Modeling and Simulation of Thermal Performance of Solar-Assisted Air Conditioning System under Iraq Climate

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ABSTRACT

In Iraq most of the small buildings deployed a conventional air conditioning technology which typically uses electrically driven compressor systems which exhibits several clear disadvantages such as high energy consumption, high electricity at peak loads. In this work a thermal performance of air conditioning system combined with a solar collector is investigated theoretically. The hybrid air conditioner consists of a semi hermetic compressor, water cooled shell and tube condenser, thermal expansion valve and coil with tank evaporator. The theoretical analysis included a simulation for the solar assisted air-conditioning system using EES software to analyze the effect of different parameters on the power consumption of compressor and the performance of system. The results show that refrigeration capacity is increased from 2.7 kW to 4.4kW, as the evaporating temperature increased from 3 to 18 ºC. Also the power consumption is increased from 0.89 kW to 1.08 kW. So the COP of the system is increased from 3.068 to 4.117. The power consumption is increased from 0.897 kW to 1.031 kW as the condensing temperature increased from 35 ºC to 45 ºC. While the COP is decreased from 3.89 to 3.1. The power consumption is decreased from 1.05 kW to 0.7kW as the solar radiation intensity increased from 300 W/m^2 to 1000 W/m^2, while the COP is increased from 3.15 to 4.8. A comparison between the simulation and available experimental data showed acceptable agreement.

Keywords: solar assisted air conditioning system, solar collector, thermal performance, simulation program, EES.

تقييم الاداء الحراري لمنظومة تكييف هواء تعمل بمساعدة الطاقة الشمسية في العراق نظَرياً

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الخلاصة:

معظم المباني الصغيرة والمتوسطة في العراق تعتمد على انظمة التبريد ذات الضاغط الكهربائي والذي من عيوبه الاستهلاك العالي للطاقة الكهربائية عند حمل الذروة. في هذا العمل تم التحقق نظريا من الاداء الحراري لمنظومة تكييف الهواء المركبة مع مجمع شمسي. يتكون مكيف الهواء الهجين من ضاغط شبه مقفل، مكثف نوع الاسطوانة والانبوب المبرد بالماء، صمام تمدد حراري ومبخر نوع الخزان والملف. شمل التحليل النظري محاكاة لمنظومة تكييف الهواء التي تعمل بمساعدة الطاقة الشمسية باستخدام برنامج EES لتحليل تأثير عوامل مختلفة على استهلاك طاقة الضاغط وأداء المنظومة. بينت النتائج ان سعة التثليج قد زادت من 2,7 الى 4,4 كيلو واط مع زيادة درجة حرارة التبخير من 3 الى 18 درجة مئوية وكذلك فان الطاقة المستهلكة قد زادت من 0.89 الى 1.08 كيلو واط , والذي ادى الى زيادة معامل اداء المنظومة من 3,068 الى4,117. ان

الطاقة المستهلكة قد ازدادت من 0,897 الى 1,031 كيلو واط مع زيادة درجة حرارة التكثيف من 35 الى 45 درجة مئوية. بينما معامل الاداء قد انخفض من من 3,89 الى 3,1. ان الطاقة المستهلكة قد انخفضت من 1,05 الى 0,7 كيلو واط عند زيادة الاشعاع الشمسي 300 الى 1000 واط\م^2, بينما معامل اداء المنظومة قد ازداد من 3,15 الى 4,8. واظهرت المقارنة بين النتائج التجريبية والنظرية اتفاق مقبول بينهما.

كلمات رئيسية: مكيف هواء بمساعدة الطاقة الشمسية ، مجمع شمسي , اداء حراري , برنامج محاكاة .

1. INTRODUCTION:

The increasing consumption of energy in buildings on air-conditioning systems has initiated a great deal of research for energy savings. With the consolidation of the demand for human comfort, around 48% of energy is consumed in commercial and public buildings due to air conditioners, usually by driving electrical vapour compression chillers **Lamberts, 1999**. The use of solar thermal energy for air-conditioning in hot and sunny climate is a promising application of solar collectors in buildings. The main advantage of the solar air conditioning system is that in solar air conditioning applications solar gains and cooling loads occur at the same time and on the seasonal level. Figures 1 and 2 show the main variations of solar radiation and maximum temperature and relative humidity for Baghdad station **, Al-Salihi Ali et al, 2010**. Iraq is one of the countries with the highest electricity production from oil during 2008 **IEA, 2011**. **Vakiloroaya et al. 2012** described and studied a newly-developed solar-assisted air-conditioning system. They proposed a new discharge bypass line connected with an inline solenoid valve, installed after compressor to regulate the refrigerant flow rate passing through a hot water storage tank. Experimental test results show that the daily energy saving is about 38.6%, while the indoor temperature and relative humidity are within comfort ranges. **Ha et al. 2012** analyzed the performance of a newly solar-assisted direct expansion air conditioner and demonstrate the capability of energy savings and ecological conservation. They equipped a flat collector storage system as well as immersed piped coil with the direct expansion evaporator to raise the superheat temperature entering the variable speed compressor, causing the smaller duty cycle of the compressor and the slight increase in the suction pressure, and hence, reduce the energy consumptions. Water in the solar collector is in a contact with the collector absorbing surface so the heat is transferred to the water, then to the refrigerant in an immersed heat exchanger. Once the space has achieved the desired temperature, the compressor turns off while space cooling will continue till the pressure of the refrigerant at the circulation loop fails for saving the desired temperature. The advantage of the system is that the heat can be imparted in the refrigerant via the flat plate collector and therefore the compressor can remain off longer. The process justifies up to 40% of energy savings.

In the current study the thermal performance of solar assisted air conditioning system was investigated theoretically under Iraqi climate conditions. Theoretical tests were studied by varying the parameters to investigate their effects on the thermal performance of the solar assisted air conditioning system such as cooling water flow rate, heating water flow rate, ambient temperature and solar radiation intensity.

2. THE MATHEMATICAL MODEL

The following general assumptions have been adopted: 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 undergoes polytrophic compression with a constant polytrophic index (n). Vii. Isenthalpic expansion process is considered. The main components covered in the model are compressor, expansion valve, water-refrigerant condenser, evaporator and liquid solar collector. **Fig.3** shows the system configuration used in the simulation.

2.1 Compressor

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

Piston Displacement volume per cylinder is calculated as:

$$V_d = \frac{\pi\, D^2\, L\, N}{4} \quad (1)$$

Volumetric efficiency of compressor is calculated as:

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

where C is the clearance volume ratio and obtainable from the manufacturer's data.

Refrigerant mass flow rate can be evaluated such that:

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

Compressor work input can be written as:

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

where η_{mech} is the compressor mechanical efficiency.

Refrigerant discharge temperature and specific volume are determined such that:

$$T_{r,o} = T_{r,i}\left(\frac{p_o}{p_i}\right)^{\frac{n-1}{n}} \quad (5)$$

$$v_o = v_i\left(\frac{p_i}{p_o}\right)^{\frac{1}{n}} \quad (6)$$

2.2 Water Cooled Condenser

The water condenser is a shell and tube heat exchanger, with water in the tube and the refrigerant around it. 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:

$$Q_r = \dot{m}_r\left(h_{r,i} - h_{r,o}\right) \quad (7)$$

The heat absorbed by the water is given as:

$$Q_w = M\, C_{p,w}\, \frac{dT_w}{dt} + \dot{m}_w C_{p,w}\left(T_{w,o} - T_{w,i}\right) \tag{8}$$

where the first term refers to the heat gain by the water in the storage tank, while the second term refers to the heat gain by the circulating water. The first term would be zero if there is no change of water temperature in the tank over time, while the second term would be zero if there is no water supplied or discharged from the tank.

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:

$$\dot{m}_r\left(h_{r,i} - h_{r,o}\right) = M\, C_{p,w}\, \frac{dT_w}{dt} + \dot{m}_w C_{p,w}\left(T_{w,o} - T_{w,i}\right) \tag{9}$$

The heat released by the condenser can be written as:

$$Q_{cond} = \varepsilon\, C_{min}\left(T_{h,i} - T_{c,i}\right) \tag{10}$$

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 work, $T_{h,i}$ refers to the temperature of superheated refrigerant and $T_{c,i}$ refers to the temperature of feed water. Effectiveness-NTU method is used to determine the effectiveness of condensing coil. Effectiveness, ε, is the ratio of actual heat transfer rate of a heat exchanger to the maximum possible heat transfer rate. ε can be expressed as follow:

$$\varepsilon = \frac{actual\ heat\ transfer}{maximum\ possible\ heat\ transfer} = \frac{Q_{cond,evap}}{Q_{\max(cond,evap)}} \tag{11}$$

C_{min} is the heat capacity of the fluid with the lowest value. Two different heat capacities C_h and C_c can be defined for hot or cold circulating fluid respectively such as C_{min} is determined as follows:

$$C_h = \dot{m}_h C_{p,h} \tag{12}$$

$$C_c = \dot{m}_c C_{p,c} \tag{13}$$

If $C_h > C_c$, $C_{min} = C_c$ and $C_{max} = C_h$. However, if $C_c > C_h$, $C_{min} = C_h$ and $C_{max} = C_c$. In heat exchanger equations, it is useful to describe heat capacity ratio, C as:

$$C = \frac{C_{min}}{C_{max}} \tag{14}$$

2.3 Thermostatic Expansion Valve

The thermostatic expansion valve is used to bring the pressure and temperature of the refrigerant from a high pressure and high temperature state to a low pressure and low temperature state. The expansion valve is also used to regulate the flow rate of the refrigerant to control the level of superheat and to ensure complete evaporation of the refrigerant in the evaporator.

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

$$h_{r,i} = h_{r,o} \tag{15}$$

2.4 Evaporator

The heat transferred through the evaporator was determined by calculating the refrigerant enthalpy increase, and energy lost by the water as seen below:

$$Q_{evap} = \dot{m}_w cp_w \left(T_{w,i} - T_{w,o}\right) \tag{16}$$

$$Q_{evap} = \dot{m}_r \left(h_{r,o} - h_{r,i}\right) \tag{17}$$

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. In order to find the heat transfer through the evaporator, the greatest heat transfer must be calculated, which is dependent on the greatest temperature difference between both sides of the heat exchanger, then limited by the minimum capacitance rate, i.e.,

$$q_{max} = \left(\dot{m}\, c_p\right)_{min} \left(T_{h,i} - T_{c,i}\right) \tag{18}$$

2.5 Water-in-glass 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 as shown in **fig.4**.

The heat rate gained by the water is given by:

$$Q = \dot{m} c_{p,w} (T_o - T_i) \tag{19}$$

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

$$\eta = \frac{\dot{m} c_{p,w} (T_o - T_i)}{A_c\, I} \tag{20}$$

2.6 Cooling Tower

Cooling towers are heat exchangers which are used to dissipate large heat load to the atmosphere. They are equipment's used to reduce the temperature of a water stream by extracting heat from water and rejecting it to the atmosphere. They are used in a variety such as power generation and refrigeration. Cooling towers are designed for industrial plants for various purposes and sizes to provide cool water.

3. EES SOFTWARE

Engineering Equation Solver is thermodynamic based software developed by William Beckman and Sanford Klein of the University of Wisconsin and is academically and commercially available from F-Chart Software. EES will solve for enthalpy. The enthalpy at the different states can be used to solve for quantities like heat transfer, power consumption and system performance as shown under the System Equations section.

Parameters of interest for solar assisted air-conditioning system: there were four system inputs that were explored when using EES. **Table (1)** below depicts all the parameters that were bracketed when exploring the possibility of the working fluids. By allowing the values to vary over a large range, the capability of the solar assisted air-conditioning system can be found.

As mentioned, the high temperature of the cycle will be determined by the performance of the solar collectors. The condenser temperature will depend on where the excess heat of the system is going to be rejected to. The program flow chart and the P-h diagram are presented in **fig.4** and **Fig.5** respectively**.**

4. RESULTS AND DISCUSSION

4.1 Effect of Evaporating Temperature on Compressor Work and Refrigerating Effect

The range of evaporating temperatures used was (3℃ to 18℃) all with constant condensing temperatures at (40°C) as shown in **figs.6 and 7**.

Fig.7 shows the relation between evaporation temperature and refrigeration capacity. It is seen that the refrigerating capacity was increased with increasing the evaporating temperature. This is due to the increase of evaporation pressure which will definitely increase the evaporator temperature the enthalpy of the refrigerant at the evaporator outlet is increased. Also the pressure ratio is decreased with increasing the evaporation pressure, which will definitely increase the volumetric efficiency and refrigerant vapour density consequently higher values of refrigerant mass flow rate was achieved.

The refrigeration capacity is increased from 2.7 kW to 4.4kW, as the evaporating temperature increased from 3 to 18℃.

In case of studying the effect of evaporating temperature on the compressor power consumption, it was found that the power consumption increases with increasing the evaporating due to the increase of refrigerant mass flow rate as discussed in previous paragraphs which overcomes the decrease of enthalpy difference with increasing the evaporation pressure the power consumption is increased from 0.89 kW to 1.08 kW as the evaporating temperature increase from 3 to 18℃. Consequently due to the reasons the COP of the system was increased with increasing the evaporating temperature as shown in **fig.6**. The coefficient of performance of the system is increased from 3.068 to 4.117, as the evaporation temperature increased from 3 to 18℃.

4.2 Effect of Condensing Temperature on the Compressor Work and Refrigerating Effect

Condensing temperature (Tcon) is also an important operating parameter that influences the system performance. Evaporating temperature is kept constant at values of (9 ºC) and the condensing temperature was ranged from 35 ºC to 45 ºC.as shown in **figs.8 and 9**.

Fig.9 shows the relation between condensation temperature and refrigeration capacity. It is seen that the refrigerating capacity was decreased with increasing the condensation temperature. This is due to the increase of condensation pressure which will definitely increase the pressure ratio, which will definitely decrease the volumetric efficiency and refrigerant vapour density consequently lower values of refrigerant mass flow rate was achieved.

In case of studying the effect of condensing temperature on the compressor power consumption, it was found that the power consumption increases with increasing the condensing due to the increase of refrigerant mass flow rate as discussed in previous paragraphs which overcomes the decrease of enthalpy difference with increasing the evaporation pressure the power consumption is increased from 0.8966 kW to 1.031 kW as the condensing temperature increased from 35 ºC to 45 ºC as shown in **fig.8**. Consequently due to these reasons the COP of the system was decreased with increasing the condensation temperature as shown in **fig.8**. The coefficient of performance of the system is decreased from 3.89 to 3.1, as the condensation temperature increased from 35 to 45℃.

4.3 Effect of Refrigerant Mass Flow Rate on Compressor Work and Refrigerating Effect

The rate of refrigerant mass flow rate was used in all experiments (83, 75, 66, 58, 50 and 41) kg/hr, the condensing temperature was kept constant at 40℃ and the evaporating temperature was kept constant at 9℃. The results are shown in **fig.10** and **fig.11**.

Fig.11 shows the refrigerating capacity is increased with increasing the refrigerant mass flow rate. This result trend can be explained according to the **eq.16**. Also the pressure ratio is decreased, which will definitely increase the volumetric efficiency, the refrigerant mass flow rate higher values. The refrigeration capacity is increased from 1.6 kW to 3.34kW as the refrigerant mass flow rate increase from 41 to 83 kg/hr.

In case of studying the effect of refrigerant mass flow rate on the compressor power consumption, it is found that the power consumption increases with increasing the refrigerant mass flow rate, the power consumption is increased from 0.48 kW to 0.97 kW as the refrigerant mass flow rate increase from 41 to 83 kg/hr, which is in the better agreement with **eq.4**. The COP is equal theoretically as shown in **fig.10**.

4.4 Effect of the Solar Radiation on the Compressor Work and Thermal Performance

Since the solar radiation intensity plays a major role on the collector heat gain, the effect of this parameter on the system performance is analyzed and presented in this section. The solar radiation intensity was used in all experiments (300, 400, 500, 600, 700, 800, 900 and 1000) W/m^2, the condensing temperature was kept constant at 40℃ and the evaporating temperature was kept constant at 9℃. The results are shown in **fig.12** and **fig.13**.

Fig.12 shows the variation of collector useful heat gain with the radiation intensity. It is seen that as the solar radiation is increased the useful energy gain is increased.

As shown in **fig.13** the power consumption is decreased from 1.05 kW to 0.7 kW as the solar radiation intensity increased from 300 W/m^2 to 1000 W/m^2, while the COP is increased from 3.15 to 4.8.

4.5 Comparison between Theoretical and Experimental Works

Figs.14 to 16 shows comparisons between the experimental and the theoretical results of power consumption and the coefficient of performance of the system. **Fig.14** shows a comparison regarding the influence of the evaporator water temperature on the power consumption and the coefficient of performance of the system. The deviation between the experimental and theoretical power consumption and the performance of the system is about 0.03 and 0.12 respectively. As it is clear from this figure the measured and the predicted values are in a reasonable agreement.

In **Fig.15** the measured power consumption and the performance of the system regarding the variation of with condensing temperature are compared to the predicted results. The deviation between the experimental and theoretical power consumption and the coefficient of performance of the system is about 0.17 and 0.25 respectively.

Fig.16 shows a comparison between the measured and predicted values regarding the influence of refrigerant mass flow rate on power consumption, cooling capacity and COP of the system. The deviation between the experimental and theoretical power consumption, cooling capacity and the performance of the system is about 0.35, 0.07 and 0.43 respectively.

These figures show that there is a reasonable agreement between the measured and predicted power consumption and cooling capacity as well as between the measured and predicted COP.

Generally, this theoretical work indicates that EES software is a good tool to predict the performance of solar assisted air conditioning system.

5. CONCLUSIONS

According to the discussed results, several conclusions can be extracted, these are:

1- Evaporating temperature is an important operating parameter that influences the system performance. An increase in evaporating temperature affects the COP of the system negatively. The refrigeration capacity is increased from 2.7 kW to 4.4 kW as the evaporating temperature increased from 3 to 18℃. The power consumption is increased from 0.89 kW to 1.08 kW as the evaporating temperature increased from 3 to 18 ℃. Consequently the COP of the system was increased from 3.068 to 4.117, as the evaporation temperature increased from 3 to 18℃.

2- The refrigerating capacity was decreased from 3.5 kW to 3.225 kW with increasing the condensation temperature from 35 to 45℃. The power consumption is increased from 0.897 kW to 1.031 kW as the condensing temperature increased from 35 ºC to 45 ºC. Consequently the COP of the system is decreased from 3.89 to 3.1.

3- Increasing the collector heat gain leads to decrease the compressor power consumption from 1.05 kW when the heat gain of 0.255 kW at a solar radiation of 300 W/m^2 to 0.7 kW when the heat gain of 1.9 kW at a solar radiation of 1000 W/m^2, consequently the COP increased from 3.15 to 4.8.

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NOMENCLATURE

A= area, m^2.

C_p= specific heat at constant pressure, kJ/kg ^oC.

C= clearance volumetric ratio, dimensionless.

D= Bore of compressor, m.

F_R= collector heat removal factor, dimensionless.

H= specific enthalpy, kJ/kg.

I= solar radiation, W/m^2.

L= Stroke of compressor, m.

$\dot{m}$= mass flow rate, kg/hr.

N= Rotational speed of compressor, rpm.

n= polytropic index of compressor, dimensionless.
P= pressure, Pa.
Q_u= useful Energy gain/rejected, Watt.
T= temperature, ^oC.
W_{comp}= compressor power consumption, kW.
U = Overall heat transfer coefficient, $W/m^2\ ^oC$.

Greek letter

υ= Specific volume, m^3/kg.
η= Efficiency, dimensionless.
ϵ= Effectiveness, dimensionless.
ρ= Density, kg/m^3.
$\tau\alpha$= Transmitance – absorptance, dimensionless.

Sub-Script

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

Table 1. Governing Factors for Parametric Study.

Parameter to be explored	Minimum value	Maximum value	Governing element
Refrigerant mass flow rate kg/hr	40	80	Air-conditioning loop
Set room temperature ^oC	15	30	Air-conditioning loop
Ambient temperature ^oC	40	50	Air-conditioning loop
Radiation intensity W/m^2	400	1000	Solar collector

Table 2. Temperature distribution for the studying apparatus

Symbol	Meaning
1	Compressor suction
2	Compressor discharge
3	Condenser inlet
4	Condenser outlet
5	Evaporator inlet
6	Evaporator outlet
7	Water outlet of the storage tank
8	Water inlet of the storage tank
9	Water outlet of the condenser
10	Water inlet of the condenser
11	Water outlet of the evaporator
12	Water inlet of the evaporator

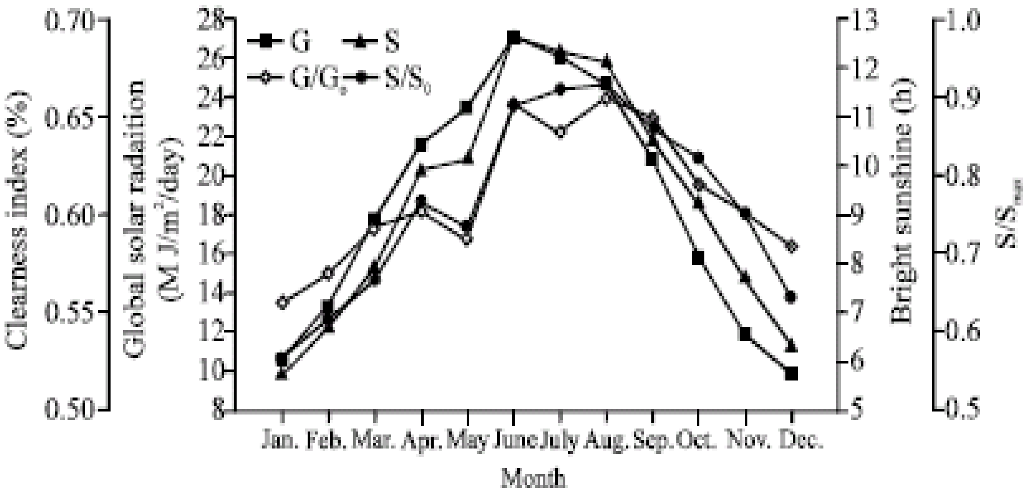


Figure 1. Monthly mean variation of a global solar radiation, bright sunshine, clearness index and S/Smax for Baghdad station

AL-Salihi Ali et al, 2010.

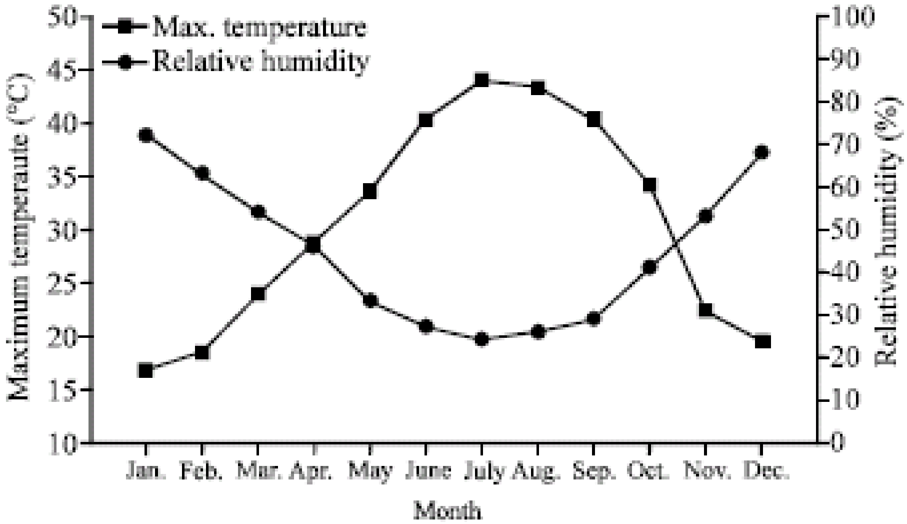
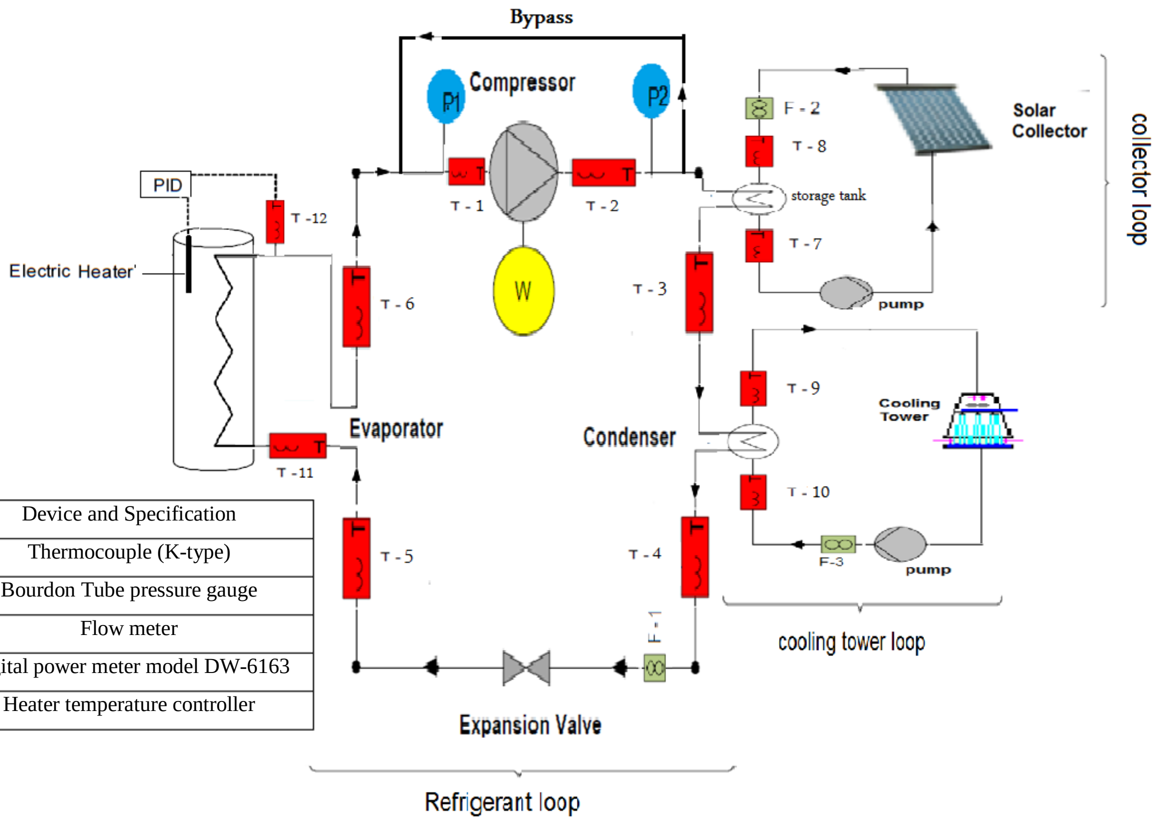


Figure 2. Monthly mean variation of maximum temperature, relative and relative humidity for Baghdad station

Al-Salihi Ali et al, 2010.



Part	Device and Specification
T_1-T_{12}	Thermocouple (K-type)
P_1,P_2	Bourdon Tube pressure gauge
F1-F3	Flow meter
W	Digital power meter model DW-6163
PID	Heater temperature controller

Figure 3. System configuration used in the simulation.

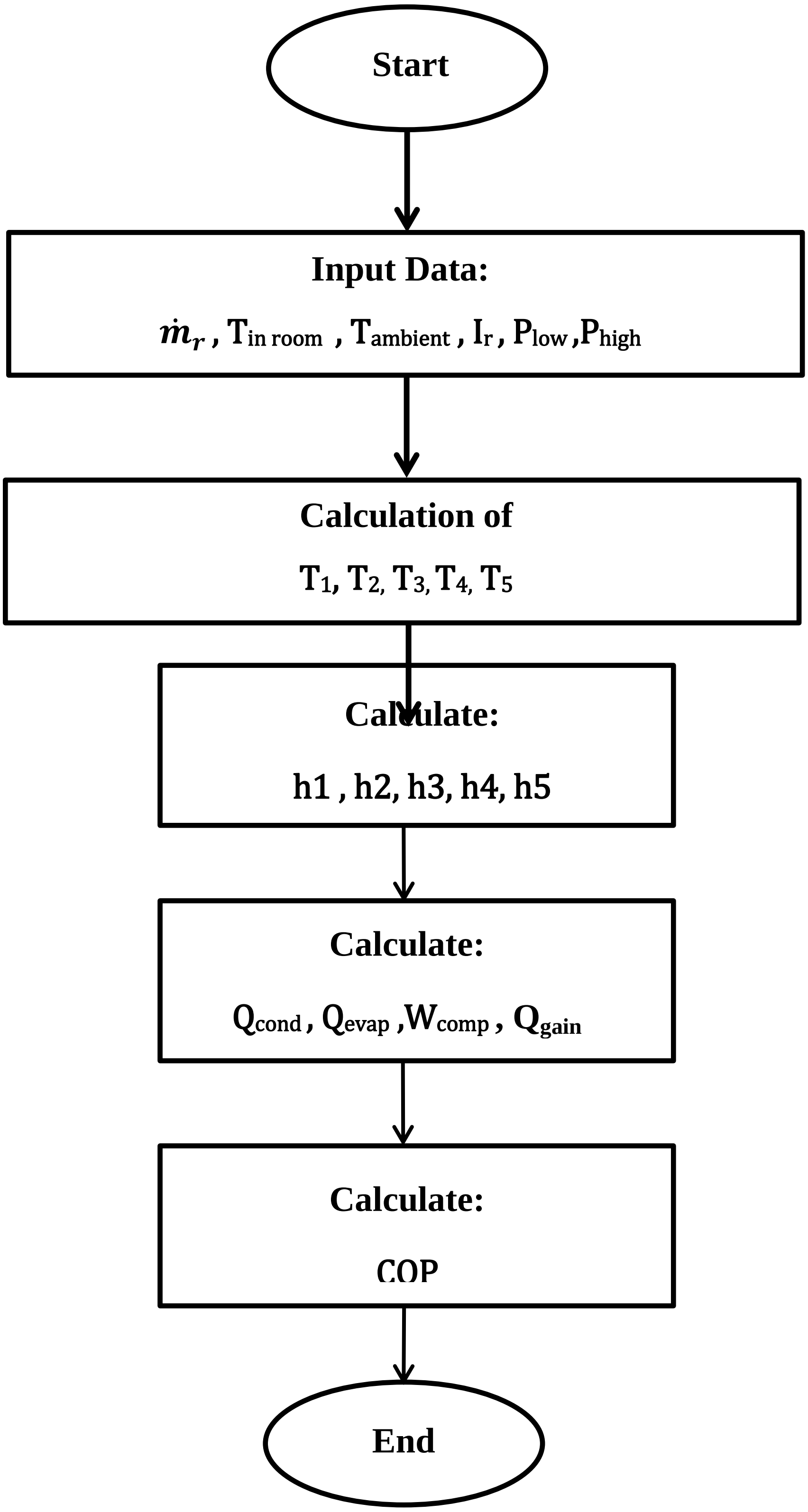


Figure 4. Program Flow Chart.

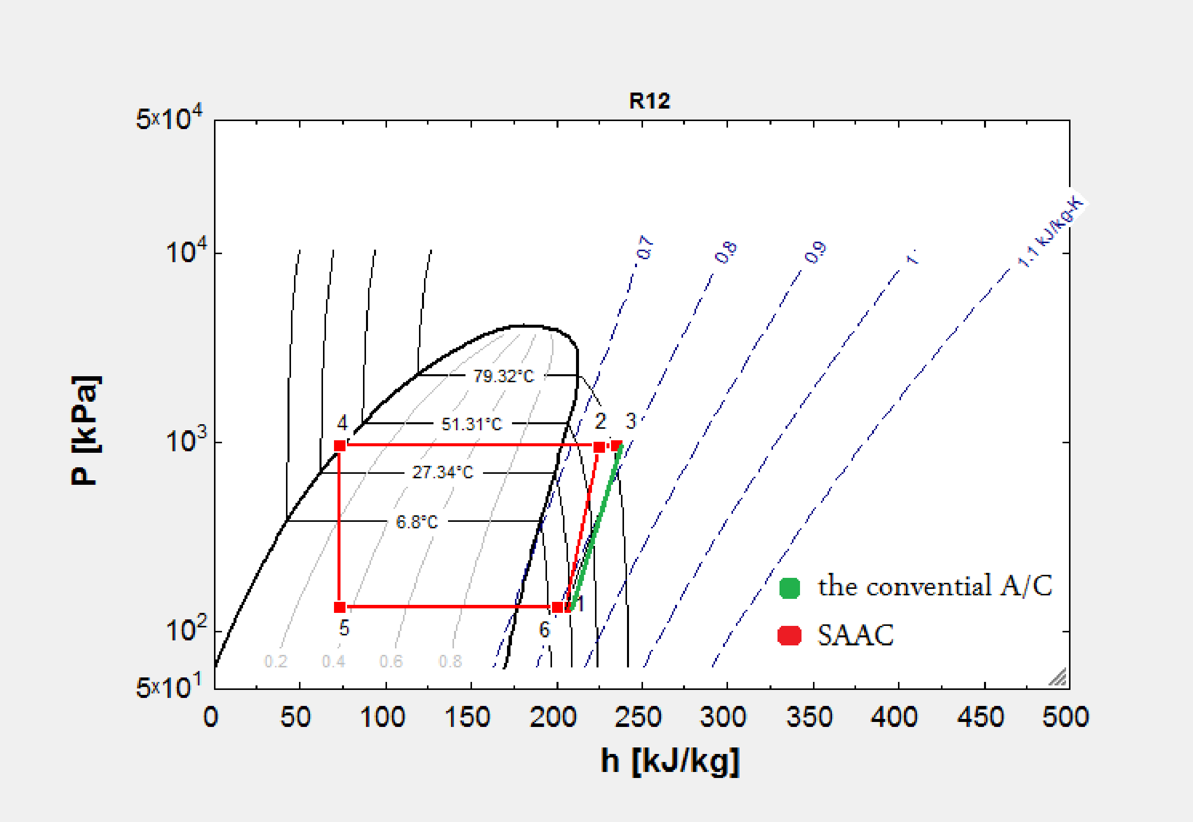


Figure 5. The P-h diagram of Solar Assisted air conditioning system

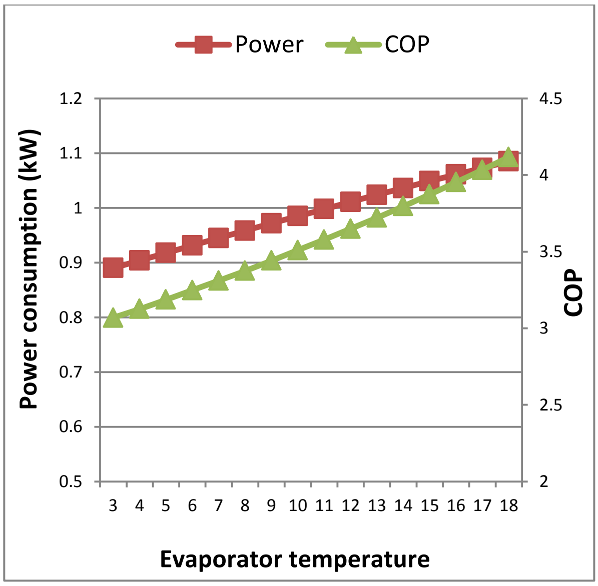


Figure 6. Variation of compressor power and COP with evaporator temperatures.

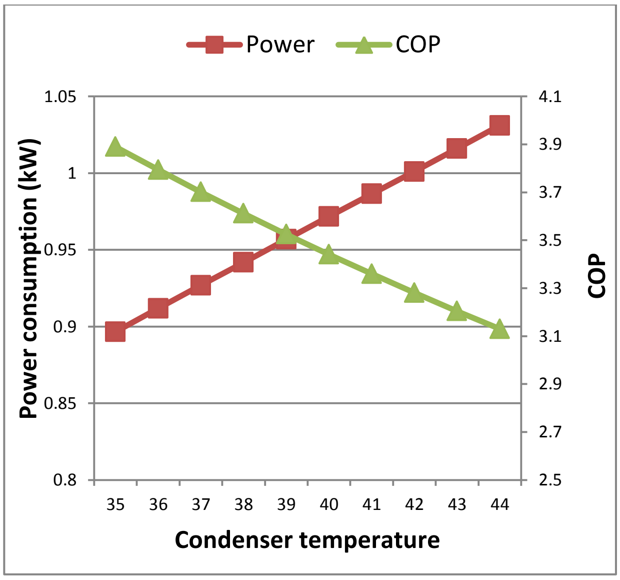


Figure 8. Variation of compressor power and COP with condensing temperatures.

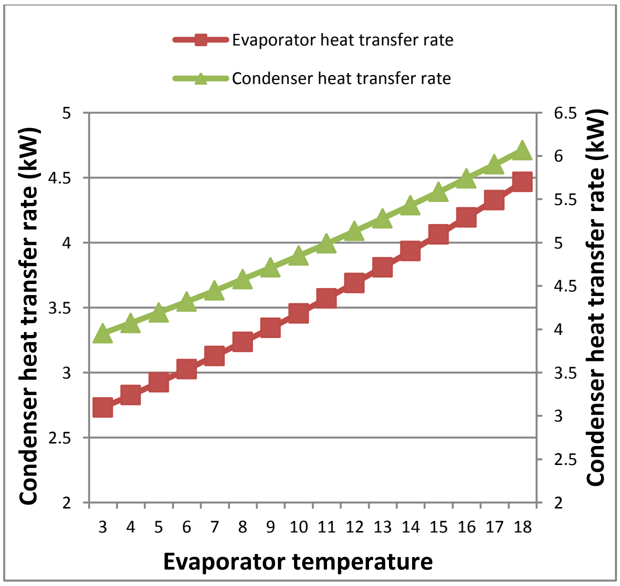


Figure 7. Variation of condenser heat transfer rate and cooling capacity with evaporator temperatures.

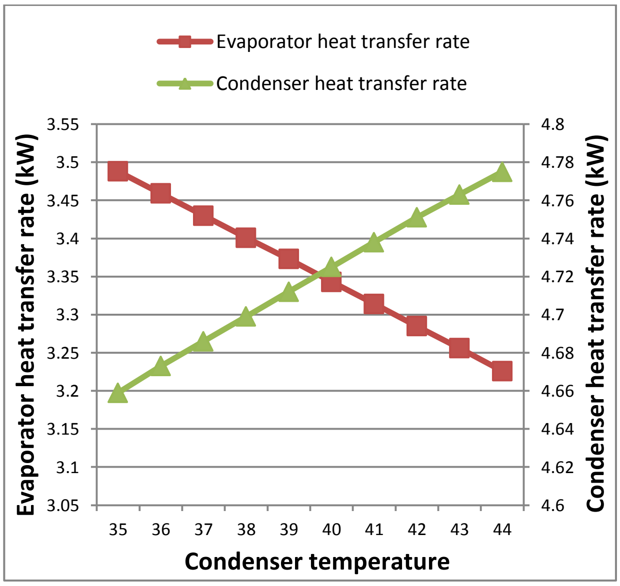


Figure 9. Variation of condenser heat transfer rate and cooling capacity with condensing temperatures.

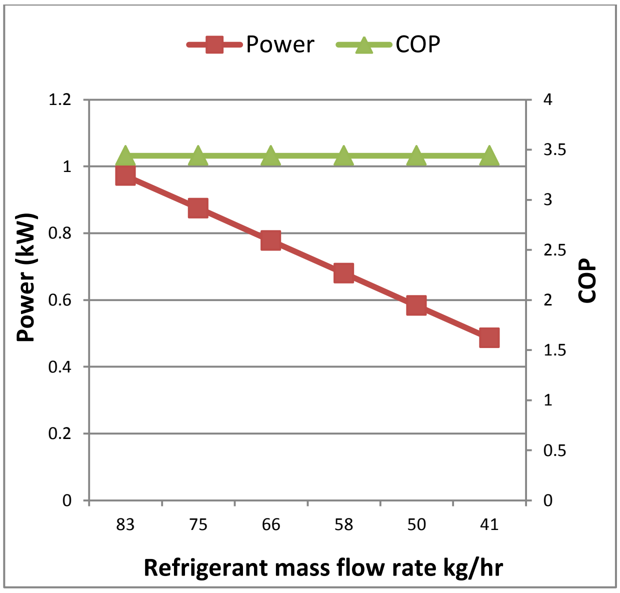


Figure 10.Variation of compressor power and COP with refrigerant mass flow rate.

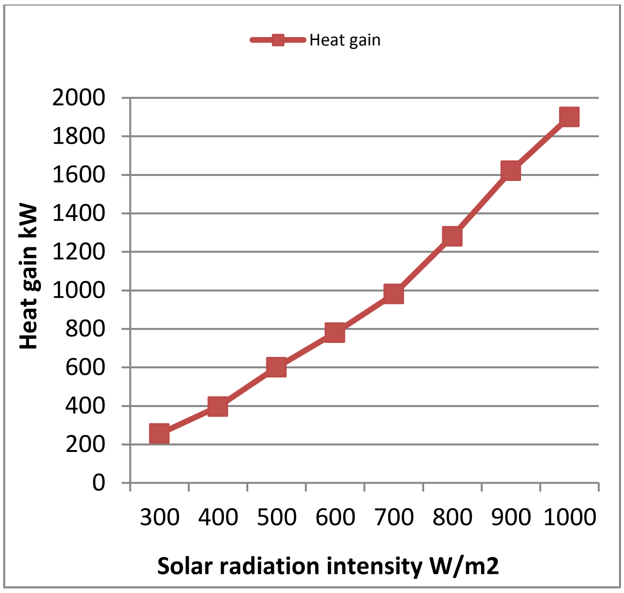


Figure 12. Variation of heat gain with solar radiation intensity.

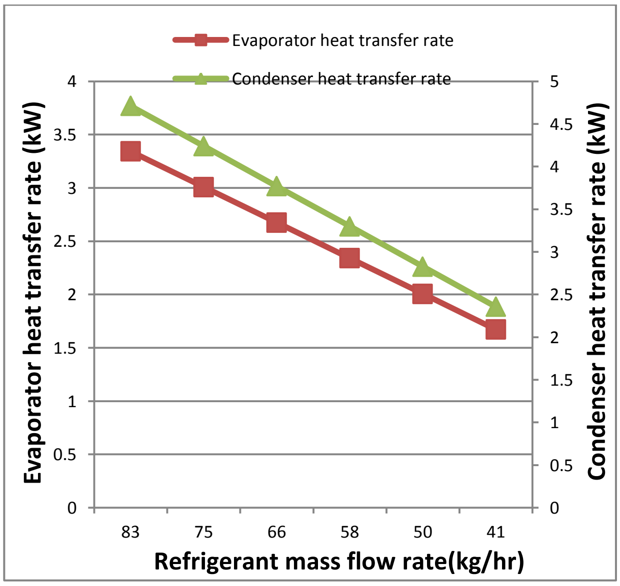


Figure 11. Variation of condenser heat transfer rate and cooling capacity with refrigerant mass flow rate.

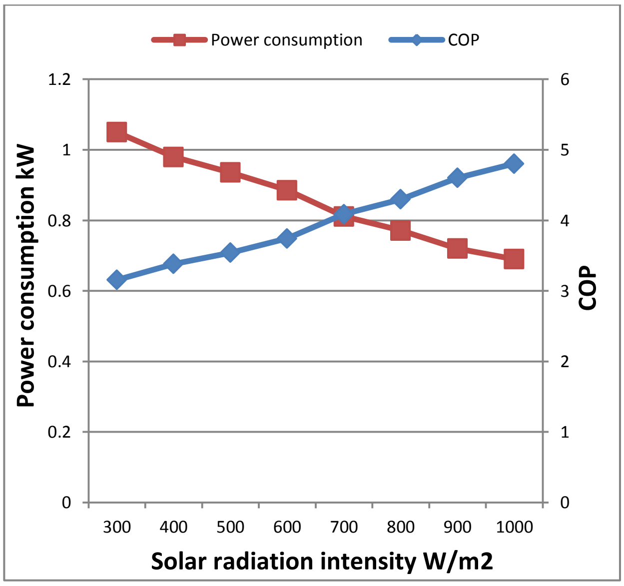
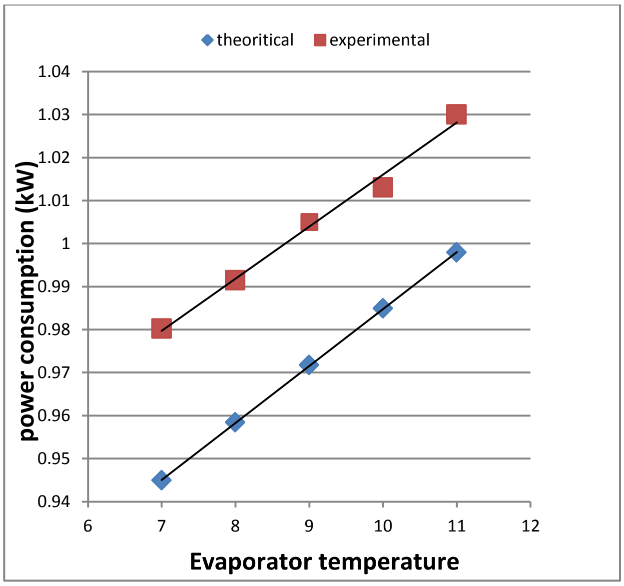
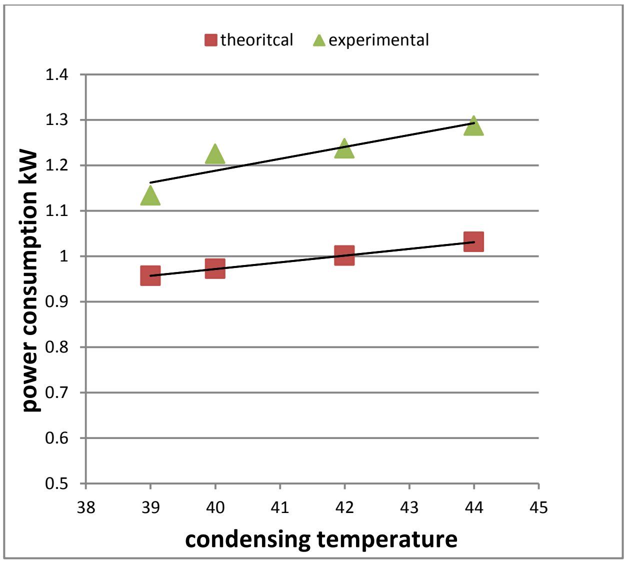


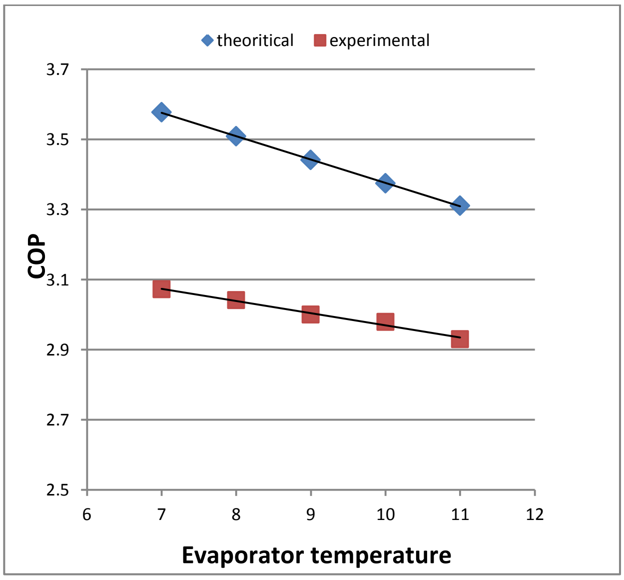
Figure 13. Variation of power consumption and thermal performance of the system with solar radiation intensity.



A. The power consumption

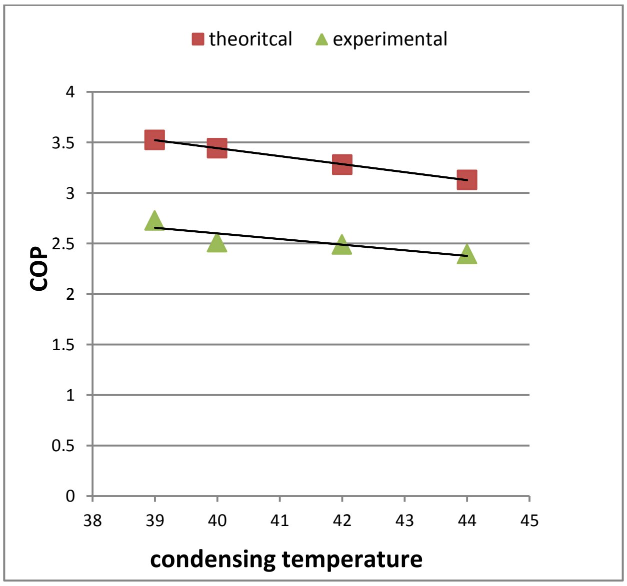


A. The power consumption



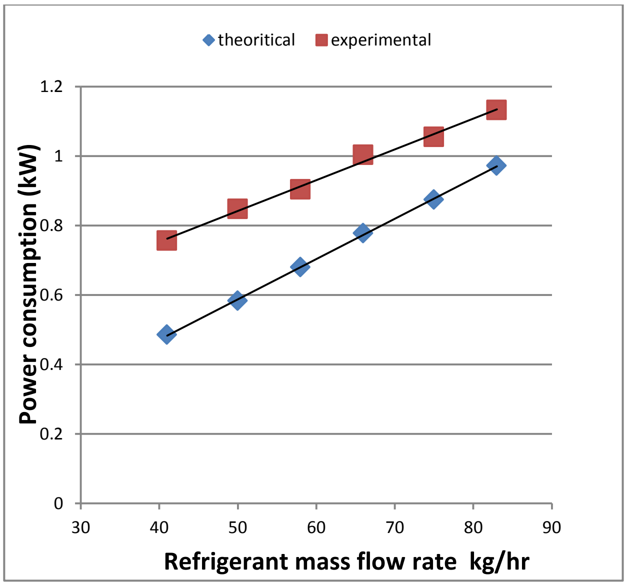
B. COP

Figure 14. Comparison between the experimental and theoretical regarding the influence of evaporator water temperature on: A. The power consumption B. COP

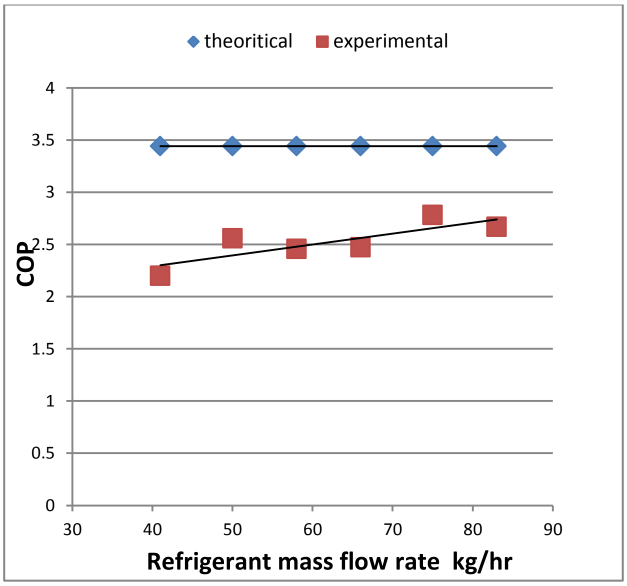


B.COP

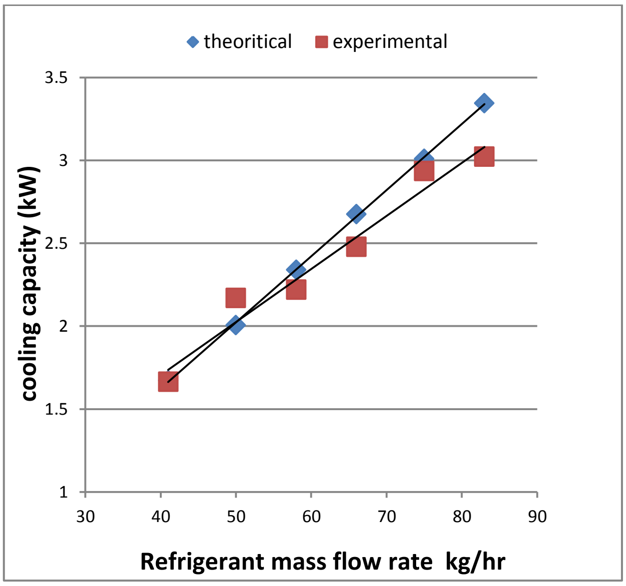
Figure 15. Comparison between the experimental and theoretical regarding the influence of condensing temperature on: A. the power consumption B. COP



A. Power consumption



B. COP



C. Cooling capacity

Figure 16. Comparison between the experimental and theoretical regarding the influence of refrigerant mass flow rate on: A. the power consumption B. COP c. the cooling capacity