IMPROVEMENT OF AN ABSORPTION HEAT TRANSFORMER COMBINED WITH PROPANE VAPOR COMPRESSION HEAT PUMP IN INDUSTRIAL PROCESS

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Abstract

This paper presents a simulation concept of a single-stage Lithium Bromide-Water absorption heat transformer (AHT) performance when a propane vapor compression heat pump (VCHP) is coupled with. The heat rejected at the AHT condenser could be upgraded and transferred to the AHT evaporator then the overall COP of the system could be improved. The parameters considered are the mass circulation ratio in the AHT cycle, the temperatures of the AHT condenser and evaporator including that of the waste heat.

From the simulation results, it could be seen that the overall COP of the AHT before improvement does not exceed 0.5 because of with heat rejection at the AHT condenser. By coupling the AHT with the propane VCHP, the overall COP could be increase to be around 0.8 for the AHT absorber temperature at around 73.45 $^{\circ}C$.

***Keywords**:* Absorption heat transformer, Vapor compression heat pump, Heating, Waste heat.

1. Introduction

Absorption heat transformer (AHT) is one method for upgrading low temperature heat to a higher temperature heat. In a conventional AHT, the low temperature heat is absorbed at the AHT generator and the AHT evaporator. The heat is delivered at the AHT absorber at a higher temperature, while the AHT condenser rejects heat at a lower temperature. Theoretical and experimental studies of the AHT has been reported by various literatures. Kiatsiriroat et al. [1] reported thermal performance of a LiBr-Water AHT for upgrading low temperature heat such as waste heat from industrial processes or solar heat. The coefficient of performance (COP) was not exceeded 0.5 because there was high heat rejection at the AHT condenser. Kiatsiriroat et al. [2], also recommended a method for improvement the overall COP of AHT by combining a R-11 vapor compression heat pump (VCHP) to recover the heat rejected from the AHT condenser and it was used to supplied at the AHT generator or evaporator. The overall COP of this work was increased around 1.0. However, using of R-11 refrigerant affects the environment, such as global warming potential and ozone layer depletion potential. Therefore, new refrigerant should be undertaken.

The main objective of this work is to study a method for improvement the thermal performance of a H_2O-LiBr AHT by combining a VCHP to recover the heat rejected from the AHT condenser. For VCHP, the appropriate natural working fluid has been selected for upgrading the waste heat of industrial process to AHT.

2. Mathematical model

Figure 1 shows a schematic diagram of an AHT coupling with a VCHP and the whole unit is called Compression/Absorption Heat Transformer (CAHT). The heat rejected at the AHT condenser is recovered by a VCHP upgraded and supplied to the AHT evaporator.

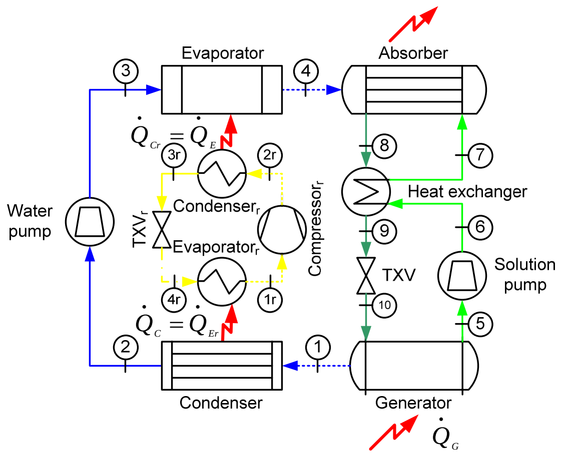

Figure 1. Diagram of the CAHT cycle.

The basic equations for the behavior of each component in the AHT cycle are as follows:

- Generator

$$Q_G = \dot{m}_1 h_1 + \dot{m}_5 h_5 - \dot{m}_{10} h_{10} \tag{1}$$

$$\dot{m}_{10} = \dot{m}_1 + \dot{m}_5 \tag{2}$$

$$\dot{m}_{10} X_{10} = \dot{m}_5 X_5, (X_1 = 0) \tag{3}$$

From equation (2) and (3),

$$\dot{m}_5 = \frac{\dot{m}_1 X_{10}}{X_5 - X_{10}} \tag{4}$$

and

$$\dot{m}_{10} = \frac{\dot{m}_1 X_5}{X_5 - X_{10}} \tag{5}$$

- Condenser

$$Q_C = \dot{m}_{ref}(h_1 - h_2) \tag{6}$$

$$\dot{m}_{ref} = \dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 \tag{7}$$

- Pump

$$W_P = (P_H - P_L)\frac{v_2 \dot{m}_2}{\eta} \tag{8}$$

$$W_{SP} = (P_H - P_L)\frac{v_5 \dot{m}_5}{\eta} \tag{9}$$

$h_2 = h_3$; W_{P1} is very small compared with Q_G (10)

$h_5 = h_6$; W_{P2} is very small compared with Q_G (11)

- Evaporator

$$Q_E = \dot{m}_{ref}(h_4 - h_3) \tag{12}$$

- Absorber

$$Q_A = \dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_8 h_8 \tag{13}$$

$$\dot{m}_8 = \dot{m}_4 + \dot{m}_7 \tag{14}$$

$$\dot{m}_8 X_8 = \dot{m}_7 X_7 \tag{15}$$

- Heat exchanger

$$Q_{HX,\max} = (\dot{m}c_p)_{\min}(T_8 - T_6) = (\dot{m}c_p)_{\min}\Delta T_i \tag{16}$$

$$\varepsilon_{HX} = \frac{Q_{HX,act}}{Q_{HX,\max}} \tag{17}$$

$$Q_{HX,act} = \dot{m}_8 c_{p8}(T_8 - T_9) = \dot{m}_6 c_{p6}(T_7 - T_6) \tag{18}$$

$$\dot{m}_8 = \dot{m}_9 \tag{19}$$

$$\dot{m}_6 = \dot{m}_7 \tag{20}$$

- Expansion valve

$h_9 = h_{10}$ (Throttling process) (21)

- Coefficient of performance (COP)

$$COP_{AHT} = \frac{Q_A}{Q_E + Q_G + W_{P1} + W_{P2}} \tag{22}$$

The basic equations for the behavior of each component in the VCHP cycle are as follows:

- Evaporator$_r$

$$Q_{E,r} = \dot{m}_{ref,r}(h_{1r} - h_{4r}) \tag{23}$$

$$\dot{m}_{ref,r} = \dot{m}_{1r} = \dot{m}_{2r} = \dot{m}_{3r} = \dot{m}_{4r} \tag{24}$$

- Compressor$_r$

$$W_{Comp,r} = \dot{m}_{ref,r}(h_{2r} - h_{1r}) \tag{25}$$

$$s_{1r} = s_{2r} \quad \text{(Isentropic process)} \tag{26}$$

$$\eta_{Comp,r} = \frac{h'_{2r} - h_{1r}}{h_{2r} - h_{1r}} \tag{27}$$

- Condenser$_r$

$$Q_{C,r} = \dot{m}_{ref,r}(h_{2r} - h_{3r}) \tag{28}$$

- Expansion valve$_r$

$$h_{3r} = h_{4r} \quad \text{(Throttling process)} \tag{29}$$

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

$$COP_{CAHT} = \frac{Q_A}{Q_G + W_{P1} + W_{P2} + W_{Cr}} \tag{30}$$

3. Selection Natural Working Fluids of the VCHP as Heat Recovery Unit

Chloroflourocarbon (CFCs) working fluids used in refrigeration and heat pump applications need to be replaced due to their influence in depletion of the stratospheric ozone layers. Therefore, Natural working fluids for heat pump have been considered as working fluid in the VCHP. Four working fluids, R-290 (Propane), R-600 (Butane), R-600a (Isobutane) and R-717 (Ammonia) for heat pump have been considered as working fluid in the VCHP [3]. The methods of choosing an appropriate working fluid for the VCHP as heat recovery unit used are physical property comparison and cycle simulation under a preferred working condition. R-22 is used as reference working fluid since it is widely used in heat pump applications. The working conditions for the evaluation are:

1. Designed the VCHP evaporator temperature is at 30 $^{\circ}C$.
2. Total cooling capacity is 10 kW.
3. Required hot water temperature at 55-60 $^{\circ}C$ (the VCHP condenser temperature is at 65 $^{\circ}C$)
4. No pressure drops at the VCHP condenser and the VCHP evaporator.
5. Isentropic efficiency of compressor is 80%.
6. Degree of superheating is 4.0 $^{\circ}C$.
7. Degree of subcooling is 3.5 $^{\circ}C$.
8. The properties of working fluids are based upon REFPROP [4].

From the calculation, it could be seen that R-290 gives the heat pump performance close to that of R-22 for generating hot water at about 55-60 $^{\circ}C$ and moreover, it could be drop-in replaced in a R-22 compressor. There are also some experiments on heat pumps with R22 and R-290. The operating conditions of the heat pumps are shown in Table 1.

Table 1. Experimental data of R-22 and R-290 heat pump comparison at upgrading heat 55 $^{\circ}C$.

Operating Conditions	R-22	R-290
High pressure ($PSIG$)	340	370
Low pressure ($PSIG$)	175	240
Inlet refrigerant of evaporator ($^{\circ}C$)	28.89	35.56
Inlet refrigerant of compressor ($^{\circ}C$)	37.16	39.50
Inlet refrigerant of condenser ($^{\circ}C$)	68.43	61.20
Inlet refrigerant of TXV ($^{\circ}C$)	29.86	37.96
Supplied hot water inlet ($^{\circ}C$)	34.78	40.68

Operating Conditions	R-22	R-290
Supplied hot water outlet $(^\circ C)$	32.09	34.79
Refrigerant mass flow rate (g/s)	53.75	29.90
Electrical power (kW)	1.19	1.18
COP_{HP} (-)	9.26	8.20

From experiments on heat pumps with R22 and R-290. The operating conditions of the heat pumps are shown in Table 1. It could be seen that the heat pump operates with R-22 and R-290 has similar results. Therefore, propane is selected as the working fluid of VCHP in this study.

4. Results of the CATH performances

A CAHT as presented in Figure 1 is considered. The AHT side has H_2O-LiBr as a working pair while the VCHP takes propane as the working fluid. The working conditions for the evaluation are

1. Waste heat of industrial process is hot water temperature at 50-70 $^\circ C$.
2. Supplied hot water flow rate is 1 $litre/s$.
3. Minimum concentration difference between strong and weak LiBr-water solutions is 2 % of LiBr.
4. No pressure drops at the AHT condenser, the AHT generator, the AHT evaporator, the AHT absorber and the AHT heat exchanger.
5. Isentropic efficiency of pump is 85%
6. The properties of H_2O-LiBr solution are taken from [5-7].

Figure 2 shows the variations of the overall COP for a given AHT condenser temperature (T_C). Decreasing T_C affects the overall COP only slightly but a wider range of waste heat temperature could be used for upgrading low temperature heat.

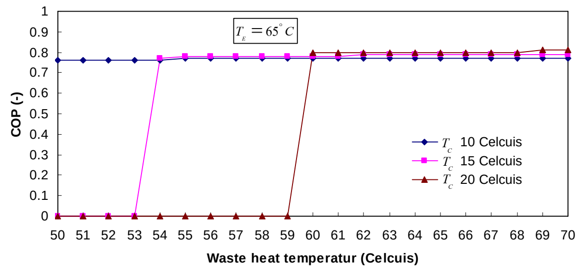

Figure 2. The effect of the AHT T_C on the overall COP.

Figure 3 shows the AHT absorber temperature (T_A) with the supplied hot water temperature in the normal AHT cycle and the CAHT cycle. It could be seen that T_A of the CAHT is nearly constant at 73.45 $^\circ C$ which that of normal AHT varies with the hot water temperature. In AHT, Increasing hot water temperature affects T_A increased.

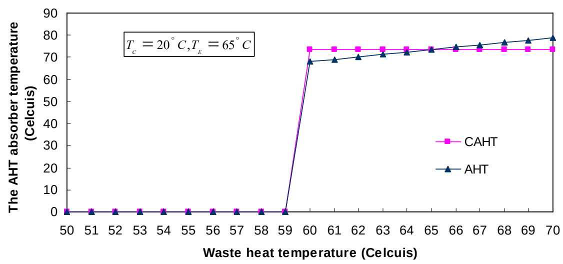

Figure 3. T_A of the normal AHT and the CAHT.

Figure 4 shows the overall COP with the supplied hot water temperature in the normal AHT cycle and the CAHT cycle. It could be seen that the overall COP of the CAHT is higher than that of the normal AHT, because the heat rejected in the condenser is recovered which the energy supplied for VCHP is not high. The overall COP improvement performed by the CAHT compared is around 80% higher than that of the normal AHT.

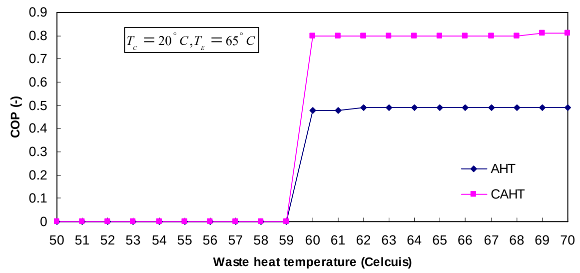

Figure 4. The overall COP of the normal AHT and the CAHT.

5. Conclusions

From this study, we can conclude the following:

1. The suitable natural working fluid of the VCHP is propane, its give the temperature for supplying heat to AHT at around 55-60 $^\circ C$.
2. The CAHT can produce approximately constant at 73.45 $^\circ C$ of the AHT absorber temperature.
3. The overall COP of the AHT is around 0.5.
4. The AHT combined with VCHP can increase the overall COP by approximately 80% over that of the AHT.

6. Acknowledgement

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7. Appendix

7.1 Density of lithium bromide-water solutions [6]

For Solution temperature t < 250 $^\circ C$

Concentration 30 < X < 65% LiBr

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

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

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

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

$M_s = 0.086845,\ kg/mole$

Table of Coefficients C_{ji}

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

7.2 Heat capacity of lithium bromide-water solutions [7]

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

Concentration 40 < X < 65% LiBr

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

A_0 = 3.462023 B_0 = 1.3499 E -3

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

8. References

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9. Nomenclature

Nomenclature

Cp	Heat capacity $(kJ/kg \cdot K)$
COP	Coefficient of performance, (-)
h	Enthalpy, (kJ/kg)
m	Mass flow rate, (kg/s)
P	Pressure, (Bar)
Q	Heat rate, (kW)
v	Specific volume, (m^3/kg)
s	Entropy, $(kJ/kg \cdot K)$
Sb	Subcooling, $(^{\circ}C)$
Sh	Superheating, $(^{\circ}C)$
W	Work, (kW)
X	Concentrate, $(\% LiBr)$

Greek symbol

η	Pump efficiency, $(\%)$
ε	Effectiveness, $(\%)$
ρ	Density, (kg/m^3)

Subscript

A	Absorber
act	Actual
$bulk$	Bulk temperature
C	Condenser
$Comp$	Compressor
e	Super heat
E	Evaporator
H	High
HP	Heat pump
HX	Heat exchanger
i	Inlet
L	Low
$\max$	Maximum
$\min$	Minimum
o	Outlet
P	Pump
r	Compression cycle
ref	Refrigerant
S	Start
SP	Solution pump