

Improvement of an Absorption Heat Transformer Performance for Upgrading Low Temperature Heat by Coupling with a Vapor Compression Heat Pump

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ABSTRACT

This paper presents a concept of a single-stage H_2O -LiBr absorption heat transformer (AHT) when it is coupled with a vapor compression heat pump (VCHP) for upgrading low temperature heat. Heat rejected at the AHT condenser is recovered by the VCHP and transferred to the AHT evaporator. For the VCHP, different types of working fluid have been considered and R-123 has been selected due to its appropriate pressure and temperature with high COP for supplying heat at around 80-90 °C.

A set of simulations has been carried out for the H_2O -LiBr AHT coupled with the R-123 VCHP to upgrade heat from hot water stream at various temperatures. With this technique, the supplied heat could be taken at the AHT generator only, not at the evaporator and the generator as the normal AHT. It could be seen that the overall COP of the combined cycle would be around 0.8, compared with 0.5 of the normal AHT. For the VCHP cycle, when it is coupled with the AHT cycle, the COP is also higher than that of a common one in upgrading the low temperature heat. For the combined cycle, lower the condensing temperature (T_C) gives a wider range of supplied hot water temperature at the generator. Higher the evaporating temperature (T_E) results in higher the absorber temperature and the overall COP.

Key words: Absorption heat transformer, Vapor compression heat pump, Simulation

INTRODUCTION

A technique to upgrade low temperature heat to a higher temperature could be taken by heat pump. The common method could be performed by a vapor compression heat pump (VCHP) of which the concept is shown in Figure 1.

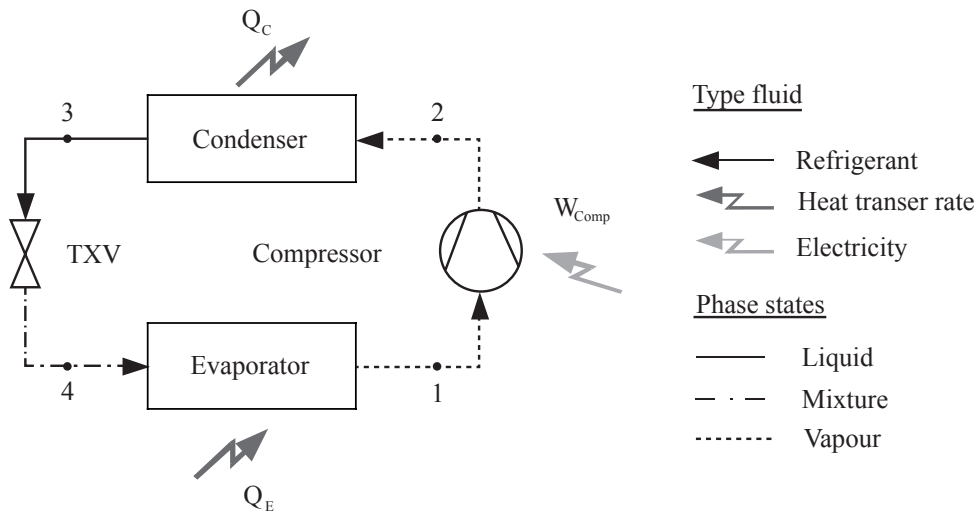


Figure 1. A concept of a vapor compression heat pump for upgrading low temperature heat.

A stream of liquid at a low temperature level supplies heat to the evaporator of the VCHP, then the heat will be upgraded and generated at the condenser. This technique is commonly used to upgrade low temperature waste heat or solar heat for water heating in hotels or hospitals (Burapha and Kiatsiriroat, 2008; and Chaiyat and Kiatsiriroat, 2010).

Absorption heat transformer (AHT) is one method for upgrading low temperature heat to a higher temperature level. The schematic diagram of the AHT cycle is shown in Figure 2.

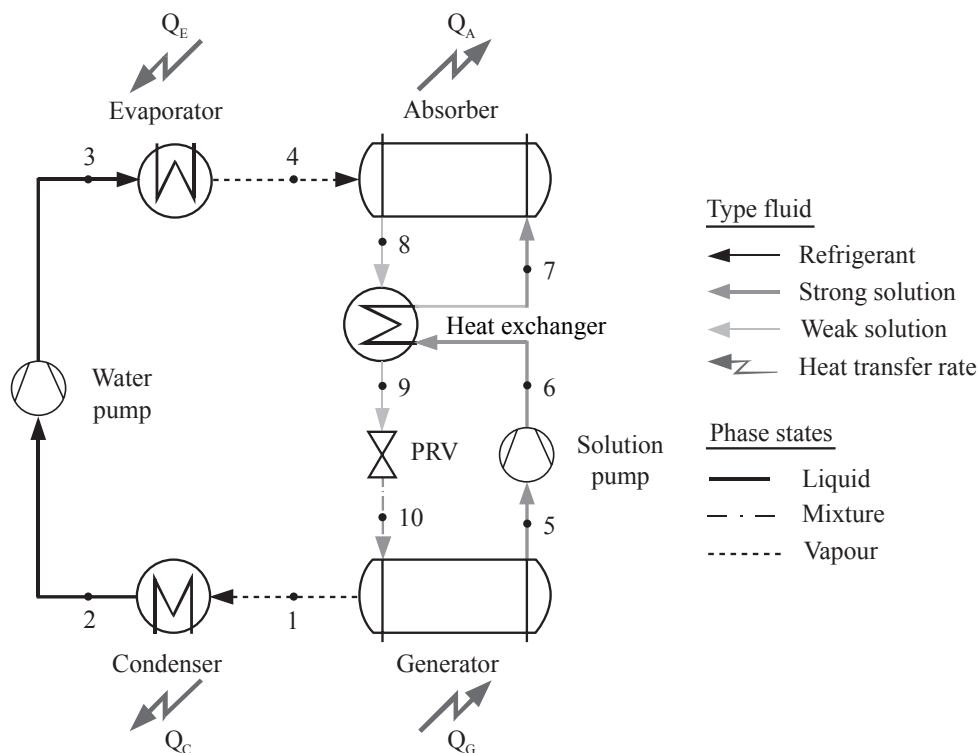


Figure 2. A schematic diagram of an absorption heat transformer (AHT).

In a conventional AHT, low temperature heat is absorbed at the AHT generator and the AHT evaporator and 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 have been reported by various literatures. Kiatsiriroat et al., (1986) reported thermal performance of a 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 AHT condenser. Xuehu et al., (2002) also reported test results of the first industrial-scale H_2O -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 with a mean COP of 0.47. The payback was approximately 2 years. Sotelo and Romero (2009) presented a heat transformer absorption cycle operating with water-CarrolTM 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.

In this study, a method to improve the thermal performance of a single-stage $\text{H}_2\text{O-LiBr}$ AHT by combining a VCHP to recover the heat rejected from the AHT condenser and supplied back at the AHT evaporator is considered. With this approach, input heat such as solar heat could supply at the AHT generator only, not at the AHT evaporator and generator. Besides, the amount of supplied heat could be less, then the number of the solar collector could be reduced. For the VCHP, an appropriate working fluid has been selected. The overall COP of the AHT coupling with the selected VCHP will be considered and compared with those of the common AHT and the common VCHP.

MATERIALS AND METHODS

From Figure 2, at the AHT generator, binary liquid mixture consisting 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 AHT generator and the AHT 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 for the behavior of each component in the AHT cycle are as follows:

• Generator

$$Q_G = \dot{m}_1 h_1 + \dot{m}_5 h_5 - \dot{m}_{10} h_{10}, \quad (1)$$

$$\dot{m}_{10} = \dot{m}_1 + \dot{m}_5, \quad (2)$$

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

From equations (2) and (3),

$$\dot{m}_5 = \frac{\dot{m}_1 X_{10}}{X_5 - X_{10}}, \quad (4)$$

and

$$\dot{m}_5 = \frac{\dot{m}_1 X_{10}}{X_5 - X_{10}}. \quad (5)$$

• Condenser

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

$$\dot{m}_{\text{ref}} = \dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4. \quad (7)$$

• Pump and solution pump

$$W_P = (P_E - P_C) \frac{v_2 \dot{m}_2}{\eta_P}, \quad (8)$$

$$W_P = (P_E - P_C) \frac{v_5 \dot{m}_5}{\eta_{SP}}, \quad (9)$$

$$h_2 \approx h_3, \quad (10)$$

$$h_5 \approx h_6. \quad (11)$$

• Evaporator

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

• Absorber

$$Q_A = \dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_8 h_8, \quad (13)$$

$$\dot{m}_8 = \dot{m}_4 + \dot{m}_7, \quad (14)$$

$$\dot{m}_8 X_8 = \dot{m}_7 X_7. \quad (15)$$

• Heat exchanger

$$Q_{HX} = \dot{m}_8 C_{p8} (T_8 - T_9) = \dot{m}_6 C_{p6} (T_7 - T_6) = \epsilon_{HX} (mCp)_{\min} (T_8 - T_6), \quad (16)$$

$$\dot{m}_8 = \dot{m}_9, \quad (17)$$

$$\dot{m}_6 = \dot{m}_7. \quad (18)$$

• Expansion valve

$$h_9 = h_{10} \text{ (Throttling process)}. \quad (19)$$

• Flow ratio (FR)

$$FR = \frac{\dot{m}_5}{\dot{m}_{\text{ref}}}. \quad (20)$$

• Gross temperature life (GTL)

$$GTL = T_8 - T_4. \quad (21)$$

• Coefficient of performance (COP)

$$COP_{\text{AHT}} = \frac{Q_A}{Q_E + Q_G + W_P + W_{SP}}. \quad (22)$$

Figure 3 shows a schematic diagram of an AHT coupling with a VCHP and the combined cycle 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.

The basic equations for the behavior of each component in the VCHP cycle as presented in Figure 3 are as follows:

- Evaporator_r

$$Q_{Er} = \dot{m}_r(h_{1r} - h_{4r}), \quad (23)$$

$$\dot{m}_r = \dot{m}_{1r} = \dot{m}_{2r} = \dot{m}_{3r} = \dot{m}_{4r} \quad (24)$$

- Compressor_r

$$W_{Comp} = \dot{m}_r(h_{2r} - h_{1r}), \quad (25)$$

$$S_{1r} = S_{2r} \text{ (Isentropic process)}, \quad (26)$$

$$\eta_{Comp} = \frac{h'_{2r} - h_{1r}}{h_{2r} - h_{1r}}. \quad (27)$$

- Condenser_r

$$Q_{Cr} = \dot{m}_r(h_{2r} - h_{3r}). \quad (28)$$

- Expansion valve_r

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

- Coefficient of performance (COP)

$$COP_{VCHP} = \frac{Q_{Cr}}{W_{Comp}}. \quad (30)$$

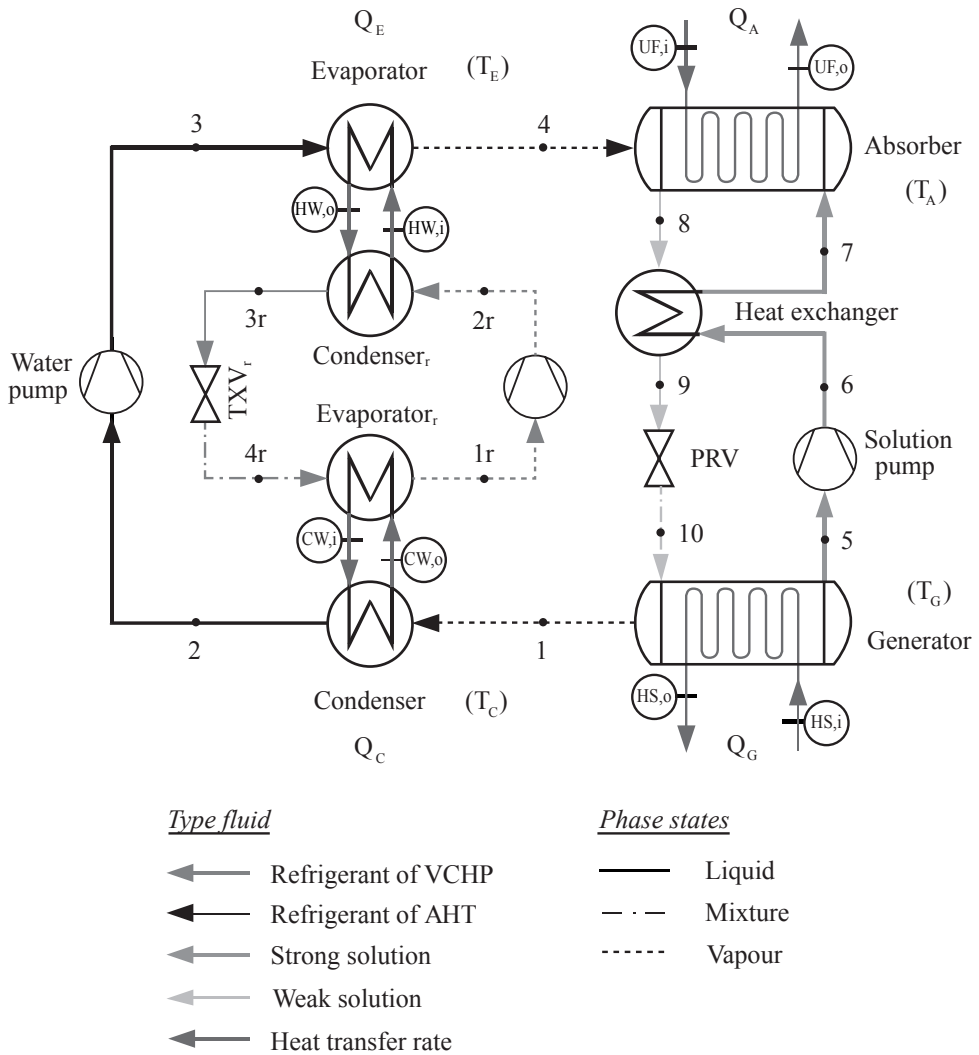


Figure 3. Diagram of the CAHT cycle.

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

$$\text{COP}_{\text{CAHT}} = \frac{Q_A}{Q_G + W_P + W_{\text{SP}} + W_{\text{Comp}}} \quad (31)$$

It could be seen that W_{Comp} in equation (31) is less than Q_E in equation (22), thus the COP_{CAHT} is higher than the COP_{AHT} .

Table 1. Physical properties of working fluids.

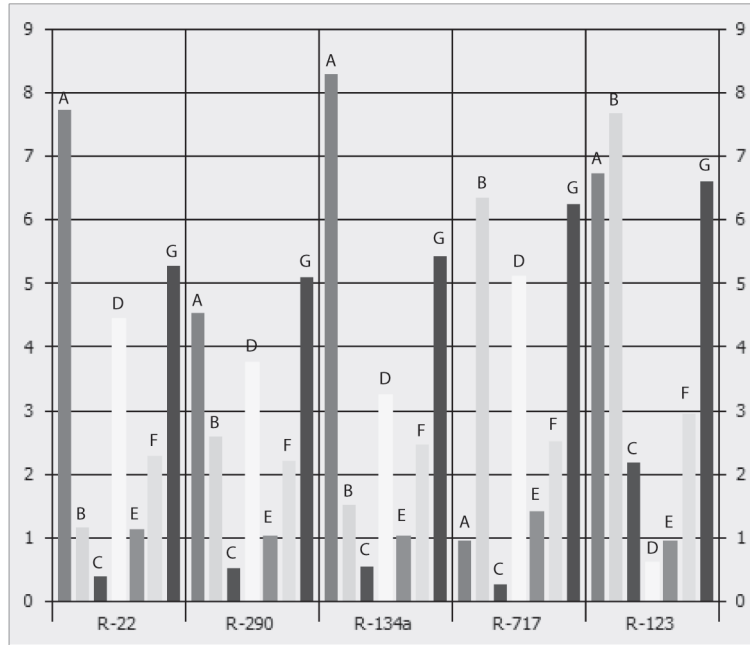
Working Fluid	R-22	R-290	R-134a	R-717	R-123
Chemical formulae	CHClF_2	C_3H_8	$\text{CF}_3\text{CH}_2\text{F}$	NH_3	CHCl_2CF_3
Molecular mass (kg/kmol)	86.46	44.10	102.03	17.03	152.93
Critical temperature ($^{\circ}\text{C}$)	96.14	96.68	101.06	132.25	183.68
Critical pressure (MPa)	4.99	4.25	4.06	11.33	3.66
Critical density (kg/m^3)	523.84	218.50	511.90	225.00	550.00
Boiling point ($^{\circ}\text{C}$)	-40.81	-42.09	-26.07	-33.33	27.82
Latent heat of vaporization at 40°C (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

Selection Working Fluid of the VCHP

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) for heat pump have been considered as working fluid in the VCHP Table 1 shows physical properties of the working fluids (NIST, 2000). The working conditions for the evaluation are:

1. The VCHP evaporator temperature (T_{Er}) is at 40°C .
2. Total cooling capacity (Q_{Er}) is 10 kW.
3. Required hot water temperature ($T_{\text{HW,o}}$) is around $80\text{--}85^{\circ}\text{C}$ (the VCHP condenser temperature (T_{Cr}) is at 90°C)
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 5.0°C .
7. Degree of subcooling (SC) is 5.0°C .
8. The properties of working fluids are based upon REFPROP (NIST, 2000).

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. Figure 4 shows the results of the selected refrigerants.



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

Figure 4. The results for the selected refrigerants.

Mass of refrigerant per unit heat output indicates the amount of refrigerant used in the VCHP cycle compared with the amount of heat generated. If the value is high, it means that for the same amount of generated heat, high amount of refrigerant is needed which results in a large scale of the components and high compression work. From Figure 4, it could be seen that R-290 and R-717 seem to be appropriate since the values of mass of refrigerant per unit heat output are around 3/5 and 1/8 of R-22. However, these refrigerants are flammable and R-717 is quite toxic and not compatible with copper which is a general material used in the heat pump.

The volume flow rate of refrigerant at the compressor inlet should be selected to be matched with the displacement volume of compressor. If the value is high, a big size of compressor is required. Figure 4 also shows that R-123 gives a high value which is around 600% of that from R-22 which means that the displacement volume of its compressor is 6 times of R-22 compressor. Therefore, if R-123 is taken, an open-type compressor should be used. From Figure 4, it could be seen that R-717 gives the best solution but the user should be aware of the corrosion of the refrigerant and the suitable lubricant.

Discharge pressure is the maximum pressure of refrigerant in the heat pump cycle. If the value is high, the thickness of coil and the fittings should have a special design which results in high initial investment. Moreover, high compression work is consumed. It could be seen that R-123 gives the best solution since the pressure is lowest compared with the other refrigerants. Discharge temperature of refrigerant at the compressor discharge is the maximum temperature in the heat pump cycle. If this is too high, the lubricant will not be stable. R-123, R-134a and propane are suitable for this case. The latter one is flammable, thus it is not appropriate to be used at a high temperature.

Pressure ratio is the ratio of the condenser pressure to the evaporator pressure. If this value is high, the compressor will consume high power input. From Figure 4, propane gives the lowest value while R-123 gives the highest one.

Figure 4 also shows the ideal heating COP_{hp} of the heat pump with different types of refrigerant. From the Figure, it could be seen that the best refrigerant is R-123 which gives the highest COP.

From the above 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 80-85°C due to its low maximum pressure for the heat pump compressor, and highest COP is obtained.

Working Conditions for the CAHT Analysis

All calculations of the CAHT are based on the systems presented in Figure 3. The water-Lithium Bromide is the working pair of the AHT and R-123 is the refrigerant of the VCHP. The working conditions for the evaluation are:

1. Supplied heat of the AHT is hot water temperature ($T_{HS,i}$) at around 50-95 .
2. Supplied hot water flow rate ($\dot{m}_{HS}$) is 1 liter/s.
3. Minimum concentration of weak H_2O -LiBr solution (X_{min}) is 45% LiBr.
4. Minimum concentration difference of strong and weak H_2O -LiBr solution is 2% LiBr.
5. No pressure drops at the AHT condenser, the AHT generator, the AHT evaporator, the AHT absorber and the AHT heat exchanger.
6. Isentropic efficiency of water pump (η_p) and solution pump (η_{sp}) is 85%.
7. Effectiveness of the AHT heat exchanger (ϵ_{HX}) is 85%.
8. Temperature difference between the outlet supplied hot water and the AHT generator is 5°C.
9. Temperature difference between the outlet useful water and the AHT absorber is 5°C.
10. Temperature difference between the outlet cooling water and the AHT condenser is 5°C.
11. Temperature difference between the outlet supplied hot water and the AHT evaporator is 5°C.
12. The properties of H_2O -LiBr solution are shown in Appendix.

Figure 5 shows the steps for calculating the CAHT performance.

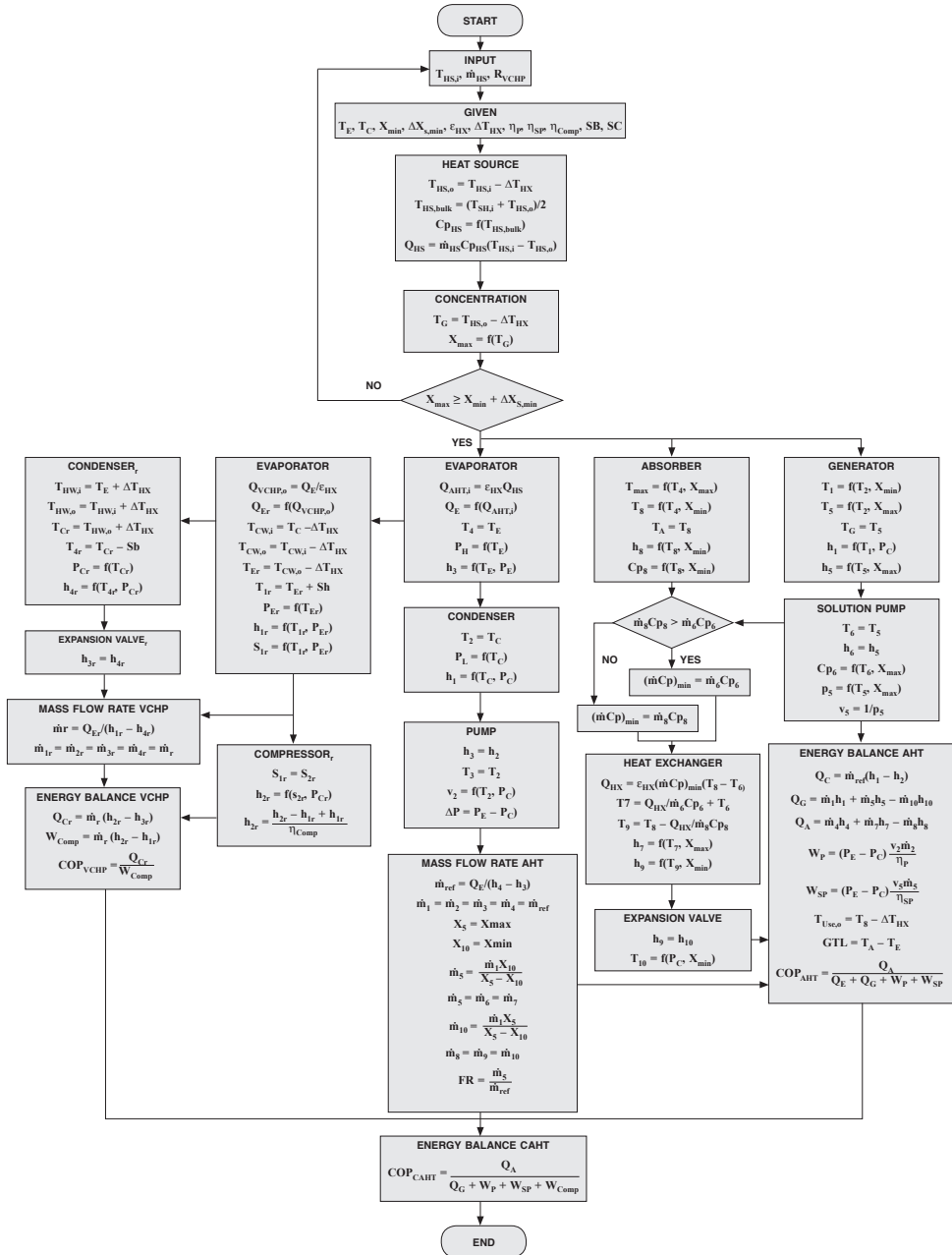


Figure 5. Flow chart of the simulation program for evaluating the CAHT performance.

RESULTS AND DISCUSSION

Effect of the AHT condenser temperature on the CAHT performance

Figure 6 shows the absorber temperature (T_A) with various values of the supplied hot water temperature at the generator of the normal AHT cycle and the CAHT cycle. It could be seen that T_A of the CAHT is nearly constant at around 100°C because the AHT evaporator temperature (T_E) is constant (about 80°C) while that of the normal AHT varies with the hot water temperature. It could also be seen that, for the CAHT, the condensing temperature (T_C) affects T_A only slightly. For lower, T_C , the CAHT cycle could operate with a wide range of hot water temperature.

Figure 7 shows the variations of the overall COP of the CAHT cycle at various hot water temperatures. The overall COP of the CAHT cycle is nearly constant with various supplied hot water temperatures at the generator. Compared to the COP of the AHT cycle, the CAHT performance is around 1.6 times of the AHT value (COP of the AHT ≈ 0.5). Higher the condensing temperature, T_C , results in higher the COP of the CAHT due to the less power at the VCHP compressor.

Effect of the AHT evaporator temperature on the CAHT performance

Figure 8 shows the variation of the CAHT absorber temperature (T_A) with the evaporator temperature (T_E). It could be seen that T_A increases significantly with T_E . Higher T_E will give higher pressure in the absorber which results in higher T_A value.

Similarly, the overall COP of the CAHT cycle could be improved by decreasing T_E . Lower T_E requires less VCHP compression work (W_{Comp}) which results in higher COP of the CAHT cycle. The results are shown in Figure 9.

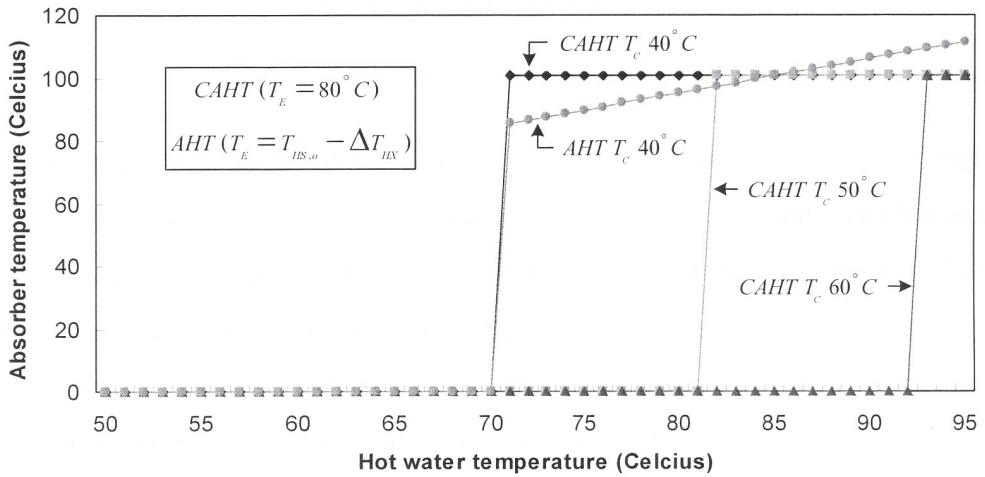


Figure 6. Effect of the supplied hot water temperature on the absorber temperature for the normal AHT and the CAHT.

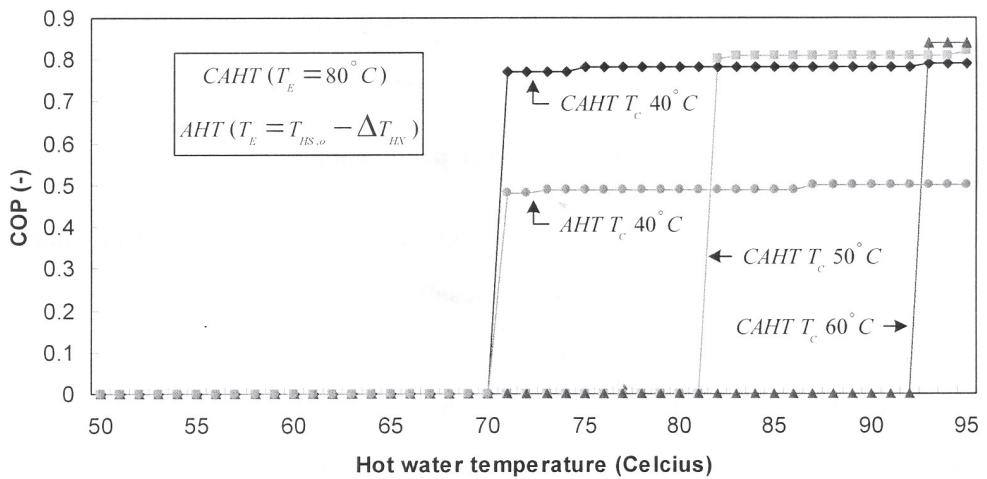


Figure 7. The overall COP on T_c of the normal AHT and the CAHT.

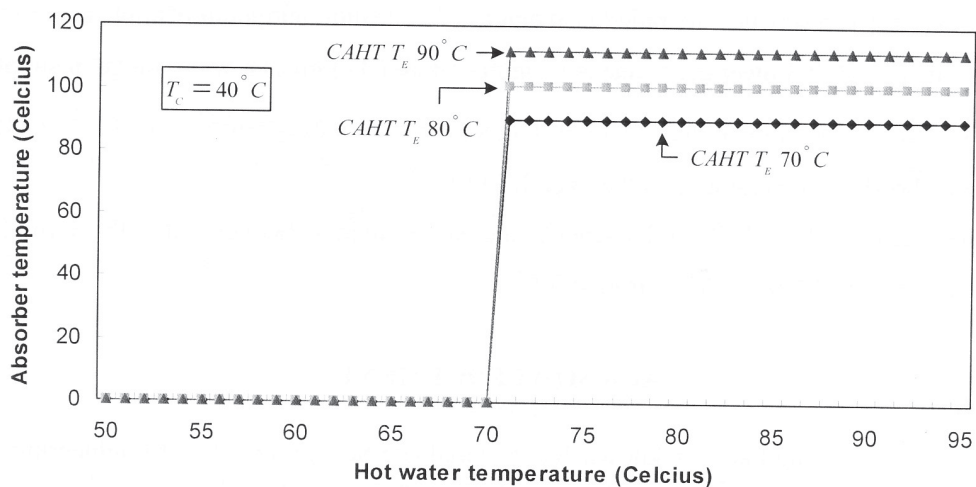


Figure 8. The effect of the AHT T_E on the AHT T_A of the CAHT.

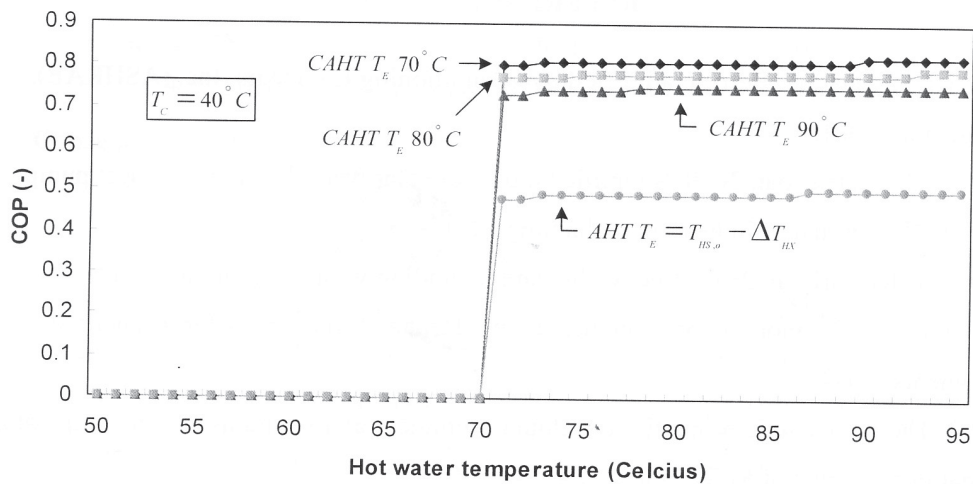


Figure 9. The overall COP on T_E of the normal AHT and the CAHT.

CONCLUSION

From this study, the conclusions are as follows:

1. The suitable working fluid of the VCHP is R-123 due to its low operating pressure and high COP of the VCHP cycle for supplying heat at around 80-90°C.
2. The CAHT can produce upgraded heat at a nearly constant temperature at the absorber and the overall COP. Lower the condensing temperature (T_C) gives a wider range of supplied hot water temperature at the generator. Higher the evaporating temperature (T_E) results in high absorber temperature and the overall COP.
3. The overall COP of the CAHT cycle can increase around 80% over that of the normal AHT (the overall COP of the AHT around 0.5).

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APPENDIX

A.1 Enthalpy-Concentration and Temperature for lithium bromide-water solutions (ASHRAE, 2001)

For Concentration $x < 40\% \text{LiBr}$

Solution temperature range $15 < t < 165^\circ\text{C}$

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

For Concentration $40 \leq x < 70\% \text{LiBr}$

Solution temperature range $15 < t < 165^\circ$

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

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

A.2 Solution Temperature-Refrigerant Temperature and Saturation pressure (ASHRAE, 2001)

For Refrigerant $-15 < t' < 110^\circ\text{C}$

Solution temperature $5 < t < 175^\circ\text{C}$

Concentration $45 < X < 70\% \text{LiBr}$

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

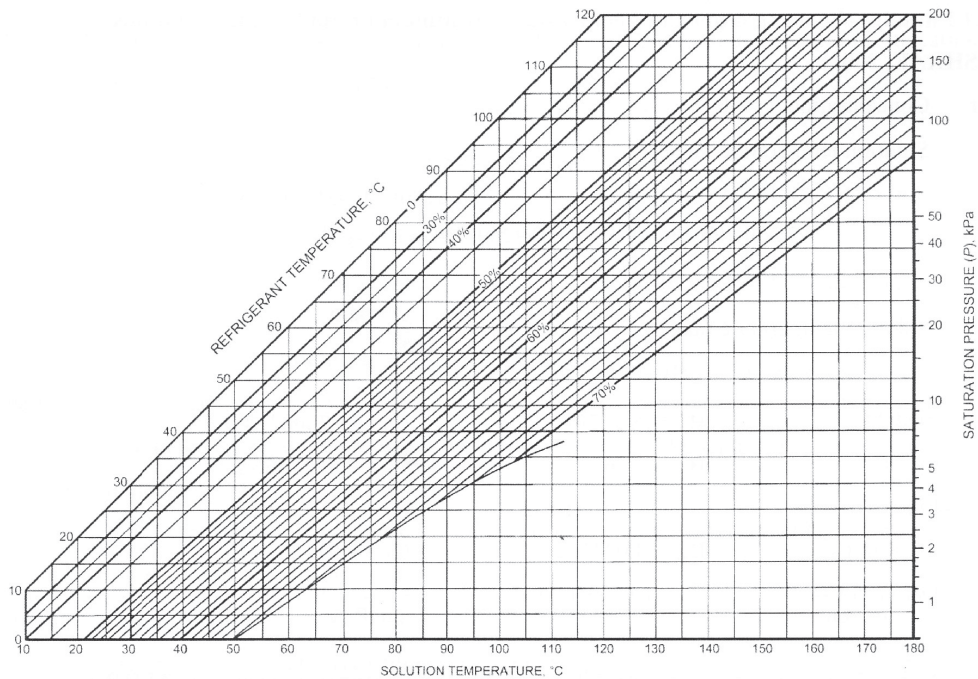
$$t' = (t - \sum_0^3 B_n X^n) / \sum_0^3 A_n X^n, ^\circ\text{C}$$

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

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

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

A.3 Equilibrium Chart for Aqueous Lithium Bromide Solutions (ASHRAE, 2001)



A.4 Density of lithium bromide-water solutions (Khairulin et al., 2006)

For Solution temperature $t < 250^{\circ}\text{C}$

Concentration $30 < X < 65\%\text{LiBr}$

$$\tilde{n}(t,m) = \tilde{n}_o(t)[1+d_o(t)m+d_1(t)m^{1.5}+d_2(t)m^2], \text{kg/m}^3$$

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

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

$$\tilde{n}_o(t) = \text{Density of pure water, kg/m}^3$$

$$M_s = 0.086845 \text{ kg/mole}$$

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

A.5 Heat capacity of lithium bromide-water solutions (Kaita, 2000)

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

Concentration $40 < X < 65\%\text{LiBr}$

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

$$A_0 = 3.462023$$

$$B_0 = 1.3499 \text{ E-3}$$

$$A_1 = -2.679895 \text{ E-2}$$

$$B_1 = -6.55 \text{ E-6}$$

Nomenclature

A	Area,
C _p	Heat capacity, (kJ/kg·K)
COP	Coefficient of performance, (-)
h	Enthalpy, (kJ/kg)
LMTD	Log mean temperature difference, (-)
m	Mass flow rate, (kg/s)
P	Pressure, (bar)
Q	Heat rate, (kW)
R	Refrigerant, (-)
v	Specific volume, (m ³ /kg)
s	Entropy, (kJ/kg·K)
SC	Subcooling, (°C)
SH	Superheating, (°C)
T	Temperature, (°C)
U	Overall heat transfer coefficient, (W/m ² ·K)
W	Work, (kW)
X	Concentrate, (%LiBr)

Greek symbol

ç	Efficiency, (%)
ã	Effectiveness, (%)
ñ	Density, (kg/m ³)

Subscript

A	Absorber
act	Actual
bulk	Bulk temperature
C	Condenser
Comp	Compressor
CW	Cooling water
e	Super heat
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
SP	Solution pump
UF	Useful