Power Generation from Low Grade Solar Energy Using Organic Rankine Cycle Driven By an Absorption Heat Transformer System

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Abstract:

Power generation using commercially available Organic Rankine Cycle (ORC) systems is expanding in a fast manner, however solar power generation applications are still limited. One way to extend solar power generation applications can be using low cost flat plate collectors. Hot water at low temperatures (80-100^{o}C) is obtainable from flat plat collectors. In this study a single stage water-Lithium Bromide absorption heat transformer (AHT) system was proposed to boost the temperature of collector outlet. Analysing AHT system showed COP performances between 52% and 55%, with a temperature lift of 30^{o}C. Upgraded energy was used to drive an ORC engine. Proposed system was found to be suitable for extending a parabolic trough solar collector field which is used to produce 40kW peak power with a commercially available ORC system and organic working fluid. Since the ORC system is designed to operate at 100^{o}C evaporating temperature, it is worth to use the AHT when heat source temperature is below 110^{o}C, however when the collector outlet temperature exceeds 110^{o}C, the AHT system decreases overall efficiency for no reason.

Keywords:

Absorption Heat Transformer, Organic Rankine Cycle, Solar Energy.

1. Introduction

Absorption heat transformers (AHT) increase temperature of low grade energy by losing a portion of the energy through condenser. Upgraded energy can be used for distillation, desalination or in an industrial process requiring high temperatures [1-3]. Using AHT systems in power generation applications has not been studied extensively. Paloso and Mohanty [4] investigated Organic Rankine Cycle (ORC) cascaded with an AHT and found AHT-ORC system offers the advantage of smaller turbine in the expense of more geothermal fluid and total heat exchanger area for 1 kW electric output. Mali et al. [5] showed 0.11 % additional power can be obtained by employing AHT in the feed water heating train of regenerative Rankine steam power plant. Fiaschi et al. [6] proposed AHT as a way to boost utilization of low temperature geothermal sources and showed utilization of geothermal fluids under 120^{o}C is possible by using the single stage AHT system.

Several studies have been carried out in the field of low temperature solar ORCs due to increasing popularity of ORC systems in the recent years. Delgado-Torres and Garcia-Rodriguez [7] optimized collector aperture area per net mechanical power and they found the minimum value of 11.9 m^2/kW for the regenerative ORC using an evacuated tube collector, R245ca fluid and heat transfer fluid configuration. Marion et al. [8] studied efficiency of single and double glazed flat plate collectors (FPC) used in the subcritical ORC with direct vapour generation configuration and concluded the effect of reducing heat loss in the collectors is more important than the choice of working fluid. Wang et al. [9] analysed daily performance of a solar-driven regenerative ORC. Their results indicated higher evaporating temperatures and turbine inlet pressures improve the system performance. Mavrou et al. [10] compared yearly performance of different organic working fluid mixtures used in a solar ORC and a mixture of neopentane-2-fluoromethoxy-2-methylpropane and neopentane at 70% neopentane is found to be most efficient in terms of generated power, operating duration and cycle efficiency.

The number of micro solar power system installations is increasing because of the availability of commercial ORC systems. Solar fields are generally undersized and ORC packages cannot be used in their full potential [11].Using low cost and widespread FPCs as a way to extend solar field is seen as an option but FPCs cannot supply heat source temperatures around 110^{o}C to activate commercial ORC systems [12]. Therefore combining an AHT system with an ORC system in order to boost the collector outlet temperature of FPCs is investigated in this study.

2. System Description

The global system consists of solar energy collecting flat plate collectors, AHT system and ORC. Schematic diagram of the global system is shown in Fig. 1. Although it is necessary to use heat storage for supplying heat source at relatively constant temperature to the AHT, heat storage is not modelled in the scope of this study.

An array of FPCs assembled in parallel is used to absorb sunlight and generate hot water which is firstly fed to the evaporator and then streamed to the generator of the AHT. The AHT system can be classified as a single stage water-Lithium Bromide (LiBr) absorption heat transformer and main components of the system are evaporator, generator, condenser and absorber. Direction of heat transfers, temperatures and pressures of each AHT system component is shown in Fig.2. Additional heat exchangers (HEX) including evaporator HEX, condenser of the evaporator HEX cycle and absorber HEX were originally proposed by Horuz and Kurt [3] to increase external coefficient of performance of the AHT. There is a refrigerant and water-LiBr binary mixture circuit in the AHT system. Hot water at point 2 generates superheated water vapour at point 5 and water-LiBr solution rich in LiBr at point 13 in the generator component.

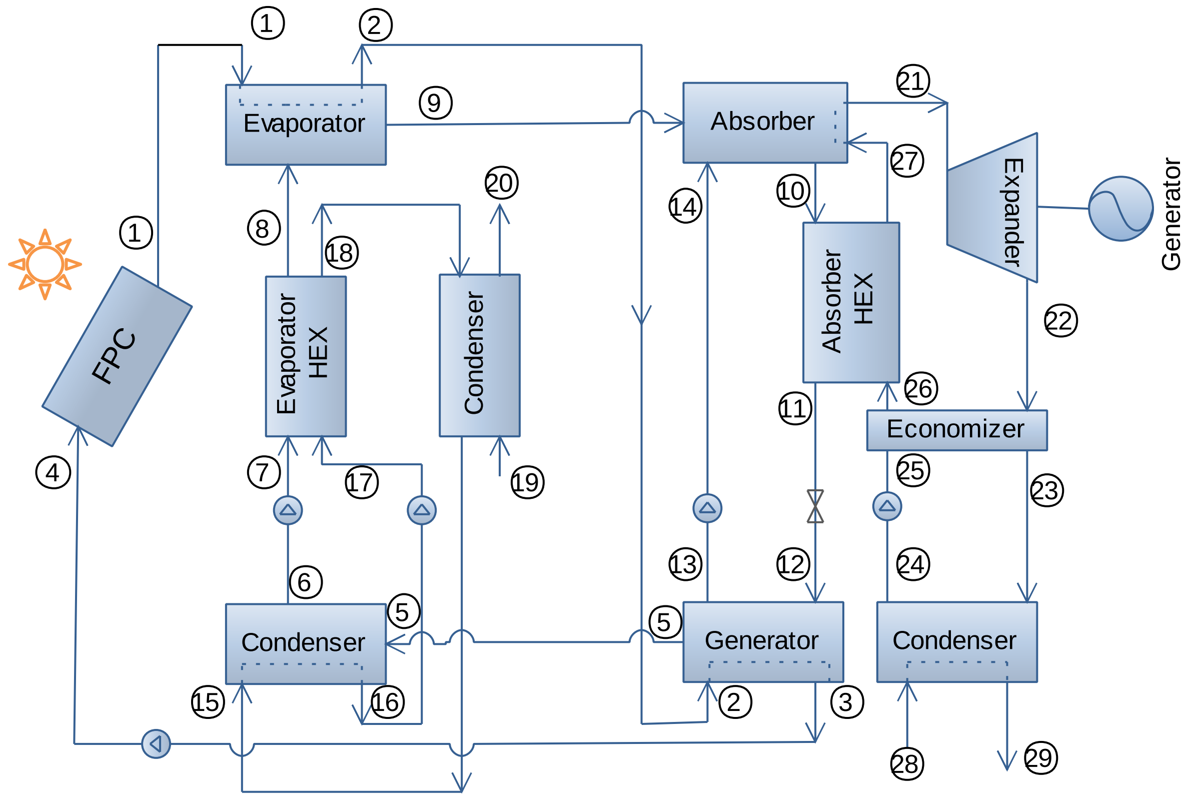

Fig. 1.Schematic diagram of the ORC driven by the solar AHT system.

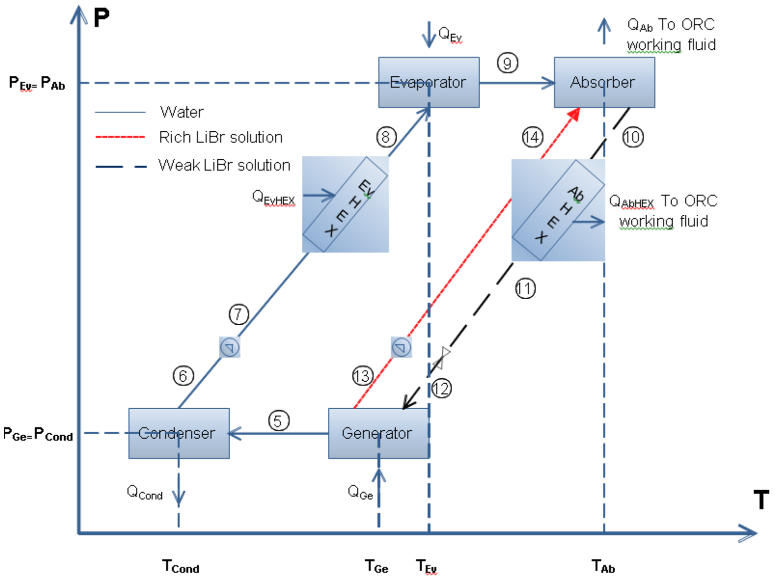

Fig. 2.Pressure and temperature diagram of the AHT system.

Rich water-LiBr solution is pumped to the absorber. Saturated vapour is produced in the evaporator by hot water at point 1 and it is absorbed in to the LiBr solution in the absorber component.

Absorption process is an exothermic reaction and produced heat is transferred to the organic working fluid. The highest temperature in the entire system is at point 10; therefore it is used to preheat organic working fluid in the absorber HEX component. The stream at point 11 is throttled and fed to generator to close the water-LiBr circuit.

The refrigerant circuit starts from point 5 which is in superheated vapour state. Superheated vapour is condensed to saturated liquid and pumped to the evaporator HEX where it is preheated in order to decrease the evaporator load. The refrigerant circuit ends after the water is evaporated to saturated vapour in the evaporator and streamed to the absorber.

ORC uses R245-fa as working fluid mainly because it is the preferred organic fluid in commercial ORC systems. Main components of the regenerative ORC process are absorber HEX, absorber, expander, economizer and condenser. As mentioned earlier, absorber and the absorber HEX of the AHT system are used for evaporating and preheating of organic working fluid, respectively. The condensed organic working fluid is pumped to the economizer, where it is preheated by desuperheating the expanded vapour. It is then preheated again and evaporated in the absorber HEX and absorber respectively before being expanded in the expander. The desuperheated fluid is condensed in the condenser.

3. Mathematical models of the subsystems

The mathematical model of the global system shown in Fig. 1 is composed of three parts. The first part is estimating solar irradiation and the useful energy from the FPCs, the second part is AHT system and the third part is ORC system.

3.1. Solar radiation and flat plate solar collector model

The hourly radiation on a tilted surface can be found by (1) incorporating hourly total radiation (I) on a horizontal surface obtainable from meteorological stations [13].

$$I_T = I_b R_b + I_d R_d + I R_r \qquad (1)$$

where I_b and I_d are the hourly beam and diffuse radiation. Hourly total radiation (I) is sum of hourly beam and diffuse radiation on a horizontal surface. R_b, R_d and R_r are tilt factors for beam radiation, diffuse radiation and reflected radiation. They are given as follows:

$$R_b = \frac{Cos(\theta)}{Cos(\theta_z)} = \frac{Sin(\delta)Sin(\phi-\beta)+Cos(\delta)Cos(\omega)Cos(\phi-\beta)}{Sin(\phi)Sin(\delta)+Cos(\phi)Cos(\delta)Cos(\omega)}, \quad (2)$$

where the declination angle is calculated by Cooper formulation:

$$\delta = 23.45 Sin\left(\tfrac{360}{365}(284+n)\right), \quad (3)$$

$$R_d = \frac{1+Cos(\beta)}{2}, \quad (4)$$

$$R_r = \rho\frac{1-Cos(\beta)}{2}, \quad (5)$$

ρ is defined as reflectivity ratio and taken as 0.2 in the literature [14].

Diffuse radiation is rarely measured and there are several models to find it. Liu Jordan [15] isotropic model expressed as in (6) is used to find hourly diffuse radiation on a horizontal surface.

$$I_d = I_0(0.384 - 0.416 k_t) \quad (6)$$

where $k_t=I/I_o$ and

$$I_0 = \frac{12\times 3600}{\pi} G_{sc}\left(1+0.033Cos(\tfrac{360n}{365})\right)\left(Cos(\phi)Cos(\delta)(Sin(\omega_2)-Sin(\omega_1)) + \frac{\pi(\omega_2-\omega_1)}{180}Sin(\phi)Sin(\delta)\right)10^{-6}, \quad (7)$$

where G_{sc}=1367 W/m^2 is solar radiation constant.

Useful energy obtained from the solar collectors is found from the empirical correlation [7]:

$$\dot{Q}_{col} = G\,A_a\left(\eta_{0a} - a_{1a}\frac{T_f - T_{amb}}{G} - a_{2a}\frac{(T_f - T_{amb})^2}{G}\right) \quad (8)$$

where A_a is the aperture area of the collector, T_{amb} is ambient air temperature, T_f is mean temperature of the fluid inside the collector, η_{0a} is optical efficiency a_{1a} and a_{2a} are heat loss coefficients. Coefficients are taken from the database of a collector certification body [16]. T_f is considered to be average of inlet and outlet temperatures. G is instantaneous radiation and given as follows $G=I_T x10^6/3600$.

3.2. Absorption heat transformer system model

The AHT system is analysed under assumptions [3] listed below

- Analyses are made under steady state and thermodynamic equilibrium conditions.
- The water-LiBr solution at the absorber and generator outlets is saturated.
- The water at the condenser and evaporator outlets is saturated.
- Heat and pressure losses are neglected.
- Pump work is considered to be negligible.
- Only on-design analyses are made in this study with determining outlet temperatures of main components of the AHT system as design parameters.
- Total heat transfer rate into the AHT system is known and given as

$$\dot{Q}_{Ev} + \dot{Q}_{Ge} = \dot{Q}_{Col} \quad (9)$$

Mass balance equation for the condenser and evaporator can be formulated as follows:

$$\sum \dot{m}_i = \sum \dot{m}_o \quad (10)$$

Mass balance equation for the absorber and generator can be formulated as follows:

$$\sum \dot{m}_i x_i = \sum \dot{m}_o x_o \qquad (11)$$

where x is the mass fraction of LiBr in the solution.

Energy balance of the each component of the AHT system can be written as follows:

$$\sum \dot{m}_i h_i + \sum \dot{Q}_{in} = \sum \dot{m}_o h_o + \sum \dot{Q}_{out} \qquad (12)$$

An important parameter in the analysis of AHT systems is flow ratio which is ratio of strong rich solution flow rate to the refrigerant flow rate. Flow ratio for the AHT system is given as follows:

$$f = \frac{\dot{m}_{13}}{\dot{m}_5} \qquad (13)$$

Another important performance indicator is coefficient of performance (COP). It is the ratio of heat supplied to the ORC to the heat supplied to the AHT. It is expressed as follows:

$$COP_{AHT} = \frac{\dot{Q}_{AbHEX} + \dot{Q}_{Ab}}{\dot{Q}_{Ev} + \dot{Q}_{Ge}} \qquad (14)$$

3.3. Organic Rankine Cycle system

The ORC system is analysed under assumptions listed below.

- Analyses are made under steady state conditions.
- Heat and pressure losses are considered to be negligible.
- Total heat transfer rate into the AHT system is known and given as

$$\dot{Q}_{Ev} + \dot{Q}_{Ge} = \dot{Q}_{Col} \qquad (15)$$

- Expander inlet (point 21) is assumed to be in the saturated vapour stage since superheating does not contribute to the thermal efficiency of the ORC cycle [17].
- Condenser outlet is saturated liquid.
- Pinch point temperature differences, efficiency of the economizer, pump and expander are determined as design parameters.

Mass balances of every ORC component and energy balances of the ORC heat exchangers can be expressed just as the AHT system formulated by (10) and (12), respectively.

Total heat input to the ORC is given by

$$\dot{Q}_{ORC} = \dot{Q}_{AbHEX} + \dot{Q}_{Ab} \qquad (16)$$

The expander power output is given as follows

$$W_{Exp} = \dot{m}_{Orc}(h_{21} - h_{22}) \qquad (17)$$

Isentropic efficiency of the expander is given by

$$\eta_{Exp} = \frac{h_{22} - h_{21}}{h_{22} - h_{21s}} \qquad (18)$$

The power consumed in the pump is given as follows

$$W_{Pump} = \dot{m}_{Orc}(h_{25} - h_{24}) \qquad (19)$$

Isentropic efficiency of the pump is given by

$$\eta_{Pump} = \frac{h_{25s} - h_{24}}{h_{25} - h_{24}} \tag{20}$$

Thermal efficiency of the ORC cycle is evaluated by

$$\eta_{ORC} = \frac{W_{\exp} - W_{pump}}{m_{Orc}(h_{21} - h_{26})} \tag{21}$$

The performance of the entire system is given by

$$\eta_{Sys} = \frac{Q_c}{I_T} COP_{AHT}\ \eta_{ORC} \tag{22}$$

4. Results and Discussion

4.1. Solar subsystem

Solar radiation simulation takes total hourly solar radiation on a horizontal surface and average ambient temperature of a southern city (N36.59^o,E35.18^o) of Turkey on June 17 of a typical meteorological year (TMY) as inputs for a base case analysis. Tilt angle of the collectors is taken as 37^o since the optimum value of tilt angle for yearly performance is found to be around latitude angle in the literature [18].Hourly variation of ambient temperature and instant solar radiation on a tilted surface is shown in Fig.3-a.

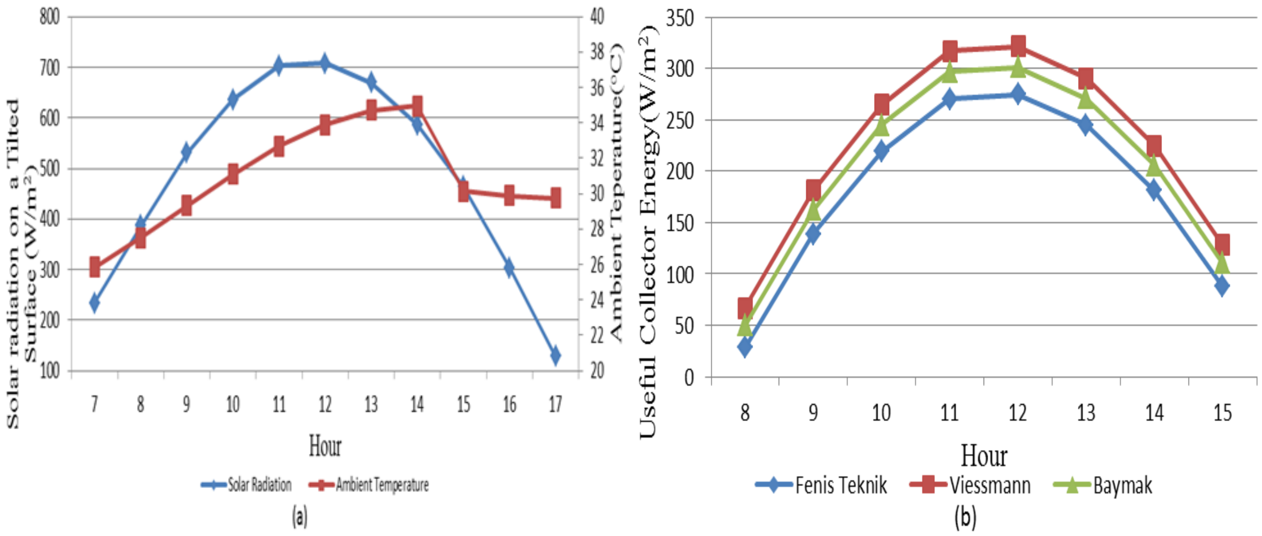

Fig. 3. a)Hourly variation of solar radiation on a tilted surface and ambient temperature, b) Hourly variation of useful energy gathered by solar collectors.

Useful energy obtained from the collectors listed in Table 1 is shown in Fig.3-b. Datasheets of the considered collectors are available in Solar Keymark database [16]. Useful energy follows the same trend as solar radiation, reaching its peak at noon and dropping sharply until the sun set. The most efficient Viessmann branded collector is considered to be assembled to form collector aperture area of 400 m^2 for the rest of simulations.

Table 1.Optical efficiency (η_{0a}) and heat loss coefficients (a_{1a},a_{2a}) for the solar collectors.

Solar Collector Model	η_{0a}	a_{1a}, W/m^{2}K	a_{2a}, W/m^{2}K^2
Fenis Teknik Solartek RA-90-CU-250	0.764	3.86	0.024
ViessmannVitosol 100 2.3	0.791	3.94	0.0122
Baymak Advanced XL	0.78	4.63	0.0036

4.2. AHT subsystem

AHT simulation takes temperatures as design parameters and finds heat transfer in each component, flow ratio, mass flow rates and COP_{AHT}. Since simulation code is written in EES built-in functions are used to find thermophysical properties of water and LiBr-Water solution. Collector outlet temperature of T_1=90°C, evaporator temperature of T_9=80°C, generator temperature of T_5= T_9-7 °C, condenser temperature of T_6=25°C, Absorber temperature of T_{10}=130°C, Evaporator HEX temperature of T_8=65°C and Absorber HEX temperature of T_{11}=65°C are set as design parameters for sensitivity analysis.

The effect of evaporator temperature (T_9) on the AHT system while keeping other design parameters fixed is shown in Fig.4-a. COP of the AHT decreases from 56% to 53% with the increase of evaporator temperature, this result is not in agreement with literature [3] due to the definition of COP_{AHT} in this study and decrasing collector efficiency at high temperatures. As evaporator temperature increases, sum of the absorber and absorber HEX heat transfer rate decreases. The effect of absorber temperature (T_{10}) on the AHT system while keeping other design parameters fixed is shown in Fig.4-b. As absorber temperature increases COP_{AHT} remains almost same, and sum of the absorber and absorber HEX heat transfer rate also exhibits similar trend.

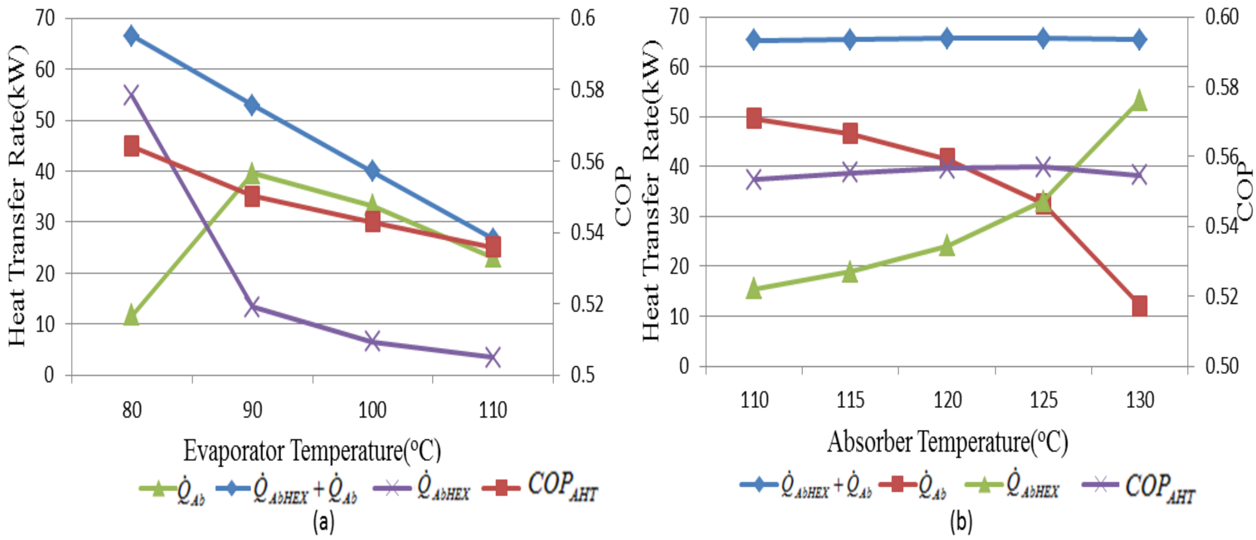

Fig. 4. a) The effect of evaporator temperature on heat transfer rates and COP_{AHT},
b) The effect of evaporator temperature on heat transfer rates and COP_{AHT}.

If condenser temperature (T_6) rises COP_{AHT} value increases as presented in Fig.5-a but as can be seen from Fig.5-b the flow ratio (f) also increases steeply after 25°C. Since f can also be expressed as $x_{13}/(x_{13}-x_{10})$, the higher flow ratio means a very small difference between x_{13} an x_{10}.In the AHT system x_{13} must be greater than $x_{12}$to continue the absorption process, condenser temperature higher than 33°C degree completely stops absorption cycle under preset design parameters.

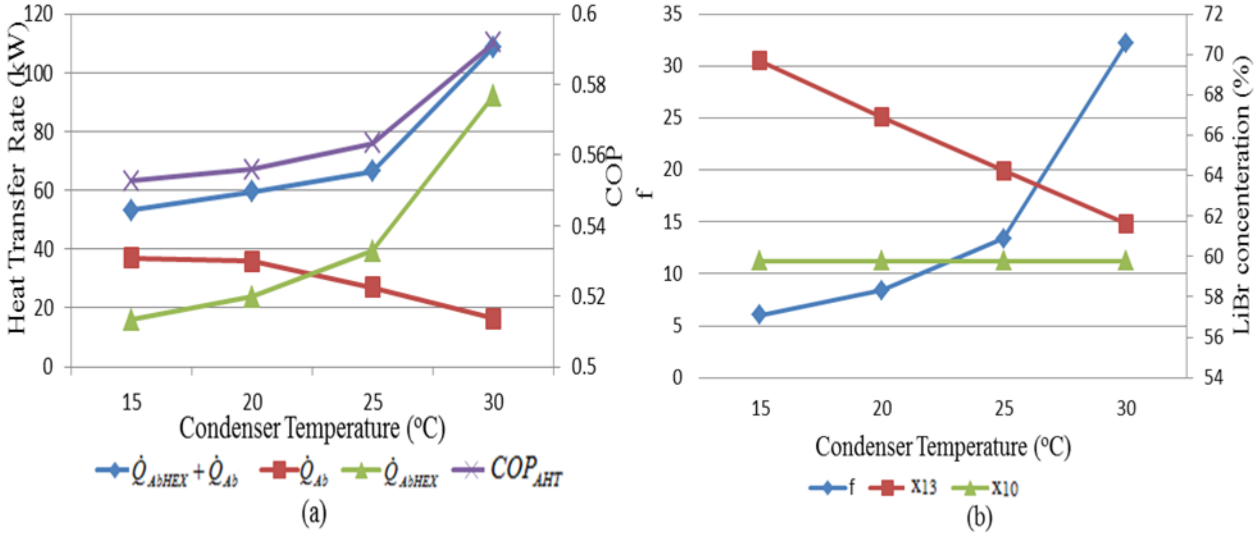

Fig. 5. a) The effect of condenser temperature on heat transfer rates and COP_{AHT}, b) The effect of condenser temperature on flow ratio.

4.3. ORC subsystem

ORC simulation takes heat transfer rate from absorber HEX and absorber as an input from the AHT system. Pinch point temperature difference of evaporation is defined as $\Delta T_e=T_{10}-T_{21}$and it is shown in Fig.6-a with the T-s diagram of the ORC system. The pinch point temperature difference of condensation, ΔT_c is also shown in Fig.6-a. Thermophysical properties of R245-fa working fluid is obtained from EES software.

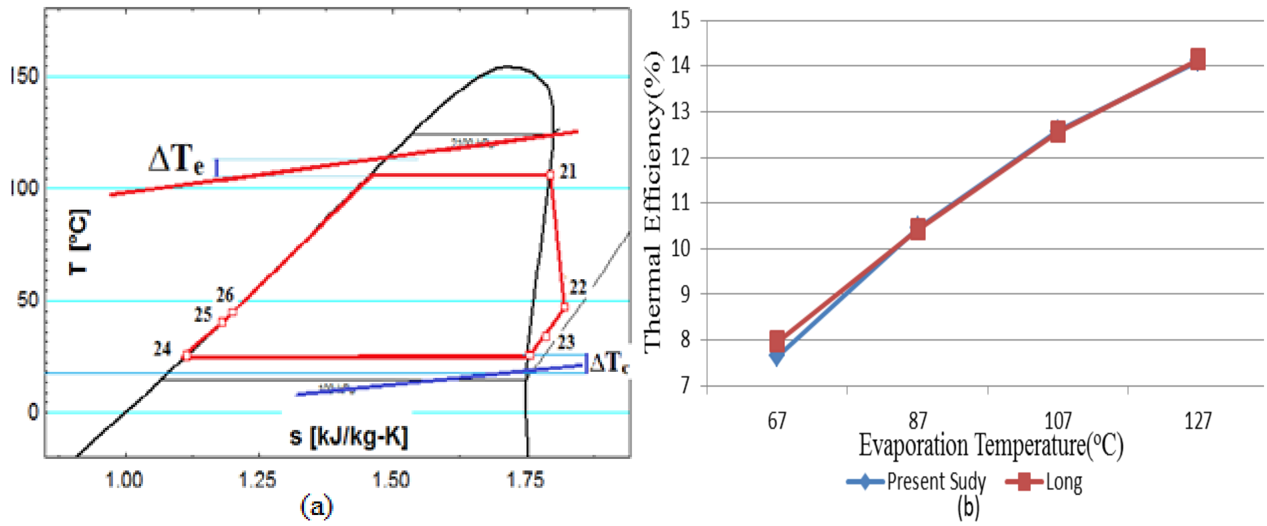

Fig. 6. a)T-s diagram of a typical ORC cycle, b)Comparison of thermal efficiency between present study and Long et al.[19] .

ORC model used in this study incorporates design parameters listed in Table 2.

Table 2.Design parameters of the ORC subsystem

Evaporation and condensation pinch point temperature [^{o}C]	10
Expander and pump isentropic efficiency	0.8
Economizer efficiency	0.7
Evaporation temperature[^{o}C]	100
Condensation temperature[^{o}C]	25

The ORC cycle thermal efficiency found by the model in this study is compared with [19]; there is an excellent agreement between them as shown in Fig.6-b

4.4. Overall system

The performance of proposed solar driven AHT-ORC system, located in a southern city of Turkey on June 17 of a TMY at 12 o'clock, is simulated for four cases and results are shown in Table 2. Different collector outlet temperatures were analysed at cases 1-4 and case 5 is a special case in which the AHT system is removed and collector outlet is directly coupled to the ORC system.

As the collector outlet temperature increases useful energy gained from collector decreases because of increased heat loses in the collectors. Collector outlet temperature which can also be described as operating temperature of collectors is the dominant parameter determining collected energy from the sun, heat input to the ORC system and expander power. Keeping collector operating temperature as low as possible is a good operating strategy Flow ratio (f),mass flow rate of collector, mass flow rate of ORC tend to decrease with increase of collector outlet temperature. Small values of flow ratio and mass flow rates mean smaller heat exchangers and expander.

Additional heat exchangers, namely evaporator HEX and absorber HEX contributes to the COP_{AHT} and makes COP_{AHT} values up to 0.55 possible. This COP_{AHT} value is greater than the conventional COP value of 0.5 in single stage absorption heat transformers. Moreover using additional heat exchangers and streaming collector outlet to the evaporator and the generator in a sequence lead to higher absorber and absorber HEX heat transfer rate capacities.

Overall system efficiency and expander output is at best in the case 5 since the AHT system in other cases adds an additional loss to the overall system. However heat source temperatures lower than 110^{o}C cannot satisfy the minimum temperature of activation requirement of commercial ORC systems, therefore boosting the collector outlet temperature of flat plate collectors by the proposed AHT system is possible.

5. Conclusions

A solar sourced AHT-ORC system for generating electricity is proposed and simple thermodynamic models are used to evaluate energetic performance of the system. Absorber and absorber HEX of the AHT system are employed as boiler and preheater of the ORC, respectively. Evaporator HEX and Absorber HEX increase COP_{AHT} as well as heat transfer rate at elevated temperature level of the AHT system. It is found to be possible to expand the solar field of an undersized micro solar power system with FPCs by using the AHT system. Operating FPCs at low outlet temperature of 80^{o}C results in best overall efficiency with a temperature lift of 30^{o}C. Implementing case 1 brings the additional power about 10 kW which will eventually contribute to the solar fraction of the micro solar power system. Using the proposed AHT system with additional heat exchangers will also increase total heat exchanger area and initial costs but thermoeconomic analysis is beyond the scope of this study.AHT is worth to use when heat source temperature is below 110^{o}C, however when the collector outlet temperature exceeds 110^{o}C, the AHT system decreases overall efficiency for no reason.

Table 3.Solar sourced AHT-ORC system simulation for various cases

	Case 1	Case 2	Case 3	Case 4	Case 5
Collector Outlet Temperature (T_1)[^oC]	80	90	100	110	110
Solar Field Aperture Area (A_a) [m^2]	400	400	400	400	400
Evaporator Temperature (T_9)[^oC]	70	80	90	100	-
Generator Temperature (T_5)[^oC]	66	76	86	96	-
Condenser Temperature (T_6,T_{24}) [^oC]	25	25	25	25	25
Evaporator HEX and Absorber HEX Temperature (T_8,T_{11}) [^oC]	60	60	60	60	-
Absorber Temperature (T_{10})[^oC]	110	110	110	110	-
ORC evaporation temperature (T_{21}) [^oC]	100	100	100	100	100
Simulation Results					
$\dot{Q}_{col}$ [kW]	138.7	117.9	96.21	73.52	73.52
$\dot{Q}_{Ev}$ [kW]	62.5	54.19	44.79	34.58	-
$\dot{Q}_{EvHEX}$ [kW]	3.85	3.317	2.723	2.089	-
$\dot{Q}_{Ge}$ [kW]	76.16	63.73	51.42	38.94	-
$\dot{Q}_{Cond}$ [kW]	66.28	57.49	47.54	36.73	-
$\dot{Q}_{AbHEX}+\dot{Q}_{Ab}$ [kW]	76.23	63.75	51.39	38.875	-
W_{Exp} [kW]	10.88	9.1	7.335	5.55	11.05
f	15.73	3.742	1.895	1.185	-
x_{13}(%)	59.73	64.24	68.5	72.58	-
x_{10}(%)	56.16	50.69	44.84	39.37	-
COP_{AHT}	0.55	0.5406	0.5342	0.5288	-
η_{ORC}	0.138	0.138	0.138	0.138	0.145
η_{Sys}	0.03757	0.03142	0.02533	0.01916	0.038
$\dot{m}_{Col}$ [kg/s]	1.84	1.562	1.271	0.9684	1.267
$\dot{m}_5$ [kg/s]	0.02632	0.02265	0.0186	0.01426	-
$\dot{m}_{Orc}$ [kg/s]	0.3427	0.2874	0.2317	0.1753	0.43

Nomenclature

A_a collector aperture area,(m^2)

f flow ratio

G_{sc} solar constant, (W/m^2)

G instantaneous radiation on a tilted surface ,(W/m^2)

h enthalpy,(kJ/kg)

I_T total radiation on a tilted surface, (MJ/m^2-hour)

I_0 extraterrestrial radiation on a horizontal surface, (MJ/m^2-hour)

I total radiation on a horizontal surface, (MJ/m^2-hour)

I_b beam radiation on a horizontal surface, (MJ/m^2-hour)

I_d diffuse radiation on a horizontal surface,(MJ/m^2-hour)

k_t clearness index

$\dot{m}$ mass flow rate,(kg/s)

n day of the year

$\dot{Q}$ heat transfer rate, (kW)

R_b beam radiation tilt factor

R_d diffuse radiation tilt factor

R_r reflected radiation tilt factor

T temperature, (^{o}C)

x mass fraction of LiBr in the water-LiBr solution

Greek symbols

η efficiency

δ declination angle

θ angle of incidence

θ_z zenith angle

Φ latitude

β tilt angle

ω hour angle

ρ reflectivity ratio

Subscripts and superscripts

1...n Thermodynamic point of the cycle

0a optical efficiency of the collector

1a first heat loss coefficients of the collector

2a second heat loss coefficients of the collector

Ab absorber

AbHEX additional absorber heat exchanger

AHT absorption heat transformer

col collector

Cond condenser

exp expander

Ev evaporator

EvHEX additional evaporator heat exchanger

Exp expander

Ge generator

i inlet

o outlet

ORC organic Rankine cycle

Sys system

s isentropic state

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