Experimental Evaluation of Thermal Performance of Solar Assisted Vapor Compression Heat Pump

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ABSTRACT

The thermal performance of indirect expansion solar assisted heat pump, IX-SAHP, was investigated experimentally under Iraqi climate. An Indirect-Solar Assisted Heat Pump system was designed, built, instrumented and tested. Experimental tests were conducted by varying the controlling parameters to investigate their effects on the thermal performance of the IX-SAHP such as cooling water flow rate, heating water flow rate, ambient temperature and solar radiation intensity. The investigation covered values of cooling water flow rate of (2, 3, 4, 5 l/min) and heating water flow rate of (2, 3, 4, 5 l/min) under meteorological condition of Baghdad from November 2014 to January 2015.

The results indicated that the performance of the IX-SAHP is not dependent on the heating water flow rate. On the contrary of heating water flow rate, cooling water flow rate has significant effect on the thermal performance of the system. The COP of the heat pump system is decreased with increasing cooling water flow rate. The collector heat gain increase with increasing the solar radiation and ambient temperature, this leads to increase in COP from 2.2 to 2.39 as the ambient temperature and solar radiation increase from 9.9℃ and 268 W/m^2 to 14.9 ℃ and 689 W/m^2 respectively.

Key words: heat pump, indirect expansion, solar assisted heat pump.

تقييم الاداء الحراري لمضخة حرارية من نوع ضغط البخار بمساعدة الطاقة الشمسية عملياً

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الخلاصة

تم في هذا البحث دراسة الاداء الحراري لمضخة حرارية تعمل بمساعدة الطاقة الشمسية من نوع التمدد غير مباشرعملياً. تم تصميم و بناء مضخة حرارية تعمل بمساعدة الطاقة الشمسية للاستخدام في بيئة بغداد مع كافة اجهزة القياس. تم اجراء إختبارات عملية بتغيير العوامل الحاكمة في أداء المضخة لحرارية للتحري عن تأثيراتها على خصائص المضخة. الدراسة غطت قيم معدل تدفق مياه التبريد (2، 3، 4 و 5 لتر / دقيقة) ومعدل تدفق مياه التسخين (2، 3، 4 و 5 لتر / دقيقة) تحت ظروف الجوية لبغداد من تشرين الثاني 2014 إلى كانون الثاني عام 2015

بينت النتائج العملية أن أداء المضخة الحرارية لا يعتمد على معدل تدفق ماء التسخين , وعلى العكس من معدل تدفق مياه التسخين , يكون لمعدل تدفق ماء التبريد تأثير واضح على الأداء الحراري للنظام. حيث انخفض معامل الاداء الحراري للمضخة الحرارية مع زيادة معدل تدفق ماء التبريد . ان الحرارة المكتسبة من الجامع الشمسي قد ازدادت مع زيادة الاشعاع الشمسي الساقط ودرجة الحرارة المحيطة مما ادى الى زيادة معامل الاداء الحراري للمضخة من 2.2 الى 2.39 مع ارتفاع درجة الحرارة المحيطة و الإشعاع الشمسي من 9.9°C و 268 W / m^2 إلى 14.9°C و 689 W / m^2 على التوالي.

الكلمات الرئيسية : مضخة حرارية, تمدد غير مباشر , مضخة حرارية بمساعدة الطاقة الشمسية.

1. INTRODUCTION

Heat pump systems are heat-generating devices that can be used to heat water to be used in either domestic hot water or space heating applications. The coefficients of performance (COP) of HP systems depend on many factors, such as the temperature of low-energy source, the temperature of delivered useful heat, the working medium used, the characteristics of components of HP systems, etc. Among the above mentioned, the temperature of the evaporator is a key factor **Kara, et al., 2008**.To increase the evaporating temperature, combination of solar collector and heat pump could be used to ease many of the cons of either Solar Domestic Water Heaters (SDWHs) and Heat Pump systems operating independently.

There are two main types of heat pump assisted solar systems that have been studied in the past; these are Direct and Indirect Expansion Solar Assisted Heat Pumps.

In a ‘direct” (or direct-expansion, DX) system, heat pump refrigerant is *directly* circulated through a solar collector that acts as the system’s evaporator. Solar energy absorbed in the collector/evaporator is transferred to the load via the heat pump’s condenser.

For IX-SAHP systems, there are many possible system configurations. Unlike the DX-SAHP systems, the solar collector does not act as the evaporator for the heat pump, but rather the heat pump is integrated into the design as a closed unit.

During operation, solar and ambient energy absorbed by the solar collectors is transported by the water to a heat exchanger that acts as the evaporator for the heat pump. Low pressure refrigerant circulating through heat exchanger/evaporator absorbs the heat input, vaporizes and superheats. The superheated vapour is then compressed to a high pressure by an electrically driven compressor where it flows into another heat exchanger that acts as the heat pump condenser.

The use of solar assisted heat pump (SAHP) increased as the commercial solar collectors entered into the market. Several researchers have developed and tested various SAHP models. **Freeman, et al., 1978.** investigated theoretically a combined solar heat pump system for residential space, and domestic water heating using the TRNSYS simulation software. They investigated three combined solar heat pump systems; parallel, series and dual.

These systems were compared with the results of a conventional solar space heating system, and a conventional heat pump space heating system. The annual COPs for the parallel, dual source, series and conventional heat pump systems were found to be 2.0, 2.5, 2.8 and 2.1 respectively. **Morgan, 1981**. studied theoretically and experimentally the performance of a series direct configuration heat pump system for water heating. Throughout the test, Morgan found average COP values ranging between 2.5 and 3.5. **Chaturvedi,** and **Shen, 1984**. investigated the steady state performance of a direct expansion SAHP for water heating experimentally, the evaporator of the heat pump was a bare plate collector. The experimental results showed that high collector efficiencies of between 40 – 70% were feasible with bare collectors operating under ambient winter conditions in northern America. COPs for the system ranged from 2 to 3, which were 30 –50% higher than the heat pump operating in fan mode. **Morrison, 1994**. investigated experimentally the performance of heat pump water heaters with solar boosted evaporators, and compared the results with a TRNSYS model simulating the annual performance. He proposed an integral system called an integral direct expansion solar assisted heat pump (IDX-SAHP), in which the evaporator and condenser were coupled with the storage tank, which needed to be installed outdoors. The results showed good agreement between the model and experimental results, and showed a relatively constant COP and energy savings throughout the year. **Kuang, et al., 2002**. experimentally investigated a solar heating system with a water source heat pump. It is concluded that the thermal storage tank is an important component in solar heating systems, which can modulate the mismatch between solar radiation and the heating load. **Li, et al., 2007**. investigated DX-SAHP using DXSAHP with constant speed rotary compressor, immerse condenser coil, aluminum plate collector/evaporator. The coefficient of performance of the "DX-SAHPWH" system can be up to 6.61 during daytime and 3.11 at a rainy night. The seasonal average value of the COP and the collector efficiency can be reached to 5.25 and 1.08, respectively. **Bridgeman,** and **Harrison, 2008**. produced a prototype of an IX-SAHP system based on the system modeled in TRNSYS by **Freeman, 1997**. to validate the simulation findings. Constant source temperature tests were performed by them in order to compare to the simulation results obtained from the TRNSYS model by **Freeman, 1997**. It was found that the TRNSYS model over predicted the COP of the heat pump due to over prediction of the effectiveness of the two heat exchangers. Once this was corrected in the TRNSYS model, the compressor power consumption and COP were within approximately 3% of the experimental results.

In the current study the thermal performance of indirect expansion solar assisted heat pump, IX-SAHP, was investigated experimentally under Iraqi climate. An Indirect-Solar Assisted Heat Pump system was designed, built, instrumented and tested. Experimental tests were conducted by varying the controlling parameters to investigate their effects on the thermal performance of the IX-SAHP such as cooling water flow rate, heating water flow rate, ambient temperature and solar radiation intensity. The investigation covered values of cooling water flow rate of (2, 3, 4, 5 l/min) and heating water flow rate of (2, 3, 4, 5 l/min) under meteorological condition of Baghdad from November 2014 to January 2015.

2. MATHEMATICAL MODEL

The mathematical modeling of the proposed IX-SAHP system to predict the thermodynamic performance is simplified based on the following general assumptions: i. Quasi-steady state conditions are approximated within the chosen time interval. ii. Pressure drop and heat loss in the connecting pipes are neglected. iii. Frictional losses in the evaporator and the condenser are negligible. iv. A good thermal insulation over the refrigerant loop is assumed, i.e. thermal loss to the surroundings is neglected. v. Kinetic and potential energy changes are assumed to be insignificant. vi. The refrigerant undergoing polytrophic compression with a constant polytrophic index (n). vii. Isenthalpic expansion process is considered.

2.1. Compressor

The various equations are expressed as follows from **Stoecker,** and **Jones, 1982**.

Piston Displacement per cylinder (V_d) is calculated as:

$$V_d = \frac{\pi \, D^2 \, L}{4} \tag{1}$$

Where (D) is the pore of the cylinder (m), (L) is the stroke of the cylinder (m).
Volumetric efficiency of compressor (η_v) is calculated as:

$$\eta_v = 1 + C - C\left(\frac{p_o}{p_i}\right)^{\frac{1}{n}} \tag{2}$$

where C is the clearance volume ratio and obtainable from the manufacturer's data.
Mass flow rate of the refrigerant through the compressor ($\dot{m}_r$) is expressed as:

$$\dot{m}_r = \frac{V_d \, N \, \eta_v}{60 \, v_i} \tag{3}$$

where N is the speed of the compressor (rpm).
And finally the compressor work (W_{comp}) can be calculated as:

$$W_{comp} = \frac{p_i \, v_i \, \dot{m}_r}{\eta} \left(\frac{n}{n-1}\right)\left[\left(\frac{p_o}{p_i}\right)^{\frac{n-1}{n}} - 1\right] \tag{4}$$

2.2. Water Cooled Condenser

Refrigerant releases heat to the water in the condenser, and becomes saturated or sub-cooled, cold water is also supplied to the condenser and hot water flows out to maintain the water temperatures and allows for more hot water production.
The heat rejected by the refrigerant is given as:

$$Q_r = \dot{m}_r\left(h_{r,i} - h_{r,o}\right) \tag{5}$$

The heat absorbed by the water is given as:

$$Q_w = \dot{m}_w C_{p,w}\left(T_{w,o} - T_{w,i}\right) \tag{6}$$

Assuming negligible heat loss from the water tank to the ambient, the heat transfer from the refrigerant to the water should be balanced as follows:

$$\dot{m}_r\left(h_{r,i} - h_{r,o}\right) = \dot{m}_w C_{p,w}\left(T_{w,o} - T_{w,i}\right) \tag{7}$$

The heat released by the condenser can also be written as:

$$Q_{cond} = \varepsilon \, C_{min}\left(T_{h,i} - T_{c,i}\right) \tag{8}$$

where $T_{h,i}$ is the inlet temperature of the hot fluid and $T_{c,i}$is the inlet temperature of the cold fluid. In present study, $T_{h,i}$ refers to the temperature of superheated refrigerant and $T_{c,i}$ refers to the temperature of feed water. In the equation, effectiveness-NTU method is used to determine the effectiveness of the condenser. Effectiveness, ε, is the ratio of actual heat transfer rate of a heat exchanger to the maximum possible heat transfer rate. ε can be express as follow (**Incropera, F.P., and DeWitt, D.P., 2002**):

$$\varepsilon = \frac{actual\ heat\ transfer}{maximum\ possible\ heat\ transfer} = \frac{Q_{cond,evap}}{Q_{\max(cond,evap)}} \quad (9)$$

To determine the maximum possible heat transfer rate in a heat exchanger, we first recognize that the maximum temperature difference in a heat exchanger is the difference between the inlet temperatures of the hot and cold fluids. That is,

$$\Delta T_{max} = \left(T_{h,i} - T_{c,i}\right) \quad (10)$$

The maximum heat transfer in a heat exchanger will get if (1) the temperature of the cold fluid is raised to the inlet temperature of the hot fluid or (2) the temperature of the hot fluid is lowered to the inlet temperature of the cold fluid. These two limiting conditions will not be reached simultaneously only in the case of matching the heat capacities of the fluids (i.e. $C_c = C_h$). When $C_c \neq C_h$, which is usually the case, the fluid with the smaller heat capacity rate will experience a larger temperature change, and thus it will be the first to experience the maximum temperature, at which point the heat transfer will come to a halt. Therefore, the maximum heat transfer rate is:

$$Q_{max} = C_{min}\left(T_{h,i} - T_{c,i}\right) \quad (11)$$

where C_{min} is the smaller of $C_h = \dot{m}_h C_{p,h}$ and $C_c = \dot{m}_c C_{p,c}$.
To calculate the effectiveness of the heat exchanger in the two phase region, we note that with a phase change process $\frac{c_{min}}{c_{max}} = 0$. Thus for the water condenser, the effectiveness is given as follows:

$$\varepsilon = 1 - e^{-\frac{U\,A}{c_{min}}} \quad (12)$$

where $\frac{U\,A_s}{c_{min}}$ is called the number of transfer units NTU .

2.3. Expansion Valve

An isenthalpic expansion process is assumed for the thermostatic expansion valve as there is no heat or work input or output, given as:

$$h_{r,i} = h_{r,o} \quad (13)$$

2.4 Evaporator

The heat transferred through the evaporator was determined by calculating the increasing in the refrigerant enthalpy and the energy reject from the water as seen below:

$$Q_{evap} = \dot{m}_w cp_w (T_{w,i} - T_{w,o}) \quad (14)$$

$$Q_{evap} = \dot{m}_r (h_{r,o} - h_{r,i}) \quad (15)$$

where $T_{w,i}$ and $T_{w,o}$ correspond to the inlet and outlet temperatures of water in the evaporator, and $h_{r,o}$ and $h_{r,i}$ are the enthalpies of the refrigerant at the outlet and inlet on the refrigerant side of the evaporator.

2.5 Evacuated Tube Solar Collector

The heat transfer in this collector is driven purely by natural circulation of water through the single-ended tubes. Water in the tubes is heated by solar radiation, rises to the storage tank and is replaced by colder water from the tank.
The heat rate gained by the water is given by:

$$Q = \dot{m} c_{p,w} (T_o - T_i) \quad (16)$$

The collector efficiency can be defined as the useful heat output from the collector to the input solar irradiation (G) received on the surface of the collector, and is:

$$\eta = \frac{\dot{m} c_{p,w} (T_o - T_i)}{A_c\, G} \quad (17)$$

1. EXPERIMENTAL APPARATUS AND METHOD

A schematic diagram of the system is shown in **Fig.1**, depicts the main components of the IX-SAHP setup which consists of three different process fluid loops; the collector loop, heat pump loop, and cooling tower loop. The collector loop consists of evacuated tube solar collector, circulating water pump and heat exchanger. The evacuated tube solar collector consists of (20) evacuated tube made of borosilicate glass 180 cm-long evacuated tube providing hot water to a (112 liter) horizontal storage tank. A centrifugal pump with 27 l/min and (25 m) head was used to circulate the water in the collector loop. It is fixed between the solar collector and the heat exchanger (Evaporator), the final component in the charge loop was a heat exchanger, which was used to transfer energy from water to the refrigerant.
The heat pump loop was connected in series between the collector loop and the cooling tower loop and used R-12 as the working fluid. Heat was transferred between the loops with the use of two heat exchangers. The other two fundamental components of the heat pump loop were a semi hermetically sealed reciprocating compressor, and a thermostatic expansion valve which throttled the flow and maintained a constant superheat temperature exiting the evaporator. The high pressure side of the heat pump was constructed using 3/8" refrigeration grade copper tube, and the low side was constructed using 5/8” copper tube. The heat pump loop prior to being fully installed is displayed in **Fig.2.**
The evaporator consists of a copper coil with 15m length and 3/8" outer diameter immersed in a tank of water of (37cm diameter, and 50cm height) with a capacity of 53 liters. The evaporator bent into a spiral coil filling the whole height of the galvanized steel tank to maximize heat exchange. The condenser is a shell and tube of eight tubes with two passes heat exchanger, the cold water flows inside the copper tubes and the refrigerant passes around the tubes. The characteristics of solar collector, evaporator and the condenser are given in **Table1**.

Cooling Tower loop consists of a cross- flow water type cooling tower, heat exchanger (condenser) which was discussed above and a circulating water pump. Cooling tower is used to cool water that used to condense the refrigerant.
In order to determine the performance of the prototype under various conditions, the apparatus was equipped with the necessary instrumentation.
Many measuring devices were used to sense the variation of cooling water temperatures & flow rate, heating water temperatures & flow rate, refrigerant temperatures, low and high pressures along the heat pump , electrical voltage & current, air temperatures and Solar radiation intensity. A schematic diagram of the apparatus is shown in **Fig.3**, demonstrating the positioning of the instrumentation within the prototype. **Table 2** lists the instrumentation used in the experiment.
Two types of tests were conducted: constant water flow rate in the condenser, and constant water flow rate in the solar collector.
Condenser constant water flow rate tests were for 4 l/min and for four different flow rates of water in solar collector namely: 2, 3, 4 and 5 l/min.
Solar collector constant water flow rates tests were also for 4 l/min and for four different flow rates of water in the condenser namely: 2, 3, 4 and 5 l/min.

4. EXPERIMENTAL PROCEDURE

The evaporator heat transfer rate was calculated for both heat lost from the water, and the heat gained by the refrigerant. The heat lost by the water was calculated as:

$$Q_e = \dot{m}_w \, c_{p,w}(T_5 - T_6) \tag{18}$$

where $\dot{m}_w$ is the water flow rate, $c_{p,w}$ is the heat capacity of water, T_5 and T_6 are the inlet and outlet evaporator water temperatures.
The condenser heat transfer rate was calculated as:

$$Q_c = \dot{m}_w \, c_{p,w}(T_8 - T_7) \tag{19}$$

where T_7 and T_8 are the inlet and outlet condenser water temperatures.
The heat transferred through the evaporator and the condenser can be evaluated as the energy gained by the refrigerant such that:

$$Q_c = \dot{m}_r \,(h_2 - h_3) \tag{20}$$

$$Q_e = \dot{m}_r \,(h_1 - h_4) \tag{21}$$

where $\dot{m}_r$ is the refrigerant flow rate of the, and $h_1$through h_4 represent the enthalpies at the corresponding points throughout the heat pump loop.
The heat pump COP is determined as:

$$COP = \frac{Q_c}{W_{comp}} \tag{22}$$

where Q_c is calculated based on the energy rejected from the refrigerant, and W_{comp} is the compressor work measured by power meter.

5. RESULTS AND DISCUSSION

5.1 The Effect of Collector Loop Water Flow Rate

The testing on the effect of collector loop water flow on the performance of the heat pump system was carried out under four flow rates (2, 3, 4 and 5 l/min), these tests were run at constant water flow rate at the condenser at 4 l/min.
It is shown that the effect of different collector loop flow rates on the discharge temperature and collector useful heat gain in **Fig.4** to **Fig.7**, belong the inlet temperature**,** it wasn't constant but it is decreased with time as it is shown because it depends on the supplier (evaporator). The outlet temperature is affected by the amount of mass flow rate, where the outlet temperature decreases with increasing the water flow rate. This is due to the solar heat that is delivered to the tank which is almost the same for all the cases and then from the tank to the load, in case of high flow rate, the heat which is collected in the tank by thermosyphon will get out by the withdrawal water as much as possible the other flow rates, and make up water will come in the tank to balance the heat in the tank, so it will not let the tank temperature rise up higher. (In case the collector working with same solar radiation and ambient conditions, whenever the flow rate is reduced the outlet temperature will rise up and vice versa).
On the other hand the inlet temperature wasn't constant, so the water temperature difference $(T_{w,out} - T_{w,in})$ will decrease as the collector water flow rate increase, while the useful heat gain from the collector is increased slightly with increasing in water flow rate because the increasing in flow rate overcome the decreasing in water temperature difference.
As the solar collector is coupled directly with the evaporator, any increase in collector heat gain will lead to increase in evaporator heat transfer rate. Due to a small increase in the collector heat gain as the flow rate increases from 2l/min to 5 l/min and due to the losses from the pipes and the evaporator tank with the environment, the increase in the evaporator heat transfer rate is very little, as the evaporator heat transfer rate increases from 1.61 kW at $\dot{m}$ =2 l/min to 1.69 kW at $\dot{m}$ =5 l/min, so the condenser heat transfer rate and COP are not significantly affected, this behavior was also mentioned by (**Bridgeman et al, 2008**).
The COP's for all four tests are shown in **Fig.8**, from this figure it is seen that the COP's throughout the tests were essentially equal. The COP=2.07 at $\dot{m}$ =2 l/min and equal to 2.104 at $\dot{m}$ =5 l/min.

5.2. The Effect of Condenser Water Flow Rate.

It is shown that the effect of different condenser water flow rate on the condenser water outlet and inlet temperatures, and the condenser heat transfer rate in **Fig.9** to **Fig.12**.
Belong the condenser inlet temperature, it is relatively constant (varies slightly $\pm$2 ºC) with time because it was supplied by the cooling tower. The outlet temperature is affected strongly by the mass flow rate, the condenser outlet water temperature decreases with increasing the water flow rate and vice versa. It is seen that the condenser heat transfer rate of the heat pump system decreased with increasing condenser water flow rate due to the increases in water flow rate will result in decreasing the temperature difference of the water, where the condenser heat transfer rate decreased from 2.18kW at $\dot{m}$ =2 l/min to 1.87 kW at $\dot{m}$ =5 l/min, therefore the

evaporator heat transfer rate will also decrease with increasing condenser water flow rate. As a result of this the COP of the system decreased with increasing condenser water flow rate. The COP dropped smoothly as the condenser water flow increased from 2l/min to 3 l/min, and then the drop tended to be continue as the condenser water flow increased from 3l/ min to 5 l/min as shown in **Fig.13.** This behavior was also mentioned by **(Hongbing Chen et al, 2011)**. The COP was decreased from 2.27 at $\dot{m}$ =2 l/min to 2.0 at $\dot{m}$ =5 l/min.

5.3. Effect of Solar Radiation and Ambient Temperature.

Fig.14 to **Fig.20** describe the effect of solar radiation and ambient temperature on the collector heat gain, condenser and evaporator heat transfer rate and COP of the system respectively. These tests where run under the meteorological condition of Baghdad on 20^{th} of January 2015, the collector and condenser water flow rates were kept constant at 4 l/m through the tests.

Fig.14 and **Fig.15** show that the solar water heater gain increased with increasing in solar radiation and ambient temperature, the heat gain is increased from 1.56 kW at 9.9℃ and 268 W/m^2 to 1.68 kW at 14.9 ℃ and 689 W/m^2.

As previously discussed any increase in solar water heater gain will lead to increase in evaporator heat transfer rate due to the increase of evaporating temperature, so the evaporator heat transfer rate increase from 1.69 kW to 1.75 kW as the ambient temperature and solar radiation increase from 9.9℃ and 268 W/m^2 to 14.9 ℃ and 689 W/m^2 respectively.

The condenser heat transfer rate was also increased from 2.14 kW at 9.9℃ and 268 W/m^2 to 2.27 kW at 14.9 ℃ and 689 W/m^2 as shown in **Fig.19**, this leads to increase in COP from 2.2 to 2.39 as the ambient temperature and solar radiation increase from 9.9℃ and 268 W/m^2 to 14.9 ℃ and 689 W/m^2 respectively as shown in **Fig.19** and **Fig.20.**

6. CONCLUSIONS

The thermal performance of an indirect expansion solar assisted heat pump was investigated experimentally. It is represented by the evaporator and condenser heat transfer rates and the coefficient of performance COP.

1. The COPs calculated for different collector loop flow rates were essentially the same, meaning that the flow rate should be selected based on the optimum flow rate for the collector.
2. Condenser water flow rate has a significant effect on the thermal performance of the system. The COP of the heat pump system decreased with increasing condenser water flow rate. The COP dropped from 2.27 to 2.0, as the condenser water flow rate increasing from 2 l/min to 5 l/min.
3. The collector heat gain increases with increasing in solar radiation and ambient temperature, this leads to increase in evaporator heat transfer. The evaporator heat transfer rate is increased from 1.69 kW to 1.75 kW as the ambient temperature and solar radiation increase from 9.9℃ and 268 W/m^2 to 14.9 ℃ and 689 W/m^2 respectively. The condenser heat transfer rate is increased from 2.14 kW at 9.9℃ and 268 W/m^2 to 2.27 kW at 14.9 ℃ and 689 W/m^2, this leads to increase in COP from 2.2 to 2.39 as the ambient temperature and solar radiation increase from 9.9℃ and 268 W/m^2 to 14.9 ℃ and 689 W/m^2 respectively.

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NOMENCLATURE

A	area	(m^2)
C_p	specific heat at constant pressure	(kJ/kg ^oC)

C	clearance volumetric ratio	(-----)
F_R	collector heat removal factor	(-----)
h	specific enthalpy	(kJ/kg)
I	solar radiation	(W/m^2)
$\dot{m}$	mass flow rate	(kg/hr)
n	polytropic index of compressor	(----)
p	pressure	(Pa)
Q_u	useful Energy gain/rejected	(Watt)
T	temperature	(^oC)
W_{comp}	compressor power consumption	(kW)
L	Stroke of compressor	(m)
D	Bore of compressor	(m)
N	Rotational speed of compressor	(rpm)
U	Overall heat transfer coefficient	($w/m^2\,^oC$)

Greek letter

υ	Specific volume	(m^3/kg)
η	Efficiency	(-------)
ϵ	Effectiveness	(-------)
ρ	Density	(kg/m^3)
$\tau\alpha$	Transmitance - absorptance	(------)

Sub-Script

i	Inlet , inner
o	Outlet , outer
r	Refrigerant
w	Water
m	Mean
u	Useful

Abbreviations

COP	Coefficient of performance
HP	Heat Pump
HPWHs	Heat Pump Water heating system
WHs	Water heating systems
SAHP	Solar Assisted Heat Pump
DX- SAHP	Direct Expansion-Solar Assisted Heat Pump
IX- SAHP	Indirect Expansion-Solar Assisted Heat Pump

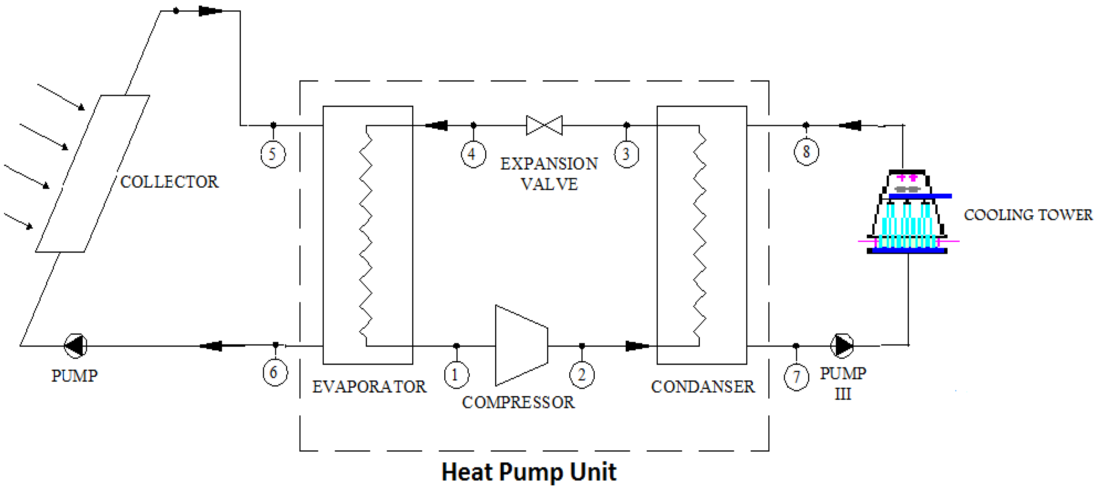

Figure1. Schematic of IX-SAHP.

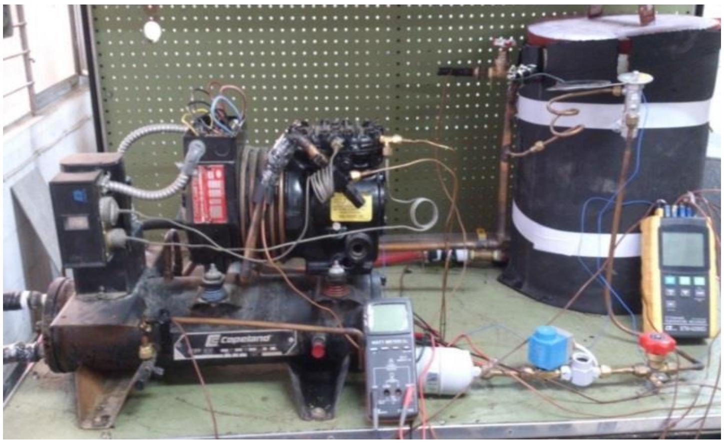

Figure2. Photograph of the experimental apparatus.

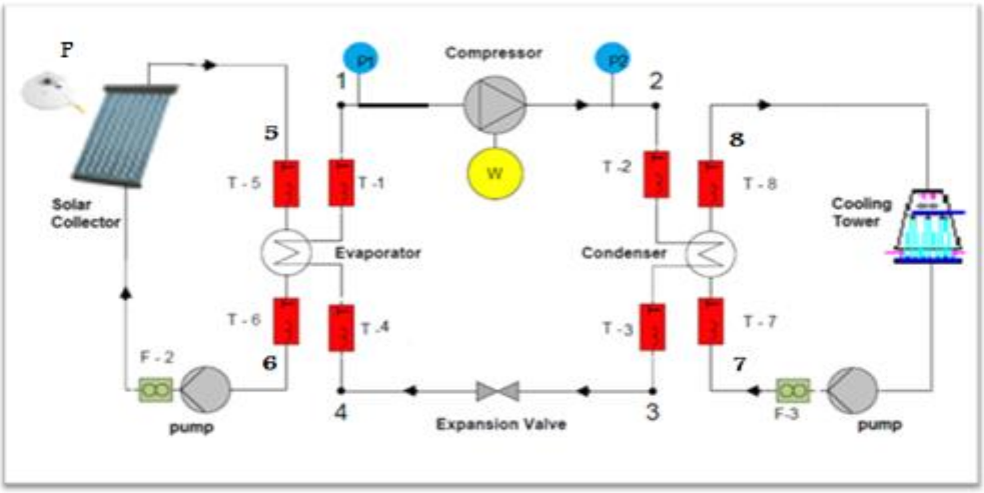

Figure 3. Schematic diagram of the IX-SAHP experimental rig.

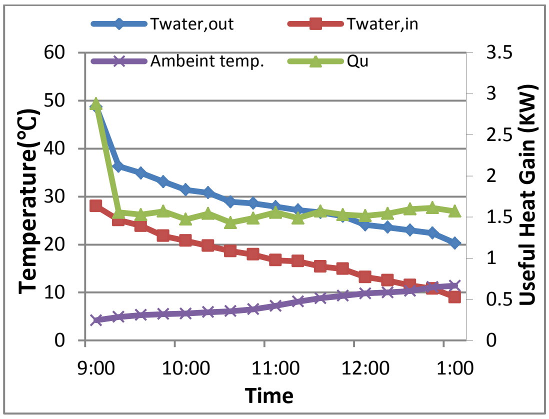

Figure4. Variation of ambient temperature, water temperature and collector heat gain with time for collector loop flow rate 2 l/min for 19th November 2014 in Baghdad city.

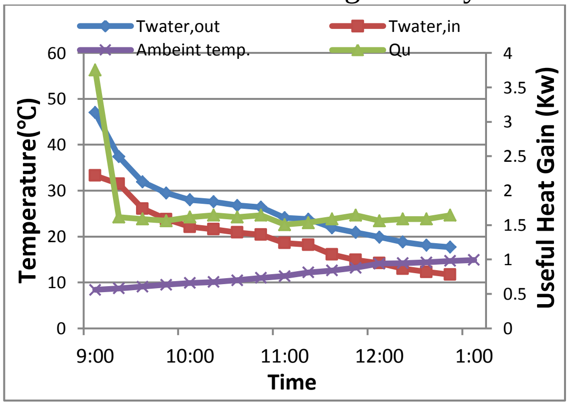

Figure5. Variation of ambient temperature, water temperature and collector heat gain with time for collector loop flow rate 4 l/min for 2nd December 2014 in Baghdad city.

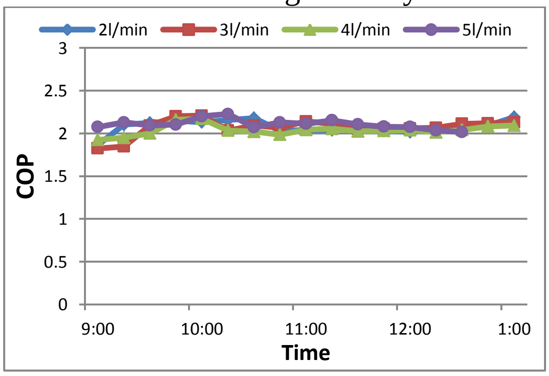

Figure6. COP's at different flow rates in the collector.

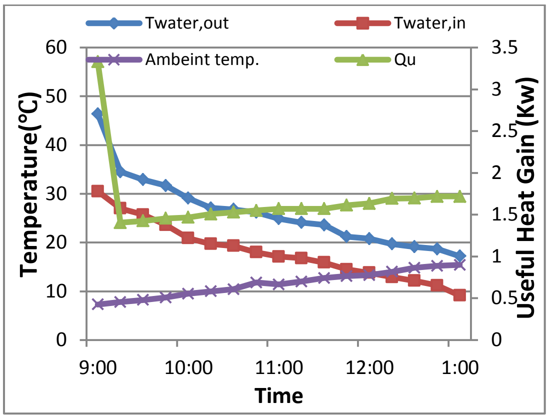

Figure7. Variation of ambient temperature, water temperature and collector heat gain with time for collector loop flow rate 3 l/min for 23th November 2014 in Baghdad city.

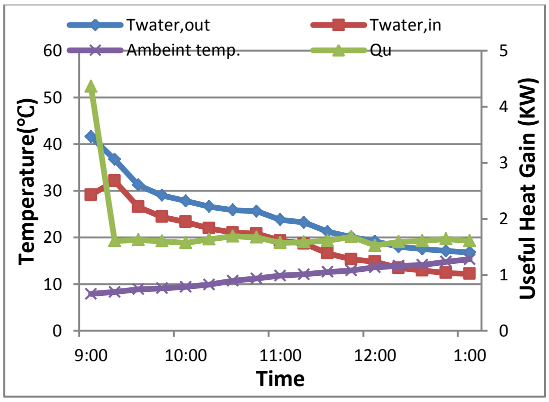

Figure8. Variation of ambient temperature, water temperature and collector heat gain with time for collector loop flow rate 5 l/min for 7th December 2014 in Baghdad city.

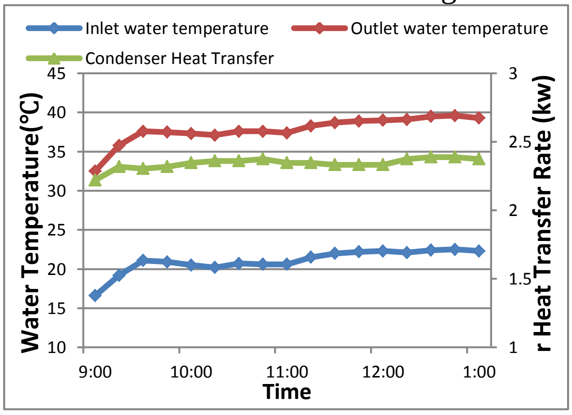

Figure9. Variation of condenser water temperatures and condenser heat transfer with time for condenser mass flow rate 2 l/min for 17th December 2014 in Baghdad city.

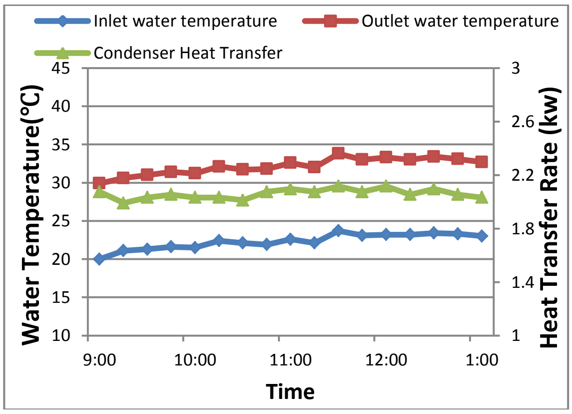

Figure10. Variation of condenser water temperatures and condenser heat transfer with time for condenser mass flow rate 3 l/min for 23th December 2014 in Baghdad city.

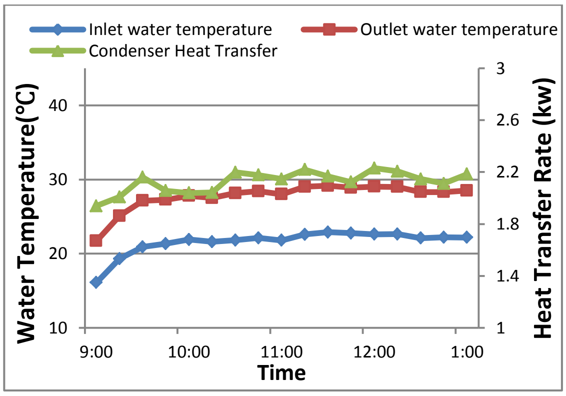

Figure11. Variation of condenser water temperatures and condenser heat transfer with time for condenser mass flow rate 5 l/min for 5th January 2015.

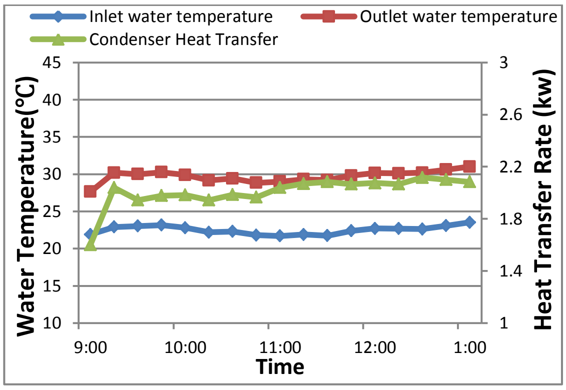

Figure13. Variation of condenser water temperatures and condenser heat transfer with time for condenser mass flow rate 4 l/min for 4th January 2015.

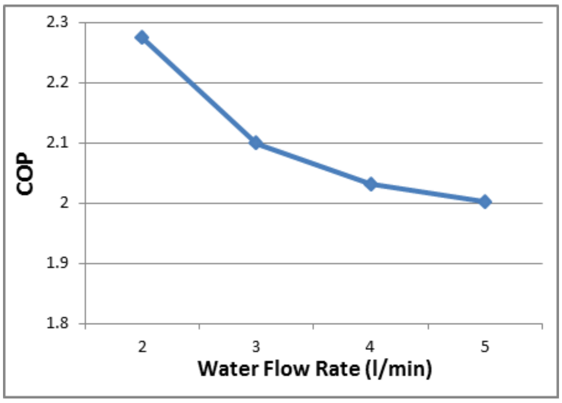

Figure14. Variation of COP with condenser water flow rates.

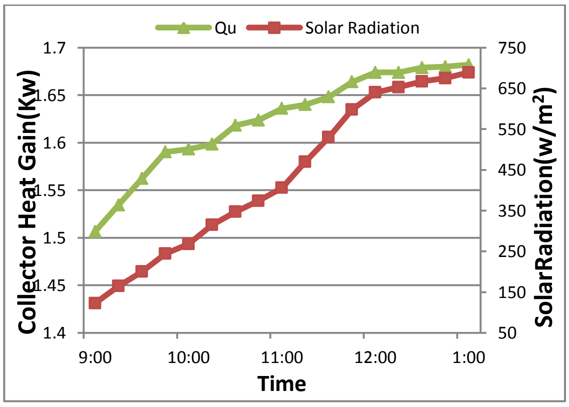

Figure12. Variation of collector heat gain and solar radiation with time for 20th January 2015 in Baghdad city.

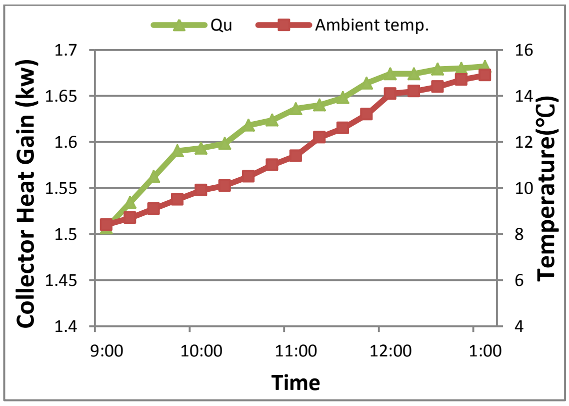

Figure15. Variation of collector heat gain and ambient temperature with time for 20th January 2015 in Baghdad city.

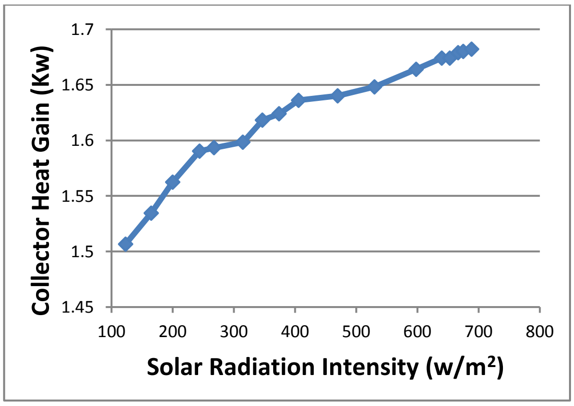

Figure16. Variation of collector heat gain with solar radiation intensity.

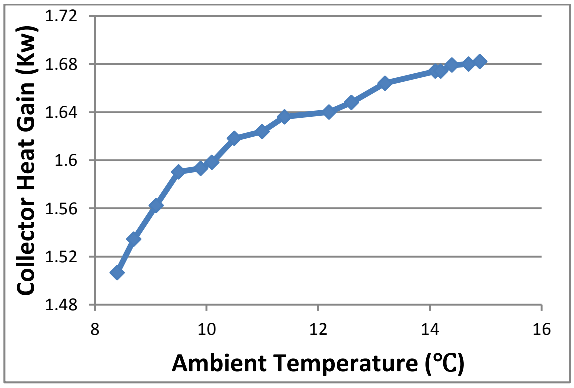

Figure19 Variation of collector heat gain with ambient temperature.

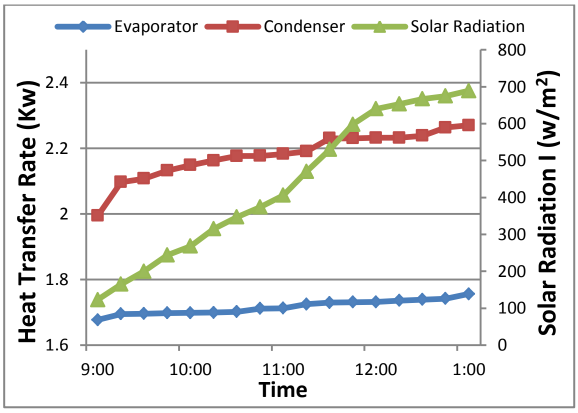

Figure17. Variation of evaporator, condenser heat transfer rates and solar radiation with time for 20th January 2015.

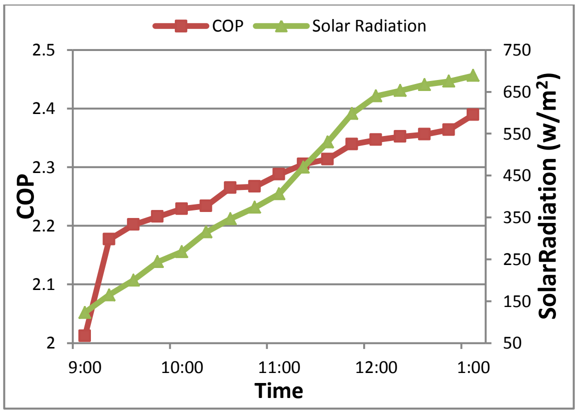

Figure20. Variation COP and solar radiation intensity with time for 20th January 2015.

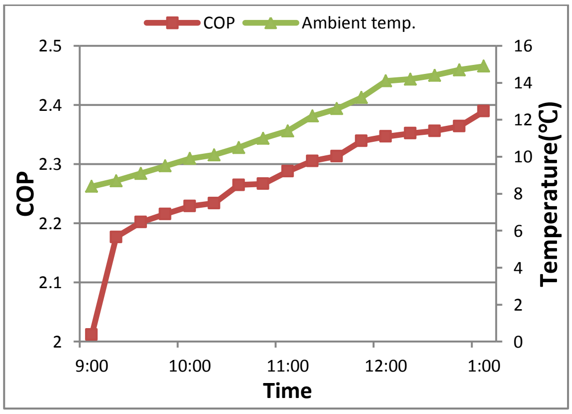

Figure18. Variation of COP and ambient temperature with time for 20th January 2015.

Table1. Characteristics of Solar collector, Evaporator and the Condenser.

Name	Remarks	
Solar Collector	Length (nominal)	176cm /180cm
	Outer tube diameter	5.8cm
	Inner tube diameter	4.4cm
	Glass thickness	0.16cm
	Thermal expansion	$3.3x10^{-6}$ ^{o}C
	Material	Borosilicate Glass 3.
	Absorptive Coating	Graded Al-N/Al
	Absorptance	>92% (AM1.5)
	Emittance	<8% ($80^{o}C$)
	Vacuum	$P<5x10^{-3}$ Pa
	Stagnation Temperatur	>$200^{o}C$
	Heat Loss	<0.8W/(m^2.K)
	Maximum Strength	0.8 MPa
Condenser	shell and tube with eight two passes tubes	
Evaporator	copper coil with 15m length and 3/8" outer diameter immersed in a tank of water of (37cm diameter, and 50cm height) with a capacity of 53 liters	

Table2. List of instrumentation used.

Part	Device and Specification
T_1-T_8	Thermocouple (T-type)
P_1,P_2	Bourdon Tube pressure gauge
F_2,F_3	Flow meter (rotameter)
W	Digital power meter model DW-6163
P	Kipp and Zonen class one pyranometer model CMP22