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Full Length Research Paper

Upgrading of Low Temperature Heat with Absorption Heat Transformer for Generating Electricity by Organic Rankine Cycle

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This study presents a concept to generate electricity from low temperature heat by using an absorption heat transformer (AHT) coupled with an Organic Rankine Cycle (ORC). The prototypes of a 10 kW AHT system and a 20 kW ORC system were tested and evaluated the thermal performances which were used to predict the system performance of the modified system. From experiment, it could be seen that the H_2O-LiBr AHT could upgrade the final temperature to be around 90-110°C at coefficient of performance (COP) of 0.44. For the R-245fa ORC system, it could be found that the system efficiency was around 8.5% at supplied water temperature higher than 90°C. From prediction results, it could be shown that a 250 kW AHT system was selected to generate electricity with the 20 kW ORC system. For economic results, saving cost and payback period of the integrated unit based on the electricity cost of Thailand were 5,945.82 USD/y and 15.96 y, respectively. The value of levelized electricity costs (LEC) of the modified system was 0.547 USD/kWh. From the above results, it could be concluded that electricity process by using the AHT and ORC systems to upgrade and transfer heat to power was beneficial in term of energy efficiency, but was deficient in term of economic results.

Keywords: Absorption heat transformer; Organic Rankin Cycle; Thermal performance; Low temperature heat; Simplified model

INTRODUCTION

An absorption heat transformer (AHT) is one method for upgrading a low temperature heat to a higher temperature level. In the conventional AHT, low temperature heat is absorbed at the generator and the evaporator while high temperature heat is delivered at the absorber and there is waste heat rejected at the

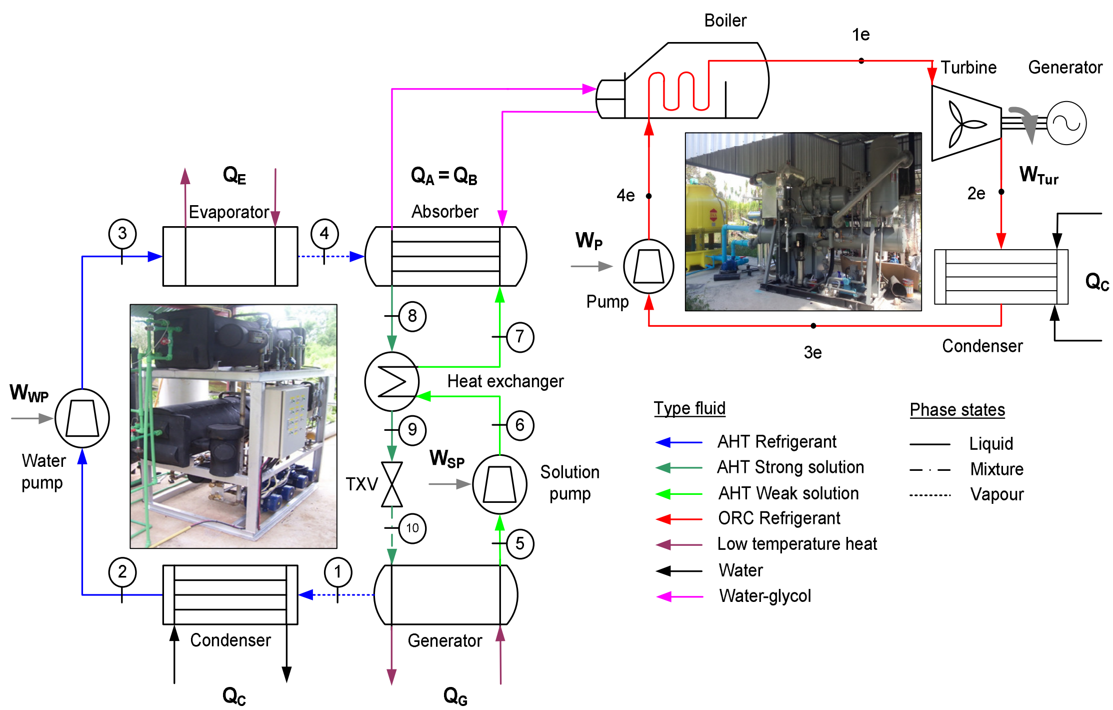

Figure 1. Schematic diagram of absorption heat transformer combined with Organic Rankine Cycle.

condenser. Studies on energy efficiency of the AHT have been reported by various literatures. Kiatsiriroat et al. (1986), Horuz and Kurt (2010), Xuehu et al. (2002) reported thermal performance of H_2O-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 condenser. Sotsil et al (2009). and Zhang and Hu (2012) presented heat transformer absorption cycle operating with water-Carrol^{TM} mixture which had a higher solubility than H_2O-LiBr mixture. It could be found that the COP was higher and less crystallization risk was obtained. Sozen (2003) reported performance of AHT that was used to increase a solar pond's temperature. It could be seen that the exergy losses of the AHT were high at the absorber and the generator. The COP and the maximum temperature were around 0.4 and 150 $^{\circ}$C, respectively. Sencan et al. (2005), Rivara et al. (2009), Donnellan et al. (2013-2014) and Khamooshi et al. (2004) conducted first and second laws of absorption heat transformer at COP and ECOP did not higher than 0.25 and 0.38, respectively. Chaiyat and Kiatsiriroat (2011) reported a novel of absorption heat transformer coupling with R-123 single-stage vapor compression heat pump. The overall COP of modified system could be increased to be around 0.8. The technique of Chaiyat and Kiatsiriroat (2011) was similarly with Junga et at. (2014), Nordtvedt (2005) and Van et al. (2014), which represented a Compression/Absorption Heat Pump (CAHP) and those system used compressor in the ammonia-water absorption cycle.

From the above reviews, it could be noted that the AHT could upgrade low and medium temperature mediums. But in electricity process, the AHT system coupled with an Organic Rankine Cycle (ORC) was not presented in the recent resources as referred from Hettiarachchi et al. (2007), Schuster et al. (2005), Guo et al. (2011), Sauret et al. (2011), Liu et al. (2012), Edrisi et al. (2013), Li et al. (2013) and Rodriguez et al. (2009), which reported the ORC system by using geothermal energy. There were

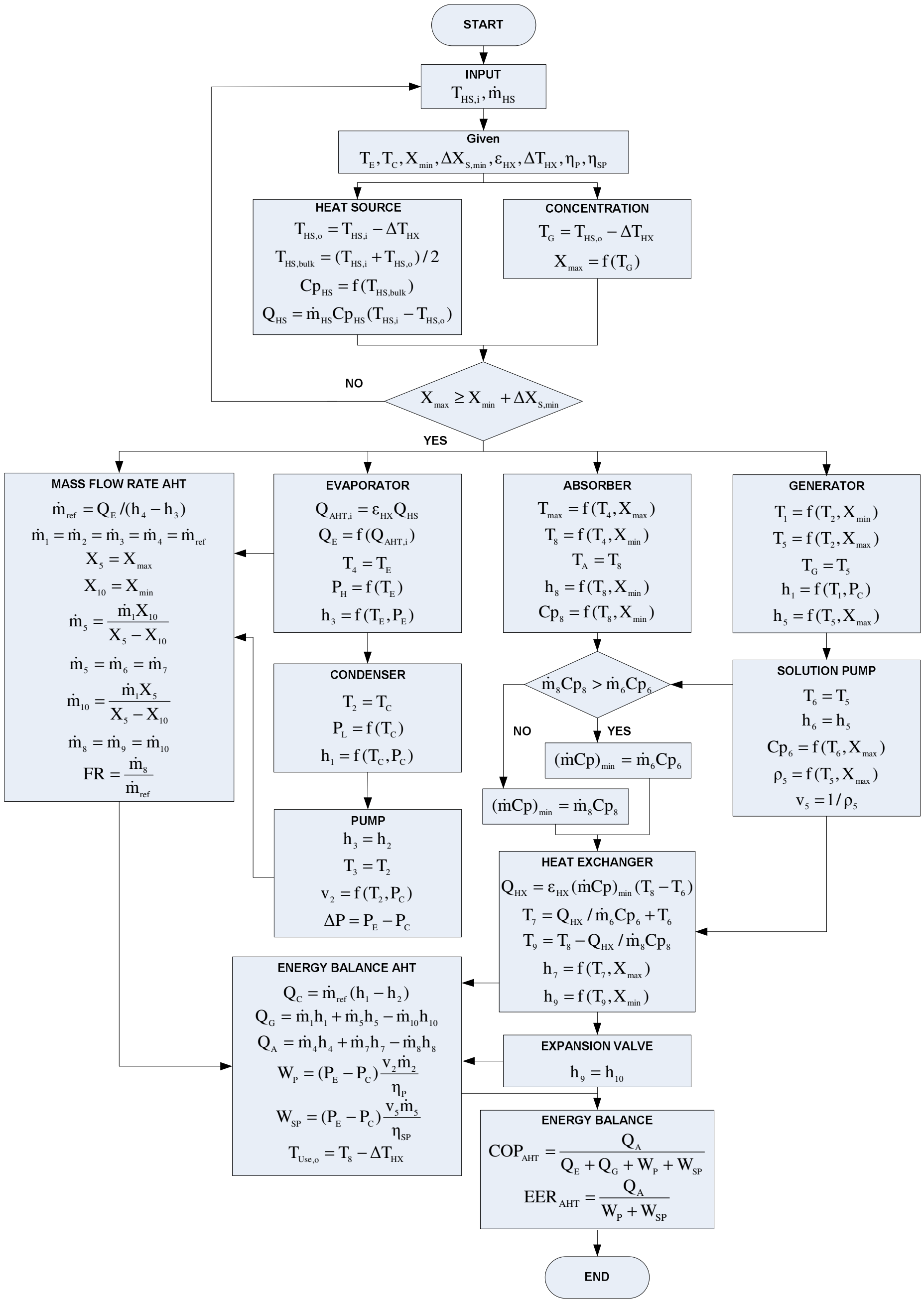

Figure 2. Flow chart of the simulation program for evaluating performance of the AHT.

some reports on the ORC with different heat source such as waste heat Hung (2001), solar thermal by Achary et al. (1983) and Jing et al. (2010), biomass by Drescher and Bruggemann (2007) and etc.

An interesting approach, a method to upgrade low temperature heat by the AHT system and supplied to the ORC system was considered. Input heats such as solar heat and waste heat could generate electricity at low temperature heat sources. The experimental results of each technology were performed to study the integrated system performances.

The aims of this study were as follows:

1. Performance analysis of the AHT and ORC systems based on the testing results.

Table 1. Description of main components of the AHT system.

Component	Type	Specification
Generator	Flooded shell and tube heat exchanger	• Capacity 10.3 kW • Heating area 1.02 m^2
Condenser	Shell and tube heat exchanger	• Capacity 10.6 kW • Heating area 0.42 m^2
Absorber	Flooded shell and tube heat exchanger	• Capacity 10 kW • Heating area 1.44 m^2
Evaporator	Shell and tube heat exchanger	• Capacity 10.8 kW • Heating area 1.16 m^2
Pressure relief device	Orifice type	• Capacity 10 kW • Pressure ratio 6.00
Lithium bromide	-	• Main content 50-55% • Light yellow transparent liquid
Solution pump	Inline pump	• Flow rate 0.6-3.7 m^3/h • Maximum pressure 10 bar • Capacity 78 W

Table 2. The description of elements operating with the ORC system.

Components	Descriptions
Output power	• Gross power 20 kW • Net power 16 kW • 3 Phase, 380 V, 50 Hz
Expander	• Semi-hermetic twin screw type expander • Displacement volume 3000 rpm
Evaporator	• SUS 316 plate type heat exchanger
Condenser	• Shell and tube heat exchanger • Shell: carbon steel 12 in x 3 m • Tube: 3/4 in copper tube
Oil separator	• Vertical type oil separator • Oil tank 18 in diameter 0.7 m
Oil pump	• Viking heavy duty oil pump GG4195 • Motor: 3 hp, 3 phase, 380 V, 50Hz
Liquid pump	• Vertical multi-stage centrifugal pump VFD drive • Motor: 2 hp, 3 phase, 380 V, 50 Hz

2. Systematic determination of optimum design parameters of the combined unit.

3. Economic results of the modified system in term of payback period.

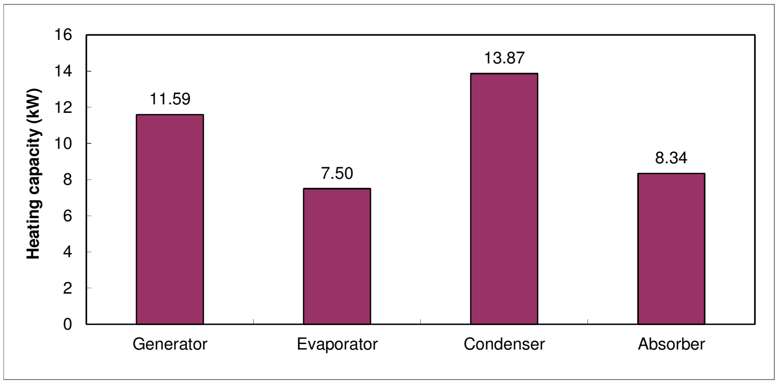

Figure 3. Heating capacities of the main components of the AHT system.

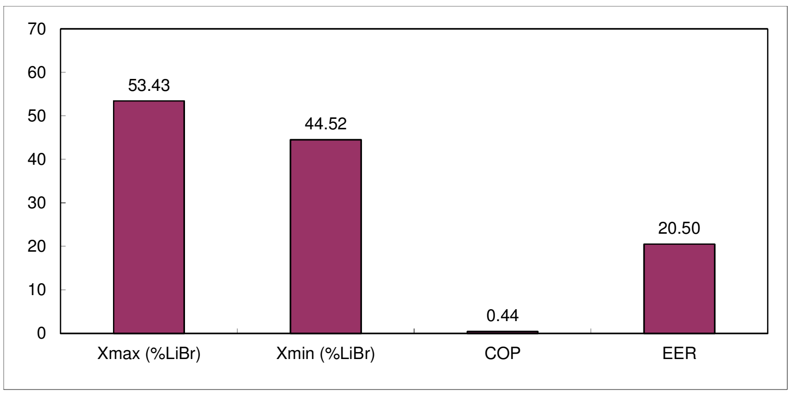

Figure 4. H_20-LiBr concentration and the cycle efficiency of the AHT system.

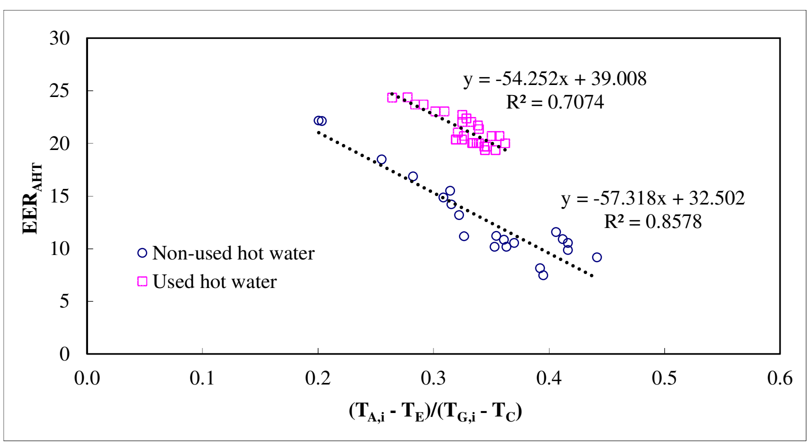

Figure 5. Simplified model of the AHT system.

It could be noted that based on the abovementioned literature review, there is no sufficient knowledge for this topics in recent literatures.

System Description

Fig. 1 shows a schematic sketch of a general AHT

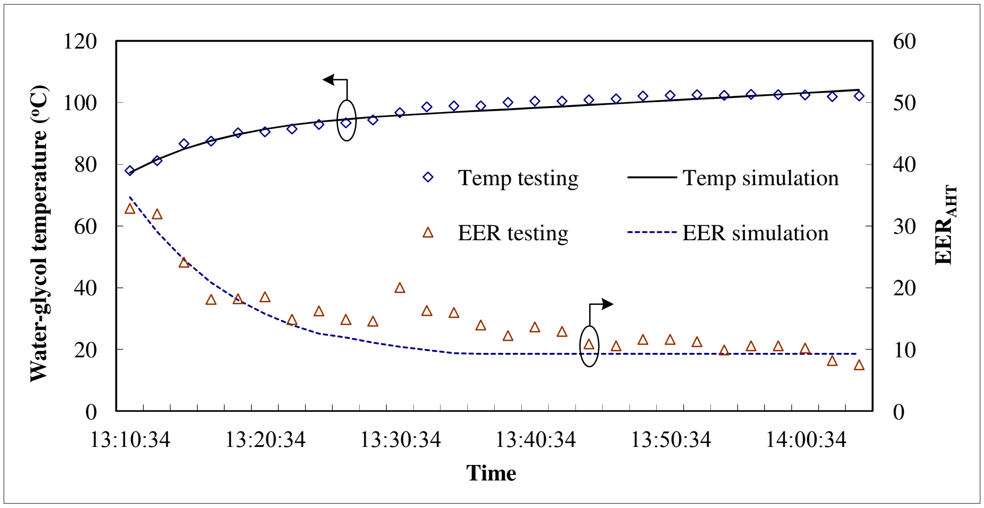

Figure 6. Comparison results of the measured data and the simulation results of the AHT system at non-used hot water.

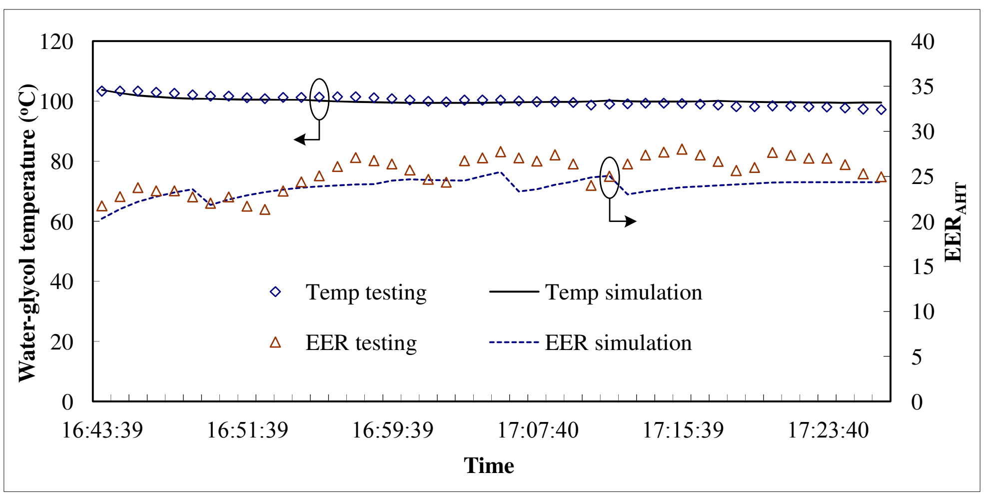

Figure 7. Comparison results of the measured data and the simulation results of the AHT system at used hot water.

system. Low temperature heat is supplied to the generator and the evaporator at temperature around 60-80^{o}C. At the 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 and a generator temperature at state 1. The vapor condenses in the condenser at a condensing temperature to be liquid at state 2 and rejected heat at a lower temperature around 25-40^{o}C. After that the absorbate in liquid phase is pumped to the evaporator at state 3 of which the pressure is higher than that of the condenser. The evaporator is heated at the evaporating temperature and the absorbate in a form of vapor enters the absorber which has the same pressure as the evaporator at state 4. Meanwhile liquid mixture from the generator, at state 5

Table 3 Testing results of the R-245fa ORC system at varying the inlet hot water temperature.

Descriptions	Data			
Hot water inlet ($T_{HW,i}$)	116	107.8	97	℃
Hot water outlet ($T_{HW,o}$)	89.8	81	75	℃
Heat source capacity (Q_B)	243.2	248.2	203.4	kW
Cool water inlet ($T_{CW,i}$)	28	28	28	℃
Cool water outlet ($T_{CW,o}$)	35	35	35	℃
Condenser temperature (T_C)	37.1	37	37	℃
Heat sink capacity (Q_C)	219.0	215.6	210.9	kW
Turbine inlet pressure (P_{High})	1,097.1	1,120.0	1,074.0	kPa-Abs
Turbine inlet temperature (T_{1e})	93.7	94.6	92.8	℃
Turbine outlet pressure (P_{Low})	227.4	227.4	227.0	kPa-Abs
Turbine outlet temperature (T_{2e})	75.0	70.6	70.6	℃
Pumping power (W_P)	1.78	1.90	1.19	kW
Oil power (W_{OP})	1.40	1.40	1.40	kW
Lift temperature ($T_{HW,i} – T_{CW,i}$)	88	79.8	69	℃
Gross power (W_{Tur})	21.50	21.36	16.70	kW
Cycle efficiency (η_{ORC})	8.73	8.49	8.11	%

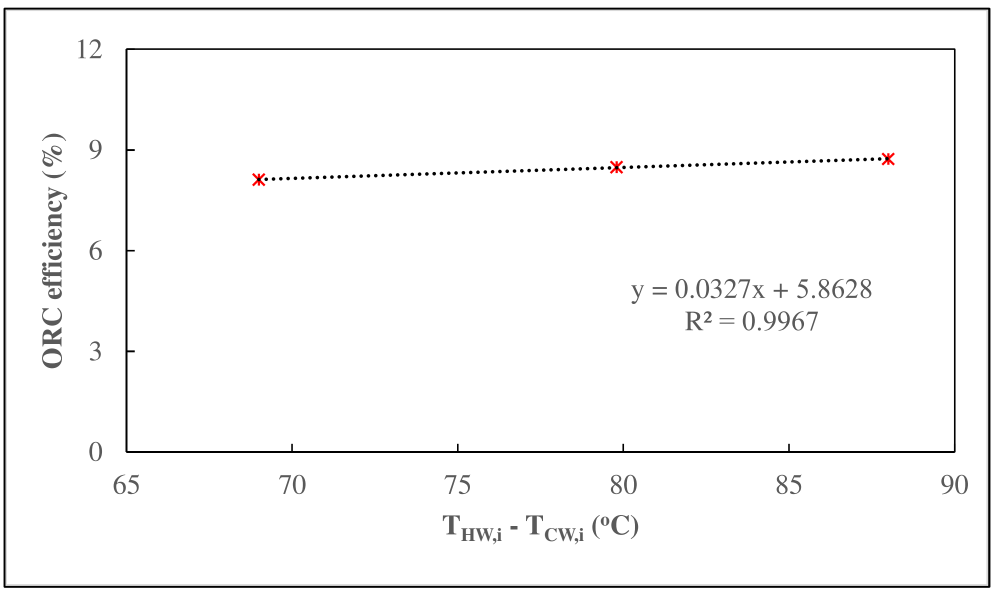

Figure 8. Simplified model of the ORC system.

is pumped through a heat exchanger (state 6) into the absorber to a high pressure at state 7. In the 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 which is higher than those at the generator and the evaporator (around 90-110^{o}C). This liberated heat is the useful output of the AHT system. The weak solution at state 8 from the absorber is then throttled to a low pressure through the heat exchanger at state 9 into the generator again at state 10 and new cycle restarts.

Fig. 1 also shows the AHT system combined with the ORC system in series connection. Upgraded heat from the AHT system is supplied to the ORC system at the boiler at temperature around 90-120°C. The working fluid at the boiler is boiled at the high pressure and temperature (state 1e). After that the working fluid in the vapor phase enters to the turbine for producing the

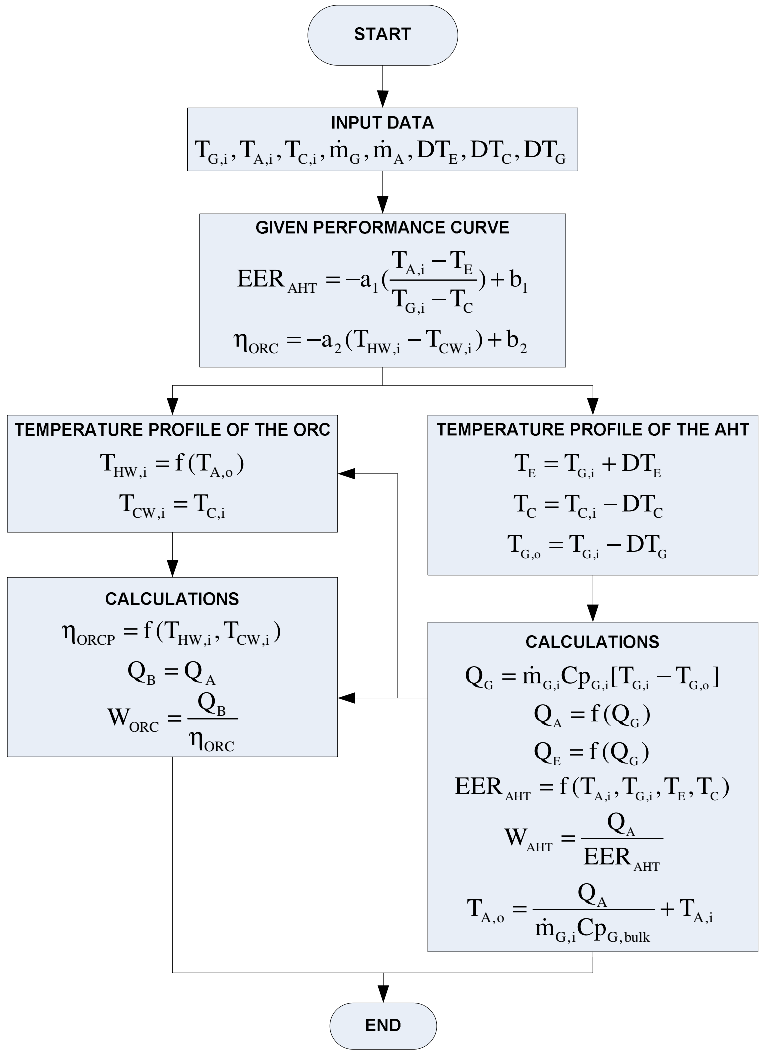

Fig. 9 Flow chart for simulation of the AHT and the ORC system by using performance curves.

electricity at the generator. Next, the working fluid at the low pressure (state 2e) is condensed at the condenser (state 3e) at the condensing temperature around 25-40°C. The working fluid in liquid phase is pumped (state 4e) to the boiler and the new cycle is started again.

The basic equations of the ORC system as shown in Figure 1 are as follows:

- Boiler

$$Q_B = \dot{m}_{ref}(h_{1e} - h_{4e}), \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

- Condenser

$$Q_C = \dot{m}_{ref}(h_{2e} - h_{3e}), \quad \ldots\ldots\ldots\ldots\ldots \quad (2)$$

- Pump

$$W_P = \dot{m}_{ref}(h_{4e} - h_{3e}), \quad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

- Turbine

$$W_{Tur} = \dot{m}_{ref}(h_{1e} - h_{2e}) \ldots\ldots\ldots, \ldots \quad (4)$$

- Isentropic efficiency of turbine

$$\eta_{s,Tur} = \frac{h_{1e} - h_{2e}}{h_{1e} - h_{2e,s}}, \quad \ldots\ldots\ldots\ldots\ldots \quad (5)$$

- Isentropic efficiency of pump

$$\eta_{s,P} = \frac{h_{4e,s} - h_{3e}}{h_{4e} - h_{3e}}, \ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

- System efficiency

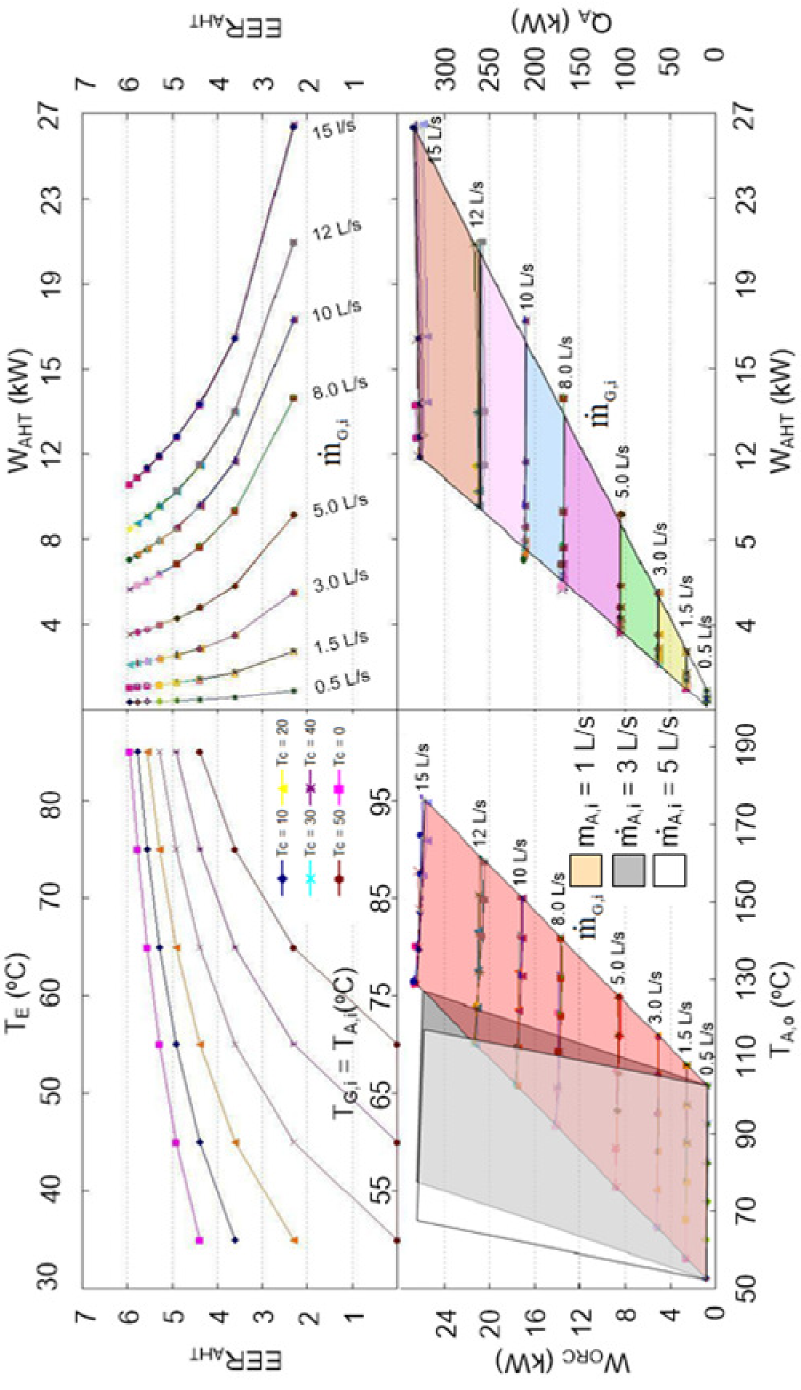

Figure 10. The prediction of the thermal performance of the AHT and the ORC systems.

$$\eta_{th} = \frac{W_{Tur}}{Q_B + W_P} \quad \text{.............................} \quad (7)$$

Fig. 2 shows steps for calculating the thermal performance of the AHT system as shown in Fig. 1.

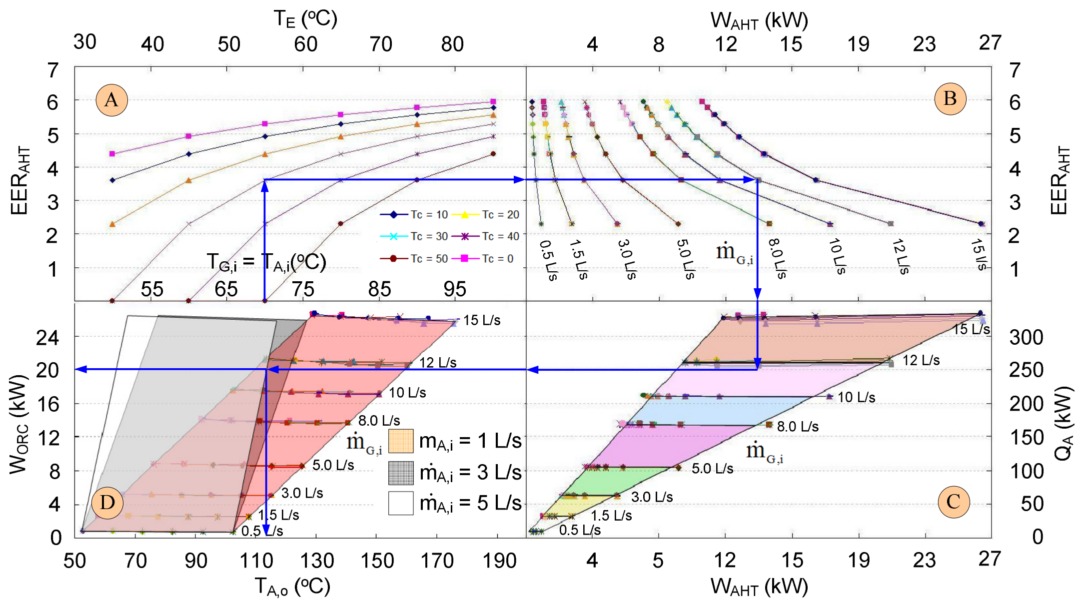

Figure 11. The prediction results for projecting the AHT performance to generate electricity 20 kW of the ORC system.

Table 4 The economics results

Descriptions	Normal system	Modified unit
Working time (h/d) - On-peak period (9.00-22.00, h/d) - Off-peak period (22.00-9.00, h/d)	 13 11	 13 11
The electrical cost - On-peak period at 9.00-22.00 o'clock (USD/y) - Off-peak period at 22.00-9.00 o'clock (USD/y)	 7,913.33 2,886.44	 4,055.58 1,479.30
Cost of the ORC system 20 kW_e (USD)	-	50,000.00
Cost of the AHT system 250 kW_{th} (USD) (Chaiyat, 2008)	-	44,878.57
Total capital cost (USD)	-	94,878.57
Saving cost (USD/y)	-	5,945.82
Payback period (y)	-	15.96
Levelized electricity cost (USD/kWh)	-	0.565

MATERIALS AND METHODS

For the experimental procedures, the constructed of the AHT and ORC units were tested its thermal performances to upgrade heat and generate electricity, respectively. The objective of this experiments were to find out a simplified model which was the correlation between the input parameters and the thermal efficiency of the AHT and ORC systems. For the simplified model, thermal performance could be predicted under various operating conditions and decreased the complicated simulation compared with the old procedure too.

For the AHT system, a 10 kW H_2O-LiBr absorption heat transformer upgraded fluid temperature to be around 90-110°C and 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 around 40% of glycol. The descriptions of the AHT components were shown in Table 1.

For the ORC system, a 20 kW R-245fa ORC system as shown in Fig. 1 was tested and carried out the simplified model. Table 2 shows the descriptions of the main components of the ORC system.

Results and Discussions

Thermal performance of the AHT system

Fig. 3 shows the average heating capacities of the absorption elements. It could be seen that the supplied heats at the generator and the evaporator were 11.59 and 7.50 kW, respectively. The upgraded heat at the absorber was 8.34 kW, while the rejected heat at the condenser was 13.87 kW.

Fig. 4 shows the strong (X_{max}) and weak (X_{min}) concentrations of H_2O-LiBr solutions and the overall COP including the overall energy efficiency ratio (EER) of the AHT system. It could be seen that the overall COP and EER of the prototype were 0.44 and 20.50, respectively. For concentration, X_{max} and X_{min} of the AHT prototype were 0.53 and 0.45 %LiBr, respectively. Moreover, it could be found that the experimental results of in this study was similarly with the other researches.

Fig. 5 shows the simplified model by correlation of EER_{AHT} with $(T_{A,i} - T_E)/(T_{G,i} - T_C)$, when liquid mixture in the storage tank was used and non-used.

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

For used hot water condition:

$$EER_{AHT} = -54.252(T_{A,i} - T_E)/(T_{G,i} - T_C) + 39.008, \quad (8)$$

For non-used hot water condition:

$$EER_{AHT} = -57.318(T_{A,i} - T_E)/(T_{G,i} - T_C) + 32.502. \quad (9)$$

In this study, the simplified model as shown in Fig. 5 will be used to project the system performance. Thus, precision of the prediction results from the experimental empirical equation was verified. The results were shown in Figs. 6 and 7 in cases of non-used and used hot water in 200 liter of storage tank (useful heat at the absorber around 10 kW), respectively. It could be seen that the both simulation results agreed well with those of the experimental data. Therefore, projection of the 10 kW AHT performance to a bigger scale of AHT system was appropriate to be used with the ORC performance in the next part.

Thermal performance of the ORC system

From the testing results, the 20 kW ORC system with using R-245fa as working fluid was tested and measured in laboratory. Hot water temperature varying 90-120^{o}C was supplied to ORC system at the boiler. While, cool water temperature around 25-35^{o}C (the typical climate conditions in Thailand) was pumped to the condenser. The testing results of the ORC system was shown in Table 3. It could be found that the average system efficiency of R-245fa ORC system was around 8.5%. Moreover, it could be noted that when hot water temperature entering the boiler increased, the ORC efficiency increased linearly too, which followed the Carnot efficiency concept.

Fig. 8 shows the simplified model by correlation of η_{ORC} with $T_{HW,i} - T_{CW,i}$. The empirical correlation of the ORC performance could be:

$$\eta_{ORC} = 0.0327(T_{HW,i} - T_{CW,i}) + 5.8628. \quad (10)$$

Projection of the thermal performance

In this topic, projection of the 10 kW AHT performance to a bigger scale of AHT was presented with the ORC efficiency. Fig. 9 shows steps of calculation for the analyses of the AHT and the ORC cycles with the simplified models. Performance correlations of EER_{AHT} and η_{ORC} with the operating temperatures were given. Fig. 10 shows the simulated result, when the R-245fa ORC system was coupled with the H_2O-LiBr AHT system. The thermal performance of the modified unit could be estimated. The input parameters were hot water temperature entering the AHT generator and the AHT absorber ($T_{G,i}$ and $T_{A,i}$), the AHT condenser temperature (T_C), the AHT evaporator temperature (T_E) and flow rate of hot water entering the CAHT system $(\dot{m}_{G,i})$. For the output parameters, the heat rate at the AHT absorber (Q_A), the upgraded temperature of the working fluid at the AHT absorber ($T_{A,o}$) and the generating electricity of the ORC system could be evaluated.

Fig. 11 shows the prediction results for projecting the AHT performance to generate electricity. It could be found that a 250 kW of AHT system was chosen to couple with the 20 kW ORC system. In addition, it could be seen that the modified system could be used a low temperature heat at 70^{o}C for upgrading heat to be 110^{o}C and generating electricity at 20 kW, respectively.

Economic result

From the previous results, the modified system was used to partial support an electrical power consumption of a large scale industrial, which had low temperature waste heat. Thus, payback period was determined in economic evaluation based on the initial conditions as follows:

- Operating period at 24 h/d and 350 d/y.
- The capacity of ORC system was 20 kW.
- Capital cost of ORC system was 2,500 USD/kW based on system world price (Industrial Technology Research Institute, 2012).
- Electricity charge (Time of use rate: TOU) of Thailand, type 4.2.3: a large scale industrial (Electricity cost, 2014 and Ft rate, 2014).

Table 4 shows the economic results of the method to upgrade low temperature heat by the AHT system cascade with the ORC system to generate electricity. It could be seen that saving electricity cost and payback period of the modified system were around 5,945.82 USD/y and 15.96 y, respectively. In additional, the levelized electricity cost (LEC) of the AHT-ORC system was 0.565 USD/kWh.

From the above results, it could be concluded that the electricity process by using the AHT and ORC systems to upgrade and transfer heat to power was beneficial in term of energy efficiency, but was deficient in term of economic results.

CONCLUSIONS

From this study, the conclusions were as follows:

1. The prototype of 10 kW H_2O-LiBr AHT system could be upgraded low temperature heat at 70-90°C to be around 90-110°C of the final hot water temperature at the COP_{AHT} around 0.44.
2. The prototype of 20 kW R-145fa ORC system had the system efficiency around 8.5% by supplying hot water temperature higher than 90°C.
3. The prediction results by using simplified models of the AHT and ORC systems could be simulated the system performance of the models to be close with those of the experimental results.
4. From the economic results, the saving electricity cost and the payback period of the modified system were around 5,945.82 USD/y and 15.96 y, respectively. In additional, the levelized electricity cost (LEC) of the modified system was 0.565 USD/kWh.

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Abbreviations and symbols

Nomenclature

C_p	heat capacity, (kJ/kg·K)
COP	coefficient of performance
EER	energy efficiency ratio
FR	flow ratio
h	enthalpy, (kJ/kg)
$\dot{m}$	mass flow rate, (kg/s)
P	pressure, (bar)
Q	heat rate, (kW)
s	entropy, (kJ/kg·K)
T	temperature, (°C)
W	work, (kW)
X	concentrate, (%LiBr)
Greek Symbol	
η	efficiency, (%)
ε	effectiveness, (%)
ρ	density, (kg/m^3)
Subscript	
A	absorber
B	boiler
bulk	bulk temperature
C	condenser
CW	cool water
E	evaporator
G	generator
HS	heat source
HW	hot water
HX	heat exchanger
i	inlet
max	maximum
min	minimum
o	outlet
P	pump
ref	refrigerant
s	isentropic
Tur	turbine

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