Front. Energy 2017, 11(4): 503–509
https://doi.org/10.1007/s11708-017-0515-1

RESEARCH ARTICLE

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Numerical study of a hybrid absorption-compression high temperature heat pump for industrial waste heat recovery

Abstract The present paper aims at exploring a hybrid absorption-compression heat pump (HAC-HP) to upgrade and recover the industrial waste heat in the temperature range of 60°C–120°C. The new HAC-HP system proposed has a condenser, an evaporator, and one more solution pump, compared to the conventional HAC-HP system, to allow flexible utilization of energy sources of electricity and waste heat. In the system proposed, the pressure of ammonia-water vapor desorbed in the generator can be elevated by two routes; one is via the compression of compressor while the other is via the condenser, the solution pump, and the evaporator. The results show that more ammonia-water vapor flowing through the compressor leads to a substantial higher energy efficiency due to the higher quality of electricity, however, only a slight change on the system exergy efficiency is noticed. The temperature lift increases with the increasing system recirculation flow ratio, however, the system energy and exergy efficiencies drop towards zero. The suitable operation ranges of HAC-HP are recommended for the waste heat at 60°C, 80°C, 100°C, and 120°C. The recirculation flow ratio should be lower than 9, 6, 5, and 4 respectively for these waste heat, while the temperature lifts are in the range of 9.8°C–27.7 °C, 14.9°C–44.1 °C, 24.4°C–64.1°C, and 40.7°C–85.7°C, respectively, and the system energy efficiency are 0.35–0.93, 0.32–0.90, 0.25–0.85, and 0.14–0.76.

Keywords absorption compression, high temperature heat pump, efficiency, industrial waste heat, thermodynamic analysis

Received Jun. 14, 2017; accepted Sep. 20, 2017; online Nov. 22, 2017

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1 Introduction

The industrial sector consumes about 54% of the total world's delivered energy[1)], and what is worse is that about one sixth of the total energy consumed by industrial sector is wasted as low-grade heat via radiation, exhausted gas, cooling fluid and so on [2)]. This energy can be recovered by various technologies, such as power generation via the organic Rankine cycle [1], refrigeration by absorption [2] and adsorption [3] technologies, and heat upgrade by heat pump [4], etc. Among these technologies, heat pump can efficiently pump the heat form low grade to high grade, and has been recognized as an efficient and practical solution to reduce greenhouse gas emission [5].

Compared to the conventional vapor-compression heat pump, the hybrid absorption-compression heat pump (HAC-HP) is believed to be able to work for high temperature applications with a better system performance [6]. The HAC-HP system is a combination of the conventional vapor-compression system and the absorption system, in which the use of binary mixture as working fluid leads to the non-isothermal process of absorption/desorption and relatively lower temperature difference between the heat source and the working fluid compared to the isothermal condensation/evaporation process in the conventional vapor-compression heat pump, thereby the system energy efficiency can increase due to the lower cycle irreversibility and entropy generation. Some other benefits of using this hybrid system are small swept volume of the compressor, higher heat transfer coefficient, environment-friendly refrigerant, more flexiblity in the changes of temperature and capacity, and higher delivery temperature [7].

Brunin et al. [6] compared the working domains of different heat pump technologies, and concluded that the vapor-compression system using hydrocarbon fluid and the hybrid system using ammonia-water were the only two

1) U.S. Energy Information Administration (EIA). The International Energy Outlook 2016. 2017-07-13
2) Element Energy Limited. The potential for recovering and using surplus heat from industrial. Final Report for DECC. 2014

technologies of high temperature heat pump as the abolishing of CFC and HCFC refrigerant. Minea and Chiriac [8] presented some design guidelines and operation experiences of the ammonia-water based HAC-HP system, and tested a 4.5 MW prototype in field and demonstrated the heat upgrade from 36°C industrial cooling water to 55°C useful hot water with a COP of 3.9. Kim et al. [9] experimentally tested a 10 kW HAC-HP system using ammonia-water as working fluid, and elevated the heat from 50°C to over 90°C with ammonia mass fraction in the weak solution of 0.42. Jensen et al. [10] numerically investigated the influences of ammonia mass fraction and liquid circulation ratio on the system constrains, and the results showed that the maximum heat supply temperature was 111°C when using standard refrigeration components without modifying the compressor (28 bar limitation); for high pressure ammonia based components, the maximum supply temperature could be 129°C (50 bar limitation) and for transcritical CO_2 based components, the maximum supply temperature could be 147°C (140 bar limitation). Jensen et al. [11] also thermodynamically analyzed the HAC-HP system to recover waste heat from spray-drying facilities, and conducted exergoeconomic analysis to minimize the lifetime cost. The best solution they reported was an 895 kW heat pump with ammonia mass fraction at 0.82 and a circulation ratio of 0.43. This system could generate an economic saving of €146,000 and an annual CO_2 emission reduction of 227 ton. Bourouis et al. [12] studied a single stage HAC-HP system using ternary mixture of Trifluoroethanol-water-tetraethylenglycoldimethylether (TFE-H_2O-TEGDME) as working fluid, and the results showed that the system could upgrade the waste heat form 80°C to 120°C at a COP of 6.4.

The conventional HAC-HP systems mentioned above are all based on a similar configuration with the main components of generator, absorber, solution pump, and compressor. Such system uses electricity as the main driven force to upgrade the low-grade waste heat. Though the system COP is high, the operation cost of the conventional HAC-HP system is still proportionally increased with the amount of recovered waste heat. The current paper proposes to add a condenser, an evaporator, and one more solution pump to the system to form a HAC-HP system using the compressor and the condenser-pump-evaporator together to elevate the pressure of the working fluid. There are two benefits of employing such a dual energy source system. First, the system can be more flexible on the use of different energy source, i.e., electricity and/or waste heat, depending on their availabilities. Secondly, more waste heat can be recovered as the system required the thermal inputs of both the generator and the evaporator while the required electricity can be reduced. The energy efficiency, exergy efficiency, temperature lift, and useful heat power are numerically calculated and analyzed.

2 Working principle and analysis method

Ammonia-water was used as the working fluid in the current study. The correlations developed by El-Sayed and Tribus [13] were used to determine the equilibrium liquid and vapor states of ammonia-water mixture, while Gibbs free energy formulations reported by Ziegler and Trepp [14] were used to calculate the enthalpies and entropies of ammonia-water mixture. The following hypothesises were used to simplify the numerical analysis: The cycle was operated in a steady-state; the solutions at the outlet of the condenser and the evaporator were in a saturated state; the solutions at the outlet of generator and absorber were in a saturated state; the vapor from the rectifier was in a saturated state; the throttling does not change the solution enthalpy; after throttling, the solution entering the generator was at two-phase or in a saturated liquid state; and the pressure drop and heat loss in the system were both negligible.

The schematic diagram of the HAC-HP system studied is shown in Fig. 1. The system consists of a column shape generator, a rectifier on the top of the generator, a recuperator, an absorber, a condenser, an evaporator, a compressor, two solution pumps, and a throttling valve.

There were two pressure levels in the whole cycle, a low pressure P_L in the generator, rectifier and condenser, and a high pressure P_H in the evaporator, absorber and recuperator. In the current study, the refrigerant ammonia mass fraction (after the rectifier),w_{ref}, was pre-defined at 0.9995 as suggested [15]. According to the predefined assumption, the ammonia-rich vapor at the outlet of the rectifier is in a saturated state; therefore, P_L could be determined as the saturated vapor pressure at rectification temperature, T_{rec} and ammonia mass fraction w_{ref}. Meanwhile, P_H could be determined by the saturated vapor at the outlet of the evaporator based on the waste heat temperature T_{was} and w_{ref}. Based on P_L and w_{ref}, the condensation temperature, T_{con}, could be determined based on the saturated liquid state at the outlet of the condenser. This condensation temperature should be slightly lower than the rectification temperature due to the temperature difference between the saturated liquid and vapor of ammonia-water at the same pressure and ammonia mass fraction. Then, other thermodynamic states of saturated liquid at the outlet of condenser, saturated vapors at the outlet of evaporator and rectifier could be determined.

The ammonia-water liquid at the outlet of the generator was in a saturated state, where $T_1 = T_{was}$ and $P_1 = P_L$, hence the ammonia mass fraction of the basic ammonia-water liquid (weak solution) used, w_{bas}, could be determined by the thermodynamic state equation. Thereafter, enthalpy h_1 and entropy s_1 could be calculated. The recirculation flow ratio (FR) [15] was used to calculate the mass flow rate of the ammonia-water liquid. FR is defined in Eq. (1).

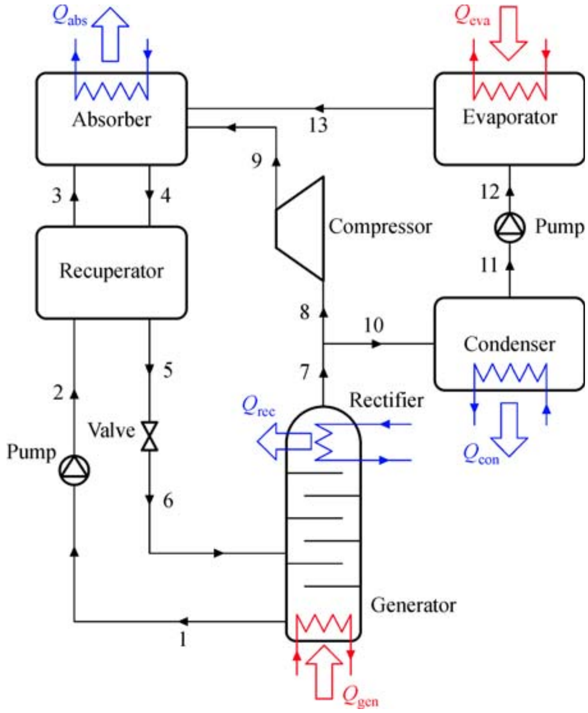

Fig. 1 Schematic diagram of hybrid absorption-compression heat pump

$$FR = \frac{\text{mass flow rate of liquid leaving generator}}{\text{mass flow rate of vapor leaving generator}} = \frac{\dot{m}_1}{\dot{m}_7}. \quad (1)$$

Then mass flow rate of the refrigerant, $\dot{m}_7$, was pre-defined at 0.01 kg/s, so that the system scale was approximately 12 kW (refrigeration power). Thereafter, the ammonia-water liquid mass flow rates $\dot{m}_1$, $\dot{m}_6$ and the ammonia-rich liquid mass fraction w_6 could be calculated based on the mass balance equations. The temperature of the useful heat, $T_{use} = T_4$, was then determined by the saturated liquid at the outlet of the absorber at the pressure of P_H and the mass fraction of pump w_4 ($w_4 = w_6$). Besides, enthalpy h_4 could be calculated.

The ammonia-water liquid from the generator was pumped from P_L to P_H. The isentropic efficiency, η_{pump1}, of this solution pump is given as

$$\eta_{pump1} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (2)$$

where h_{2s} is the enthalpy of ammonia-water liquid at the outlet of the pump if the process is isentropic. η_{pump1} was pre-defined to be 0.85 according to Ref. [16]. h_{2s} could be calculated by considering the ammonia-water liquid at pressure P_H with an entropy value of s_1. The value of h_2 could be calculated by Eq. (2). The pumping power consumed is then calculated by

$$\dot{W}_{pump1} = \dot{m}_1(h_2 - h_1). \quad (3)$$

The same method could be applied to the second pump located between condenser and evaporator to obtain $\dot{W}_{pump2}$. Meanwhile, the following two equations could be employed to solve the compression process using the similar procedure as that for the pump.

$$\eta_{com} = \frac{h_{9s} - h_8}{h_9 - h_8}, \quad (4)$$

$$\dot{W}_{com} = \dot{m}_8(h_9 - h_8), \quad (5)$$

where η_{com} was pre-defined to be 0.75 [16], and the mass flow rate of the ammonia-water vapor entering the compressor could be calculated based on the value of splitting ratio, $R_{ref} = \frac{\dot{m}_8}{\dot{m}_7}$.

For the recuperator, the thermal states of the two inlet ammonia-water liquids (T_2, h_2, T_4, h_4) were determined. Then, the outlet temperature (T_3 and T_5) and the enthalpy (h_3 and h_5) of the two liquids could be iteratively determined using heat balance equations, Eq. (6) and Eq. (7), and the logarithmic temperature difference ΔT_{LMTD} in Eq. (8).

$$\dot{Q}_{HE} = \Delta T_{LMTD} UA, \quad (6)$$

$$\dot{Q}_{HE} = \dot{m}_2(h_3 - h_2) = \dot{m}_4(h_4 - h_5), \quad (7)$$

$$\Delta T_{\mathrm{LMTD}}=\frac{T_4-T_3-(T_5-T_2)}{\ln\left(\frac{T_4-T_3}{T_5-T_2}\right)}. \tag{8}$$

The rectification reflux ratio, R_{refl}, could be used to assist the calculation of the rectification process [17]. As depicted in Fig. 2, the limited height of the rectification column leads to a steeper operation line compared to the ideal isothermal rectification line. The reflux ratio is defined as

$$R_{\mathrm{refl}}=\frac{h_{\mathrm{pole}}-h_7}{h_{\mathrm{min}}-h_7}, \tag{9}$$

where h_{min} can be calculated by

$$h_{\mathrm{min}}=h_{6\mathrm{v}}+(h_{6\mathrm{v}}-h_{6\mathrm{l}})\frac{w_{\mathrm{ref}}-w_{6\mathrm{v}}}{w_{6\mathrm{v}}-w_{6\mathrm{l}}}. \tag{10}$$

R_{refl} of 2 was used in the current study as recommended [17]. h_{pole} can be calculated by Eq. (9). The rectification heat is then

$$Q_{\mathrm{rec}}=\dot{m}_7(h_{\mathrm{pole}}-h_7). \tag{11}$$

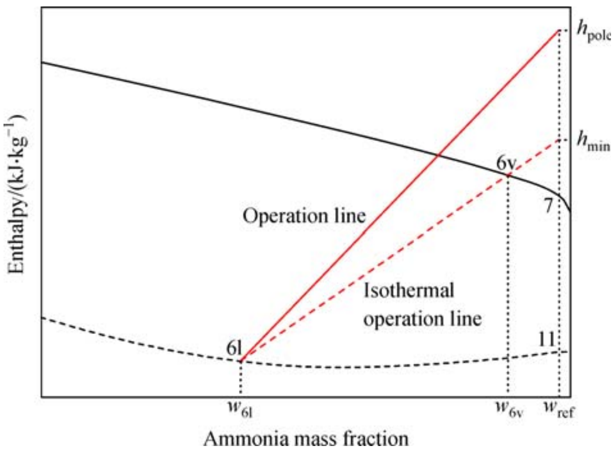

Fig. 2 Determination of the pole of rectification

The generation, evaporation and absorption heat can be calculated by

$$\dot{Q}_{\mathrm{gen}}=\dot{m}_1h_1+\dot{m}_7h_7+\dot{Q}_{\mathrm{rec}}-\dot{m}_6h_6, \tag{12}$$

$$\dot{Q}_{\mathrm{eva}}=\dot{m}_{12}(h_{13}-h_{12}), \tag{13}$$

$$\dot{Q}_{\mathrm{abs}}=\dot{m}_3h_3+\dot{m}_9h_9+\dot{m}_{13}h_{13}-\dot{m}_4h_4. \tag{14}$$

Finally, the energy efficiency and exergy efficiency of the HAC-HP cycle can be calculated by Eqs. (15) and (16), respectively.

$$\eta_{\mathrm{en}}=\frac{\dot{Q}_{\mathrm{abs}}}{\dot{Q}_{\mathrm{gen}}+\dot{Q}_{\mathrm{eva}}+\dot{W}_{\mathrm{com}}+\dot{W}_{\mathrm{pump1}}+\dot{W}_{\mathrm{pump2}}}, \tag{15}$$

$$\eta_{\mathrm{ex}}=\frac{\dot{Q}_{\mathrm{abs}}(1-T_{\mathrm{amb}}/T_{\mathrm{use}})}{(\dot{Q}_{\mathrm{gen}}+\dot{Q}_{\mathrm{eva}})(1-T_{\mathrm{amb}}/T_{\mathrm{was}})+\dot{W}_{\mathrm{com}}+\dot{W}_{\mathrm{pump1}}+\dot{W}_{\mathrm{pump2}}}. \tag{16}$$

The parameters used in the calculation are summarized in Table 1.

Table 1 Parameters used in the calculation

Parameter	Value
T_{was}/°C	60–120
FR	1–20*
R_{ref}	0–1
R_{refl}	2
η_{pump1}, η_{pump2}	0.85
η_{com}	0.75
UA/(W·K^{-1})	1000
w_{ref}	0.9995
$\dot{m}_{\mathrm{ref}}$/(kg·s^{-1})	0.01

*FR was initially set from 1 to 20. However, the feasible values of FR were limited by each working condition.

3 Results and discussion

Figure 3 is an example of the operation states of the HAC-HP system in the enthalpy-mass fraction chart, where the waste heat temperature is 80°C and FR = 7. The two pressure levels in the system are 11.1 bar and 38.9 bar while the ammonia-lean and ammonia-rich solution have a mass fraction of 0.412 and 0.485 respectively, which gives a pre-defined refrigerant mass fraction of 0.9995.

Figure 4 demonstrates the energy efficiency of the HAC-HP system proposed at a waste heat temperature of 80°C. The maximum energy efficiency obtained under the working conditions studied is approximately 0.90. This maximum value occurs under the condition of FR = 1 and R_{ref} = 0. These curves indicate that it is more efficient to use a compressor than a condenser-pump-evaporator, as can be seen from Fig. 4 that a lower refrigerant splitting ratio leads to a higher energy efficiency. For example, the highest η_{en} is about 0.90 when using the compressor only under the condition of FR = 1 while the lowest η_{en} is only about 0.43 when using the condenser-pump-evaporator only. The upgraded temperature is from 95°C to approximately 132°C, which gives a temperature lift in the range of 15°C–52°C. A higher FR leads to a smaller ammonia mass fraction difference between the ammonia-rich and the ammonia-lean solution. The saturated ammonia-water liquid from the absorber has a relatively lower ammonia mass fraction. Therefore, the equilibrium

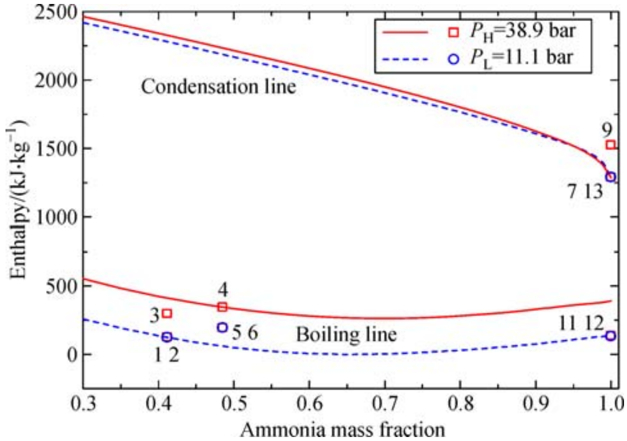

Fig. 3 Thermodynamic states of the working fluid at different points at T_{was} = 80°C and FR = 7

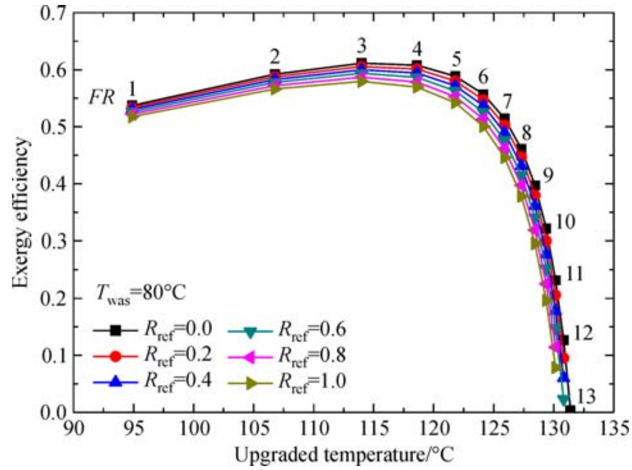

Fig. 5 Exergy efficiency of HAC-HP at a waste heat temperature of 80°C

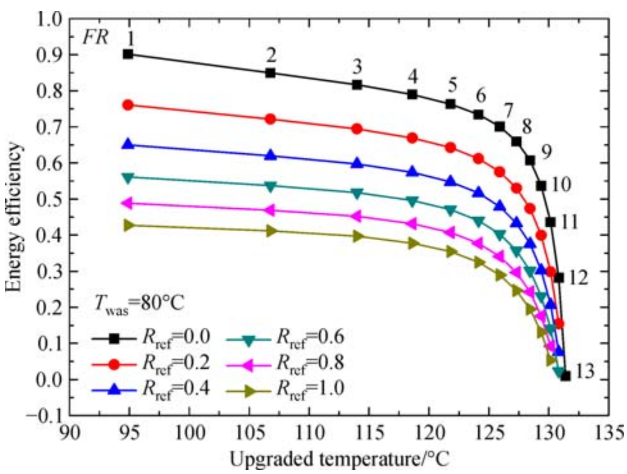

Fig. 4 Energy efficiency of HAC-HP at a waste heat temperature of 80°C

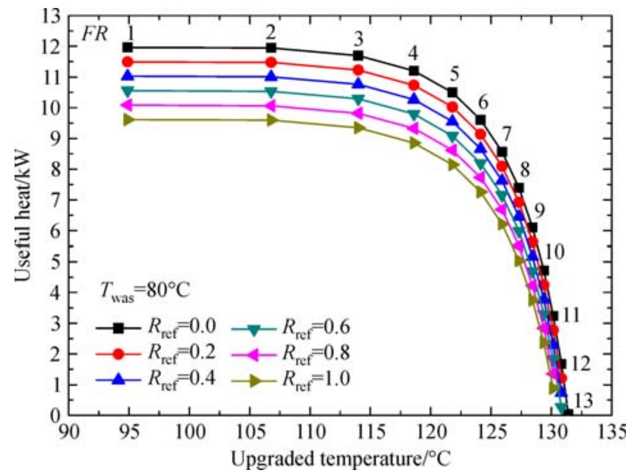

Fig. 6 Useful heat of HAC-HP at a waste heat temperature of 80°C

temperature in the absorber at a certain pressure (P_H has been determined by the evaporator) is higher. A relatively flat energy efficiency curve can be noticed when FR is smaller than 6 for all values of R_{ref}, and the change of energy efficiency is no more than 25%. Then, the energy efficiencies drop dramatically towards zero as FR is higher than 7.

Figure 5 shows the exergy efficiency of the system at a waste heat temperature of 80°C. Compared to the energy efficiency curves in Fig.4, the gap between each exergy efficiency curve having different values of R_{ref} is smaller, e.g. less than 8%. The exergy efficiency increases gently as FR increases to 3. Then, it decreases gently as FR increases to 6. And finally, it decreases violently towards zero. Based on these values of exergy efficiency, the optimal operating value of FR locates in the range of 1–6, where the exergy efficiency varies in the range of 0.50–0.61 while the corresponding energy efficiency is 0.33–0.90 and the temperature lift is 14.9°C–44.1°C.

Figure 6 illustrates the upgraded useful heat power at a waste heat temperature of 80°C. The power of useful heat is only slightly reduced as FR increases from 1 to 4, e.g., from 11.97 kW to 11.20 kW in the case of R_{ref} = 0 and from 9.6 kW to 8.9 kW in the case of R_{ref} = 1. Thereafter, the useful heat power drops significantly towards zero as shown in Fig. 6. To obtain these amounts of useful heat, the input heat and electricity power required are exhibited in Fig.7. The required electricity consumed by the compres-

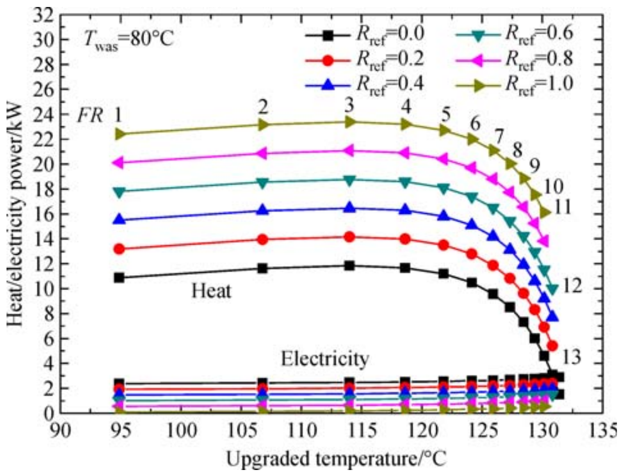

Fig. 7 Input heat and electricity power at a waste heat temperature of 80°C

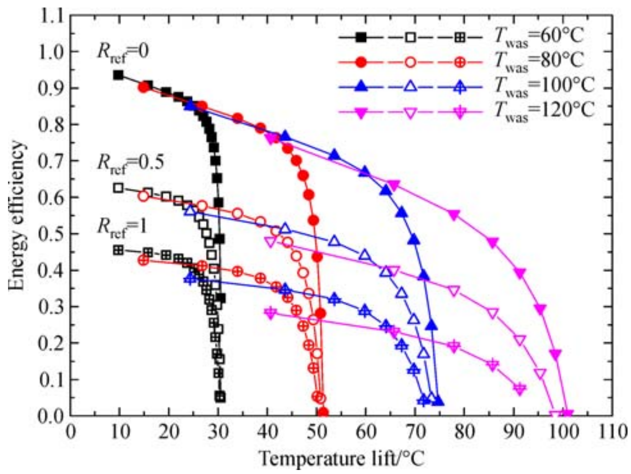

Fig. 8 Energy vs temperature lift of HAC-HP at waste heat temperatures of 60°C–120°C

sor and solution pumps is less than 3 kW, while the required heat for the generator and evaporator can be as high as nearly 24 kW. As shown in Fig. 7, when $R_{ref} = 0$, more electricity and less heat are consumed compared to that in the case of $R_{ref} = 1$. However, the extent of variation of heat is more notable than that of electricity due to the lower exergy possessed by thermal energy than electricity. More heat consumption indicates lower energy efficiency. Nevertheless, if treating the waste heat as totally free energy and only electricity is counted in Eq. (15), the system energy efficiency (or coefficient of performance) will be significantly higher and increase with the drop of R_{ref}.

Figure 8 displays the energy efficiencies of the HAC-HP system as a function of temperature lift at different waste heat temperatures from 60°C to 120°C. As shown in Fig. 8, the energy efficiency curves shift to the lower right side as the waste heat temperature increases, towards a higher temperature lift but a lower energy efficiency. For example, for the waste heat temperature of 120°C, the temperature lift is in the range of 40.7°C–101.0°C, the energy efficiency is lower than 0.76 when R_{ref} is 0 and is lower than 0.28 when R_{ref} is 1. Table 2 presents the recommended operation conditions and corresponding performances of the HAC-HP system before the violent drop of system energy and exergy efficiencies and output useful heat power.

4 Conclusions

The present study investigated a hybrid absorption-compression heat pump to upgrade and recover the industrial waste heat in the temperature range of 60°C–120°C. The system proposed can use two routes to elevate the working fluid pressure, one is via the compressor and the other is via the condenser-pump-evaporator, so that the use of energy sources of electricity and waste heat can be flexible. The major conclusions are summarized as follows.

(1) As the system recirculation flow ratio increased, the temperature lift was improved; however, the system energy and exergy efficiencies dropped violently when the recirculation flow ratio was larger than a certain value.

(2) More ammonia-water vapor flowing through compressor led to a higher system energy efficiency; however, the exergy efficiency changed little with the vapor splitting ratio.

(3) The recirculation flow ratio should be lower than 9, 6, 5, and 4 respectively as recommended for a waste heat temperature of 60°C, 80°C, 100°C, and 120°C respectively, while the temperature lifts were 9.8°C–27.7°C, 14.9°C–44.1°C, 24.4°C–64.1°C, and 40.7°C–85.7°C respectively, and the system energy efficiency were 0.35–0.93, 0.32–0.90, 0.25–0.85, and 0.14–0.76.

Table 2 Recommended *FR* and corresponding performance of HAC-HP system at different waste heat temperatures R_{ref} = 0–1

Waste heat temperature/°C	Recommended *FR*	Energy efficiency	Exergy efficiency	Useful heat/kW	Temperature lift/°C
60	< 9	0.35–0.93	0.55–0.67	7.2–11.7	9.8–27.7
80	< 6	0.32–0.90	0.50–0.61	7.3–11.9	14.9–44.1
100	< 5	0.25–0.85	0.37–0.56	5.7–11.9	24.4–64.1
120	< 4	0.14–0.76	0.21–0.50	3.5–11.2	40.7–85.7

Acknowledgements This work was supported by the Heat-STRESS project (EP/N02155X/1), funded by the Engineering and Physical Science Research Council of UK.

Notations

FR	Recirculation flow ratio
h	Enthalpy/$(J\cdot kg^{-1})$
$\dot{m}$	Mass flow rate/$(kg\cdot s^{-1})$
P	Pressure/Pa
$\dot{Q}$	Heat power/W
R	Ratio
T	Temperature/°C
ΔT_{LMTD}	Logarithmic mean temperature difference/°C
UA	Heat exchanger performance/$(W\cdot K^{-1})$
w	Mass fraction
$\dot{W}$	Electric power/W
h	Efficiency
Subscripts	
abs	Absorption
amb	Ambient
bas	Basic
com	Compressor
en	Energy
eva	Evaporation
ex	Exergy
gen	Generator
H	High pressure
HE	Heat exchanger
l	Liquid
L	Low pressure
min	Minimum
pole	Pole
pump	Pump
rec	Rectifier
ref	Refrigerant
refl	Reflux
s	Isentropic
use	Useful
v	Vapor
was	Waste

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