THEORETICAL ENERGY AND EXERGY ANALYSES OF SOLAR ASSISTED HEAT PUMP SPACE HEATING SYSTEM

by

Ibrahim ATMACA** and Sezgi KOCAK

Department of Mechanical Engineering, Faculty of Engineering,
Akdeniz University, Antalya, Turkey

Original scientific paper
DOI: 10.2298/TSCI120813024A

Due to use of alternative energy sources and energy efficient operation, heat pumps come into prominence in recent years. Especially in solar-assisted heat pumps, sizing the required system is difficult and arduous task in order to provide optimum working conditions. Therefore, in this study solar assisted indirect expanded heat pump space heating system is simulated and the results of the simulation are compared with available experimental data in the literature in order to present reliability of the model. Solar radiation values in the selected region are estimated with the simulation. The case study is applied and simulation results are given for Antalya, Turkey. Collector type and storage tank capacity effects on the consumed power of the compressor, COP of the heat pump and the overall system are estimated with the simulation, depending on the radiation data, collector surface area and the heating capacity of the space. Exergy analysis is also performed with the simulation and irreversibility, improvement potentials and exergy efficiencies of the heat pump and system components are estimated.

Key words: *solar energy, heat pump, space heating, energy, exergy*

Introduction

The issues on energy field can be solved not only with renewable energy applications but also by choosing the right equipment which uses conventional energy resources more efficiently. Right at this point, heat pump technologies come forward both with their renewable energy usage and efficient electric power consumption. Even if there are many alternatives such as air, soil or geothermal energy, the use of solar energy is a considerable interest because it leads to diminution of fossil fuel consumption and is a non-pollutant source of energy. A solar assisted heat pump system combines the heat pump technology and solar thermal energy application in mutual beneficial ways. While these combinations improve the coefficient of performance (COP) of the heat pump unit and reduce the combustion of fossil energy resource, the low temperature thermal requirements of a heat pump makes it an excellent match for the use of low temperature solar energy and adds the benefit of a smaller solar energy system.

As known, solar assisted heat pumps have two different types; direct and indirect expansion [1]. Several computer models and experimental studies for describing the performance characteristics of solar assisted heat pump systems have been proposed over literature about both direct and indirect expansion systems. Computer models were developed and experiments were carried out for direct expansion systems by Guoying *et al*. [2], Kuang *et al*. [3], Li *et al*. [4], Huang and Chyng [5], and Huang and Lee [6]. Experimental studies related

* Corresponding author; e-mail: atmaca@akdeniz.edu.tr

to energy and exergy analysis for indirect systems were made by Kuang *et al*. [7], Dikici and Akbulut [8], Ozgener and Hepbasli [9, 10], Anderson and Morrison [11], Hepbasli and Akdemir [12], and Badescu [13]. In order to improve the present understanding of the system dimensions and productive usage of solar energy, much more investigation still has to be done. As can be seen from the literature data, experimental studies focused more on indirect expansion systems, while simulation models focused on direct expansion systems. Therefore, in this study, solar assisted indirect expanded heat pump space heating system, using a refrigerant R410A, is simulated.

This study is concentrated on the simulation of the heat pump space heating system with a detailed solar energy process. Solar radiation values in the selected region are estimated with the simulation. The case study is applied and simulation results are given for Antalya, Turkey. Collector type and storage tank capacity effects on the consumed power of the compressor, COP of the heat pump and the overall system are estimated with the simulation program depending on the radiation data, collector surface area and the heating capacity of the space. Exergy analysis is also performed with the simulation and irreversibility, improvement potentials and exergy efficiencies of the heat pump and system components are estimated. Simulation results are also compared with available experimental data in the literature in order to present reliability of the model. The simulation allows the designer to change each design parameter of the system (zone, collector properties, storage tank capacity, heating load, system dimensions), and see the effects over the system before the installation. This characteristic of the study is a great advantage over the experimental studies hence allows the designer to test different system scenarios.

Energy and exergy analyses model

The schematic diagram of a solar assisted heat pump space heating system, using R410A as the working fluid, is shown in fig. 1. As it can be seen from fig. 1, evaporator draws energy from solar heated water and sends the working fluid to the compressor. The working fluid is compressed by the compressor and then enters to the condenser. On the condenser, the working fluid releases heat to the space heating water. Finally, the refrigerant is transported to the expansion valve and fed to the evaporator afterwards.

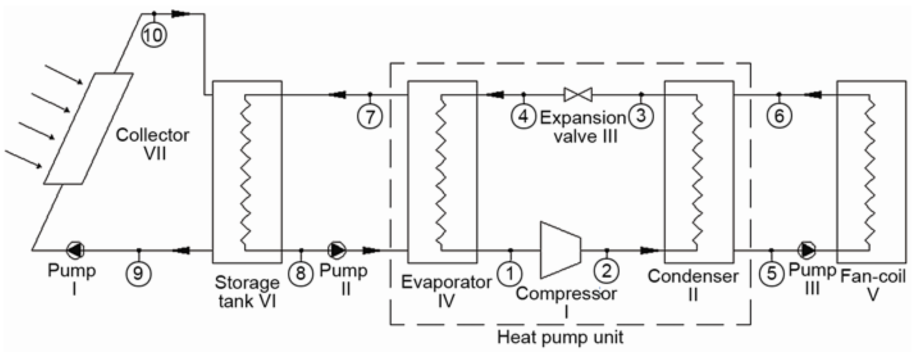


Figure 1. Schematic diagram of the solar assisted heat pump space heating system

In the analyses, steady-state, steady-flow processes are assumed and mass, energy and exergy balance equations are used to determine the heat input, the rate of irreversibility and the energy and exergy efficiencies [9, 14]. These equations are listed in tab. 1. The exergy efficiency is defined as the ratio of total exergy output to total exergy input where “out” stands

Table 1. The equations used in energy and exergy analyses [9, 14, 15]

Description	Equation
Mass balance equation	$\Sigma\dot{m}_{\text{in}} = \Sigma\dot{m}_{\text{out}}$
Energy balance equation	$\Sigma\dot{E}_{\text{in}} = \Sigma\dot{E}_{\text{out}}$
Exergy balance equation	$\Sigma\dot{E}x_{\text{in}} - \Sigma\dot{E}x_{\text{out}} - \Sigma\dot{E}x_{\text{dest}} = 0$
Exergy rate	$\dot{E}x = \dot{m}(ex)$
Specific exergy of the fluids	$ex = (h - h_o) - T_o(s - s_o)$
Exergy efficiency	$\psi = \dot{E}_{\text{out}}/\dot{E}_{\text{in}} = \dot{P}/\dot{F}$
Van Goal's improvement potential	$I\dot{P} = (1-\psi)(\dot{E}_{\text{in}} - \dot{E}_{\text{out}})$

Table 2. COP equations

Description	Equation
COP of the heat pump	$COP_{\text{HP}} = \dfrac{\dot{Q}_{\text{cond}}}{\dot{W}_{\text{comp}}}$
COP of the heat pump in terms of electrical input	$COP_{\text{HP}} = \dfrac{\dot{Q}_{\text{cond}}}{\dot{W}_{\text{comp,elec}}}$
COP of the system	$COP_{\text{system}} = \dfrac{\dot{Q}_{\text{cond}}}{\dot{W}_{\text{comp}} + \dot{W}_{\text{pumps}}}$
COP of the system in terms of electrical input	$COP_{\text{system}} = \dfrac{\dot{Q}_{\text{cond}}}{\dot{W}_{\text{comp,elec}} + \dot{W}_{\text{pumps}}}$

for "net output" or "product" or "desired value" or "benefit" and "in" refers to "given" or "used" or "fuel" [15]. The first law efficiency (COP) of the heat pump unit and the overall system can be defined in several different forms. These equations are given in tab. 2. The balance equations of mass, energy, and exergy for each component of the system are obtained by the equations given in tab. 1 and listed in tab. 3.

Several assumptions are made for the first and second law analyses of the system:

– Steady-state and steady-flow with negligible potential and kinetic energy effects and no chemical reactions are accepted for all processes.
– The directions of heat transfer to the system and work transfer from the system are given positive.
– Since the lengths of pipelines connecting the components are assumed to be short, the pressure drops are negligible.

Solar energy modeling

The collected useful energy is transferred to the hot liquid storage tank and the evaporator of the heat pump is supplied with input thermal energy. Necessary equations for calculating the solar radiation, I_T, are taken from Duffie and Beckman [16]. The total irradiance on a tilted surface under clear sky conditions is calculated by the equations depending upon R_b which is the ratio of the instantaneous direct solar radiation on a tilted surface to the instantaneous direct solar radiation on a horizontal surface:

$$R_\text{b} = \frac{\cos(\varphi-\beta)\cos\delta\cos\omega + \sin(\varphi-\beta)\sin\delta}{\cos\varphi\cos\delta\cos\omega + \sin\varphi\sin\delta} \tag{1}$$

$$I_\text{T} = I_\text{b}R_\text{b} + I_\text{d}\frac{1+\cos\beta}{2} + \left(I_\text{b} + I_\text{d}\right)\rho\frac{1-\cos\beta}{2} \tag{2}$$

Solar collectors are modeled with the formulation as suggested by Duffie and Beckman [16]:

$$\dot{Q}_\text{u} = A_\text{c}I_\text{T}\eta_\text{c} \tag{3}$$

where Q_u is the useful energy collected in system collectors, A_c – the collector area, and the collector efficiency, η_c, is:

Table 3. The balance equations for the components of the system [8, 9, 14]

Component	Mass analysis	Energy analyses
Compressor (I)	$\dot{m}_1 = \dot{m}_{2,s} = \dot{m}_{2,\text{act}} = \dot{m}_\text{ref}$	$\dot{W}_\text{comp} = \dot{m}_\text{ref}(h_{2,\text{act}} - h_1)$
Condenser (II)	$\dot{m}_2 = \dot{m}_3 = \dot{m}_\text{ref}$; $\dot{m}_5 = \dot{m}_6 = \dot{m}_\text{w,fc}$	$\dot{Q}_\text{cond} = \dot{m}_\text{w,fc} c_p (T_5 - T_6) = \dot{m}_\text{ref}(h_2 - h_3)$
Expansion valve (III)	$\dot{m}_3 = \dot{m}_4 = \dot{m}_\text{ref}$	$h_3 = h_4$
Evaporator (IV)	$\dot{m}_1 = \dot{m}_4 = \dot{m}_\text{ref}$; $\dot{m}_7 = \dot{m}_8 = \dot{m}_\text{w,st}$	$\dot{Q}_\text{evap} = \dot{m}_\text{w,st} c_p (T_8 - T_7) = \dot{m}_\text{ref}(h_1 - h_4)$
Fan-coil unit (V)	$\dot{m}_5 = \dot{m}_6 = \dot{m}_\text{w,fc}$	$\dot{Q}_\text{fc} = \dot{Q}_\text{cond} = \dot{m}_\text{w,fc} c_p (T_5 - T_6)$
Storage tank (VI)	$\dot{m}_7 = \dot{m}_8 = \dot{m}_\text{w,st}$; $\dot{m}_9 = \dot{m}_{10} = \dot{m}_\text{w,c}$	$\dot{Q}_\text{st} = \dot{Q}_\text{evap} = \dot{m}_\text{w,st} c_p (T_8 - T_7)$
Collector (VII)	$\dot{m}_9 = \dot{m}_{10} = \dot{m}_\text{w,c}$	$\dot{Q}_\text{u} = \dot{m}_\text{w,c} c_p (T_{10} - T_9)$
Component	**Irreversibility analyses**	**Exergy efficiency**
Compressor (I)	$\dot{E}x_\text{dest,comp} = \dot{m}_\text{ref}(ex_1 - ex_{2,\text{act}}) + \dot{W}_\text{comp}$	$\psi = \frac{\dot{P}}{\dot{F}} = \frac{\dot{m}_\text{ref}(ex_{2,\text{act}} - ex_1)}{\dot{W}_\text{comp}}$
Condenser (II)	$\dot{E}x_\text{dest,cond} = \dot{m}_\text{ref}(ex_2 - ex_3) + \dot{m}_\text{w,fc}(ex_6 - ex_5)$	$\psi = \frac{\dot{P}}{\dot{F}} = \frac{\dot{m}_\text{w,fc}(ex_5 - ex_6)}{\dot{m}_\text{ref}(ex_2 - ex_3)}$
Expansion valve (III)	$\dot{E}x_\text{dest,ev} = \dot{m}_\text{ref}(ex_3 - ex_4)$	$\psi = \frac{\dot{P}}{\dot{F}} = \frac{\dot{m}_\text{ref}(ex_4)}{\dot{m}_\text{ref}(ex_3)}$
Evaporator (IV)	$\dot{E}x_\text{dest,evap} = \dot{m}_\text{ref}(ex_4 - ex_1) + \dot{m}_\text{w,st}(ex_8 - ex_7)$	$\psi = \frac{\dot{P}}{\dot{F}} = \frac{\dot{m}_\text{ref}(ex_4 - ex_1)}{\dot{m}_\text{w,st}(ex_8 - ex_7)}$
Fan-coil unit (V)	$\dot{E}x_\text{dest,fc} = \dot{m}_\text{w,fc}(ex_5 - ex_6) - \dot{Q}_\text{fc}\left(1 - \frac{T_o}{T_\text{air,in}}\right)$	$\psi = \frac{\dot{P}}{\dot{F}} = \frac{\dot{Q}_\text{fc}\left(1 - \frac{T_\text{o}}{T_\text{air,in}}\right)}{\dot{m}_\text{w,fc}(ex_5 - ex_6)}$
Storage tank (VI)	$\dot{E}x_\text{dest,st} = \dot{m}_\text{w,st}(ex_7 - ex_8) + \dot{m}_\text{w,c}(ex_{10} - ex_9)$	$\psi = \frac{\dot{P}}{\dot{F}} = \frac{\dot{m}_\text{w,st}(ex_8 - ex_7)}{\dot{m}_\text{w,st}(ex_{10} - ex_9)}$
Collector (VII)	$\dot{E}x_\text{dest,st} = \dot{m}_\text{w,c}(ex_9 - ex_{10}) + A_\text{c} I_\text{T}\left(1 - \frac{T_o}{T_\text{p}}\right)$	$\psi = \frac{\dot{P}}{\dot{F}} = \frac{\dot{m}_\text{w,c}(ex_{10} - ex_9)}{A_\text{c} I_\text{T}\left(1 - \frac{T_\text{o}}{T_\text{p}}\right)}$

$$\eta_\text{c} = F_\text{R} \cdot \left[(\tau\alpha) - U_\text{L} \frac{T_\text{in} - T_\text{o}}{I_\text{T}} \right] \tag{4}$$

In this equation, T_in is the collector inlet temperature and T_o – the environment temperature. A water storage tank is placed after the solar collectors as shown in fig. 1. Perfect mixing within the tank is assumed. If the rate of heat addition and removal for a reasonable time period of Δt are assumed to be constant, equations can be written for each time interval as suggested by Duffie and Beckman [16]:

$$T_{\text{st,new}} = T_{\text{st,old}} + \frac{\Delta t}{(mc_{\text{vw}})_{\text{s}}}\left[\dot{Q}_{\text{u}} - \dot{Q}_{\text{evap}} - (UA)_{\text{s}}(T_{\text{st}} - T_{\text{o}})\right] \tag{5}$$

In eq. (5) $\dot{Q}_{\text{evap}}$ is the extracted energy from the storage tank in the evaporator, T_{s} – the main storage temperature for the period, and m – the storage tank mass. $(U.A)_{\text{s}}$ is taken as 11.1 W/K as suggested by Duffie and Beckman [16].

Simulation method

Simulation includes inputs and outputs. The inputs of the system include fan coil inlet and outlet water temperatures, compressor isentropic and mechanical-electrical efficiencies, mass flow rate circulating in the collectors and, latitude angle and altitude of the investigated region. The storage tank temperature, which is connected to storage tank mass, collector type and area, is calculated for hourly input from 08:00 to 18:00 for about 10 hours a day. Following assumptions were made in the simulation program:

– The compressor isentropic and mechanical-electrical efficiencies ($\eta_{\text{elec-mech}}$) are assumed to be 75% and 76%, respectively.
– The working fluid is considered to be saturated both at the exits of the condenser and the evaporator.
– Expansion of the working fluid is considered to be isenthalpic.
– Storage tank is assumed to be non-stratified.
– The evaporation temperature is assumed to be 10 K less than the storage tank water temperature.
– The fan-coil inlet and outlet water temperatures are taken to be constant at 50 °C and 40 °C, respectively.
– The condensing temperature is assumed to be 15 K higher than the fan-coil return water temperature.
– Flow rate circulating in the collectors are taken to be 0.135 m^3/hm^2 collector area.

Three types of collectors are considered in the simulation. Types of collectors with their characteristics are presented in tab. 4. This investigation covers the environmental and atmospheric circumstances of Antalya (□ = 36.91$^{\text{o}}$ N), Turkey. Solar insolation, I_{T}, for Antalya is presented in tab. 5. The angle of incidence of the collector is taken as 50$^{\text{o}}$ which is the optimum incidence angle for Antalya in winter. The outputs include the electric power consumed by the compressor, COP of the heat pump and overall system, irreversibility, improvement potential, and second law efficiencies of the system components.

Table 4. Solar collector characteristics [17, 18, 19]

Collector type	Collector description	F_{R} $(\tau\alpha)$	$F_{\text{R}}U_{\text{L}}$
A	Evacuated, selective surface	0.70	3.3
B	Single glazed	0.6675	5.5
C	Double glazed	0.63	4.6

Results and discussion

Simulation results are discussed in this section under design considerations which are presented above. In order to investigate the storage tank mass effect on the storage tank

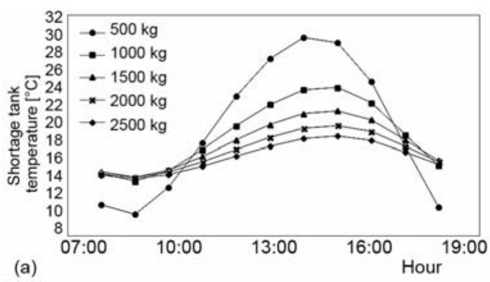
Table 5. Solar insolation, I_T [Wm^{-2}], for Antalya, Turkey

Hour	December	January	February
08:00-09:00	207.58	220.12	279.37
09:00-10:00	370.38	377.00	428.76
10:00-11:00	474.92	477.56	524.71
11:00-12:00	520.61	522.16	569.22
12:00-13:00	520.61	522.16	569.22
13:00-14:00	474.92	477.56	524.71
14:00-15:00	370.38	377.00	428.76
15:00-16:00	207.58	220.12	279.37
16:00-17:00	43.04	57.14	113.55
17:00-18:00	–	–	–

temperatures, electric consumption of compressor and heat pump COP, it is assumed that the collector area is 20 m^2 and the condenser capacity is 5 kW. Collector A is chosen as the collector type. Storage tank temperature changes according to storage tank mass for a day in January are shown in fig. 2(a). There is a reverse order between the storage tank mass and the change of the storage tank temperature. This means that increasing the storage tank mass results in less change in the storage tank temperature. As a result, the smallest tank reaches maximum and minimum temperatures in comparison among the others during day time. It can be seen that the highest storage tank temperature at the end of the working hours is supplied at the highest storage tank mass.

The changes in the electric power consumed by the compressor and heat pump COP depending on storage tank capacity are given in figs. 2(b) and (c), respectively. Based upon the low electric power consumption and high COP values, storage tank with 500 kg mass is appeared to be suitable. Although, storage tank temperature decreased too much at the end of operating hours and this is an important disadvantage. In this case, storage tanks with 1000 kg and 1500 kg mass could be preferred since they almost have the same electric power consumption and COP. In order to keep thermal inertia of the storage tank until the end of operating hours, storage tank with 1500 kg capacity can be decided as a suitable choice for the application.

In order to investigate the effects of collector type on the system performance, it is assumed that the collector area is 20 m^2, the condensner capacity is 5 kW, and storage tank



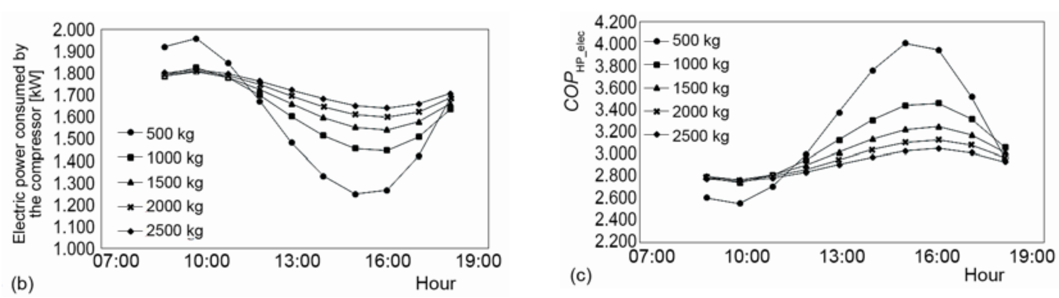


Figure 2. The effect of storage tank mass on the storage tank temperatures (a), on the electric consumption of compressor (b), and on the *COP* (c) (in January, for 5 kW condenser capacity, collector A and 20 m^2 collector area)

mass is 1500 kg. Changes in the collector efficiency for three types of collectors for a day in January are presented in fig. 3 (a). Figures 3 (b), (c), and (d) show that, when collector A that has high efficiency, is used, the storage tank temperature reaches a higher value, electric power consumption of the compressor decreases and heat pump COP increases.

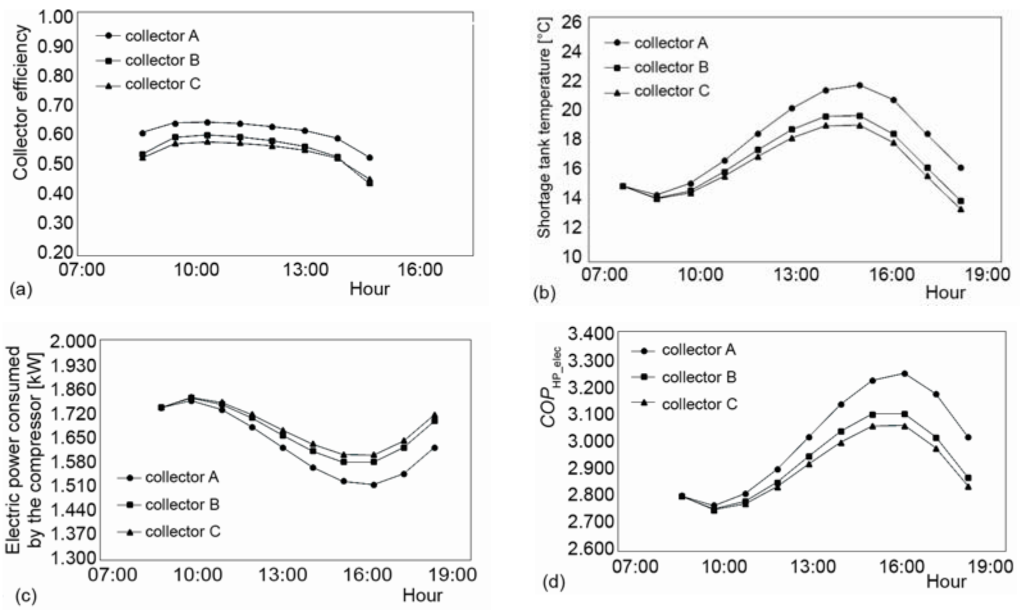


Figure 3. The effect of the collector type on the collector efficiency (a), storage tank temperature (b), electric consumption of compressor (c), and *COP* (d) (in January, for 5 kW condenser capacity, 20 m^2 collector area and 1500 kg storage tank mass)

As known, exergy is the maximum work that can be produced from the given energy under ambient conditions. Due to the exergy is the optimal use of energy, exergy analysis is a useful method to establish for the design of operation of many industrial processes. In this concept, dynamic exergy analysis is also performed with the simulation and irreversibility, improvement potentials and exergy efficiencies of the heat pump and system components are estimated. As shown in fig. 4, hourly exergy destructions of the system components for a day in January are estimated with the simulation program.

In addition to the dynamic analysis, exergy analyses depending upon the mean working parameters are also realized with the simulation. In this investigation, it is assumed that the collector area is 20 m^2, the storage tank mass is 1500 kg and collector A is chosen as a collector type. The restricted dead state temperature and pressure are taken to be 9.5 °C and 100 kPa for both working fluids R410A and water which are the mean meteorological values in January. Figure 5 and 6 represents

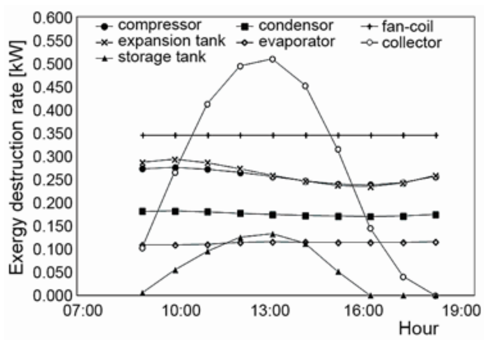


Figure 4. Hourly exergy destructions of the system components for a day in January. (for 5 kW condenser capacity, 1500 kg storage tank mass, 20 m^2 collector area and collector A)

the values for thermodynamic parameters such as exergy efficiency, product/fuel values, utilized power, improvement potential and exergy destruction rates of each component, the heat pump unit and the overall system. According to the data given in fig. 5, the exergy efficiencies on a product/fuel basis for the heat pump unit and the overall system are estimated as 82.62% and 69.85%, respectively, while the corresponding COP values are determined as 3.01 and 2.31.

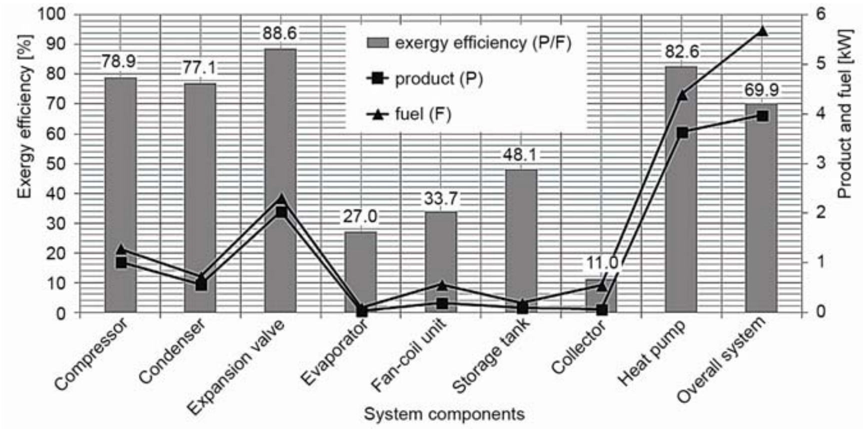


Figure 5. Comparison of exergy efficiency and product/fuel values of system components

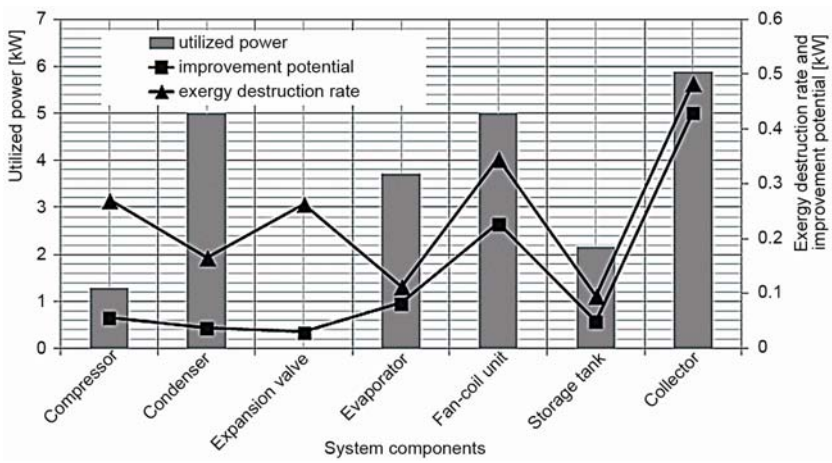


Figure 6. Comparison of utilized power, improvement potential, and exergy destruction rates of system components

It is clear from fig. 4 and fig. 6 that the greatest irreversibility occurs in compressor and expansion valve for heat pump unit and solar collectors and fan coil unit for overall system. The equipment which have the high irreversibility values especially have to be improved since components of inferior performance considerably reduce overall performance of the system. As can be seen from fig. 6, the component exergy losses of the system indicate that the greatest improvement potential is in the solar collector, followed by fan coil unit, evaporator, compressor, storage tank, condenser and expansion valve.

Obtained results have been compared with similar studies as shown in tab. 6, like those by Hepbasli [20], Ozgener and Hepbasli [9], Dikici and Akbulut [8], Hepbasli and

Akdemir [12], Kuang *et al*. [7], and Ozgener and Hepbasli [10], by taking heat pump and system COP values, exergy destruction rates of system components and exergy efficiency into consideration. Since present study and other studies were performed at different conditions, the differences between the results are normal. Nevertheless, results of this study and other studies show the same behavior. It may be concluded that the heat pump and overall system COP values obtained from the simulation are fairly close to the experimental results. Beside this, exergy destruction rates of the system components estimated from simulation show similar trends in comparison with other experimental studies. As an example, the largest irreversibility occurs generally in the compressor followed by the condenser and evaporator as confirmed by Ozgener and Hepbasli [9]. Since R410A working fluid is used in this simulation,

Table 6. Comparison of simulation results with experimental study results in the literature

Study	Content of the study	Cond. load [kW]	COP_{HP}	COP_{sys}	Exergy destruction rates [kW]							ψ_{HP} [%]	ψ_{sys} [%]
					Comp. (I)	Cond. (II)	E.valve (III)	Evap. (IV)	F.C (V)	St.tank (VI)	Coll. (VII)		
Present study simulation	R410A Solar heated water to water	5	3.009	2.312	0.271	0.166	0.263	0.113	0.346	0.096	0.484	82.6	69.9
Hepbasli [20] experimental	R410A Solar assisted ground source HP floor heating	10.01	–	–	1.25	1.33	0.47	0.34	–	–	–	72.3	69.7*
Ozgener and Hepbasli [9] experimental	R22 Solar assisted ground source to water	3.977	2.64	2.38	0.45	0.22	0.18	0.13	0.48	–	–	71.8	67.7
Dikici and Akbulut [8] experimental	R22 Solar heated water to air	3.84	3.08	–	0.827	0.232	–	0.63	–	0.096	1.92	–	–
Hepbasli and Akdemir [12] experimental	R22 Ground source to water	3.4056	–	–	0.1659	0.28	0.2169	0.0785	0.2121	–	–	–	–
Kuang *et al*. [7] experimental	Solar heated water to water	4.99	2.50-3.00	2.00-2.50	–	–	–	–	–	–	–	–	–
Ozgener and Hepbasli [10] experimental	R22 Solar assisted ground source to water	4.194	2.00-3.13	1.70-2.60	–	–	–	–	–	–	–	–	–

* Calculated from the data in regarding reference

differences between the evaporator and condenser pressures are very high. As a result, the irreversibility of expansion valve is found to be higher than the other studies.

Conclusions

In this paper, solar assisted indirect expanded heat pump space heating system using a working fluid R410A is theoretically investigated with the simulation program. Predicted results are compared with experimental literature data, and it is shown that the simulation is able to estimate both energetic and exergetic parameters of the solar assisted heat pump space heating system with reasonable accuracy. In this case, it can be said that this simulation is reliable for determining the sizing and selection of system requirements such as collector type and area and storage tank mass. As known, the exergy analysis allows to evaluate the amount of exergy destroyed for each component of the system and determine which component weights more on the overall system inefficiency. Depending on the selected collector type and area, storage tank mass and condenser capacity, exergetic analysis can be applied on the system with the simulation program and exergy destruction rate, improvement potential and exergy efficiencies of the system components can be estimated. Before installation of the system, the equipment which have the high irreversibility values may be improved and so overall performance of the system can be increased. As a result, the simulation and results is a helpful guide to engineers for design of solar assisted heat pump systems.

Acknowledgments

The authors gratefully acknowledge financial support from the Scientific Research Projects Coordination Unit of Akdeniz University (project no: 2011.02.0121.034).

Nomenclature

$\dot{E}$ – energy rate [kW]
$\dot{E}x$ – exergy rate, [kW]
ex – specific exergy, [$kJkg^{-1}$]
$\dot{F}$ – exergy rate of fuel, [kW]
F_R – collector overall heat removal factor, [–]
h – specific enthalpy, [$kJkg^{-1}$]
I – rate of instantaneous radiation, [kWm^{-2}]
$\dot{I}P$ – improvement potential, [kW]
$\dot{m}$ – mass flow rate, [kgs^{-1}]
$\dot{P}$ – exergy rate of product, [kW]
s – specific entropy, [$kJkg^{-1}K^{-1}$]
T – temperature, [°C]
$\dot{Q}$ – heat transfer rate, [kW]
U – overall heat transfer coefficient, [$Wm^{-2}K^{-1}$]
$\dot{W}$ – work rate or power, [kW]

Greek symbols

β – tilt angle of collector, [°]
δ – declination angle, [°]
η – energy efficiency, [%]
ρ – reflectance, [–]
□ – latitude angle, [°]
ψ – exergy efficiency, [%]
$(\tau\alpha)$ – effective product of transmittance-absorptance, [–]
ω – hour angle at sunrise, [°]

Subscripts

act – actual
b – direct
c – collector
comp – compressor
cond – condenser
d – diffuse
dest – destruction
elec – electric
ev – expansion valve
evap – evaporator
fc – fan-coil
HP – heat pump
in – inlet
o – dead state
out – outlet
p – plate
ref – refrigerant
sr – solar radiation
st – storage tank
sys – system
T – tilted
u – useful
w – water

References

[1] Li, H., Yang, H., Potential Application of Solar Thermal Systems for Hot Water Production in Hong Kong, *Applied Energy, 86* (2009), 2, pp. 175-180
[2] Guoying, X., *et al*., A Simulation Study on the Operating Performance of a Solar-Air Source Heat Pump Water Heater, *Applied Thermal Engineering, 26* (2006), 11-12, pp. 1257-1265
[3] Kuang, Y. H., *et al*., Study on a Direct-Expansion Solar-Assisted Heat Pump Water Heating System, *International Journal of Energy Research, 27* (2003), 5, pp. 531-548
[4] Li, Y. W., *et al*., Experimental Performance Analysis and Optimization of a Direct Expansion Solar-Assisted Heat Pump Water Heater, *Energy, 32* (2007), 8, pp. 1361-1374
[5] Huang, B. J., Chyng, J. P., Performance Characteristics of Integral Type Solar-Assisted Heat Pump, *Solar Energy, 71* (2001), 6, pp. 403-414
[6] Huang, B. J., Lee, C. P., Long-Term Performance of Solar-Assisted Heat Pump Water Heater, *Renewable Energy, 29* (2003), 4, pp. 633-639
[7] Kuang, Y. H., *et al*., Experimental Study on Solar Assisted Heat Pump System for Heat Supply, *Energy Conversion and Management, 44* (2003), 7, pp. 1089-1098
[8] Dikici, A., Akbulut, A., Performance Characteristics and Energy-Exergy Analysis of Solar-Assisted Heat Pump System, *Building and Environment, 43* (2008), 11, pp. 1961-1972
[9] Ozgener, O., Hepbasli, A., Experimental Performance Analysis of a Solar Assisted Ground-Source Heat Pump Greenhouse Heating System, *Energy and Buildings, 37* (2005), 1, pp. 101-110
[10] Ozgener, O., Hepbasli, A., Performance Analysis of a Solar-Assisted Ground-Source Heat Pump System for Greenhouse Heating: An Experimental Study, *Building and Environment, 40* (2005), 8, pp. 1040-1050
[11] Anderson, T. N., Morrison, G. L., Effect of Load Pattern on Solar-Boosted Heat Pump Water Heater Performance, *Solar Energy, 81* (2007), 11, pp. 1386-1395
[12] Hepbasli, A., Akdemir, O., Energy and Exergy Analysis of a Ground Source (Geothermal) Heat Pump System, *Energy Conversion and Management, 45* (2004), 5, pp. 737-753
[13] Badescu, V., Model of a Space Heating System Integrating a Heat Pump, Photothermal Collectors and Solar Cells, *Renewable Energy, 27* (2002), 4, pp. 489-505
[14] Dincer, I., Rosen, M. A., *Exergy, Energy, Environment and Sustainable Development,* Elsevier, Oxford, UK, 2007, pp. 91-102
[15] Gunerhan, H., Hepbasli, A., Exergetic Modeling and Performance Evaluation of Solar Water Heating Systems for Building Applications, *Energy and Buildings, 39* (2007), 5, pp. 509-516
[16] Duffie, J. A. , Beckman, W. A., *Solar Engineering of Thermal Process,* University of Wisconsin, Madison, John Willey and Sons, Interscience Publications, New York, USA, 1980
[17] Karatasou, S., *et al*., On the Calculation of Solar Utilizability for South Oriented Flat Plate Collectors Tilted to an Angle Equal to the Local Latitude, *Solar Energy, 80* (2006), 12, pp. 1600-1610
[18] Yumrutas, R., Unsal, M., Analysis of Solar Aided Heat Pump Systems with Seasonal Thermal Energy Storage in Surface Tanks, *Energy, 25* (2000), 12, pp. 1231-1243
[19] Ileri, A., Yearly Simulation of a Solar Aided R22-Degdme Absorption Heat Pump System, *Solar Energy, 55* (1995), 4, pp. 255-265
[20] Hepbasli, A., Exergetic Modeling and Assessment of Solar Assisted Domestic Hot Water Tank Integrated Ground-Source Heat Pump Systems for Residences, *Energy and Buildings, 39* (2007), 12, pp. 1211-1217

Paper submitted: August 8, 2012
Paper revised: February 12, 2013
Paper accepted: February 12, 2013