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Theoretical Performance Study of an Indirect- Expansion Solar Assisted Heat Pump

Asst. Prof. Dr. Najim Abid Jassim& Kadhim Kareem Al-Chlaihawi**

* Department of Mechanical Engineering, university of Baghdad,

** Department of Mechanical, University of Al-Qadisiya

Abstract

This work presents a theoretical performance study of an indirect- expansion solar assisted heat pump SAHP. The theoretical analysis included a simulation of the IX-SAHP using TRNSYS software. These tests were conducted by varying the controlling parameters to investigate their effects on the thermal performance of the IX-SAHP such as the evaporating and condensing temperatures, compressor speed, refrigerant type and the solar collector area. A significant effect of the compressor speed on the thermal performance of the system is indicated. An increase of compressor speed causes a decrease of coefficient of performance COP. The performance of refrigeration system largely depends upon the characteristics of the refrigerants. COP was affected by the solar collector area and type where the COP increases with increasing collector area, but for a given heating load there is a required minimum solar collector's area, also evacuated tube solar collector lead to higher COP than flat plate collector for a given collector area. Compared to the conventional heat pump system, the present indirect expansion solar assisted heat pump system has the advantage that the collector can absorb the radiation and ambient energy and lead to increase the superheat level at the compressor inlet, so that reduced the work input of compressor in comparison to the conventional heat pump system.

Keywords: indirect expansion heat pump, solar assisted heat pump, experiments.

Introduction

The threat of global warming has become a very important international issue. With today's world still relying heavily on burning fossil fuels as a primary energy source, an excessive amounts of greenhouse gases to the atmosphere were adding [West, 2010].[14]

Although greenhouse gases naturally exist in the environment, it is believed by most climate scientists that the excess amount generated from burning fossil fuels is trapping more heat than the atmosphere normally would trap, and this has caused steady "increases in global average air and ocean temperatures" [IPCC, 2007]. [7] If these adverse climate trends are not reduced, then in the long term it would "be likely to exceed the capacity of natural, managed and human systems to adapt" [IPCC, 2007]. [7] Therefore, actions must be taken to reduce the greenhouse gas emissions from burning fossil fuels. This can be achieved by using less energy and by using renewable energy sources, such as solar and wind, to assist in meeting energy demands.

The possibility of running thermal system using the energy from the sun has been receiving considerable attention in recent years. Solar energy is clean and almost inexhaustible of all known energy sources. The low temperature thermal requirement of a heat pump makes it an excellent match for the use of solar energy. A combination of solar energy and heat pump system can bring about various thermal applications for domestic and industrial use, such as water heating, solar drying, space cooling, space heating and refrigeration. Unlike thermosyphon solar water heaters, solar heat pump systems offer opportunity to upgrade low-grade energy resources from the surroundings as well as solar energy and make use of it for domestic and industrial applications [Hawlader et al., 2001]. [6]

There are two main types of heat pump assisted solar systems that have been studied in the past. These are Direct Expansion Solar Assisted Heat Pumps [DX-SAHP] and Indirect-style Solar Assisted Heat Pumps [IX-SAHP].

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 (Bridgeman and Harrison, 2008) [1], (Chandrashekar et al., 1982) [2], (Kuang et al., 2003) [9]. 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 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.

[Freeman et al., 1978] [5] Proposed an indirect SAHP system to arrive at a practical heat pump system for use in the Canadian environment. The system used a similar collector loop as atypical SDHW system, with an unglazed collector instead of an insulated single or double-glazed collector. Rather than transferring the heat to the tank directly as in a SDHW system, it transferred its heat to the heat pump loop via a compact heat exchanger which also acted as the evaporator of the heat pump. The refrigerant was then compressed in a hermetically sealed reciprocating compressor, and condensed in another compact heat exchanger connected externally to the tank. Water from the tank flowed through the condenser drawn from the bottom of the tank via natural convection.

Freeman conducted simulations of the indirect solar assisted heat pump using (TRNSYS) a transient simulation program.

The analysis used mostly component models supplied with the TRNSYS software, but also included models for the heat pump, the natural convection heat exchanger, and the heat pump controller created by Freeman.

The simulation was performed for the solar assisted heat pump under typical domestic hot water loads, for five Canadian cities; Toronto, Vancouver, Montreal, Halifax and Winnipeg. The results were compared to simulations of electric water heater systems, Solar Domestic Hot Water systems (SDHW), and air-to-water heat pump systems. Freeman's results concluded that the solar assisted heat pump gathered more energy from the environment during marginal weather conditions as well as during the winter when compared to SDHW and air-to-water heat pump systems. Other findings based on his simulations were that for the cities of Toronto, Montreal, and Winnipeg, the SAHP and SDHW systems had similar life cycle costs, and in

Halifax and Vancouver the SAHP system demonstrated life-cycle savings between 12% and 29%.

[Bridgeman and Harrison, 2008] [1] Produced a prototype of an i-HPASDHW system based on the system modeled in TRNSYS by [Freeman, 1997] [4] to validate the simulation findings.

The antifreeze collector loop is connected to the domestic hot water tank via a heat pump. This system is very similar to a standard closed loop SDHW system except that a heat pump is used rather than an external heat exchanger to transfer the collected energy to the water in the storage tank. Natural convection is used in this system to circulate the water from the domestic tank through the condenser heat exchanger of the heat pump.

To construct the prototype, [Bridgeman and Harrison, 2008] [1] used an auxiliary heater to simulate the solar collector in order to perform controlled experiments. The auxiliary heater was able to supply constant temperature outputs, as well as produce typical solar collector output temperatures to the evaporator of the heat pump in order to simulate a day of operation. The other main components used to construct the system included: a 1/3 horsepower single speed compressor, a thermostatic expansion valve, two flat plate counter flow heat exchangers for the evaporator and condenser of the heat pump, a standard hot water tank (270 L), and a variable speed pump. The collector loop used a 50%-50% glycol/water solution by volume as the working fluid, and the heat pump loop used R-134a.

Constant source temperature tests were performed by them in order to compare to the simulation results obtained from the TRNSYS model by [Freeman, 1997] [4]. It was determined 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.

[Sterling and Collins, 2011] [11] modeled Indirect-style Solar Assisted Heat Pump (i-SAHP) using the TRNSYS software and compared to a traditional solar domestic hot water (SDHW) system and an electric domestic hot water (DHW) system. All of the systems had the same load profile and delivered domestic hot water at a constant temperature. This insured that each system delivered the same amount of energy for the entire modeling period thereby creating a common basis for comparison. It was found that the dual tank i-SAHP system proved to be the most energy efficient and had the lowest annual operating cost of the three models analyzed.

In the current study the thermal performance of indirect expansion solar assisted heat pump, IX-SAHP, was investigated theoretically. The theoretical analysis included a simulation of the IX-SAHP using TRNSYS software. These tests were conducted by varying the controlling parameters to investigate their effects on the thermal performance of the IX-SAHP such as the evaporating and condensing temperatures, compressor speed, refrigerant type and the solar collector area.

2. System Descriptions

A schematic diagram of the system is shown in Figure(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. A centrifugal pump 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 collector loop was a heat exchanger, which is also common to the heat pump loop, which was used to transfer energy from the water to the refrigerant.

The heat pump loop was connected in series between the charge loop and the cooling tower loop. Heat was transferred between the loops with the use of two heat exchangers. These heat exchangers acted as the evaporator and condenser of the heat pump cycle. The other two fundamental components of the heat pump loop were a compressor, and a thermostatic expansion valve which throttled the flow and maintained a constant superheat temperature exiting the evaporator.

Cooling Tower loop consist of a cross- flow water type cooling tower, heat exchanger (condenser) which was discussed in the previous section and a circulating water pump. Cooling tower used to cool water that used to condense the refrigerant.

The system considered was modeled and simulated using TRNSYS software.

It is a very easy-to-use program that allows the user to simply add existing components from the library into a project and connect them to the other components that they interact with in order to form the desired system. Each component is based on a mathematic model that reads in various parameters specified by the user and inputs from other components in the system with which it interacts.

2.1 TRNSYS Components

The IX-SAHP system considered used the Type 668 water-to-water heat pump model in the heating mode. The component's performance was based on an external file specified by the user that contained catalog data for the capacity and power consumption of the heat pump at various combinations of inlet load and inlet source temperatures. Therefore, Type 668 simply read in the inlet load and inlet source temperatures entering the heat pump and used the external file to determine the capacity and power consumption for that situation. Linear interpolation was used by the heat pump model to determine these outputs at specific inlet conditions. Knowing the capacity of the heat pump, the mass flow rates, and the input temperatures, Type 668 calculated the outlet load and outlet source temperatures of the heat pump for each time step. The user specified how many inlet load and source temperatures were used to generate the performance table for the external file in the parameters menu. The load and source specific heats were also entered by the user here.

An evacuated tube solar collector using a quadratic efficiency curve and a biaxial Incidence Angle Modifiers (IAM) was used in the system. The collector was south-facing and had a tilt angle of 45.

Two Type 3b pumps were used to circulate water in the collector and the cooling tower. The units presented here were kept in the same form that they were used in the TRNSYS model for clarity.

System controls had to be implemented to ensure that the pumps were running only when there was sufficient energy to be collected. The Type2b differential controller for temperature was used in TRNSYS to control the solar loop of the system. It monitored the temperature of the cold water at source side for the heat pump and the outlet temperature of the solar collector to determine if there was energy to be collected. If the temperature of the water at the collector outlet was 5 or more above the temperature of the water at the source side of the heat pump, then the pump was turned on to collect this solar energy. The system would continue to run until this temperature difference fell below 2.

Type 51 cooling tower was used to cool the water that used to condense the refrigerant.

Type109-TMY2 weather file with data for Baghdad, Iraq was used for the solar assisted system considered. The data reader converted the weather information to the desired system of units and determined the direct and diffuse radiation outputs at each time step experienced by the solar collector at the specific orientation and incline. Therefore, many calculations were performed by this component throughout the simulated time.

A variety of other components were used in the TRNSYS model. One of which was a pipe model which accounted for losses through the pipe length to the environment. The pipe model was used between the water pump and the collector, and between the collector and the heat pump.

2.2 Full TRNSYS Model

The full TRNSYS model was built in TRNSYS 16, which uses a graphical user interface. Each component is placed on the workspace, and connected to interact components using the connector tool. Once each component was placed and connected in the workspace, the inputs and outputs of each component were set by connecting corresponding outputs from one component, to the input of another component. An outline of the TRNSYS model is shown in **Fig.2**, along with the inputs and outputs from each component.

3. MATHEMATICAL MODEL

A mathematical model with which the system and process parameters can be studied is of considerable importance due to the fact that a reliable mathematical model will prevent or minimize costly mistakes in the further development of the system.

3.1. Compressor

An assumption of the refrigerant undergoing polytrophic compression with a constant theoretical polytrophic index, n, is made in the formulation of the compressor model. The various equations are expressed as follows from [Stoecker and Jones, 1982] [12].

Piston Displacement per cylinder () is calculated as:

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

Volumetric efficiency of compressor is calculated as:

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

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

Refrigerant mass flow rate ($\dot{m}_r$) 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:

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

3.2. Condenser and Evaporator

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 in condenser 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.

Heat exchangers were evaluated in terms of the effectiveness – NTU method, with the effectiveness of the heat exchangers being (Incropera, F.P., and DeWitt, D.P., 2002) [8]:

$$\varepsilon = \frac{actual\ heat\ transfer}{maximum\ possible\ heat\ transfer}$$

$$= \frac{Q_{cond,evap}}{Q_{\max(cond,evap)}} \qquad (9)$$

C_{min} Is the heat capacity of the fluid with the lowest value. The system only has two working fluids, either the refrigerant or water. Hence, the two different heat capacities are C_h and C_c, respectively. C_{min} Can be determined as follows:

$$C_h = \dot{m}_h C_{p,h} \qquad (10)$$

$$C_c = \dot{m}_c C_{p,c} \qquad (11)$$

If $C_h > C_c$ then $C_{min} = C_c$ and $C_{max} = C_h$. However, if $C_c > C_h$, $C_{min} = C_h$ and $C_{max} = C_c$.

The heat rejection and addition through the condenser and evaporator take place primarily in the form of latent heat transfer. During boiling and condensation the fluid temperature remains constant and the refrigerant acts as though it has infinite specific heat, forcing the capacitance ratio, $\dot{m}\ C_{p,min}/\dot{m}\ C_{p,max}$, to zero.

This simplifies the effectiveness equation to:

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

The heat transferred through the evaporator is governed by the enthalpy increase of the refrigerant, and energy lost by the water where:

$$Q_{evap} = \dot{m}_w c p_w \left(T_{w,i} - T_{w,o}\right) \qquad (13)$$

$$Q_{evap} = \dot{m}_r \left(h_{r,o} - h_{r,i}\right) \qquad (14)$$

Where $T_{w,i}$ and $T_{w,o}$ correspond to the inlet and outlet water temperatures of the evaporator. In order to determine the heat transferred through the evaporator, the maximum heat transfer must be first calculated, which is dependent on the maximum temperature difference between the two sides of the heat exchanger, and limited by the minimum capacitance rate, ie:

$$q_{max} = (\dot{m}\ c_p)_{min}(T_{w,i} - T_{r,i}) \qquad (15)$$

Then the heat transfer can be calculated by combining Eqs. 9, 12 and 15:

$$q_{evap} = \left[1 - e^{-\frac{UA}{c_{pmin}}}\right](\dot{m}c_p)_{min}(T_{w,i} - T_{r,i}) \qquad (16)$$

3.3. 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:

$$h_{r,i} = h_{r,o} \qquad (17)$$

3.4 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:

$$Q = \dot{m}c_{p,w}(T_o - T_i) \qquad (18)$$

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:

$$\eta = \frac{\dot{m}c_{p,w}(T_o - T_i)}{A_c\ G} \qquad (19)$$

4. Results and Discussion

4.1. Effect of Refrigerant Type on the Performance.

Refrigerants are essential working substances used in refrigeration systems. The performance of refrigeration system largely depends upon the characteristics of the refrigerants. To analyze and characterize the extent of degradation of the thermal performance due to the use of several refrigerants and to identify the refrigerants that are best suited for solar heat pump applications, three working fluids,R-22(Chlorodifluoromethane), R-404A (R125/143A/134A) and R-134a(1,1,1,2 Tetrafluoroethane) for heat pump have been considered as working fluid in the SAHP. Table1 shows physical properties of the working fluids [NIST, 2000] [10].

Figures (3&4) show cooling capacity and power consumption for the three refrigerants at evaporating temperate ranges from 0 ºC to 12.5 ºC and condensing temperature constant at 30 ºC .It is seen from these figures that as the evaporator temperature increases from 0 ºC to 12.5 ºC, the value of compressor work decreases and refrigerating effect increases for all refrigerants. R404A has highest value of compressor work. R134a has minimum value of compressor work. R404A also has highest value of refrigerating effect at evaporator temperature equal to 0 ºC.

Figures (5-87) shows the variation of the condenser and evaporator heat transfer rates and coefficient of thermal performance for the several refrigerants mentioned earlier. This figure shows that R-22 yields the highest value of COP, followed by R-134a and R-404A.

4.2 Effect of Compressor Speed on the Power Consumption and the Thermal Performance.

The effect of compressor speed on the overall system performance is assessed by varying the speed of the compressor. Figures (8, 9) show the effect of compressor speed ranged from 1450 to 1750 rpm on the power consumption and the cooling capacity as the evaporating temperature increasing from -20 ºC to 5 ºC. It could be noted from these figures that, as the compressor's speed increases, the power consumption and the cooling capacity also increase but the increase in power consumption are much greater than the increase in the cooling capacity.

The power consumption is increased by 9.1% as the compressor speed increase from 1450 to 1600 rpm, while the cooling capacity is increased by 8.3%. Further increasing in compressor speed to 1750 rpm, the power consumption is increased by 16.8% and the cooling capacity is increased by 15.3%.

Figure (10) presents the effect of variation of compressor speed on the system coefficient of performance. It could be noted that, as the compressor's speed increases, the COP value reduces. It is due to the fact that the discharge temperature increases along with an increase in speed of the compressor. At lower compressor speed (1450 rpm), the IX-SAHP system could reach a COP value of 3.3, increasing compressor speed from 1450 to1600 rpm results in a reduction of coefficient of performance to 2.93 and to 2.59 at 1750 rpm, although the heat extraction rate is much higher. This is due to the fact that, increase in the compressor speed is aided only with the work input which reflects in lower COP values.

4.3 Effect of Solar Collector Area on the Thermal Performance

Figure (11) shows the variation of collector useful heat gain with area of the collector. It is seen that as the area of the collector increase the useful energy gain is increased.

Figure(12) shows the variation of COP of the system with increasing in the collector area, this figure shows that the COP enhancement of the system with increasing in the collector area from $4m^2$ to $8m^2$ is much greater than that of collector area increase from $8m^2$ to $12\ m^2$ and from $12m^2$ to $16m^2$, where the increasing in the COP equal to 15.6% as the collector area increased from $4\ m^2$ to $8\ m^2$, 8.4% as the collector area increased from $8\ m^2$ to $12\ m^2$ and 4.77% as the collector area increased from $12\ m^2$ to $16\ m^2$. It is due to the fact that the difference between the useful energy gains of the collectors with area of $12m^2$ and $16m^2$ is less than that of $4m^2$ and $8m^2$.

It can be conclude that this solar system performance is not in proportion to the collector area, collected heat is not proportional to the increase in the solar collector's area. For a given heating load there is a required minimum solar collector's area.

4.4 Effect of Solar Collector Type on the Thermal Performance

Different types of collectors can be found in TRNSYS software, to analyze the effect of solar collector type on the thermal performance of the system , two types of solar collectors used ,they are flat plate and evacuated tube solar collector which are the most commonly used .

Figure (13) shows that the useful energy gain of evacuated tube collector is greater than that of flat plate collector, as a result of this the evaporator and condenser heat transfer rates will be increased and thus the COP of the system increasing, where the COP is increased by 15.3% when evacuated tube collector is used rather than flat plate collector, as seen in Figure (14).

4.5 Comparison with Conventional Heat Pump System

A conventional vapor compression water source heat pump system throws the heat from a heat source (tap water) to the load without making an effort to recover it. Furthermore, the performance of a conventional heat pump system is greatly limited by the heat source.

Compared to the conventional heat pump system, the present indirect expansion solar assisted heat pump system has the advantage that the collector can absorb the radiation and ambient energy and lead to increase the superheat level at the compressor inlet, so that reduced the work input of compressor in comparison to the conventional heat pump system.

The results are tabulated in Table (2). As shown in this table the compressor power consumption of SAHP system is reduced by 5.3% as compared with conventional heat pump, the evaporator and condenser heat transfer rates are increased by 13.4% and 11.9% respectively. The COP is increased by 16%.

4.6 Comparison between Theoretical and Experimental Result.

Figure (15) shows a comparison between the measured & the predicted COP of the system.

The experimental result has been conducted under the meteorological conditions of Baghdad, and for constant water flow rate in the condenser and evaporator at 4 l/min.

This figure show that there is a good agreement between the measured & the predicted COP.

5. Conclusions

Effective parameters on the performance of the system was investigated which included compressor speed, refrigerant type, solar collector area and solar collector type.

It is pertinent to choose an appropriate compressor speed, it could be noted that, as the compressor's speed increases, the COP value reduces.

The performance of refrigeration system largely depends upon the characteristics of the refrigerants.

COP was affected by the solar collector area and the type where the COP increases with increasing collector area, but for a given heating load there is a required minimum solar collector's area, also evacuated tube solar collector lead to higher COP than flat plate collector for a given collector area.

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Nomenclature

A	area	(m^2)
C_p	specific heat at constant pressure	(kJ/kg °C)
T	temperature	(°C)
W_{comp}	the electrical power consumption of the compressor	(kW)
C	clearance volumetric ratio	(------)
h	specific enthalpy	(kJ/kg)
I	solar radiation	(W/m^2)
$\dot{m}$	mass flow rate	(kg/hr)
p	pressure	(Pa)
n	polytropic index of compressor	(----)
Q_u	useful Energy gain/rejected	(Watt)

Greek letter

υ	Specific volume	(m^3/kg)
η	Efficiency	(-------)
ϵ	Effectiveness	(-------)
ρ	Density	(kg/m^3)

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
i-HPASDHW	Indirect - Heat Pump Assisted Solar Domestic Water Heating
NTU	Number of transfer units
ETC	Evacuated tube collector

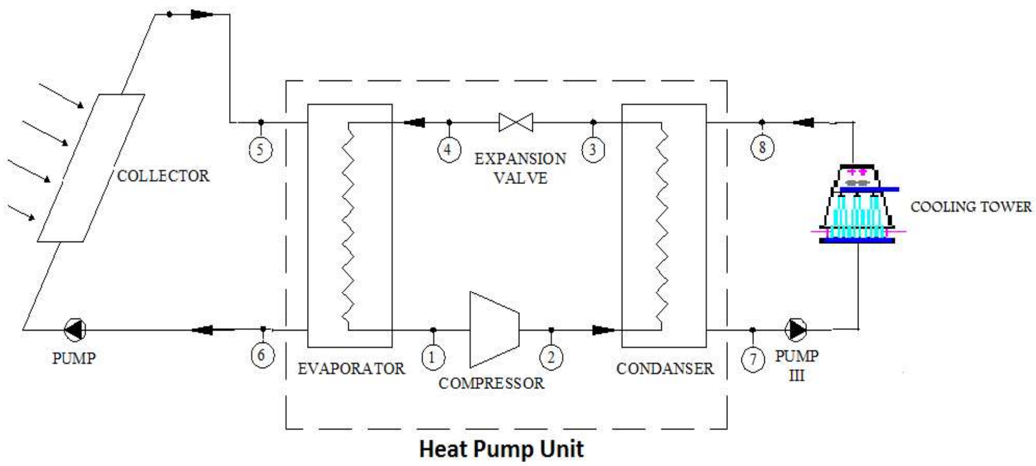
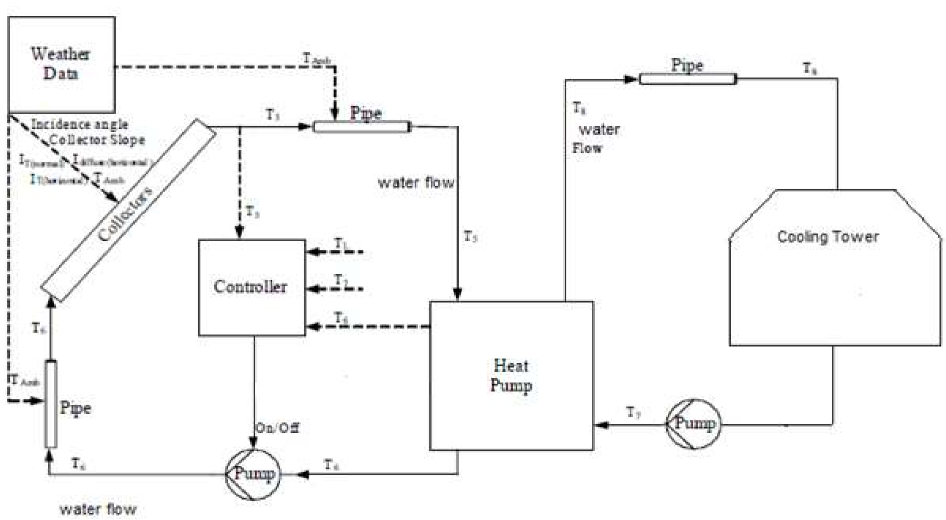


Fig. (1) Schematic of IX-SAHP



Fig(2) Outline of TRNSYS model showing component interaction.

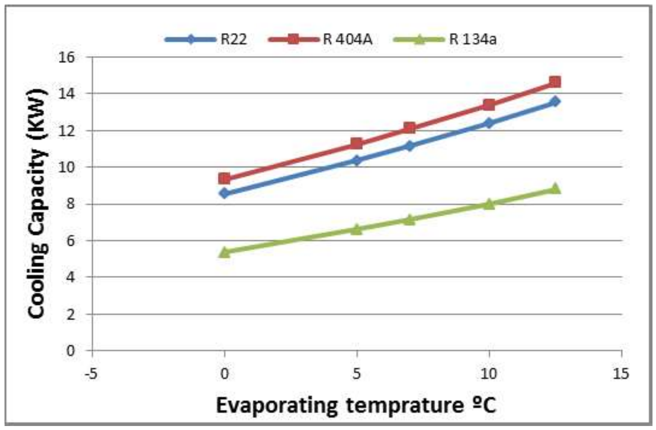


Fig.(3) Variation of cooling capacity with evaporating temperature for different refrigerants.

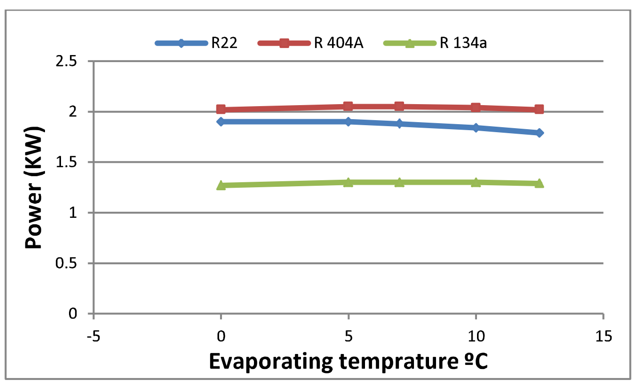


Fig.(4) Variation of compressor power with evaporating temperatures for different refrigerants.

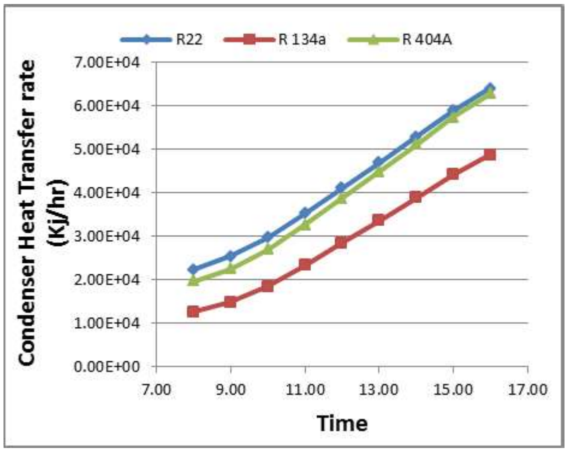


Fig. (3)Variation of condenser heat transfer rate with time for different refrigerants

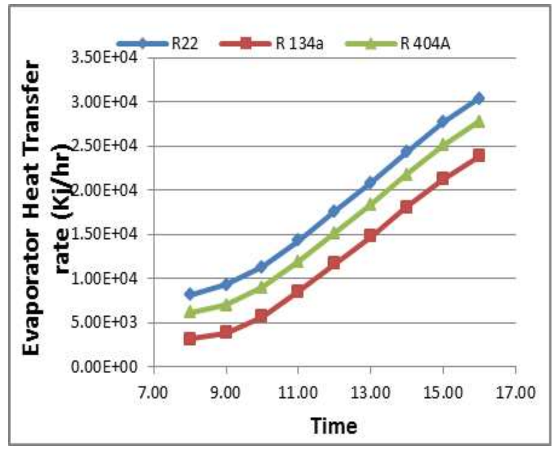


Fig. (6) Variation of evaporator heat transfer rate with time for different refrigerants

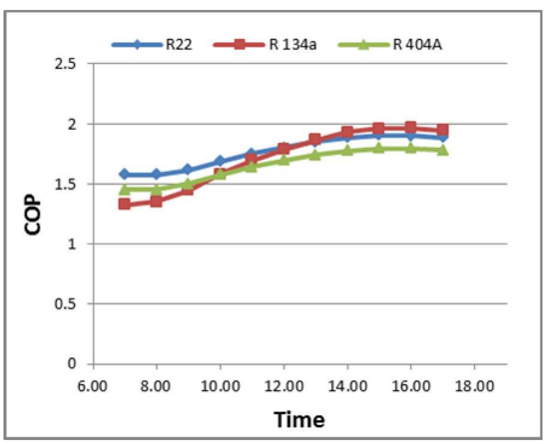


Fig. (7)Variation of COP with time for different refrigerants

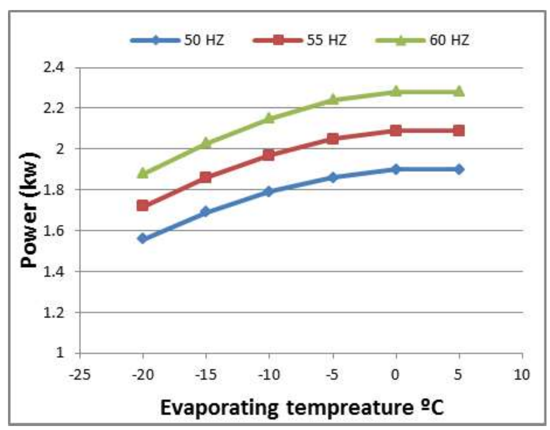
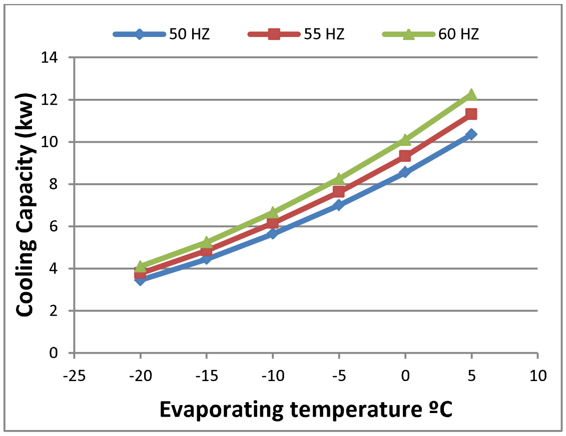


Fig. (8)Variation of compressor power with evaporating temperature for different compressor speed



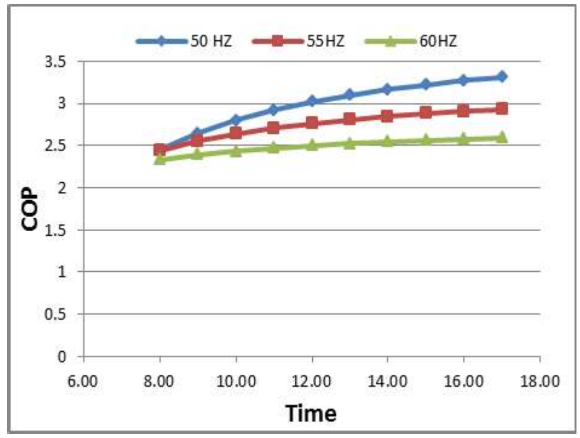
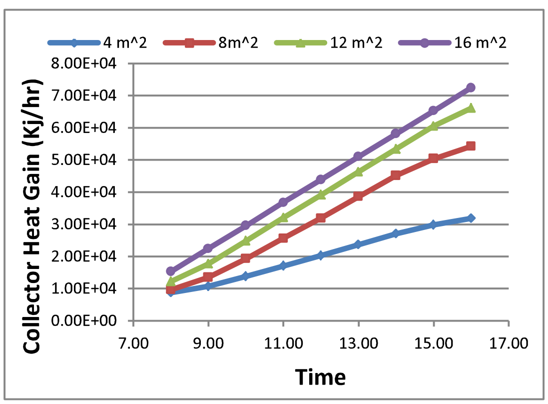


Fig. (9)Variation of cooling capacity with evaporating temperature for different compressor speed

Fig. (10) Effect of compressor speed on the COP.



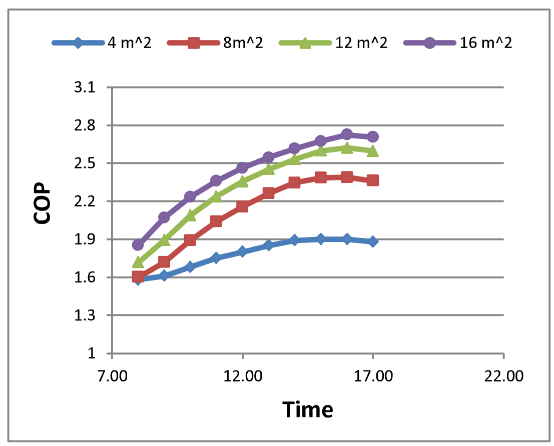


Fig. (11)Variation of collector useful energy gain with time for different collector area.

Fig. (12)Variation of COP with time at different collector area.

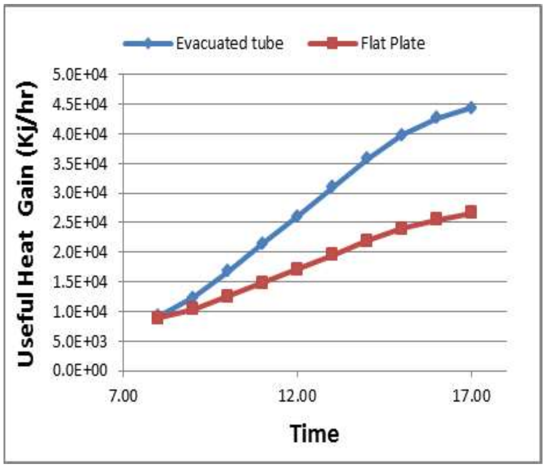


Fig. (13) Effect of collector type on the useful energy gain.

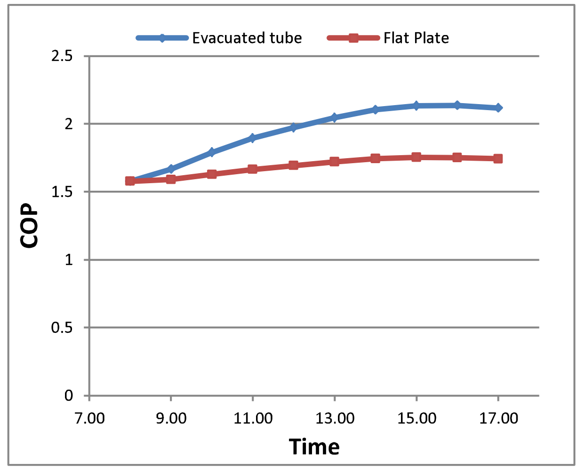


Fig. (14) Effect of collector type on the COP.

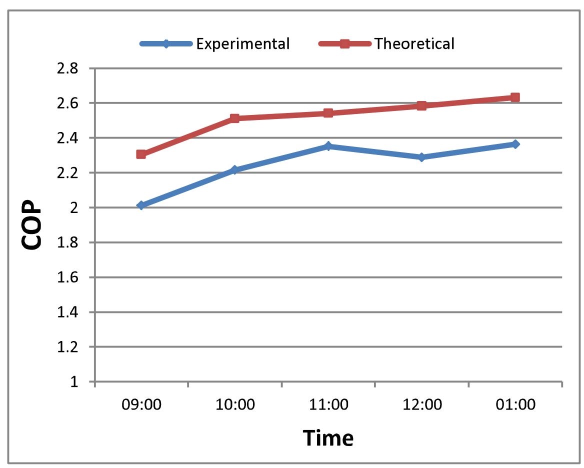


Fig. (15)Comparison between the measured and the predicted COP.

Table (1): Physical properties of the working fluids (NIST, 2000).

Working Fluid	R-22	R-134a	R-404A
Chemical formulae	CHC lF2	CF3CH2F	R125/143A/134A 44/52/4% by weight
Molecular mass (kg/kmol)	86.46	102.03	97.6
Critical temperature (˚C)	96.14	101.06	72.1
Critical pressure (MPa)	4.99	4.06	3.729
Critical density (kg/m^3)	523.84	511.90	485
Boiling point	-40.81	-26.07	-46.6
Latent heat of vaporization at 40 ˚C (kJ/kg)	164.24	160.88	123.56

Table (2): Systems performance comparison

	Power (kW)	**Q_c (kW)**	**Q_E (kW)**	**COP**
Conventional heat pump	1.03	1.92	1.48	1.87
SAHP	0.978	2.18	1.71	2.23