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ABSTRACT

The thermodynamic and thermo-economics models for three-pressure absorption-ejector hybrid refrigeration system are set up. The thermo-economical analysis model of the system is considered in two cases of high-temperature heat resources: waste heat resources and natural gas fuel are presented. The performances of the system in two modes of the running hours per year (600h and 1000h) are calculated and discussed to show the commercial perspective of the absorption-ejector hybrid refrigeration system.

INTRODUCTION

Since the novel absorption-ejector hybrid refrigeration cycles, including the three-pressure absorption-ejector hybrid refrigeration system (three-pressure AEHRS) and the four-pressure absorption-ejector hybrid refrigeration (four-pressure AEHRS), were presented by GU and YU in 1994 [1], the further characteristics of the cycles were studied with the working pair of R21-DMF [2～4]. The influences of the temperature and the pressure parameters over the performances of the refrigeration systems with CH_3OH-LiBr-$ZnCl_2$ in different conditions were then discussed [5]. Some system design parameters with LiBr-H_2O as the working pair were analyzed and optimized [6, 7]. This novel system [8] and then the optimum operating condition [9] and ejector design data [10-12] were also described and investigated. An experimental study of the three-pressure AEHRS showed that the coefficient of performance (COP) of the three-pressure AEHRS is increased by 30～60%, compared to the single-absorption refrigeration cycle and near the COP of small commercial double-effect absorption refrigeration system [13-15].

However, as a practical refrigeration system, the commercial perspective of the absorption-ejector hybrid refrigeration system lies in the thermo-economical performance, that is to say, we should consider the cost-energy tradeoff. In this paper the thermo-economical analyses of the three-pressure absorption-ejector hybrid refrigeration system and small double-effect refrigeration system are studied and compared to show which one is better and why.

COMPARISON OF COP BETWEEN THREE-PRESSURE AEHRS AND DOUBLE-EFFECT ABSORPTION SYSTEM

Figure 1 and Figure 2 are the schematics of the double-effect absorption system (DEAS) and the three-pressure absorption-ejector hybrid refrigeration system, respectively. When the comparison of COP between the three-pressure absorption-ejector hybrid refrigeration system and the small double-effect absorption refrigeration is carried out, the following assumptions are included.

(a) LiBr-H_2O as the working-pair of the two systems and the flows in two systems are steady.

(b) The total concentration difference of the solution is no more than 5.5% so that the crystallization of the solution will not occur at the outlet of the heat exchanger in the side of high concentration solution.

(c) The efficiency of the heat exchanger is η =0.9. All heat exchangers have no heat loss to the ambience.

(d) The cooling capacity of two systems is 30kW. The temperatures of generator, condenser, evaporator and absorber of two systems are 170℃, 42℃, 7℃and 40℃, respectively. It should be noted that it is reasonable to put DEAS in operation with a high-temperature of 170℃, which is an advantage compared to the single-effect absorption refrigeration system [16].

(e) The deviations of heat balance of the two systems are less than 10%, respectively.

(f) The solution rates are 12 respectively.

The simulation results of the two systems [17, 18] are shown in Figure 3. The evaluation of the performance with temperature of evaporator (T_e) shown in Figure 3 is calculated theoretically, which is higher than that in real case. The coefficient of performance of the double-effect absorption refrigeration system is slightly higher than that of the three-pressure absorption-ejector hybrid refrigeration system. The difference of COP between two systems results from that in the double-effect refrigeration system the temperature of the low-concentration solution at the outlet of the high-temperature heat exchanger reaches 154℃ after it flows through the low-temperature heat exchanger and high-temperature heat exchangers, consecutively, while in the three-pressure absorption-ejector hybrid refrigeration system the temperature of the low-concentration solution at the outlet of heat exchanger is only 138℃ The evaporate temperatures in DEAS and AEHRS all keep 7℃.

In the double-effect absorption refrigeration system, the vapor, or the primary vapor, from the high-pressure generator comes into the low-pressure generator, heating the medium-concentration solution from the high-pressure generator and then being condensed. At the outlet of low-pressure generator, the medium-concentration solution becomes high-concentration solution and the secondary vapor is produced, which, together with the condensed water of the primary vapor, flows into the condenser in which the heat rejection takes place. It is well known that the latent heat of the primary vapor from the high-pressure generator is efficiently used in the low-pressure generator to get the secondary vapor. Comparatively, the primary vapor from the generator in the three-pressure absorption-ejector hybrid refrigeration system induces the low-pressure vapor from the evaporator. The mixed vapor at the outlet of the ejector is in superheated state. Therefore, the exergy efficiencies of the two systems are different because of the different modes of using the driving energy. Here, we define the exergy efficiency of the system as the following formula:

$$Exergy\ \ efficiency = \frac{Exergy\ \ output}{Exergy\ \ input} \times 100\% \qquad (1)$$

The exergy efficiencies of the two systems are 26.18% for the double-effect absorption refrigeration system and 19.64% for the three-pressure absorption-ejector hybrid refrigeration system, respectively. So the double-effect refrigeration system is more efficient than three-pressure absorption-ejector hybrid refrigeration system in utilization of the high value energy.

Figure 3 also shows that, compared to the heat capacity of the three-pressure absorption-ejector hybrid refrigeration system, the total capacity of the two solution heat exchangers of the double-effect absorption refrigeration system increases by about 50% because of adding the low-temperature heat exchanger. In addition, considering the miniaturization of the system design, the cooling capacity beyond 30kW, the proportion of no efficacy area of heat exchangers increase, which increases the primary investment.

There are no low-pressure generator and low-temperature heat exchanger in the three-pressure absorption-ejector hybrid refrigeration system and thus the primary investment will be decreased. As a practical refrigeration system, the commercial perspective of the absorption-ejector hybrid refrigeration system lies in the thermo-economical potential; that is to say, we should consider the cost-energy tradeoff between the three-pressure absorption-ejector hybrid refrigeration system and small double-effect refrigeration system.

THERMAL ECONOMICAL MODELS

The annual total cost is evaluated, including the primary cost of equipment, running fare per year and the value depreciation of equipment. The primary investment of the system is composed of the investment of all heat exchangers, ejector (in AEHRS), all pumps for the circulation of low concentration solution and cooling water, and fans for cooling tower. The running fare is the expenditure of driving heat source, cooling water and the electricity.

For different heat resources there are different costs and thus annual total costs. Two cases, such as waste heat resources and natural gas fuel, are considered in this paper.

Thermo-Economical Model of Refrigeration Systems Using Waste Heat as Heat Source

Using low-grade energy, such as waste heat and flue gas, is one of the effective methods to save energy and improve economical benefit of the system. If the cost of waste heat is naught, the annual total cost of the two systems can be formulated as follows, respectively.

$$ATC_{dou} = \delta\left(Z_{hg} + Z_{\lg} + Z_{hex} + Z_{lex} + Z_{c} + Z_{e} + Z_{a} + Z_{ct} + Z_{ps} + Z_{fan}\right) + C_{ele} B\left(W_{ps} + W_{fan}\right) \quad (2)$$

$$ATC_{hyb} = \delta\left(Z_{g} + Z_{ex} + Z_{e} + Z_{c} + Z_{a} + Z_{ej} + Z_{ct} + Z_{ps} + Z_{fan}\right) + C_{ele} B\left(W_{ps} + W_{fan}\right) \quad (3)$$

Where δ is the depreciation ratio and can be evaluated as [11, 12]

$$\delta = \frac{i\left(1 + i\right)^{n}}{\left(1 + i\right)^{n} - 1} \quad (4)$$

The cost of heat exchangers is the function of the heat transfer area, A. The running cost of pumps and fans is the function of the power, W [19].

$$Z_{A} = p_{1} + q_{1} \cdot A^{r_{1}} \quad (5)$$

$$Z_{P} = p_{2} + q_{2} . P^{r_{2}} \quad (6)$$

Where, the subscript, *A*, in Equation (5) represents the heat transfer area of all kinds of heat exchanger; the subscript, *P*, in Equation (6) represents the power-consumption of all pumps and fans. p_{1}, q_{1}, r_{1}, p_{2}, q_{2} and r_{2} are constant.

The heat transfer area, *A*, for given fluid temperatures and flow capacities can be calculated by [10, 11]

$$A = \frac{Q}{h\left(\Delta - a \cdot \Delta t_{1} - b \cdot \Delta t_{2}\right)} \quad (7)$$

Where *Q* is the heat capacity of heat exchangers, kW; *h* is the overall heat-transfer rate, kW/(m^2.K); *Δ* is the maximum temperature difference between the two flow-streams, ℃; Δt_{1} and Δt_{2} are the temperature differences in each side of fluids, respectively. *a* and *b* are constants. *a* is 0.35 for the counter-flow, 0.65 for the parallel-flow and 0.45～0.55 for the cross-flow; *b* is equal to 0.65.

Because the primary investment of the ejector in the three-pressure absorption-ejector hybrid refrigeration system is much lower than all the heat exchangers, Z_{ej} in Equation (3) is ignored [19].

Thermo-Economical Models of Refrigeration Systems Using Natural Gas Fuel

Because of the phase-out of CFCs and HCHCs in cascade refrigeration system [20], the use of absorption refrigeration system, using heavy oil or natural gas as the fuels of heat sources, has been expanded. The refrigeration systems using natural gas fuel are considered. The costs for the consumption of natural gas fuel are added to the annual total costs of the three-pressure absorption-ejector hybrid refrigeration system and the double-effect refrigeration system, respectively, in addition to the annual total cost expressed in Equation (2) and (3). The cost of natural gas fuel is $C_{gas}BG_{gas}$, where B is the running hours per year, C_{gas} is the price of natural gas per normal cubic meter, and G_{gas} is given by

$$G_{gas} = \frac{Q_g}{\xi \cdot \hbar_g} \tag{8}$$

RESULTS AND DISCUSSIONS

In order to analyze the influences of the uncertainties of parameters, two modes of the running hours per year, such as 600h per year and 1000h per year, are analyzed [21]. These running hours resulted from the investigation in Xi'an in 1999. In each mode, the annual total costs of the two systems are calculated for given electricity price, C_{ele} and the life-year of the system, n. The life-year of the system is fixed 10 years. The exchange rate is 1 USD = 8.27 RMB. The electricity price, C_{ele}, is 0.045 USD/kW.h for Xi'an City of China in 1999. ξ and $\hbar$ are 0.85 and 35,000 kJ/Nm^3, respectively. The regions between the two curves, corresponding to two modes of running hours per year (600h per year and 1000h per year), for the three-pressure absorption-ejector refrigeration system and the double-effect absorption refrigeration system, respectively, mean the annual total cost of the two systems for the running hours per year between 600h and 1000h per year as shown in Figure 4 and Figure 5.

The cost of power-consumption is only small fraction in the annual total cost of the two systems. No cost for waste heat is assumed when waste heat is used as the heat source. The annual total cost of the two systems is strongly influenced by the primary investment and thus the depreciation rate, or the annual interest rate. So the annual interest rate is considered as a variable when waste heat is used as the heat resources of the two systems. The annual total costs of the two systems in two modes of the running hours per year are shown in Figure 4. The temperature of waste heat resource is 250℃. There is no overlap between annual total costs of the two systems in two modes of the running hours per year. This indicates that the annual total cost of the three-pressure absorption-ejector hybrid refrigeration system is lower than that of the double-effect absorption refrigeration system.

When natural gas fuel is used, the natural gas cost is large fraction in the annual total cost. So the price of natural gas, C_{gas}, is considered a variable and the relative price ratio, $\beta = \frac{C_{gas}}{C_{gas0}}$, is used to show the influence of the natural gas price on the annual total cost, as shown in Figure 5. C_{gas0}=0.134 USD/Nm^3 for Xi'an City in 1999; C_{gas} is the changeable price in the market. The temperature of the heat resource (high-temperature gas) is 800℃ β_1、 β_2 and β_3 are three crossing points of the annual total cost curves of the two systems in two modes of the running hours per year. There is an overlap between the two regions of the annual total cost of the two systems for different running hours per year. This means that the annual total cost of the three-pressure absorption-ejector refrigeration system is competitive to that of the double-effect absorption refrigeration system in the shadow zone shown in Figure 5.

CONCLUSIONS

The comparisons of COP and the cyclic characteristics between the three-pressure absorption-ejector hybrid refrigeration system and small double-effect absorption refrigeration system are carried out. The coefficient of performance of the three-pressure absorption-ejector refrigeration system is up to 0.9-1.0 and is slightly lower than that of the commercial double-effect absorption refrigeration system.

The thermo-economical analysis models of the two systems are presented. The thermo-economical performances in two modes of the running hours per year (600h and 1000h), using waste heat resources and natural gas fuel as the high-temperature heat sources, respectively, are calculated.

No cost for waste heat is assumed when waste heat is used as the heat source. The annual total cost of the two systems is strongly influenced by the primary investment and thus the depreciation rate, or the annual interest rate. The annual total cost of the three-pressure absorption-ejector hybrid refrigeration system is lower than that of the double-effect absorption refrigeration system.

When natural gas fuel is used, the natural gas cost is large fraction in the annual total cost. There is an overlap between the two regions of annual total costs of the two systems for different running hours per year. This means that the annual total cost of the three-pressure absorption-ejector refrigeration system is competitive to that of the double-effect absorption refrigeration system in the overlapped zone.

ACKNOWLEDGEMENT

The National Nature Science Foundation of China is appreciated for the financial support of the work in this paper, under the project No. 59806010.

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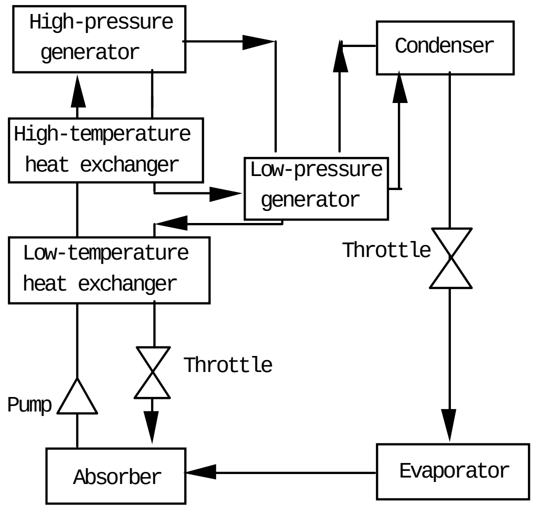

Figure 1 Schematics of double-effect absorption refrigeration system

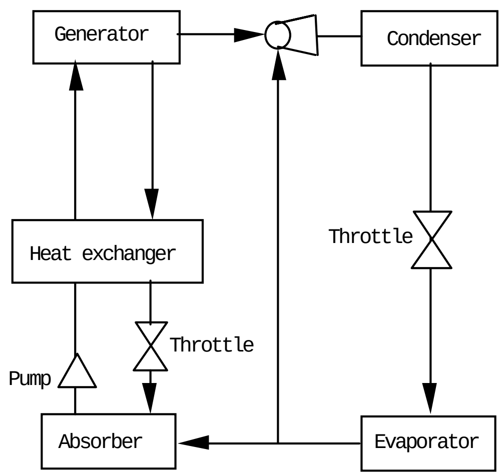

Figure 2 Schematics of three-pressure absorption-ejector hybrid refrigeration system

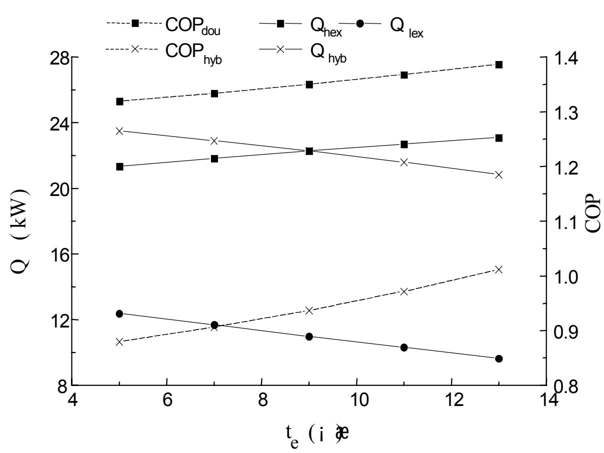

Figure 3 COP and the heat capacity of the heat exchangers at different evaporator temperatures

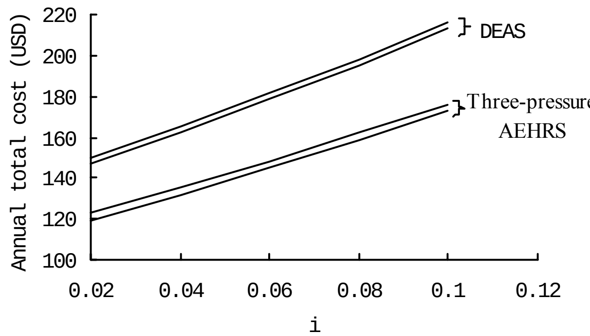

Figure 4 Annual total costs of the two systems (waste heat)

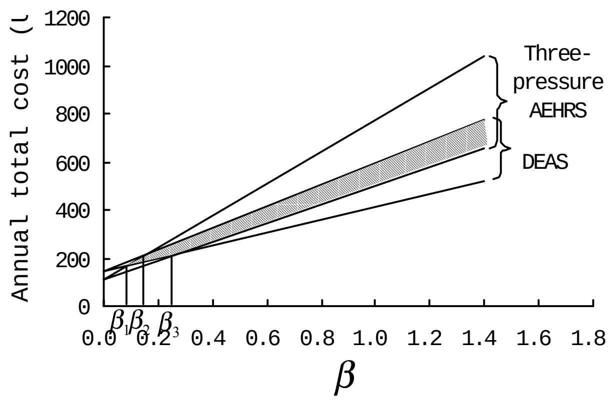

Figure 5 Annual total costs of the two systems (natural gas fuel)