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Performance Comparison of Absorption Heat Pump and Refrigeration in Low Temperature Waste Heat Recovery

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Absorption heat pump and absorption refrigeration cycle are widely used in industries as two of the most important low-temperature waste heat recovery technologies. In order to compare their energy and economic performances, this paper establishes the models of one effect LiBr –H_2O absorption heat pump and one effect LiBr -H_2O absorption refrigeration cycle in the process simulation software Aspen PLUS. The two systems are simulated for waste heat source under temperature from 95 °C to 125 °C saturated water to output 130 °C saturated steam (absorption heat pump), or 7 and 10 °C cold water (absorption refrigeration). The energy and economic performances of the two technologies are compared and analyzed through three parameters, coefficient of performance, exergy efficiency, and operation cost per unit exergy output. The results show that the performance of heat pump is better than that of refrigeration cycle under each of these evaluation indexes. The work in this paper aims to provide some theoretical support and technology selection suggestion for low temperature waste heat recovery.

1. Introduction

The energy-intensive chemical industry combusts fossil fuels to provide heat and cold energy for production, which is accompanied by a large amount of low temperature waste heat discharged. Recovering the waste heat can greatly reduce the consumption of fossil energy. There exist lots of low-temperature waste heat recovery technologies, among which absorption refrigeration and heat pumps are widely used due to their good energy saving, and emission reduction effects. The research on the two technologies mainly focuses on three aspects: new working pairs, improvement of absorption performance, and development of new cycles (Wang, 2016).

Comparing different technologies can provide guideline for industries to choose a suitable one. Little and Garimella (2011) compared five kinds of thermodynamic cycles to produce cold energy, high-grade heat or mechanical work by using waste heat under the same input (60 °C and 120 °C). It is found that the refrigeration performance of an organic Rankine cycle is better than that of a Maloney-Robertson cycle, while that of an absorption refrigerator is better than that of an organic Rankine-vapor compression cycle. Farshi et al. (2012) studied and compared three kinds of double effect LiBr-H_2O absorption refrigeration systems with the same refrigeration capacity, and pointed out advantages and disadvantages of different configurations of absorption refrigeration systems. Bor et al. (2015) studied different types of heat pumps and power cycles based on economic benefits. They found that in the temperature range of 45 - 60 °C, heat pumps have better economic performance than that of power cycles, while organic Rankine and Kalina cycles become competitive at the residual heat temperature above 130 °C. According to the Composite Curve of waste heat streams, Wang et al. (2017) divided the waste heat of multistream into three types: straight, convex and concave. The thermodynamic performance of organic Rankine and Kalina cycles were compared and it was found that the performance was related to the shape of the Composite Curve of waste heat. Wang et al. (2018) compared a Kalina cycle and a LiBr-H_2O absorption refrigeration cycle for low-temperature waste heat recovery, and found the suitable conditions for the two technologies. Liu et al. (2019) analyzed the comprehensive performance of transcritical Rankine cycles for waste heat recovery of diesel engine using hydrocarbons and CO_2 as working fluid. Akman and Ergin (2019) studied the operation of waste heat recovery system of chemical oil tanker based on four

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different organic Rankine cycle under different working conditions through thermodynamic, environmental and economic analysis. Ren and Wang (2019) studied and compared the thermodynamic performance of organic transcritical cycle (OTC), steam flash cycle (SFC) and steam dual-pressure cycle (SDC), analyzed the influence of main parameters on the cycle performance, and optimized the parameters with the net power output as the objective function. Wang et al. (2020) based on Pinch Analysis of a given heat exchange network, put forward the systematic method for heat pump integration into industrial processes with an exergoeconomic criterion, and analyzed the influence of different waste heat temperature and temperature rise on heat pump selection. It can be seen that up to now, the comparison of energy-saving effect between absorption heat pumps and absorption refrigeration cycles in low-temperature waste heat recovery has not been reported.
This paper establishes the models of absorption heat pumps and absorption refrigeration cycles by Aspen PLUS in section 2, and selects three analysis methods in section 3. According to the simulation results in section 4, the energy and economic performances of the two cycles are compared from three aspects, coefficient of performance, exergy efficiency and operation cost, which results in conclusions in section 5. The work in this paper can provide theoretical support and selection suggestion for low-temperature waste heat recovery.

2. Simulation models

This paper establishes simulation models with the output of 130 °C steam of the single-effect LiBr-H_2O absorption heat pump, 7 °C and 10 °C cold water of the single-effect LiBr-H_2O absorption refrigeration cycle. The waste heat temperature is taken as 95, 100, 105, 110, 115 and 120 °C. The cycle is shown in Figure 1a.
The system includes a condenser, evaporator, absorber, generator, solution pump and other accessories. The dilute LiBr solution is heated by a driven heat source in the generator. After that, the refrigerant water enters the condenser, and the high-temperature concentrated solution enters the absorber after being cooled in the solution heat exchanger and depressurized by the valve. The refrigerant vapor releases heat in the condenser and enters the evaporator after being depressurized. The working fluid vaporizes in the evaporator by absorbing the heat from the waste heat (heat pump) or releases the cold energy to the water (refrigeration cycle), then enters the absorber to be absorbed by the concentrated solution, and releases heat to the cold fluid. The dilute solution ultimately enters the generator after being heated in the solution heat exchanger to complete a cycle.

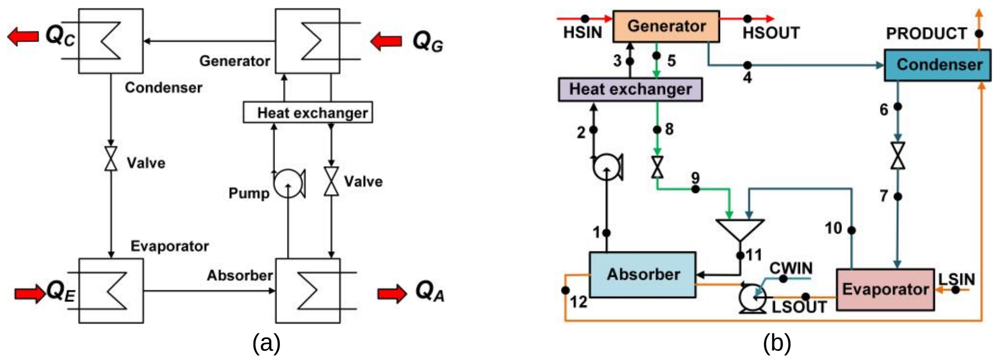

Figure 1: (a) Flowsheet of absorption heat pump / refrigeration cycle; (b) Selection of system state points

2.1 Property Methods

For pure water, the SteamNBS method is selected to calculate its related physical properties. The ELECNRTL method is specially designed for electrolyte calculation, so it is appropriate for LiBr-H_2O solution.

2.2 State points, assumptions

The following assumptions are made for the state point in this simulation. (1) The state at the outlet of the absorber is designated as state point 1, which is assumed to be saturated liquid. (2) The gas-phase state at the outlet of the generator is designated as state point 4, the liquid-phase state as state point 5, which is assumed to be saturated liquid, and state point 4 is the gas-phase in equilibrium with state point 5. (3) The outlet of hot stream of the condenser is designated as state point 6, assuming that is saturated liquid. (4) The inlet of hot stream of the evaporator is designated as state point LSIN, and the outlet is designated as LSOUT, assuming that LSIN is saturated vapor, and LSOUT is saturated liquid. (5) In the whole cycle, it is assumed that there is no pressure drop in each process. The selection of state points is shown in Figure 1b.

2.3 Models

1) Pump and valve

In Aspen Plus, the pumps of the two cycles are modeled by Pump module. The input of the pump model includes outlet pressure and efficiency (assumed as 1). The valves are modeled by Valve module. The outlet pressure of the solution valve depends on the pressure of the absorber, and the outlet pressure of the refrigerant valve depends on the pressure of the evaporator.

2) Solution heat exchanger

The model of the solution heat exchanger is simulated by the Heater module, as shown in Figure 2a. The two modules are connected by heat flow SHE-Q, whose heat duty reflects the heat transferred from the concentrated solution to the dilute solution. Generally, the heat duty of SHE-Q is uncertain. For the solution heat exchanger, only the inlet temperatures of the cold and hot streams are known. To calculate the outlet temperatures of the streams, another degree of freedom is needed. Somers (2009) took the heat exchange efficiency ε of the solution heat exchanger as 0.64, which is another degree of freedom. Its definition is shown in Eq(1).

$$\varepsilon = \frac{T_{HOTIN} - T_{HOTOUT}}{T_{HOTIN} - T_{COLDIN}} \tag{1}$$

3) Generator

The HeatX module and the Flash2 module are used to simulate the generator, as shown in Figure 2b.The temperature and heat duty of the heat source determine the flow rates of the refrigerant vapor and the concentrated solution. For the absorption heat pump, higher pressure steam is used as the heat source. For the absorption refrigeration cycle, the waste heat is used to heat the dilute solution. The HeatX module needs to specify any two variables of the vapor fraction, temperature or heat duty of the hot or cold stream at the outlet. The Flash2 module needs to specify any two of the flash temperature, flash pressure and flash heat duty. This paper specifies that the vapor separation rate at the outlet of the hot stream is 0. The latent heat of the steam is all transferred to the dilute solution in the heat exchanger. The flasher is specified adiabatically with zero heat duty. The pressure is the same as the inlet pressure. The vapor-liquid phase is separated into two phases after passing through the flasher. The vapor phase bocomes superheated vapor at the saturation temperature of the concentrated solution, and the liquid phase is saturated solution.

(a) Solution heat exchanger model (b) Generator model

Figure 2: Models of heat recovery units

4) Condenser

Two Heater modules are used to simulate the condenser, which are connected by the heat flow CON-Q, as shown in Figure 3a.The saturated refrigerant vapor changes into saturated liquid after passing through the condenser. Its temperature depends on the inlet and outlet temperatures of the cold stream. For the absorption heat pump, the inlet temperature of the cold stream is known. The heat transfer temperature difference is taken as 7 °C to calculate the saturated refrigerant liquid temperature and the saturation pressure. For the absorption refrigeration cycle, the outlet temperature of the saturated liquid of the condenser is generally 2 - 5 °C higher than that of the cold water. The cold water flows seriesly from the absorber to the condenser. When the outlet temperature of the cold water is specified, the saturated refrigerant liquid temperature at the outlet of the condenser is also determined, and so is the condensation pressure. The pressure of the condenser is equal to that of the generator, which determines the pressure at the high-pressure side of the whole cycle.

5) Evaporator

The evaporator is simulated by HeatX module. For the absorption heat pump, since the heat supplied to the evaporator is certain, the temperature of the saturated refrigerant vapor can be determined under a certain minimum heat transfer temperature difference. For the absorption refrigeration cycle, the temperature of the saturated refrigerant vapor at the outlet of the evaporator depends on the cooling temperature required.

Generally, the evaporation temperature is 2 - 5 °C lower than the outlet temperature of the refrigerated water. The evaporation temperature of the LiBr-H_2O refrigeration cycle varies from 0 °C to the ambient temperature theoretically. The minimum temperature of the evaporator is generally specified as 5 °C. The temperature of the refrigerated water is generally higher than 5 °C. In this paper, 7 °C and 10 °C are adopted. The pressure of the evaporator is the saturation pressure of water at the evaporation temperature. The pressure of the evaporator determines the pressure at the low-pressure side of the whole cycle.

6) Absorber

The Mixer module is used to simulate the adiabatic mixing process of the concentrated solution and refrigerant vapor, and the Heater module is used to simulate the heat release process in the mixing process, which is similar to the evaporator, as shown in Figure 3b. The input of the Mixer is the outlet pressure during adiabatic mixing. Since the pressure of the concentrated solution is the same as that of the refrigerant vapor from the solution valve, this process is a constant-pressure process. The pressure drop of the Mixer is specified as 0 Pa, and the solution pressure after mixing is equal to the low pressure of the whole cycle. The pressure of the absorber is generally about 27 - 80 Pa lower than that of the evaporator (Dai,1996). During the heat exchange, the latent heat of the vapor in the mixture is completely exchanged.

(a) Condenser model (b) Absorber model

Figure 3: Models of mass transfer related units

3. Analysis methods

In this section, the coefficient of performance, exergy efficiency, and operation costs per unit exergy output are used to evaluate the performance of the absorption heat pump and absorption refrigeration cycle.

3.1 Coefficient of performance

The coefficient of performance（COP）of the absorption heat pump is shown in Eq(2).

$$COP = \frac{Q_{ABS} + Q_{CON}}{Q_{GEN} + Q_{EVAP} + W_x} = \frac{H_{PRODUCT}}{H_{HWIN} + W_x'} \tag{2}$$

where Q_{ABS}, Q_{CON}, Q_{GEN} and Q_{EVAP} refer to the heat duty of the absorber, condenser, generator and evaporator. W_x denotes the mechanical work input by the pump. H represents the total energy of a certain stream, whose value is equal to the product of specific enthalpy and mass flowrate. The subscripts PRODUCT and HWIN represent the output hot stream and the input waste heat source. W_x' refers to the heat provided by the high temperature heat source and the mechanical work provided by the pump.
The COP of the absorption refrigeration cycle is shown in Eq(3).

$$COP = \frac{Q_{EVAP}}{Q_{GEN} + W_x} \tag{3}$$

where Q_{EVAP} indicates the output cold capacity of the evaporator. Q_{GEN} refers to the heat absorbed by the generator. W_x refers to the mechanical work input by the pump.

3.2 Exergy efficiency

Exergy is used to represent the quality of energy because heat and cold energy have different energy levels.
The method to calculate physical exergy is shown in Eq(4).

$$E = m \cdot [h - h_0 - T_0(s - s_0)] \tag{4}$$

where m refers to the mass flow rate of streams. h and h_0 refer to the specific enthalpy of stream in the current and reference state. T_0 represents reference temperature, taken as 298.15 K. s and s_0 mean specific entropy of stream in current and reference condition.
In this paper, the exergy values of the streams are obtained by Aspen Plus simulation.
The exergy efficiency of the absorption heat pump is shown in Eq(5).

$$\eta=\frac{E_{PRODUCT}-E_{CWIN}}{E_{HWIN}-E_{HWOUT}+E_x} \tag{5}$$

where E refers to the exergy of streams. The subscripts PRODUCT, CWIN, HWOUT and HWIN represent the output hot stream, the input cold stream, and the input and output waste heat. E_x refers to the exergy of the low pressure steam input.
For absorption refrigeration cycle, its exergy efficiency is calculated as shown in Eq(6).

$$\eta=\frac{E_{CWOUT}-E_{CWIN}}{E_{HWIN}-E_{HWOUT}} \tag{6}$$

where the subscripts CWOUT and CWIN represent the output and input refrigerated water.

3.3 Operating cost

Since the basic components of the two systems are the same and the investment costs are similar, this paper mainly considers the operating cost per unit exergy output to evaluate the economic performance. Operating cost includes the costs of electricity, water and low-pressure steam. In this study, the electricity fee is taken as 0.8 yuan/(kW·h), the water cost is taken as 5 yuan/t, and the cost of 170 °C steam is taken as 180 yuan/t. The annual operating time is 8, 000 h.

4. Results

According to the calculation results based on the simulation model in Aspen PLUS, the performance coefficient, exergy efficiency and operating cost per unit exergy output of the absorption heat pump and refrigeration cycle under different working conditions are summarized in Table 1.

Table 1: calculation results

	Wwaste heat temperature (°C)	COP (1)	Exergy efficiency (%)	Operating costs per unit exergy output (yuan/kWh)
Heat pump	95	1.00	78.10	0.40
(130 °C steam)	100	1.00	80.69	0.60
	105	1.00	80.60	0.69
	110	1.00	79.47	0.68
	115	1.00	83.59	0.69
	120	1.00	84.38	0.68
Refrigeration	95	0.33	9.26	9.86
cycle	100	0.63	17.45	6.20
(7 °C water)	105	0.71	19.36	5.52
	110	0.73	19.22	5.57
	115	0.74	18.88	5.63
	120	0.75	18.36	5.38
Refrigeration	95	0.40	8.83	12.02
cycle	100	0.66	14.42	8.72
(10 °C water)	105	0.72	15.59	7.96
	110	0.75	15.51	8.02
	115	0.76	15.25	8.09
	120	0.76	14.80	7.75

From Table 1 the following can be easily seen.
(1) When the the waste heat temperature increases, the COP of the absorption heat pump remains to be 1.00, while that of the absorption refrigeration cycle increases, and achives 0.75 (7 °C) and 0.76 (10 °C) at 120 °C. The COP of the refrigeration cycle is always lower than that of the heat pump.

(2) The exergy efficiency of the absorption heat pump increases with the temperature of the waste heat, which achives 84.38 % at 120 °C. The refrigeration cycle with 7 °C water output and 10 °C water output can both reach the highest efficiency when the temperature of the waste heat is 105 °C, which are 19.36 % and 15.59 %. The efficiency of the absorption heat pump is always higher than that of the absorption refrigeration cycle.
(3) The operating cost per unit exergy output of the absorption heat pump increases with the increase of the waste heat temperature, while that of the refrigeration cycle decreases with the increase of the waste heat temperature. The operating cost of the heat pump is always lower than that of the absorption refrigeration cycle.

5. Conclusions

In order to select an appropriate technology to recover low-temperature waste heat, this paper compares the absorption heat pump and refrigeration cycle based on the performance coefficient, exergy efficiency and operating costs per unit exergy output. The conclusions are as follows.
(1) With increasing temperature of the waste heat, the COP of the absorption heat pump remains unchanged, while that of the absorption refrigeration cycle increases with the waste heat temperature.
(2) The exergy efficiency of the absorption heat pump increases with the temperature of the waste heat source, and that of the refrigeration cycle has a maximum value relative to the temperature of the waste heat.
(3) The operating cost per unit exergy output of the absorption heat pump increases with the waste heat temperature, while that of the refrigeration cycle decreases with the temperature.
(4) The absorption heat pump is better than the absorption refrigeration cycle in low-temperature waste heat recovery from the thermodynamic and economic aspects.
In fact, these two technologies both are widely used in waste heat recovery, but their application occasions are different. When selecting waste heat recovery technologies, both the temperature of the waste heat source and the demand of the user side should be considered. The future work should explore more working conditions and more user demand to provide suggestions for industrial selection.

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