Engng.J.CMU.(2014) 21 (2), 10-20

วารสารวิศวกรรมศาสตร์
มหาวิทยาลัยเชียงใหม่
Engineering Journal
Chiang Mai University

UPGRADING OF SOLAR HEAT BY AN ABSORPTION HEAT TRANSFORMER ASSISTING WITH A VAPOR COMPRESSION HEAT PUMP

Nattaporn Chaiyat[1] and Tanongkiat Kiatsiriroat[2]
[1] College of Renewable Energy, Maejo University, Chiang Mai, Thailand
[2] Department of Mechanical Engineering, Chiang Mai University, Chiang Mai, Thailand
Email: benz178@hotmail.com

Abstract

In tropical area, even solar radiation level is rather high but diffuse solar radiation component is also very significant thus only solar flat-pate solar collector could be competitive with conventional energy for heat generation with a temperature not over $70^\circ C$. For higher temperature applications, a technique to boost-up the temperature is needed.

This paper presents simulation results of a H_2O-LiBr absorption heat transformer (AHT) performance having a R123 vapor compression heat pump (VCHP) recovering heat at the AHT condenser which is transferred to the AHT evaporator. The heat capacity at the AHT absorber is 10 kW. The unit is used to upgrade heat from a solar water heating system by using a set of flat-plate solar collectors in parallel connection each having an area of 2 m^2 with $F_R(\tau\alpha)$ and $F_R U_L$ of 0.802 and 10.37 $W/m^2 \cdot K$, respectively. The weather data of Chiang Mai, Thailand are the input information for the calculation.

For the AHT with VCHP assisted, the number of the solar collectors is around 18 units which is about 50% of that without the VCHP. The temperature at the AHT absorber could be booted up at a higher temperature level and the COP of the modified AHT is about 0.8 compared with 0.5 of the normal AHT when the AHT absorber temperature is in a range of 80-120$^\circ C$.

1. Introduction

Absorption heat transformer (AHT) is a method for upgrading low temperature heat to a higher temperature level. In a conventional AHT, low temperature heat is absorbed at the AHT generator and the AHT evaporator while high temperature heat is delivered at the AHT absorber and there is waste heat rejected at the AHT condenser. Theoretical and experimental studies of the AHT have been reported by various literatures. Kiatsiriroat et al. [1] reported thermal performance of a water-LiBr AHT for upgrading low temperature heat such as waste heat from industrial processes or solar heat. The coefficient of performance (COP) did not exceed 0.5 because there was a high heat rejection at the AHT condenser. Florides et al. [2] modeled and simulated an absorption solar cooling system in Cyprus which used 3 types of solar collectors, flat plate solar collectors, compound parabolic collectors (CPC) and evacuated tube collectors for comparison by the TRNSYS simulation program. It could be seen that the compound parabolic collector was appropriate for solar absorption cooling in a house during the whole year. The final optimized system consisted of a 15 m^2 compound parabolic collector tilted 30° from the horizontal plane and a 600 L hot water storage tank. Xuehu et al. [3] also reported the test results of an industrial-scale water-LiBr AHT in China which was used to recover waste heat released from organic vapor at $98^\circ C$ in a synthetic rubber plant. The recovered heat was used to heat hot water from $95^\circ C$ to $110^\circ C$. The AHT system was operating with a heat rate of 5,000 kW with a mean COP of 0.47. Rivera et al. [4] presented a single-stage and advanced AHT

Received 26 March 2013
Accepted 16 September 2013

operating with water-LiBr and water-Carrol™ mixtures to increase the temperature of the useful heat produced by solar ponds. The results showed that the single-stage and the double AHT increased solar pond's temperature until 50 °C at COP about 0.48 and 100 °C at COP about 0.33, respectively. Sotsil Silva Sotelo et al. [5] presented an AHT cycle operating with water-Carrol™ mixture which had a higher solubility than aqueous Lithium Bromide mixture. It could be found that the coefficient of performance was higher and less crystallization risk was obtained compared with the water-Lithium Bromide solution.

It could be seen that the COP of the H_2O-LiBr AHT could not be over 0.5 due to the heat rejected at the AHT condenser. If this one could be recovered and supplied back to the AHT evaporator then the COP could be increased. Therefore, in this study a method to improve thermal performance of a single-stage H_2O-LiBr AHT by combining a VCHP to recover the heat rejected from the AHT condenser which was supplied back to the AHT evaporator is considered. With this approach, input heat form solar collectors could be supplied at the AHT generator only, thus the number of the solar collectors could be reduced.

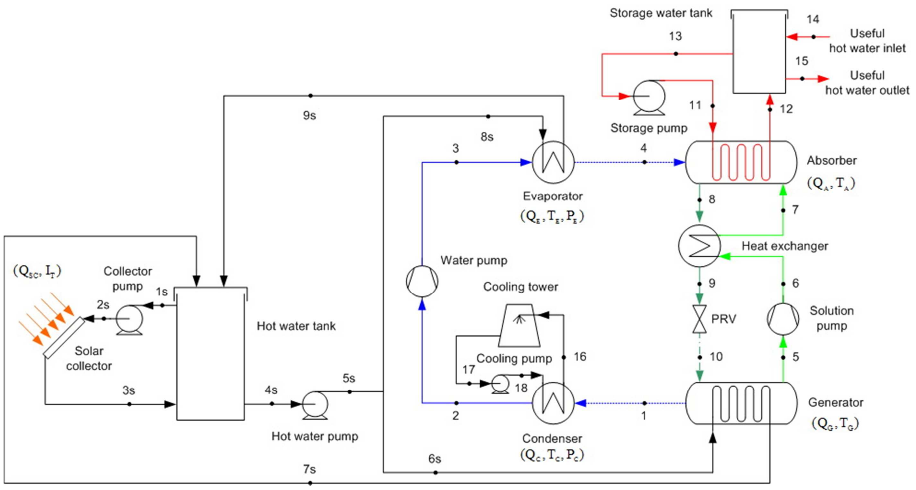


Figure 1. Schematic diagram of a solar absorption heat transformer.

2. System Descriptions and Equations

Figure 1 shows a schematic sketch of a solar-absorption heat transformer (Solar-AHT). Solar heat is supplied to the AHT generator and evaporator at a medium temperature (around 60-80 °C) and rejected heat at a lower temperature (around 35-45 °C) at the AHT condenser. A higher temperature heat (around 90-110 °C) is obtained at the AHT absorber.

The basic equations of a solar water heating system as shown in Figure 1 are as follows:

- Solar collector

$$Q_{SC} = \dot{m}_{SC} Cp_{SC} (T_{SC,o} - T_{SC,i}), \tag{1}$$

$$Q_{SC} = F_R(\tau\alpha) I_t A_{SC} - F_R U_L A_{SC} (T_{SC,i} - T_a). \tag{2}$$

For solar collectors in series connection

$$(F_R(\tau\alpha))_{Series} = F_R(\tau\alpha)\left[\frac{1-(1-K)^N}{NK}\right], \tag{3}$$

$$(F_R U_L)_{Series} = F_R U_L \left[\frac{1-(1-K)^N}{NK}\right], \tag{4}$$

where

$$K = \frac{A_{SC}(F_R U_L)_{Single\ unit}}{\dot{m}_{SC} Cp_{SC}}. \tag{5}$$

- Auxiliary heat from electric heater

$$Q_{Aux} = M_{ST} Cp_{bulk} \frac{\Delta T}{\Delta t}. \tag{6}$$

- Supplied heat of useful hot water at hot water tank

$$Q_{Sup} = \dot{m}_{Sup} Cp_{bulk} (T_{Sup,o} - T_{Sup,i}). \tag{7}$$

• Heat loss at hot water tank

$$Q_{Loss} = UA_{Tank}(T_{1s} - T_a). \tag{8}$$

• Hot water tank

$$Q_{ST} = M_{ST}Cp_{ST}(\frac{dT}{dt}). \tag{9}$$

Using numerical method, the water temperature could be calculated from,

$$T_{ST}^{t+\Delta t} = T_{ST}^{t} + \frac{Q_{ST}\Delta t}{M_{ST}Cp_{ST}}, \tag{10}$$

$$T_{ST}^{t+\Delta t} = T_{ST}^{t} + \frac{(Q_{SC} - Q_{loss} - Q_{Sup})\Delta t}{M_{ST}Cp_{ST}}. \tag{11}$$

With an auxiliary heater, the temperature becomes,

$$T_{ST}^{t+\Delta t} = T_{ST}^{t} + \frac{(Q_{SC} + Q_{Aux} - Q_{loss} - Q_{Sup})\Delta t}{M_{ST}Cp_{ST}}. \tag{12}$$

At the AHT generator, binary liquid mixture consists of a volatile component (absorbate) and a less volatile component (absorbent) is heated at a medium temperature. Part of the absorbate boils at a low pressure (P_C) and a generator temperature (T_G) at state 1. The vapor condenses in the AHT condenser at a condenser temperature (T_C) to be liquid at state 2. After that the absorbate in liquid phase is pumped to the AHT evaporator at state 3 of which the pressure (P_E) is higher than that of the AHT condenser. The AHT evaporator is heated at a medium temperature (T_E) and the absorbate in a form of vapor enters the AHT absorber which has the same pressure as the AHT evaporator at state 4. Meanwhile liquid mixture from the AHT generator, at state 5 is pumped through a heat exchanger (state 6) into the AHT absorber to a high pressure at state 7. In the AHT absorber, the strong solution absorbs the absorbate vapor and the weak solution leaves the absorber at state 8. During absorption process, heat is released at a high temperature (T_A) which is higher than those at the generator and the evaporator. This liberated heat is the useful output of the AHT. The weak solution at state 8 from the AHT absorber is then throttled to a low pressure through the AHT heat exchanger at state 9 into the AHT generator again at state 10 and new cycle restarts.

The basic equations of each component in the AHT cycle as shown in Figure 1 are as follows:

• Generator

$$Q_G = \dot{m}_1 h_1 + \dot{m}_5 h_5 - \dot{m}_{10} h_{10}, \tag{13}$$

$$\dot{m}_{10} = \dot{m}_1 + \dot{m}_5, \tag{14}$$

$$\dot{m}_{10} X_{10} = \dot{m}_5 X_5, (X_1 = 0). \tag{15}$$

From equations (2) and (3),

$$\dot{m}_5 = \frac{\dot{m}_1 X_{10}}{X_5 - X_{10}}, \tag{16}$$

and

$$\dot{m}_{10} = \frac{\dot{m}_1 X_5}{X_5 - X_{10}}. \tag{17}$$

• Condenser

$$Q_C = \dot{m}_{ref}(h_1 - h_2), \tag{18}$$

$$\dot{m}_{ref} = \dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4. \tag{19}$$

Pump and solution pump

$$W_P = (P_E - P_C)\frac{v_2 \dot{m}_2}{\eta_P}, \tag{20}$$

$$W_{SP} = (P_E - P_C)\frac{v_5 \dot{m}_5}{\eta_{SP}}, \tag{21}$$

$$h_2 \approx h_3, \tag{22}$$

$$h_5 \approx h_6. \tag{23}$$

• Evaporator

$$Q_E = \dot{m}_{ref}(h_4 - h_3). \tag{24}$$

• Absorber

$$Q_A = \dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_8 h_8, \tag{25}$$

$$\dot{m}_8 = \dot{m}_4 + \dot{m}_7, \tag{26}$$

$$\dot{m}_8 X_8 = \dot{m}_7 X_7 \ . \tag{27}$$

• Heat exchanger

$$Q_{HX} = \dot{m}_8 Cp_8 (T_8 - T_9) = \dot{m}_6 Cp_6 (T_7 - T_6), \tag{28a}$$

$$Q_{HX} = \varepsilon_{HX} (mCp)_{min} (T_8 - T_6), \tag{28b}$$

$$\dot{m}_8 = \dot{m}_9 \ , \tag{29}$$

$$\dot{m}_6 = \dot{m}_7 \ . \tag{30}$$

• Expansion valve

$$h_9 = h_{10} \ , \text{(throttling process).} \tag{31}$$

• Flow ratio (FR)

$$FR_{AHT} = \frac{\dot{m}_8}{\dot{m}_{ref}} \ . \tag{32}$$

• Coefficient of performance (COP)

$$COP_{AHT} = \frac{Q_A}{Q_E + Q_G + W_P + W_{SP}} \ . \tag{33}$$

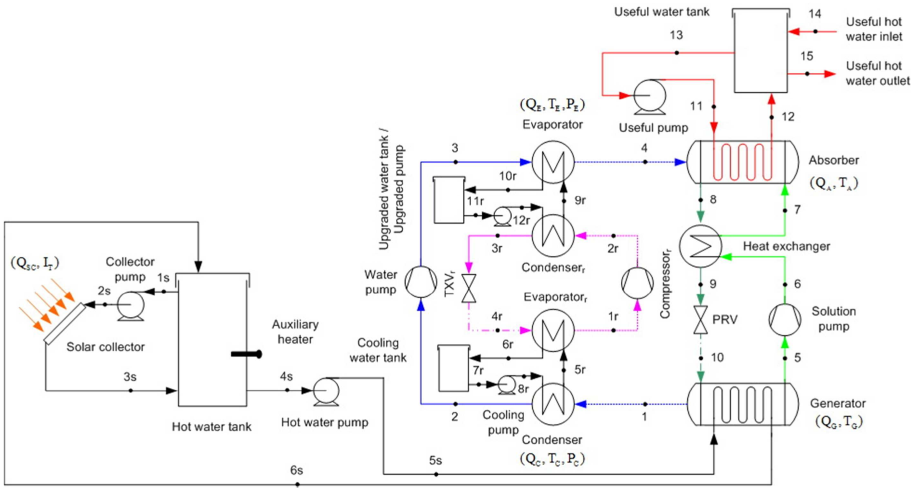


Figure 2. Schematic diagram of a solar vapor compression heat pump-absorption heat transformer.

Figure 2 shows a schematic diagram of an AHT coupling with a VCHP which is called compression/absorption heat transformer (CAHT). The heat rejected at the AHT condenser is recovered by the VCHP then the heat is upgraded and generated back to the AHT evaporator. Besides, the number of solar collectors could be less and the overall COP could be increased since the compression work is low compared with the heat loss at the AHT condenser.

R-123 is selected as the working fluid of the VCHP due to its low compression work at a high temperature range and the cycle pressure ratio is not high. The basic equations for the behavior of each component in the VCHP cycle as presented in Figure 2 are as follows:

• Evaporator$_r$

$$Q_{Er} = \dot{m}_r (h_{1r} - h_{4r}), \tag{34}$$

$$\dot{m}_r = \dot{m}_{1r} = \dot{m}_{2r} = \dot{m}_{3r} = \dot{m}_{4r} \, . \tag{35}$$

• Compressor$_r$

$$W_{Comp} = \dot{m}_r (h_{2r} - h_{1r}), \tag{36}$$

$$s_{1r} = s_{2r} \ , \text{(Isentropic process),} \tag{37}$$

$$\eta_{Comp} = \frac{h'_{2r} - h_{1r}}{h_{2r} - h_{1r}} \ . \tag{38}$$

• Condenser$_r$

$$Q_{Cr} = \dot{m}_r (h_{2r} - h_{3r}) \ . \tag{39}$$

• Expansion valve$_r$

$$h_{3r} = h_{4r} \ , \text{(Throttling process).} \tag{40}$$

Then the overall coefficient of performance (COP) of the CAHT will be:

$$COP_{CAHT} = \frac{Q_A}{Q_G + W_P + W_{SP} + W_{Comp}}. \qquad (41)$$

When the AHT and the CAHT are used to upgrade heat from flat-plate solar collectors, the solar heat is supplied at the evaporator and the generator of the AHT for the previous case and only at the generator for the latter one. The temperature inlet of solar collector is assumed to be $5^\circ C$ higher than that of the AHT evaporator and generator. The supplied heat rate could be

For the conventional AHT:

$$Q_E = A_{C,E}\,[\,F_R(\tau\alpha)I_T - F_R U_L(T_E + 5 - T_a)\,]. \qquad (42a)$$

For the AHT and the CAHT:

$$Q_G = A_{C,G}\,[\,F_R(\tau\alpha)I_T - F_R U_L(T_G + 5 - T_a)\,]. \qquad (42b)$$

3. Operating Conditions and Assumptions

A solar hot water unit is used to supply heat to an AHT and a CAHT. H_2O-LiBr is the working pair of the AHT and R-123 is the refrigerant of the VCHP. The working conditions for the evaluation are:

<u>Solar water heating system</u>

1. The solar radiation (I_T) used for the simulation is the mean solar radiation level of Chiang Mai, Thailand, as shown in appendix, [6].
2. The ambient temperature (T_{amb}) used for the simulation is the mean temperature of Chiang Mai, Thailand, as shown in appendix, [7].
3. Water flow rate $(\dot{m}_{SC})$ in each solar collector is 0.043 L/s.
4. Hot water tank capacities of the Solar-AHT and the Solar-CAHT are 3,000 L and 1,500 L, respectively.
5. Solar collector is flat-plat type having $F_R(\tau\alpha)$ and $F_R(U_L)$ of 0.802 and 10.37 $W/m^2 \cdot K$, respectively, [8].
6. Each collector is in parallel connection and each has an area of $2\,m^2$.

<u>The AHT system</u>

1. Useful heat leaving the AHT absorber is 10 kW.
2. Minimum concentration of weak H_2O-LiBr solution (X_{min}) is 45 %LiBr.
3. Maximum flow ratio (FR) for starting is lower than 20.
4. No pressure drops at the AHT condenser, the AHT generator, the AHT evaporator, the AHT absorber and the AHT heat exchanger.
5. Isentropic efficiencies of water pump (η_P) and solution pump (η_{SP}) are 85%.
6. Effectiveness of the AHT heat exchanger (ε_{HX}) is 85%.
7. The properties of H_2O-LiBr solution are taken from Refs [9-12].

<u>The VCHP system</u>

1. No pressure drops at the VCHP condenser and the VCHP evaporator.
2. Isentropic efficiency of compressor (η_{Comp}) is 80%.
3. Degree of superheating (SH) is $5.0^\circ C$.
4. Degree of subcooling (SC) is $5.0^\circ C$.
5. The properties of R-123 are based upon REFPROP [13].

In this study, a R-123 VCHP is used to couple with the AHT. In our experiment, the VCHP performance has been undertaken. It could be found that its coefficient of performance (COP_{HP} - a ratio of heat at its condenser and its electrical power consumption) is a function of water temperature entering the VCHP evaporator ($T_{CW,i}$) and water temperature entering the VCHP condenser ($T_{HW,i}$). The performance curve of the R-123 heat pump system is shown in Figure 3.

The COP_{HP} of the R-123 VCHP is in a form of:

$$COP_{HP} = -0.1065(T_{HW,i} - T_{CW,i}) + 5.8922, \quad (kW_{th}/kW_{elec}) \qquad (43)$$

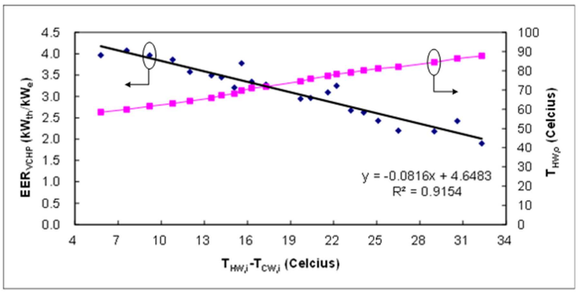


Figure 3. Performance curve of R-123 vapor compression heat pump system.

4. Results and Discussions

Figure 4 shows hot water temperature leaving solar water heating system (T_{HW}) of the Solar-AHT at various numbers of solar collector units between 30-50 units each in parallel connection at 3,000 L of hot water tank with the solar radiation (I_T) and the ambient temperature (T_{amb}) of April, Chiang Mai, Thailand. It could be noted that the AHT could be operated when the hot water temperature from the solar water heating system is over 70 °C then the minimum units of solar collectors for supplying heat to the AHT is about 35 units. The system could operate continuously about 8 h/d. The Solar-AHT supplies medium heat at the evaporator and the generator about 20 kW and generates upgraded heat about 10 kW at a higher temperature level at the AHT absorber.

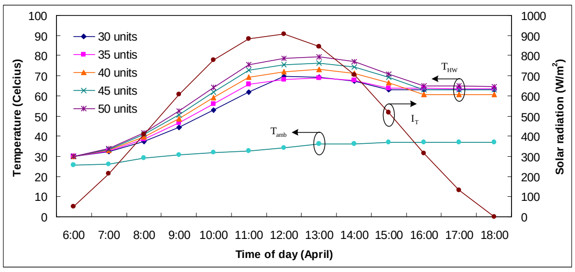


Figure 4. T_{HW}, T_{amb} and I_T of the solar-AHT at various the number of solar collectors during time of a day in April.

Figure 5 shows temperatures of the solar-AHT components for 35 units of flat-plate solar collector during a day in April. The AHT could operate at around 11 a.m. when the hot water temperature is over 65 °C. It could be seen that the absorption system could upgrade heat at the absorber over 80 °C. T_C is nearly constant while T_G, T_E and T_A vary with the hot water temperature leaving the solar water heating system. Figure 6 shows the heat rates for all components of the system.

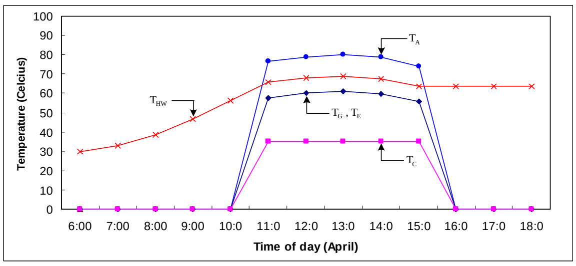


Figure 5. Temperatures of the solar-AHT components for 35 units of flat-plat solar collectors during time of a day in April.

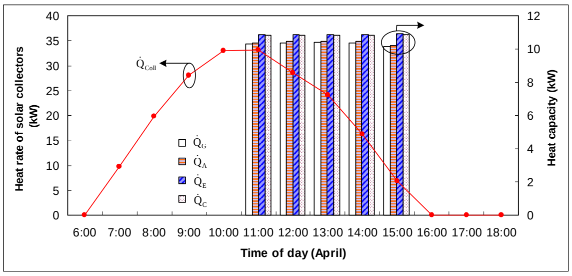


Figure 6. Heat rates at the solar-AHT components for 35 units of flat-plat solar collectors during time of a day in April.

Figure 7 shows the temperature profiles of the solar-CAHT components. It could be seen that T_E and T_A are constant while T_G varies with T_{HW} from solar water heating system. The heat rates from solar heat and the heating capacities of the AHT components are shown in Figure 8. It could be noted that the system requires the solar collectors only 18 units.

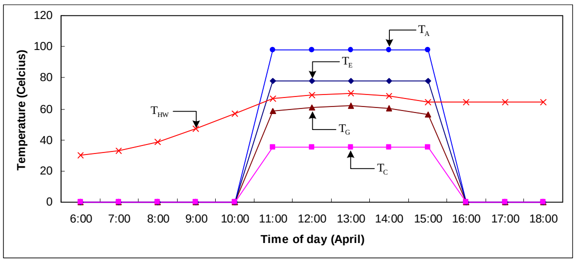


Figure 7. Temperatures of the solar-CAHT components for 18 units of flat-plat solar collectors during time of a day in April.

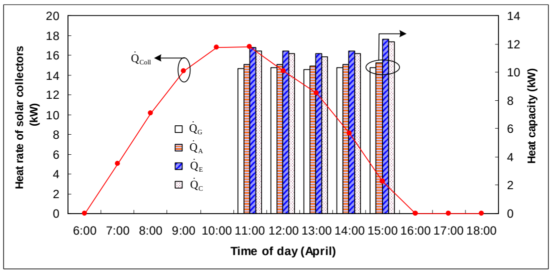


Figure 8. Heat rates of the solar-CAHT components for 18 units of flat-plat solar collectors during time of a day in April.

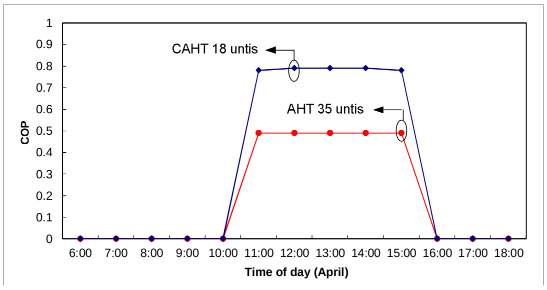


Figure 9. Comparison of the overall COP of the Solar-AHT and the Solar-CAHT on I_T during time of a day in April.

Figure 9 shows the overall COP of the normal Solar-AHT and the Solar-CAHT during time of day in April. It could be found that the overall COP of the Solar-CAHT increases around 60% which is 0.8 compared with that of the Solar-AHT which is 0.5. The number of solar collectors is 18 units compared with that of the Solar-AHT which is 35 units.

5. Conclusions

From this study, the conclusions are as follows:

1. The temperature at the AHT absorber could be booted up to a higher level than that by the normal AHT.
2. For 10 kW heat rate with the absorber temperature over 80 °C, the number of the solar collectors units of the solar-CAHT could be decreased about 50% which is 18 units instead of 35 units of the normal solar-AHT.
3. The overall COP of the solar-CAHT could be increased around 60% which is around 0.8 compared with around 0.5 of the normal solar-AHT.

6. Acknowledgement

The authors would like to acknowledge the Graduate School, Faculty of Engineering, Chiang Mai University, the Office of the Higher Education Commission, Ministry of Education, Thailand and Energy Policy and Planning Office (EPPO), Ministry of Energy, Thailand and College of Renewable Energy, Maejo University, Thailand for supporting the facilities for this study.

References

[1] Kiatsiriroat, T., Bhattacharya, S.C., Wibulswas, P. Upgrading Heat by a Reversed Absorption Heat Pump. *Applied Thermal Engineering*, 1986; 25: 175-186.

[2] Florides, G.A., Kalogirou, S.A., Tassou, S.A., Wrobel, L.C,. Modeling and Simulation of an Absorption Solar Cooling System for Cyprus. *Solar Energy*, 2002: 72: 43-51.

[3] Xuehu, M., Jiabin, C., Songping, L., Qingyun, S., Aiming, L., Wei, L., Jiayan, Z., Guojun, Z., Zhihao, F. Application of absorption heat transformer to recover waste heat from a synthetic rubber plant. *Appl Therm Eng*, 2002; 25:797-806.

[4] Rivera, W., Cardoso, M.J., Romero, R.J. Single-Stage and Advanced Absorption Heat Transformers Operating with Lithium Bromide Mixtures Used to Increase Solar Pond's Temperature. *Solar Energy Materials & Solar Cells*, 2001; 70:321-333.

[5] Sotelo, S.S., Romero, R.J. Improvement of Recovery Energy in the Absorption Heat Transformer Process Using Water-Carrol™ for Steam Generation. The ninth International Conference on Chemical & Process Engineering, 2009, 9, Italy.

[6] NASA Surface meteorology and Solar Energy. RETScreen Data. [Online] available: http://eosweb.larc.nasa.gov/sse/RETScreen/. (July, 2010).

[7] Thai Meteorological Department. The ambient temperature at Chiang Mai. [Online] available: http://www.tmd.go.th/thailand.php. (July, 2010).

[8] Sanguantrakarnkul, P. Sizing of a Solar Water Heating System for Abattoir. MEng thesis, Chiang Mai University, Chiang Mai, Thailand, 2006.

[9] American Society of Heating. Refrigerating and Air-Conditioning Engineers. Inc. (ASHRAE). Refrigerants. Atlanta, 2001.

[10] Khairulin, R.A., Gruzdev, V.A., Stankus, S.V., Verba, O.I. Experimental study of the density of aqueous solutions of lithium bromide at temperature of up to 250°C in the range of mass concentrations from 30 to 65 %. *Thermophysics and Aeromechanics*, 2006 ; 13: 575-583.

[11] Kaita, Y. Thermodynamic properties of lithium bromide-water solutions at high temperatures. *International Journol of Refrigeration*, 2000 ; 24 : 374-390.

[12] Feuerecker, G., Scharfe, J., Greiter, I., Frank, C., Alfeld, G. Measurement of thermophysical properties of LiBr-solutions at high temperatures and concentrations. International Absorption Heat Pump Conference ASME, 1993 ; 31:493-499.
[13] National Institute of Standards and Technology, Inc. (NIST, REFPROP Version 7). Thermodynamic Properties of Refrigerants and Refrigerant Mixtures Software. America, 2000.
[14] Provincial Electricity Authority. The rate of electricity cost. [Online] available: http://www.pea.co.th/. (July, 2010).
[15] Krungthai Bank Public Company Limited. Minimum Retail Rate (MRR). [Online] available: http://www.ktb.co.th. (July, 2010).

Appendix

A. Enthalpy-Concentration and Temperature for lithium bromide-water solutions [9]

For Concentration $x < 40$ %LiBr

Solution temperature range $15 < t < 165$ °C

$$h = 21.4817157 - 2.38366711X + 3.90458186t + 0.03625001X^2 + 5.25010607 \times 10^{-4} t^2 - 0.0369249939tX, \ \mathrm{kJ/kg}$$

For Concentration $40 \leq x < 70$ %LiBr

Solution temperature range $15 < t < 165$ °C

$$h = \sum_0^4 A_n X^n + t\sum_0^4 B_n X^n + t^2 \sum_0^4 C_n X^n, \ \mathrm{kJ/kg}$$

A_0 = -2024.33	B_0 = 18.2829	C_0 = -3.7008214 E-2
A_1 = 163.309	B_1 = -1.1691757	C_1 = 2.8877666 E-3
A_2 = -4.88161	B_2 = 3.248041 E-2	C_2 = -8.1313015 E-5
A_3 = 6.302948 E-2	B_3 = -4.034184 E-4	C_3 = 9.9116628 E-7
A_4 = -2.913705 E-4	B_4 = 1.8520569 E-6	C_4 = -4.4441207 E-9

B. Solution Temperature-Refrigerant Temperature and Saturation pressure [9]

For Refrigerant $-15 < t' < 110$ °C

Solution temperature $5 < t < 175$ °C

Concentration $45 < X < 70$ %LiBr

$$t = \sum_0^3 B_n X^n + t' \sum_0^3 A_n X^n, \ ^\circ\mathrm{C}$$

$$t' = (t - \sum_0^3 B_n X^n) / \sum_0^3 A_n X^n, \ ^\circ\mathrm{C}$$

$$\log P = C + D/T' + E/T'^2, \ P = \mathrm{kPa}; \ T' = \mathrm{K}$$

$$T' = \frac{-2E}{D + [D^2 - 4E(C - \log P)]^{0.5}}$$

A_0 = -2.00755	B_0 = 124.937	C = 7.05
A_1 = 0.16976	B_1 = -7.71649	D = -1596.49
A_2 = -3.133362 E-3	B_2 = 0.152286	E = -104095.5
A_3 = 1.97668 E-5	B_3 = -7.9509 E-4	

C. Density of lithium bromide-water solutions [10]

For Solution temperature $t < 250$ °C

Concentration $30 < X < 65$ %LiBr

$$\rho(t, m) = \rho_0(t)[1 + d_0(t)m + d_1(t)m^{1.5} + d_2(t)m^2], \ \mathrm{kg/m^3}$$

$$m = w / M_s(1 - w), \ \mathrm{mole/kg}$$

$$d_j(t) = \sum_{i=0}^{4} C_{ji} t^i$$

$$\rho_0(t) = \text{ Density of pure water, } \; kg/m^3$$

$$M_s = 0.086845, \; kg/mole$$

Table of Coefficients C_{ji}

j/i	0	1	2	3	4
0	6.9979 E-2	-9.36591 E-5	1.1770035 E-6	-2.829722 E-9	7.963374 E-12
1	-7.30855 E-3	1.78947 E-5	-3.458841 E-8	-8.88725 E-10	1.085224 E-12
2	1.811867 E-4	-1.9292 E-6	-1.565022 E-8	2.082693 E-10	-3.761121 E-13

D. Heat capacity of lithium bromide-water solutions [11]

For Solution temperature $40 < t < 210\ ^\circ C$

Concentration $40 < X < 65$ %LiBr

$$Cp = (A_0 + A_1 X) + (B_0 + B_1 X)t, \; kJ/kg \cdot {}^\circ C$$

$A_0 = 3.462023$ $B_0 = 1.3499$ E-3

$A_1 = -2.679895$ E-2 $B_1 = -6.55$ E-6

E. Entropy of lithium bromide-water solutions [12]

For Solution temperature $40 < t < 210\ ^\circ C$

Concentration $40 < X < 65$ %LiBr

$$S = \sum_{i=0}^{3} \sum_{j=0}^{3} B_{ij} X^j T^i, \; kJ/kg \cdot K$$

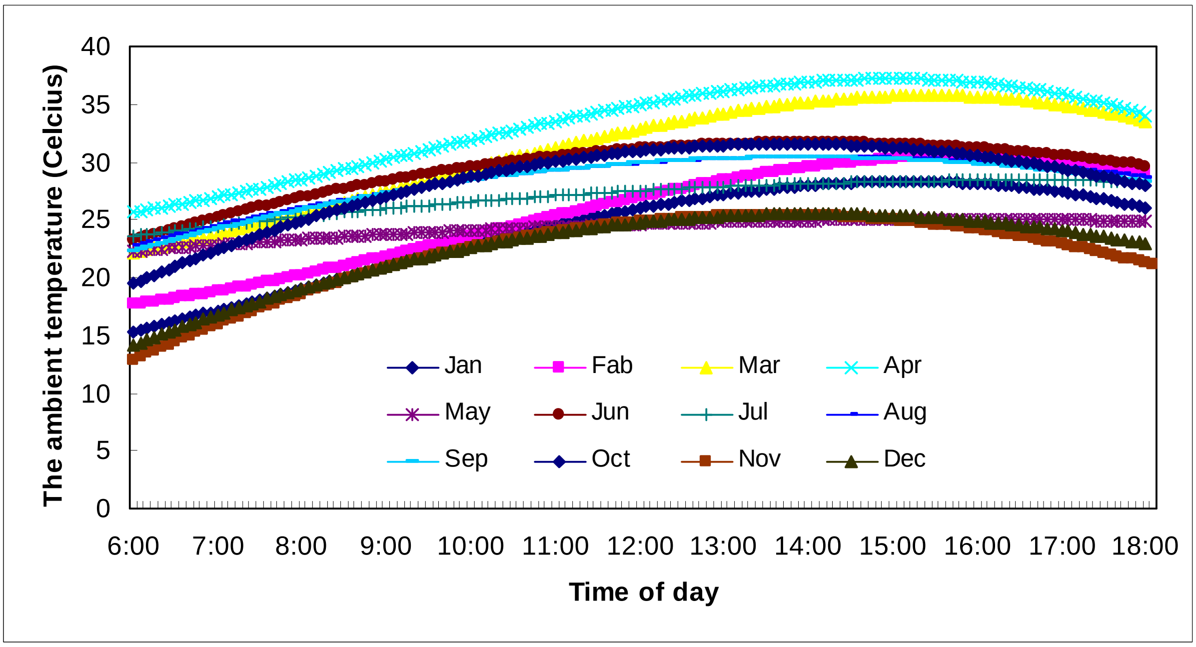
Table of Coefficients B_{ij}

i	B_{i0}	B_{i1}	B_{i2}	B_{i3}
0	5.127558 E-01	-1.393954 E-02	2.924145 E-05	9.035697 E-07
1	1.226780 E-02	-9.156820 E-05	1.820453 E-08	-7.991806 E-10
2	-1.364895 E-05	1.068904 E-07	-1.381109 E-09	1.529784 E-11
3	1.021501 E-08	0	0	0

F. The average solar radiation of Chiang Mai, Thailand [14].

Month	Jan	Feb	Mar	Apr	May	Jun
$I_T \; (MJ/m^2 \cdot d)$	17.82	20.34	21.71	22.36	19.69	16.88
Month	Jul	Aug	Sep	Oct	Nov	Dec
$I_T \; (MJ/m^2 \cdot d)$	15.66	15.23	15.77	15.73	15.84	16.45

G. The ambient temperature of Chiang Mai, Thailand [15].



Nomenclature

A	area, (m^2)
AC	annual cost, (Baht)
Cp	heat capacity, $(kJ/kg \cdot K)$
COP	coefficient of performance
i	annual discount rate on loans
I_T	solar radiation, (W/m^2)
m	mass flow rate, (kg/s)
N	operation life of the system in consideration, (y)
P	pressure, (bar)
Q	heat rate, (kW)
R	refrigerant
v	specific volume, (m^3/kg)
s	entropy, $(kJ/kg \cdot K)$
SC	subcooling, $(^\circ C)$
SH	superheating, $(^\circ C)$
t	time, (s)
T	temperature, $(^\circ C)$
U	overall heat transfer coefficient, $(W/m^2 \cdot K)$
W	work, (kW)
X	concentrate, (%LiBr)

Greek symbol

η	efficiency, (%)
ε	effectiveness, (%)
ρ	density, (kg/m^3)

Subscript

A	absorber

Aux	auxiliary heat
act	actual
amb	ambient
bulk	bulk temperature
C	condenser
Coll	solar collector
Comp	compressor
CW	cooling water
e	super heat
elec	electric
E	evaporator
H	high
HS	heat source
HW	hot water
HX	heat exchanger
i	inlet
L	low
max	maximum
min	minimum
o	outlet
P	pump
r	compression cycle
ref	refrigerant
S	start
SC	solar collector
SP	solution pump
ST	storage tank
Sup	supply
UF	useful