for the purpose of set point.

VI. Results

Many experiments were conducted on the experimental apparatus to examine the effect of the changing parameters on the performance of the IX-SAHP. These experiments involve changing the surface area by add porous media at different sites to solar air collector with different air mass flow rate of solar collector, the temperature of air from collector which exchange with evaporator and also recognize the effect of solar irradiation and ambient...
temperature. The effect of these changed parameters can be seen in Figures (8) to (18).

VI.i Effect of Ambient Temperature, Heat Gain and Solar Radiation on the Thermal Performance.

A range of tests were performed to determine the characteristics of the SAHP under different conditions. Five models of solar air collector with two air mass flow rates of tests were run. For each test, the compressor power consumption was measured, and the heat transfer rates and COP were calculated based on the measured values.

Since the solar radiation intensity and ambient temperature play the main role on the heat gain of the solar collector and the effect of these parameters on the performance of the heat pump system is analyzed and presented in this section. Figures (8) to (9) illustrate the effect of heat gain and solar radiation on the power consumption of compressor and COP of the system respectively. These experiments were run under the meteorological conditions of Baghdad through winter 2019. In examining the effect of solar radiation on the power consumption of compressor at full heat pump on the compressor, it was found the power consumption of compressor was reduced from 0.9 kW at solar radiation and heat gain 398 W/m$^2$ and 0.12 kW to 0.69 kW at solar radiation and heat gain 997W/m and 20.448 kW, this leads to rising in COP from 4.53 to 6.3 as shown in Figures (10) to (13).

![Figure (8) Comparison the effect of Ambient temperature (C), Solar Radiation (w/m$^2$) & Heat Gain on COP with day hours for models V and IV at air mass flow rate (0.02172 kg/s)](image)
Figure (9) Comparison the effect of Ambient temperature (C), Solar Radiation (w/m²) & Heat Gain on Power consumption (w) with day hours for model V at air mass flow rate (0.02172 kg/s)

Figure (10) Comparison the effect of collector configuration on COP with day hours for all models at air mass flow rate (0.01086 kg/s)

Figure (11) Comparison the effect of collector configuration on COP with day hours for all model at air mass flow rate (0.02172 kg/s)
VI.ii. Effect of porous media position and configuration on the Thermal Performance.

The COP be more when the porous media was exist in (the lower channel) of the solar air collector because of the useful energy of the air in this case be more than in case of (in upper channel), as result the average COP of Models (V) (in lower channel) are more than in Models (IV) (in upper channel) by (4.9%) and the power consumption decrease by (5.1%) at mass flow rate 0.02172 kg/s as shown in Figures (14) and (15).

The COP increases when the useful energy increases. The heat energy obtained by the flowing air in the upper pass from the glass covers is used to preheat it and also decreases the temperature of glass covers which in turn decreases the heat losses to the surroundings and hence the performance of such heaters has been found to be superior compared to conventional solar air
heater wherein air flows in one pass either above or below the absorber plate. It is noted that the COP of dual solar air heater with porous media is higher than that of dual solar air heater without porous media. This is because of the increase in heat transfer area per volume due to an increase in thermal capacity inside the solar heater.

VI.iii. Effect of Collector Air Mass Flow Rate

When the working fluid (air) flow at velocity (0.02172 kg/s), it will not take enough time to receive the dissipation heat from the absorber plate and porous media by convection, that’s lead to decreases in air temperature raise of air comparing with velocity of (0.01086 kg/s), but the useful energy and the thermal efficiency be more according previous explain, also the convective heat transfer coefficient of air be more with high velocity.

The COP for solar assistant heat pump increases when the air mass flow rate increases due to increase in heat gain while the power consumption
decreases when increase the mass flow rate due to the compressor required less work as shown in Figures (16) and (17).

![Figure (16) Comparison the influence of air mass flow rate on COP with day hours for model V](image1)

![Figure (17) Comparison the influence of air mass flow rate on power consumption with day hours for model V](image2)

**VI.iv. Comparison with Conventional Heat Pump System**

The performance of a traditional heat pump system is considerably limited by the heat source. In this research, a try was performed to recover the evaporator heat and investigate the performance of a traditional heat pump with a further heat source: solar energy, the existing indirect expansion solar assisted heat pump system has the benefit that the solar air collector can absorb the ambient and radiation energy and lead to increase the superheat level at the inlet of the compressor, so that decreased the work input of the compressor in contrast to the traditional heat pump system.

A set of experiments tests were conducted to assess the performance of the system and subsequent comparison of the two systems. The results are tabulated in Table (3) As shown in this table the power consumption of...
compressor for SAHP system is decreased by 34.7% as compared with conventional heat pump also, the heat transfer rates condenser and evaporator also the COP are increased by 7.2% and 25.5% respectively.

**Table (3): System performance comparison**

|                     | Power (KW) | $Q_C$ (KW) | $Q_E$ (KW) | COP   |
|---------------------|------------|------------|------------|-------|
| Conventional heat pump | 1.15       | 3.91       | 2.77       | 3.36  |
| SAHP                | 0.78       | 4.33       | 3.45       | 5.57  |

From table can be noticed the power consumption of compressor of SAHP is importantly less than the power consumption compressor of HP. In other words, the evaporator of SAHP experienced higher air temperatures with respect to the evaporator of HP which consequently reduced the work of the compressor.

**VI.v. Comparison of the current work with other works**

In this section, the results of the current experimental work are compared with other results of previous experimental work. The comparison is not a quantities comparison but, it is a qualitative comparison, i.e. the agreement was achieved in the performance, features, and the tendency of the curves but there is no agreement in the values with the previous works and this was seen in Figure (18).

Figure (18) shows a comparison between the present experimental results including COP with respect to time for the experimental results (A) presented by (Salmanzadeh, et al, 2019).

![Graph showing COP comparison](image)
VII. Conclusion

In this study, the concept of solar assisted heat pump was presented by combination of a solar air collector with and without porous media at several configurations and an air-to-air heat pump. In this configuration, inlet air of evaporator was preheated by solar air collector.

The heat gain for solar air collector was increased with increasing in an ambient temperature and solar radiation, this leading to rising in the heat transfer rate of evaporator due to the increasing temperature of evaporating, so the heat transfer rate of evaporator rises from 1.77 kW to 1.89 kW as the solar radiation and ambient temperature rise from 10.1°C and 268 W/m² to 13.8 °C and 689 W/m² respectively, this drives to increase in the coefficient of performance from 2.2 to 2.39 as the solar radiation and ambient temperature rise from 9.9°C and 268 W/m² to 14.9 °C and 689 W/m² respectively.

Mass flow rate of air from collector crossing through the evaporator has a great influence on the thermal characteristics of the solar assistant heat pump system. The COF of the solar assistant heat pump reduced with the increasing the mass flow rate of air crossing through the evaporator. The coefficient of performance increased from 1 to 3.2, responding to the air mass flow rate passing through the evaporator decreasing from (0.0217 kg/s) to (0.01086kg/s).

The current experimental results achieve a good qualitative agreement with the previous works in the characteristics performance and the tendency of the curves. Additional future work can be done to make use of Cloud Computing technology and Particle Swarm Optimization algorithm to optimize the employed system performance with highly accurate results.
References

I. Brumbaugh, J. E. (2004). Audel. HVAC Fundamentals. Volume 3: Air Conditioning, Heat Pumps and Distribution Systems. Wiley Publishing. 105.

II. Caglar, A. and C. Yamali, Performance analysis of a solar-assisted heat pump with an evacuated tubular collector for domestic heating. Energy and Buildings, 2012. 54(0): p. 22-28.

III. Chaturvedi, S. K., and Shen, J. Y., 1982, "Analysis of Two-Phase Flow Solar Collectors with Application to Heat Pumps," Journal of Solar Energy Engineering, 104 pp. 358.

IV. Chaturvedi, S.K., D.T. Chen, and A. Kheireddine, Thermal performance of a variable capacity direct expansion solar assisted heat pump. Energy Conver Manage, 1998. 39(3-4): p. 181-191.

V. Dincer, I. K. (2010). Refrigeration Systems and Applications (2nd Edition). New Jersey: Wiley.

VI. Duffie, J., & Beckman, W. (2006). Solar Engineering of Thermal Processes. (3rd, Ed.) Hoboken, NJ: John Wiley & Sons, Inc.

VII. El-Sebaii, A. A., et al. "Investigation of thermal performance of-double pass-flat and v-corrugated plate solar air heaters." Energy 36.2 (2011): 1076-1086

VIII. Freeman, G. A., 1997, "Indirect Solar-Assisted Heat Pumps for Application in the Canadian Environment," Master's thesis, Queen's University.

IX. Garg, H. P. Advances in Solar Energy Technology: Volume 3 Heating, Agricultural and Photovoltaic Applications of Solar Energy. Springer Science & Business Media, (1987).

X. Gorozabel Chata, F.B., S.K. Chaturvedi, and A. Almogbel, Analysis of direct expansion solar assisted heat pump using different refrigerants. Energy Conversion and Management, 2005. 46(15-16): p. 2614-2624

XI. Hawlader MNA, Chou SK, Ullah MZ. The performance of a solar heat pump water heating system. Appl Thermal Energy 2001; 21:1049.

XII. Jassim, Najim Abed, and Kadhim Kareem Al-Chlaihani. "Experimental Evaluation of Thermal Performance of Solar Assisted Vapor Compression Heat Pump." Journal of Engineering 21.11 (2015): 145-160.

XIII. Kara O, Ulgen K, Hepbasli A. Exergetic assessment of direct-expansion solar-assisted heat pump systems: review and modeling. Renewable and Sustai-able Energy Reviews 2008;12(5):1383-401.
XIV. Kush, E. A., 1980, "Performance of Heat Pumps at Elevated Evaporating Temperatures - with Application to Solar Input," Journal of Solar Energy Engineering, 102 pp. 203-210

XV. Li, Y.W., et al., Experimental performance analysis on a direct expansion solar-assisted heat pump water heater. Applied Thermal Engineering, 2007. 27(17-18): p. 2858-2868

XVI. Ong, K. S. "Thermal performance of solar air heaters Experimental correlation." Solar Energy 55.3 (1995): 209-220.

XVII. Smil, V., “Energy in Nature and Society: General Energetics of Complex Systems”, (2008).

XVIII. Safijahanshahi, Esmaeil, and Mazyar Salmanzadeh. "Performance simulation of combined heat pump with unglazed transpired solar collector." Solar Energy 180 (2019): 575-593.

XIX. Struckmann, Fabio. "Analysis of a flat-plate solar collector." Heat and Mass Transport, Project Report, 2008MVK160 (2008).

XX. Vladimir Soldo, Tonko Curko, Igor Balen, Thermal Performance of a Direct Expansion Solar Assisted Heat Pump, International Refrigeration and Air Conditioning Conference. Paper 724.

XXI. Yousefi, M., & Moradali, M. (2015). Thermodynamic analysis of a direct expansion solar assisted heat pump water heater. Journal of Energy in Southern Africa, 26(2), 110-117.

XXII. Yumrutas, R. and O. Kaska, Experimental investigation of thermal performance of a solar assisted heat pump system with an energy storage. International Journal of Energy Research, 2004. 28(2): p. 163-75.