International Journal of Modern Engineering Research (IJMER)
www.ijmer.com Vol.3, Issue.4, Jul - Aug. 2013 pp-1923-1934 ISSN: 2249-6645

Upgrading of Low Temperature Solar Heat with Cascade Vapor Compression and Absorption Heat Pump

Nattaporn Chaiyat[1]
School of Renewable Energy, Maejo University, Chiang Mai, Thailand

Abstract: *The objective of this project is to study a method to upgrade a low temperature heat form solar energy by cascaded vapor compression heat pump/absorption heat pump. The modified system could be used to produce high temperature heat such as high temperature hot water with full replacement or partial support for boiler in hotel, hospital and other related industries. The input energy comes from solar energy which is clean and friendly to environment.*

In this study, a solar water heating system was designed and constructed. The unit had 10 units of flat-plate solar collectors (1 unit = 2.3 m^2) each generated hot water at a temperature range of 40-60 oC and a storage tank of 1,500 liter. After that these hot water temperature was upgraded by 2 units of R-123 vapor compression heat pumps each having a heating capacity of 10 kW. Hot water at a higher temperature of around 60-80 ^{o}C was produced and kept in a 200 liter hot water tank. Then a 10 kW water-Libr absorption heat pump upgraded the final hot water temperature to be around 90-110 ^{o}C kept in a 200 liter tank. Since the water temperature might be over the boiling point then glycol was mixed in the water with a concentration of around 40%,

Mathematical correlations of the related parameters from the experimental data could be set up and these could be used to predict outputs of the studied system under various operating conditions. The final outputs such as the system COP and the final hot water temperature simulated by the models were found to be close with those of the experimental results.

From the economic results, the modified system was used to partially support a boiler for generating hot water at 5 Ton/d compared up to fully support at 35 Ton/d. For the partially support, the energy saving and the payback period for the modified system were around 2,675,434 Baht/y (1 USD = 30.6535 Baht) and 1 year 2 months, respectively. The payback was longer with the higher load of the system.

Keywords: *Absorption heat transformer; Vapor compression heat pump; Solar collector; Boiler, Economical analysis*

I. INTRODUCTION

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. Normally, the flat-plate one will not supply heat with a temperature over 60 ^{o}C, otherwise its thermal efficiency is very low, therefor, a technique to boost-up the temperature is needed.

Absorption heat transformer (AHT) and vapor compression heat pump (VCHP) are a method for upgrading heat to a higher temperature level. For the VCHP, this technology is used to upgrade a low temperature heat (around 40-60 ºC) to a medium temperature level (around 60-80 ºC). In a conventional AHT, the absorption system is used to upgrade a medium temperature heat (around 70-80 ºC) to a high temperature level (around 90-120 ºC). 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^o 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 ^{o}C in a synthetic rubber plant. The recovered heat was used to heat hot water from 95$^-$110 ^{o}C. The AHT system was operating with a heat rate of 5,000 kW with a mean COP of 0.47. Chaiyat et al. [4] reported a concept of a single-stage H_2O-LiBr absorption heat transformer (AHT) when it was coupled with a vapor compression heat pump (VCHP) for upgrading low temperature heat (CAHT). Heat rejected at the AHT condenser was recovered by the VCHP and transferred to the AHT evaporator. It could be seen that a simulation results of the modified system could be increased around 0.8 compared with 0.5 of the normal AHT. Moreover, Chaiyat et al. [5] also reported a simulation result of a H_2O-LiBr absorption heat transformer performance having an R-123 vapor compression heat pump (CAHT). The CAHT unit was used to upgrade heat from a set of flat-plate solar collectors. It could be found the number of the solar collectors could be decreased 30 units which is about 50 % of that without the VCHP. Moreover, the COP of the modified AHT is about 0.8 compared with 0.5 of the conventional AHT. But this technique could be upgraded the maximum temperature around 90 ^{o}C.

The objective of this study is to study a method to upgrade a low temperature heat form solar energy by the vapor compression heat pump cascaded with the absorption heat pump to generate a high temperature level at over 100 ^{o}C. The modified system could be used to produce high temperature heat such as high temperature hot water with full replacement or partial support for boiler in hotel, hospital and other related industries. The input energy comes from solar energy which is clean and friendly to environment. For the VCHP, an appropriate working fluid has been selected. The seven parameters for evaluating the thermal performance of the VCHP will be considered and compared with those of the common VCHP.

II. SYSTEM DESCRIPTION

Fig.1 shows a schematic sketch of a general solar-absorption heat transformer (Solar-AHT). Solar heat is supplied to the AHT generator and the AHT evaporator at a medium temperature (around 60-80 $^{\circ}$C). 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 and rejected heat at a lower temperature (around 35-45 $^{\circ}$C). 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 the 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 (around 80-110 $^{\circ}$C). 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.

At the AHT condenser, high amount of heat rate is rejected to the environment thus the coefficient of performance (COP) of the normal AHT system is rather low. Moreover, when the solar collectors generates the high water temperature which results to its higher heat loss too.

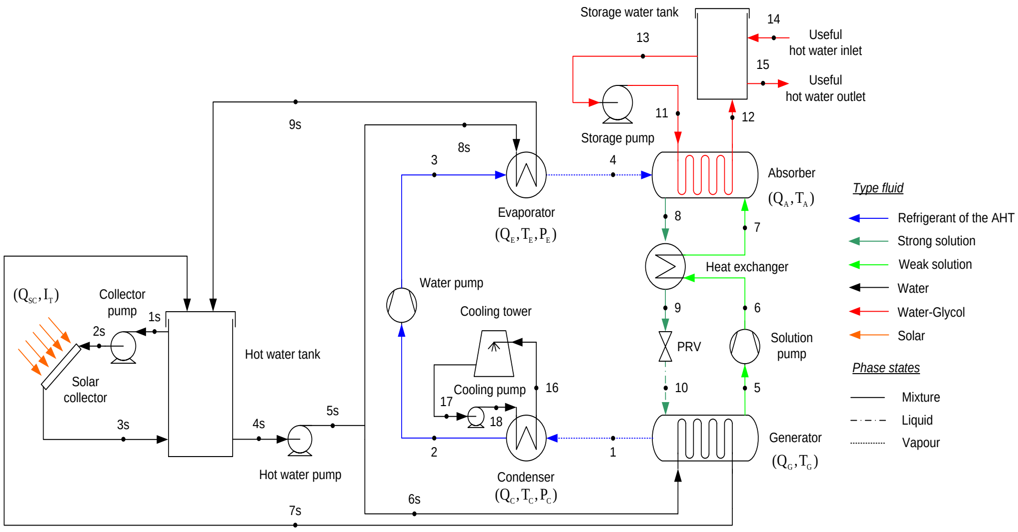


Figure 1. Schematic diagram of a solar-absorption heat transformer.

Fig.2 shows a schematic sketch of a solar water heating system (SWHS) combined with a vapor compression heat pump (VCH) cascaded an absorption heat transformer (AHT). Solar heat is supplied to the VCHP evaporator at a low temperature (around 40-60 ^{o}C) and upgraded heat at a medium temperature (around 60-80 ^{o}C) at the VCHP condenser. After that, a medium temperature heat is obtained at the AHT generator and evaporator for boosting heat to a high temperature level (around 100-120 ^{o}C) at the AHT absorber. Besides, the solar collector will supply heat at a low temperature level compared with the normal system since the solar collector operates at a higher efficiency.

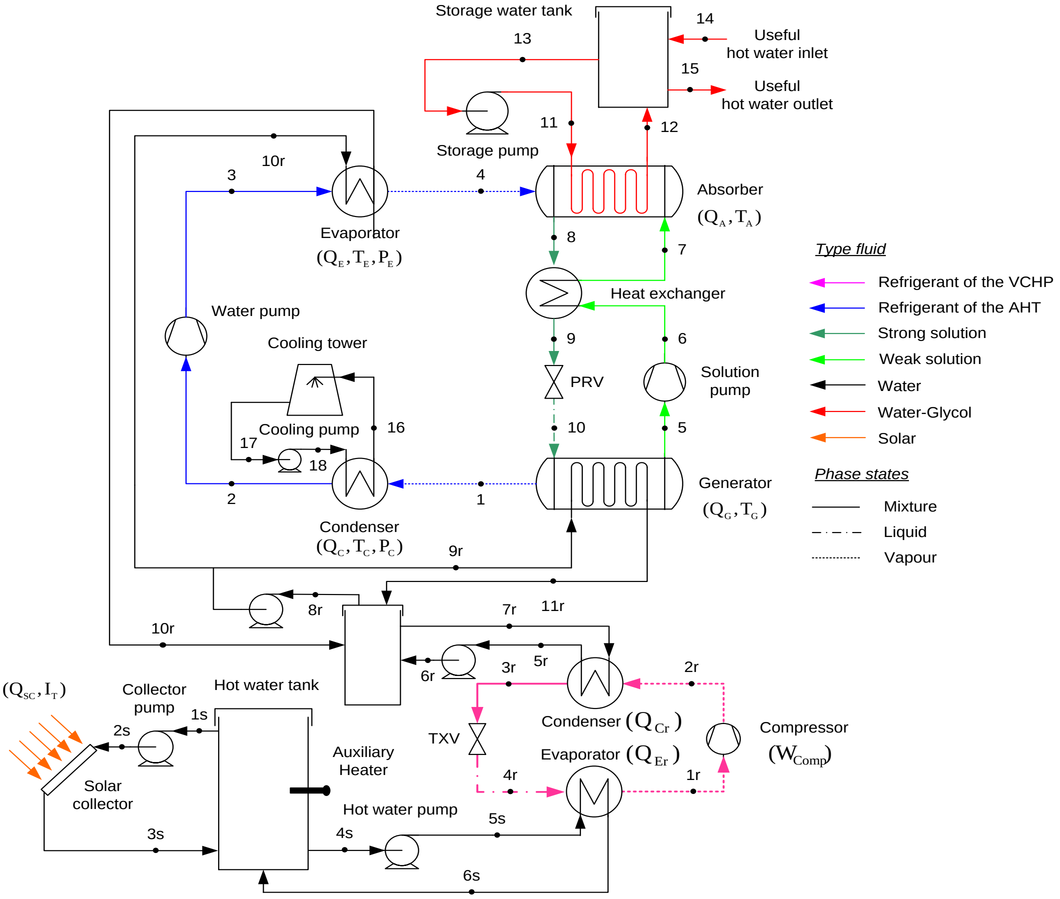


Figure 2. Schematic diagram of a solar absorption heat transformer.

III. SELECTION WORKING FLUID OF THE VCHP

For the method to select the suitable working fluid of the VCHP, the mathematical simulation has been used [6]. For the VCHP, the main components are compressor, condenser, evaporator and expansion valve as shown in Fig.2. At the VCHP evaporator, the working fluid in liquid phase is boiled at a low pressure and temperature to be vapor at state 1r. After that, the fluid in vapor phase is compressed in the compressor to state 2r and the vapor condenses in the VCHP condenser at a high pressure and temperature to be liquid at state 3r. The liquid is then throttled to a low pressure at state 4r and the temperature drops down thus the fluid could absorbed low temperature heat at the VCHP evaporator again and the new cycle restarts. The basic equations for using to simulate the behavior of each component in the VCHP cycle are as follows:

- Evaporator$_r$

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

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

- Compressor$_r$

$$W_{Comp} = \dot{m}_r (h_{2r} - h_{1r}), \tag{3}$$

$$s_{1r} = s_{2r} \quad \text{(Isentropic process)}, \tag{4}$$

$$\eta_{Comp} = \frac{h'_{2r} - h_{1r}}{h_{2r} - h_{1r}}. \tag{5}$$

- Condenser$_r$

$$Q_{Cr} = \dot{m}_r (h_{2r} - h_{3r}). \tag{6}$$

- Expansion valve$_r$

$$h_{3r} = h_{4r} \quad \text{(Throttling process)}. \tag{7}$$

- Coefficient of performance (COP)

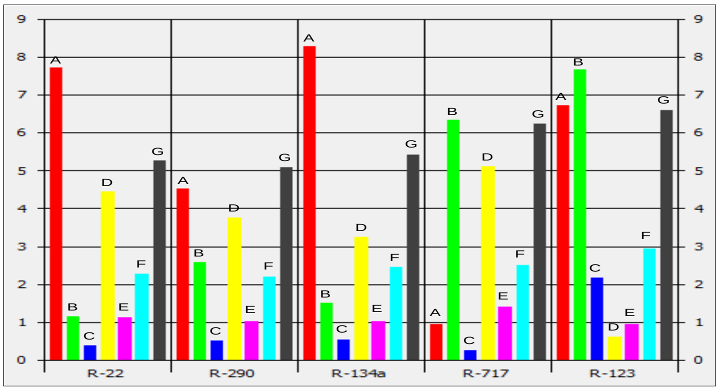
$$COP_{VCHP} = \frac{Q_{Cr}}{W_{Comp}}. \tag{8}$$

Table 1. Physical properties of working fluids.

Working Fluid	R-22	R-290	R-134a	R-717	R-123
Chemical formulae	$CHClF_2$	C_3H_8	CF_3CH_2F	NH_3	$CHCl_2CF_3$
Molecular mass (kg/kmol)	86.46	44.10	102.03	17.03	152.93
Critical temperature (^oC)	96.14	96.68	101.06	132.25	183.68
Critical pressure (MPa)	4.99	4.25	4.06	11.33	3.66
Critical density (kg/m^3)	523.84	218.50	511.90	225.00	550.00
Boiling point (^oC)	-40.81	-42.09	-26.07	-33.33	27.82
Latent heat of vaporization at 40 ^oC (kJ/kg)	164.24	302.30	160.88	1089.82	164.04
Flammability	NO	YES	NO	YES	NO
Toxicity	NO	NO	NO	YES	YES
ALT (Year, Atmosphere Life Time)	13.3	< 1	14	< 1	1.4
ODP (CO_2-related, Ozone Depletion Potential)	0.034	~0	0.0015	~0	0.02
GWP (100 Years, Global Warming Potential)	1780	0	1320	0	76

Five working fluids, R-22 (Chlorodifluoromethane), R-290 (Propane), R-134a (1,1,1,2-Tetrafluoroethane), R-717 (Ammonia) and R-123 (2,2-Dichloro-1,1,1-trifluoroethane) have been considered as working fluid in the VCHP. Table 1 shows physical properties of the working fluids [6]. The working conditions for the evaluation are:

1. The VCHP evaporator temperature (T_{Er}) is at 40 ^oC.
2. Total cooling capacity (Q_{Er}) is 10 kW.
3. The VCHP condenser temperature (T_{Cr}) is at 90 ^oC
4. No pressure drops at the VCHP condenser and the VCHP evaporator.
5. Isentropic efficiency of compressor (η_{Comp}) is 80%.
6. Degree of superheating (SH) is 5 ^oC.
7. Degree of subcooling (SC) is 5 ^oC.
8. The properties of working fluids are based upon REFPROP [6].



A) Mass of refrigerant per unit heat output, (g/kJ)
B) Vapor volume flow rate, (10^{-2} m3/kg)
C) Displacement volume, (10 m^3/h)
D) Discharge pressure, (10 bar)
E) Discharge temperature, (10^2 ^oC)
F) Pressure ratio, (-)
G) COP_{hp}, (-).

Figure 3. The results for the selected refrigerants.

The indicators used to identify the appropriate working fluid are mass of refrigerant per unit heat output, volume flow rate of refrigerant, high-side pressure, refrigerant temperature at the compressor outlet, pressure ratio and heating COP. Fig.3 shows the results of the selected refrigerants.

Form the simulation results, it could be seen that R-123 gives the suitable refrigerant in terms of energy consumption for the heat pump for generating heat at about 70-80 ^oC due to its low maximum pressure for the heat pump compressor, the cycle pressure ratio is not high and highest COP is obtained.

IV. Experimental Procedures and Simplified Model

For the experimental procedures, the constructed of solar water heating system combined with the vapor compression heat pump cascaded with the absorption heat pump AHT is tested its thermal performances to upgrade heat from the installed flat-plate solar collector. The objective of this experiment is to find out a simplified model which is the correlation between the input parameters and the thermal efficiency of the VCHP and the AHT. For the correlation model, thermal performance could be predicted under various operating conditions and decreased the complicated simulation compared with the old procedure too.

For the solar water heating system, a set of 10 unit solar collectors each in parallel connection and an auxiliary heater of 10 kW were integrated with a 1,500 liter of hot water tank for supplying heat to the absorption system at temperature around 40-60 ^{o}C. The description of each components of the solar water heating system are shown in Table 2.

Table 2. The description of the main components of the solar water heating system.

Component	Type	Specification
1. Solar collector	Flat-plate solar collector	• Area 2.3 m^2/unit • 10 units • $F_R(\tau\alpha)$ = 0.802 • F_RU_L= 10.37 W/m^2.K
2. Hot water tank	Vertical tank	• Capacity 1,500 liter • Thickness of insulator 1 in
3. Double tube heater	Water heater	• Double tube heat exchanger • Capacity 10 kW • Thickness of insulator 0.5 in

For the VCHP system, hot water temperature form solar water heating system is upgraded by 2 units of R-123 vapor compression heat pumps each having a heating capacity of 10 kW. Hot water at a higher temperature of around 60-80 ^{o}C (system could be increased hot water temperature around 20 ^{o}C) is produced and kept in a 200 liter hot water tank. The descriptions of the heat pump components are given in Table 2 and Fig.4 also shows the R-123 heat pump.

Table 3. The descriptions of the 10 kW heat pump components.

Component	Type	Specification
Compressor	Scroll compressor	Power input 1.50 A Displacement volume 12.7 m^3/h
Evaporator	Plate heat exchanger	Capacity 8.00 kW Area 1.64 m^2
Condenser	Plate heat exchanger	Capacity 10.00 kW Area 1.64 m^2
Expansion valve	Thermo static orifice 02	Capacity 10.00 kW Pressure ratio 3.00

Figure 4. 10 kW of R-123 vapor compression heat pump.

For the AHT system, a 10 kW water-Libr absorption heat pump upgrades the final hot water temperature form the VCHP system to be around 90-110 ^{o}C (system could be increased temperature around 20-30 ^{o}C) and keeps in a 200 liter tank. Since the water temperature might be over the boiling point then glycol is mixed in the water with a concentration of around 40%. The descriptions of the absorption heat transformer components are shown in Table 4 and Fig.5 shows the assembly of the absorption system.

Table 4. The description of the 10 kW absorption heat transformer.

Component	Type	Specification
1. Generator	Flooded shell and tube heat exchanger	• Capacity 10.3 kW • Weak solution 50 %LiBr • Strong solution 55 %LiBr • Generator temperature 85 ^{o}C • Tube diameter 4/8 in

Component	Type	Specification
		• Number of Tube passes 4 • Length 1.24 m • Area 1.02 m^2
2. Condenser	Shell and tube heat exchanger	• Capacity 10.6 kW • Condenser temperature 55 ^oC • Tube diameter 4/8 in • Number of Tube passes 2 • Length 1.01 m • Area 0.42 m^2
3. Absorber	Flooded shell and tube heat exchanger	• Capacity 10 kW • Weak solution 50 %LiBr • Strong solution 55 %LiBr • Absorber temperature 115 ^oC • Tube diameter 3/4 in • Number of Tube passes 6 • Length 1.1 m • Area 1.44 m^2
4. Evaporator	Shell and tube heat exchanger	• Capacity 10.8 kW • Evaporator temperature 85 ^oC • Tube diameter 4/8 in • Number of Tube passes 9 • Length 0.94 m • Area 1.16 m^2
5. Pressure relief device	Orifice type	• Capacity 10 kW • Pressure ratio 6.00
6. Lithium bromide	-	• Main content 50-55% • Light yellow transparent liquid • Chloride = 0.05% max • Sulphate = 0.05% max • Bromate = Non reaction • Ca = 0.0001% max • Mg = 0.0001% max • Na = 0.03% max • PH = 9.0-10.5 • Lithium chromate = 0.2-0.3%
7. Solution pump	Inline pump	• Flow rate 0.6-3.7 m^3/h • Maximum head 6 m • Maximum temperature 110 ^oC • Maximum pressure 10 bar • Capacity 78 W • Current 0.34 A • Voltage 230 V

Figure 5. The prototype of absorption heat transformer.

Mathematical correlations of the related parameters from the experimental data is set up and used to predict the thermal performance under various operating conditions.

For the VCHP system, the mathematical model shows the related data between an energy efficiency ratio (EER_{VCHP}) and the different temperature of the entering water temperature at the VCHP condenser and the VCHP evaporator. This correlation is called performance curve and used to predict the thermal performance of the VCHP as shown in Fig.6.

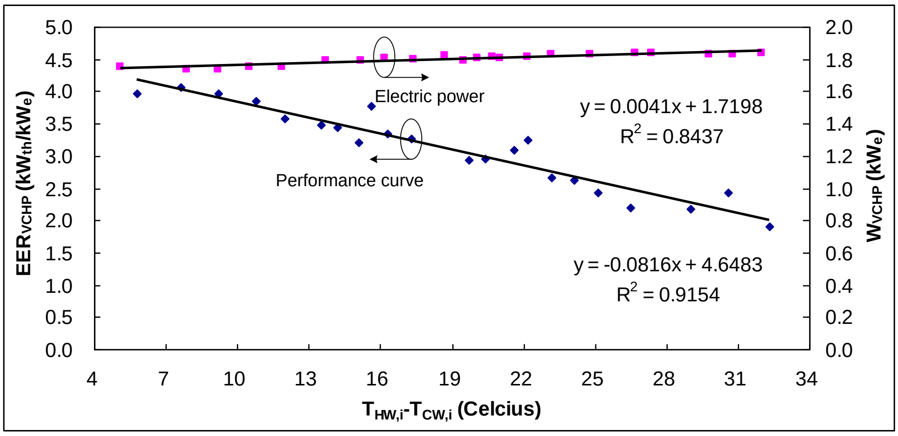


Figure 6. Performance curve of the 10 kW of R-123 vapor compression heat pump.

Fig.6 shows the correlations between thermal efficiency in term of EER_{VCHP} and the entering hot water temperature at the VCHP and the VCHP condenser. It could be found that when increased the hot water temperature different effects the EER_{VCHP} reduced because the electrical power consumption of the compressor is increased at nearly constants of heating capacity at the VCHP condenser. The mathematical model of the heat pump performances are shown as follow:

$$EER_{VCHP} = -0.0816(T_{HW,i} - T_{CW,i}) + 4.6483, \ (kW_{th}/kW_e), \quad (9)$$
$$W_{VCHP} = 0.0041(T_{HW,i} - T_{CW,i}) + 1.7193, \ (kW_e). \quad (10)$$

Fig.7 shows EER_{AHT} with $(T_{A,i} - T_E)/(T_{G,i} - T_C)$ when water in the storage tank (AHT side) is used and non-used. In both cases, use and non-use of hot water, when the value of $(T_{A,i} - T_E)/(T_{G,i} - T_C)$ increases the COP_{AHT} and the EER_{AHT} decreased due to lower extracted heat at the absorber. When hot water is used, the COP_{AHT} and EER_{AHT} are higher than those of another case since the hot water temperature in the storage tank is lower thus the absorption could supply more heat. The empirical correlations of the COP_{AHT} with $(T_{A,i} - T_E)/(T_{G,i} - T_C)$ for both cases could be:

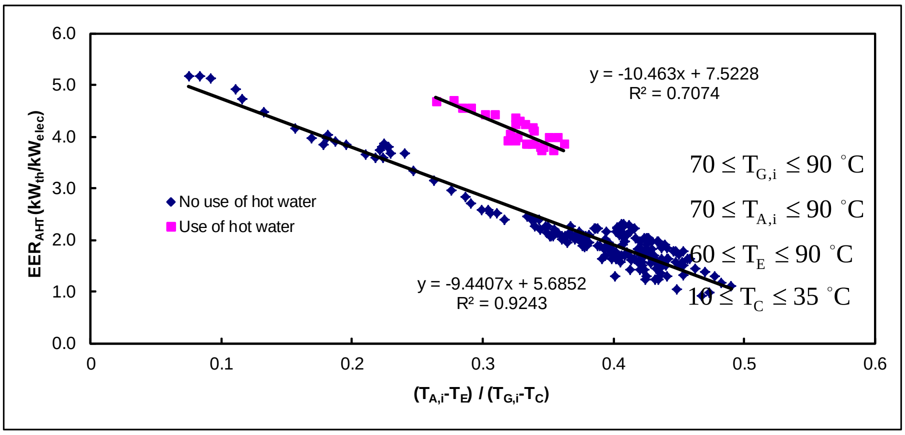


Figure 7. Effect of $(T_{A,i} - T_E)/(T_{G,i} - T_C)$ on COP_{AHT} of the CAHT at hot water temperature leaving the AHT around 100 °C from the experimental results.

For used hot water condition:
$$EER_{AHT} = -10.463(T_{A,i}-T_E)/(T_{G,i}-T_C) + 7.5228. \quad (11)$$
For non-used hot water condition:
$$EER_{AHT} = -12.577(T_{A,i}-T_E)/(T_{G,i}-T_C) + 6.7079. \quad (12)$$
These equations are valid for the following criteria:
$$70 \le T_{G,i} \le 90\,^{\circ}C,\ 70 \le T_{G,i} \le 90\,^{\circ}C,$$
$$60 \le T_E \le 90\,^{\circ}C,\ 10 \le T_C \le 35\,^{\circ}C. \quad (13)$$

Fig.8 and Fig.9 show steps of calculation for the analyses of the VCHP and the AHT cycles with the simplified models. Performance correlations of the EER and the electrical power consumption with the operating temperatures are given. With the input data which are the operating conditions, the upgraded temperature leaving the VCHP condenser and that leaving the AHT absorber are the outputs of the calculations, respectively.

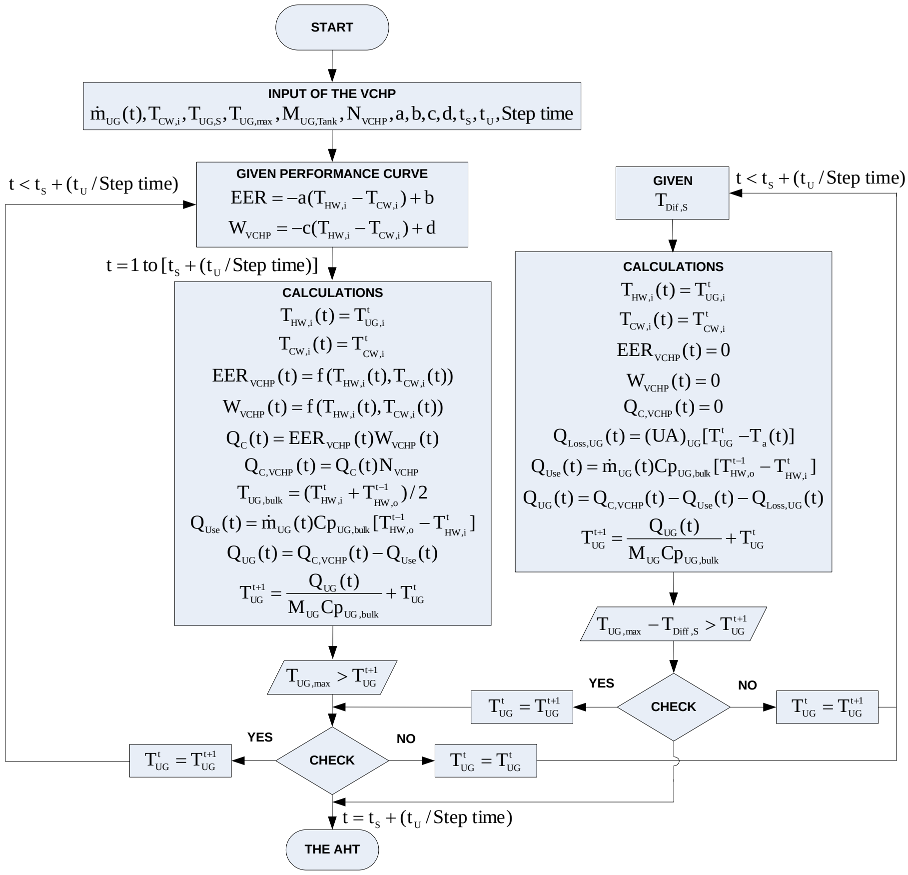


Figure 8. Flow chart for simulation of the vapor compression heat pump by using performance curve.

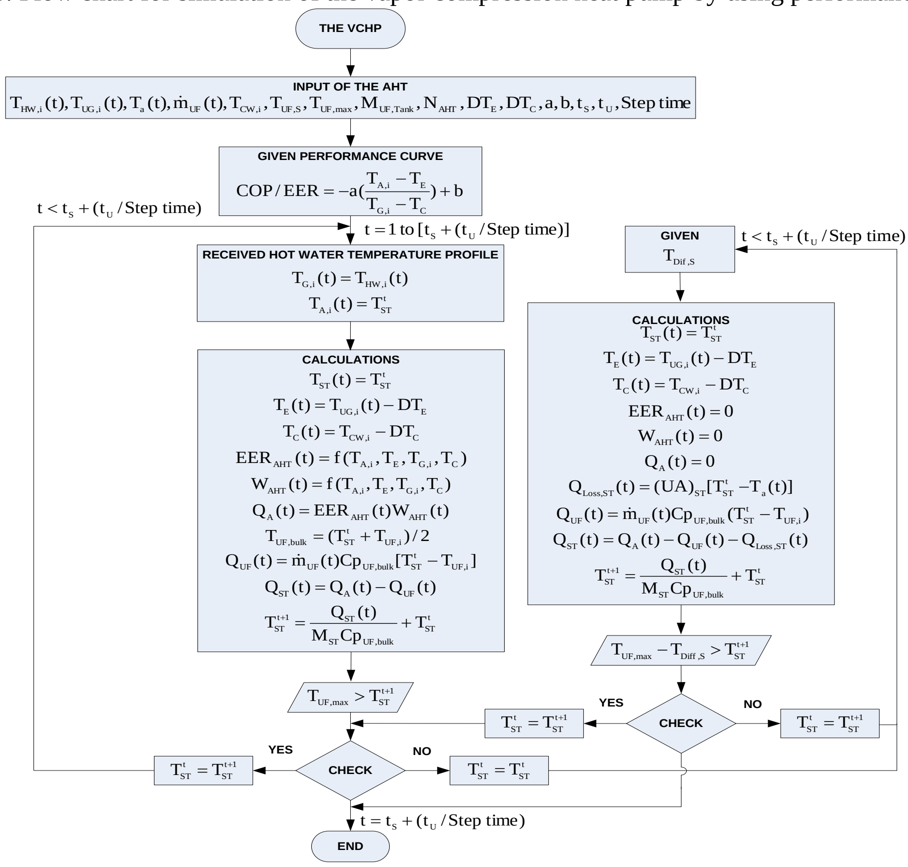


Figure 9. Flow chart for simulation of the absorption heat transformer by using performance curve.

V. Results and Discussions

The modified system of the solar water heating system combined with the VCHP system cascade with the AHT system as described above was tested. The EER of its VCHP was evaluated when the hot water of 200 liter at the AHT absorber was used at a heating capacity around 10 kW_{th}. The result was shown in Fig.10. It could be seen that the simulation results agreed well with those of the experimental data. Fig.10 also showed the water temperatures leaving the VCHP condenser and EER_{VCHP}. In this figure, hot water temperature and the EER_{VCHP} were nearly constant around 60 ^{o}C and 3.1, respectively.

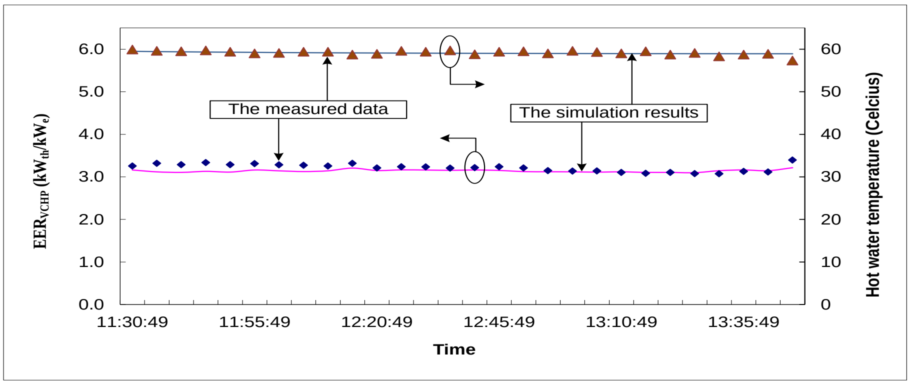


Figure 10. Comparison results of the measured data and the simulation results of hot water temperature from the R-123 VCHP system at flow rate 0.031 l/s (hot water is used at tank 200 liter)

Fig.11 shows the simulated results of the EER_{AHT} when the R-123 VCHP system is coupled with the AHT system. Since the water temperature might be over the boiling point then glycol was mixed in the water with a concentration of around 40% by mass. In this case, the generated hot water was non-used. It could be seen that the simulated results agreed well with the measured data. The hot water temperature also affected the EER_{AHT} which as the temperature increased the EER decreased. Fig.12 also shows the EER_{AHT} when the generated hot water is used at a flow rate of 0.024 l/s. It could be seen that when the hot water was used, the EER_{AHT} was higher than that of the non-used hot water because the water temperature in the storage tank was lower than the system could supply more heat rate. In this Figure, the hot water temperature in the storage tank was nearly constant at 90 $^{\circ}$C and the EER_{AHT} was nearly constant at around 4.1. The simulated results agreed well with the measured data.

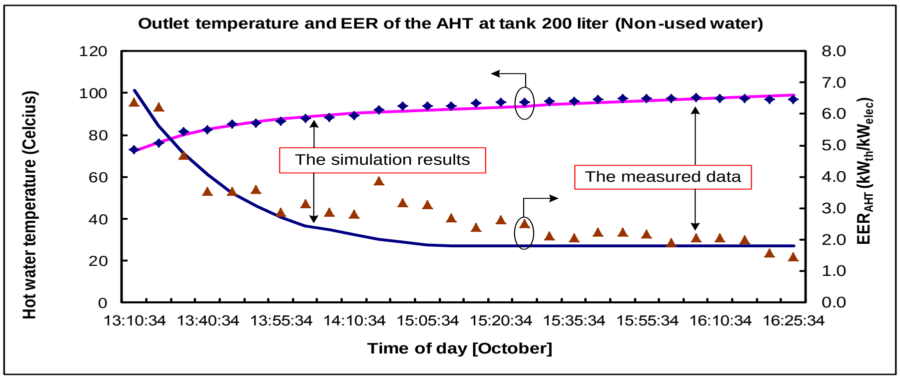


Figure 11. Comparison results of the measured data and the simulation results of the water-glycol solution temperature from the AHT system (hot water is not used at tank 200 liter)

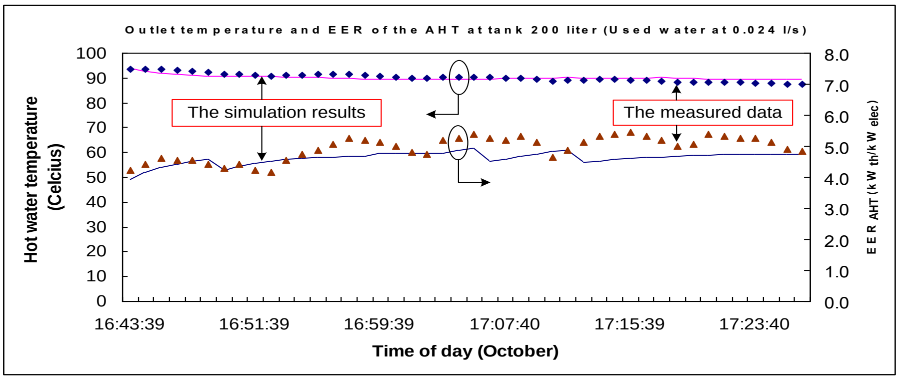


Figure 12. Comparison results of the measured data and the simulation results of water-glycol solution temperature from the AHT system at flow rate 0.024 l/s (hot water is used at tank 200 liter)

For the economic results, the modified system is used to partial support a boiler for generating hot water in a hospital. For the hospital, the boiler is normally used to generate heat in a form of steam at temperature higher 120 °C. For the some processes, the using steam temperature does not exceed 90 °C such as a drying process. Thus the modified is used to support the boiler for reducing the fossil fuel and the green house emission form fossil combustion.
The flue rate of the hospital in Chiang Mai, Thailand is selected for the simulation. A boiler using diesel and heavy oil at around 326 l/d and 2,523 l/d, respectively, is taken to generate steam at temperature around 150 °C. The conditions for the simulation are as follow:

- Operating period 15 h/d.
- Profile of hot water consumption as in **Error! Reference source not found.**
- Initial temperature of hot water ($T_{HW,S}$) in a storage tank is at 30 °C and the maximum temperature is at 85 °C.
- The rate of hot water consumption is around 35,000 l/d.
- Fill-in water temperature ($T_{Sup,i}$) is at 27 °C.

Table 5. The fuel rate of the hospital in Chiang Mai, Thailand [7].

Data	Diesel (liter)	Heavy oil (liter)
Average (l/d)	326	2,523
Total (l/d)	2,848	
Fraction of fuel (Diesel/Heavy oil)	13.08%	
Average (l/m)	10,100	78,200
Total (l/m)	88,300	

In this study, the modified unit is conducted to work with the boiler as described above to reduce the fossil fuel. Fig.13 shows the schematic skate of the boiler to generate steam 35 m^3/d. Fig.14 shows the schematic skate of the modified system operating with the boiler to produce steam and hot water at 30 m^3/d and 5 m^3/d, respectively. The economic results of the modified system shows in Table 6.

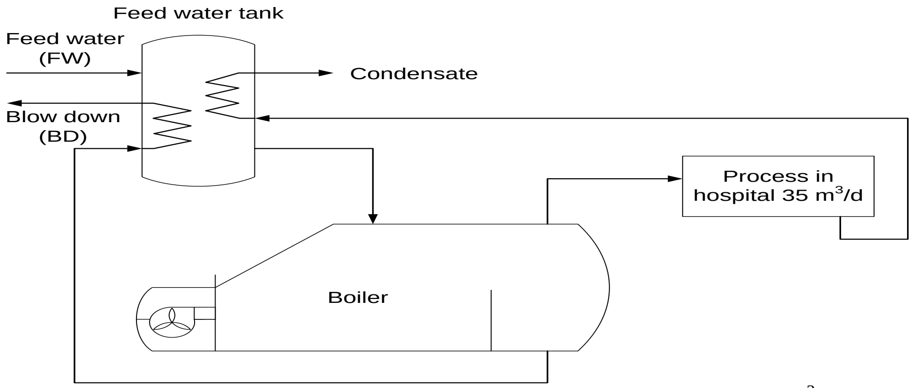


Figure 13. Schematic skate of the steam generation by boiler at 35 m^3/d.

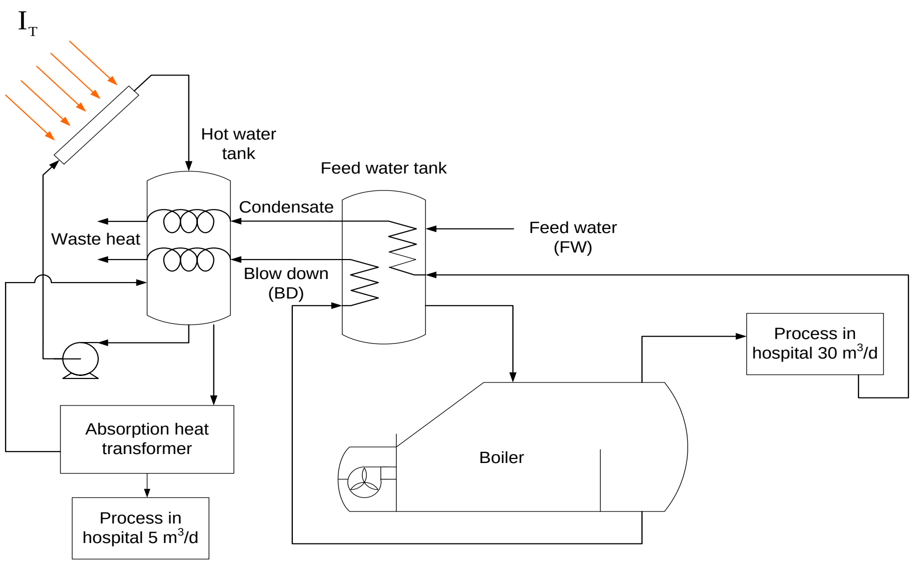


Figure 14. Schematic skate of the steam and hot water generation by boiler and the modified system at 30 m^3/h and 5 m^3/h, respectively.

Table 6. The economics results

Descriptions	The normal unit	The modified unit
Working time (h/d)	15	15
• On-peak period (9.00 - 22.00, h/d)	8	8
• Off-peak period (22.00 - 9.00, h/d)	4	4
Fuel type		
• Diesel (l/d)	373	316
• Heavy oil (l/d)	2,475	2,097
• Total (l/d)	2,848	2,413
Cost of fuel (Baht/y)	11,306,123	8,288,9906
The electrical cost [8] (Baht/y)	38,477.33	380,176.09
Cost of solar collector at 2 m^2/unit (36 units, Baht)	-	900,000
Cost of the VCHP system 20 kW (2 units, Baht)	-	1,000,000
Cost of the AHT system 10 kW (2 units, Baht)	-	1,000,000
Payback period (y)	1.12	

Note: 1 USD = 30.6535 Baht

Table 6 shows the economic results of the method to upgrade a low temperature heat form solar energy by the VCHP system cascade with the AHT system to generate heat partial the boiler. It could be seen that payback period of the modified system is around 1 y 2 m. For increasing the load form 10,000-35,000 l/d at temperature around 85 $^{\circ}$C, it could be found that the payback is longer with the higher load because the saving cost at the high load is less than the investment cost compared with the lower load as shown in Fig.15.

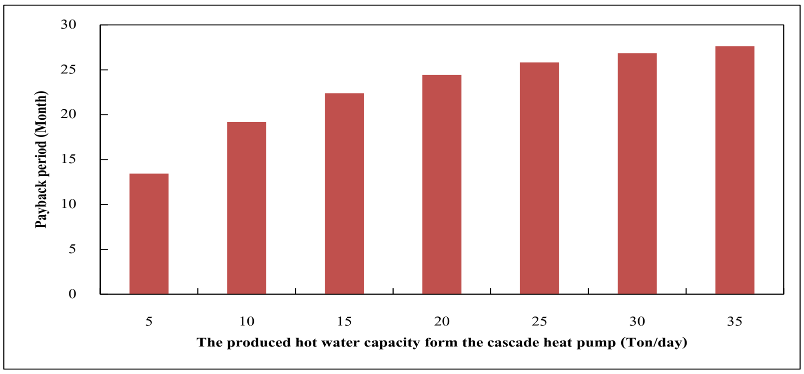


Figure 15. Comparison results of the payback period of the modified system for varying the hot water load at 10,000-35,000 l/d

VI. CONCLUSION

From this study, the conclusions are as follows:

1. The modified system could be upgraded hot water temperature around 50 ^{o}C which increases solar heat form 40-60 ^{o}C to be around 90-110 ^{o}C of the final hot water temperature.
2. The prediction results from performance curve of the modified system could be simulated the system performance such as the system EER and the final hot water temperature of the models to be close with those of the experimental results.
3. From the economic results, the modified system was used to partially support a boiler for generating hot water at 5 Ton/d compared up to fully support at 35 Ton/d. For the partially support, the energy saving and the payback period for the modified system were around 2,675,434 Baht/y and 1 year 2 months, respectively. The payback was longer with the higher load of the system.

ACKNOWLEDGEMENTS

The authors would like to thank School of Renewable Energy, Maejo University and Thermal System Research Unit, Chiang Mai University for supporting testing facilities.

NOMENCLATURE

A	Area, (m^2)
C_p	Heat capacity, (kJ/kg·K)
COP	Coefficient of performance
EER	Energy efficiency ratio, (kW_{th}/kW_e)
h	Enthalpy, (kJ/kg)
I_T	Solar radiation, (W/m^2)
m	Mass flow rate, (kg/s)

P	Pressure, (bar)
Q	Heat rate, (kW)
s	Entropy, (kJ/kg·K)
SC	Subcooling, (°C)
SH	Superheating, (°C)
t	Time, (s)
T	Temperature, (°C)
U	Overall heat transfer coefficient, ($W/m^2·K$)
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	Electric
E	Evaporator
G	Generator
H	High
HS	Heat source
HW	Hot water
HX	Heat exchanger
i	Inlet
L	Low
max	Maximum
min	Minimum
o	Outlet
r	Refrigerant
S	Start
SC	Solar collector
ST	Storage tank
Sup	Supply
th	Thermal
U	Stop using time
UF	Useful
UG	Upgraded

REFERENCES

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

[2] Florides, GA, Kalogirou, SA, Tassou, SA,Wrobel, LC. Modeling and Simulation of an Absorption Solar Cooling System for Cyprus. *Solar Energy, 72*, 2002, 43-51.

[3] Xuehu, Ma, Jiabin, C, Songping Li, Qingyun, Sha, Aiming, L, Wei, Li, Jiayan, Z, Guojun, Z, Zhihao, F. Application of absorption heat transformer to recover waste heat from a synthetic rubber plant. *Apply Thermal Energy, 25*, 2002, 797-806.

[4] Chaiyat, N, and Kiatsiriroat, T. Improvement of an Absorption Heat Transformer Performance for Upgrading Low Temperature Heat by Coupling with a Vapor Compression Heat Pump, *Chiang Mai University Journal of Natural Sciences, 10(2)*, 2011, pp.315-333.

[5] Chaiyat, N, and Kiatsiriroat, T. Upgrading of Low Temperature Solar Heat for Medium Temperature Applications by a Solar-Absorption Heat Transformer Assisting with a Vapor Compression Heat Pump, *Proc. 1st Conf. on International Symposium on Low Carbon & Renewable Energy Technology (ISLCT2010)*, Korea, 15-18 November 2010, 150.

[6] Chaiyat, N. and Chaichana, C. Drying Room from Geothermal Energy, *Proc. 8th Conf. on Heat and Mass Transfer in Thermal Equipments*, Thailand, 12-14 March 2008, 87-91

[7] Maharaj Nakorn Chiang Mai Hospital. The fuel rate of boiler, Chiang Mai University, Thailand, 2013.

[8] Provincial Electricity Authority. The rate of electricity cost. Online, http://www.pea.co.th, 2013.