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Performance analysis of different high-temperature heat pump systems for low-grade waste heat recovery

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HIGHLIGHTS

- Different heat pump systems were introduced to recover the heat of waste water.
- Thermodynamic and economic performances of those systems were analyzed and compared.
- The two-stage heat pump system with flash tank was preferred.

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ABSTRACT

Different heat pump systems were used to recover the heat from waste water with mean temperature of 45 °C and produce hot water with the temperature up to 95 °C. Those systems include single-stage vapor compression heat pump (system 1), two-stage heat pump with external heat exchanger (system 2), two-stage heat pump with refrigerant injection (system 3), two-stage heat pump with refrigerant injection and internal heat exchanger (system 4), two-stage heat pump with flash tank (system 5) and two-stage heat pump with flash tank and intercooler (system 6). Thermodynamic and economic analyses were conducted to compare the performance of each system. Results showed that the COP and exergy efficiency for both system 5 and system 6 are quite close, and much higher than those of other systems. Besides, the payback period of both system 5 and system 6 are also shorter as compared to other systems. Considering both the thermodynamic performance and economic quality of the system, system 5 is finally preferred since less initial investment is required for system 5 as compared to system 6.

1. Introduction

Heat pump is widely used in domestic and commercial buildings and industrial processes [1], which can efficiently recover waste heat and reduce greenhouse gases. Among all the types of heat pump, mechanical vapor compression heat pump is the most common one, which shows great potentials in process industries such as lumber drying, food and beverage production and petroleum refining [2–7]. For industrial applications, usually, a higher output temperature of heat pump is required. However, in order to achieve a high output temperature, the compression ratio of compressor for the single-stage heat pump is very high, which lowers the compression efficiency and degrades the heating capacity and the coefficient of performance (COP). As a result, a multistage system design, for example, two-stage vapor compression system is required to overcome those problems.

Compared to a single-stage heat pump system, the compression ratio of two-stage system is smaller and the compression efficiency is higher, so it is possible to achieve higher COP. Some researchers have already carried out both theoretical and experimental studies on the thermodynamic performance of two-stage heat pump system. Bertsch and Groll [8] simulated, designed, constructed, and tested an air-source two-stage heat pump at ambient temperatures as low as −30 °C and supply temperatures of up to 50 °C in air and water heating mode. The results showed that a coefficient of performance of 2.1 could be achieved, which were better than most currently available one-stage air-source heat pumps. Wang et al. [9] reported a double-stage coupled heat pumps (DSCHP) heating system, which couples air source heat pump (ASHP) and water source heat pump (WSHP) together with the objective to improve the working condition and heating performance of the air source heat pump (ASHP) under cold environment. The test results

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indicated that the operating characteristics of the DSCHP heating system were greatly improved and the system can offer considerable application potential in cold regions. A heat pump system with flash tank coupled with scroll compressor was also experimentally studied and compared with a system with a sub-cooler by Ma and Zhao [10] in order to find solution for system performance drop caused by large compression ratio. The results indicated that the heating capacity decreased much slower than that of a conventional air-source heat pump with the decrease of the evaporation temperature and the power input varied slightly with the evaporation temperature. In addition, it was showed that the heat pump system with a flash tank is more efficient than the system with a sub-cooler at low ambient temperatures, making it much suitable for small capacity air-source heat pump. K.J. Chua et al. [11] experimentally studied a two-stage evaporator heat pump drying system with R22 as the working fluid. It was demonstrated that up to 35% more heat could be recovered via the two-stage evaporator system in comparison to a single evaporator system.

Besides, exergy analysis was undertaken by some researchers to study the potential of improvement for the thermodynamic quality of the heat pump system. Jan et al. [12] expressed the reasons of the considerable decrease of COP of a real heat pump in comparison with the Carnot heat pump by means of the component thermodynamic efficiencies. The influences of the particular irreversible processes on the coefficient of performance of the considered heat pump were determined. They pointed out that the component efficiencies can indicate the possibilities of the improvement of the installation. Ma and Li [13] set up an exergetic model for a heat pump system with economizer coupled with scroll compressor based on the second law of thermodynamics. With this model, the influence of the intermediate pressure on the performance of the heat pump system was analyzed in detail based on the experimental data of the heat pump prototype. Hepbasli et al. [14] also assessed the performance of a heat pump utilizing geothermal resource by both energy and exergy analysis. The study indicated that the exergy analysis is a very useful tool in the performance evaluation of the heat pump system for the possibilities of thermodynamic improvement.

Although the parameters like COP and exergy efficiency can clearly present the thermodynamic performance of different heat pump systems, it is hard to evaluate the economic performance of different heat pump systems with different complexities because of their different initial investments. To date, few studies on analyzing both the thermodynamic and economic performances of the heat pump system can be found in the open literature. In this paper, five different two-stage compression heat pump systems were used to recover the heat from the oil field waste water and produce hot water with the temperature up to 95 °C. Thermodynamic and economic analysis was performed to compare the thermodynamic and economic performances of different two-stage heat pump systems with high output temperature. Parameters such as the average system power consumption, the efficiency and the payback period of each system were calculated and compared. The studies can provide guidance for selecting suitable two-stage heat pump system with high output temperature for industrial applications.

2. Systems description and analysis

Thermodynamic and economic performances of different high-temperature heat pump systems were compared. They are, respectively, single-stage heat pump used as the baseline (system 1), two-stage heat pump with external heat exchanger (system 2), two-stage heat pump with refrigerant injection (system 3), two-stage heat pump with refrigerant injection and internal heat exchanger (system 4), two-stage heat pump with flash tank (system 5) and two-stage heat pump with flash tank and intercooler (system 6). The scheme and p–h diagram for each heat pump are presented in Fig. 1. In the analysis, the oil field waste water is used as heat source in the evaporator of heat pump and the data of waste water is obtained from an oil production base in China [15]. The average temperature of waste water is 45 °C and the mass flow rate is 10 kg s^{-1}. It is assumed that the hot water produced is heated from 70 °C to 95 °C in the condenser of heat pump, which is the standard water-bath temperature range for the crude oil. R152a is selected as working fluid of heat pump and its thermodynamic properties is obtained from REFPROP 7.1 [16]. The parameters in the calculations can be found in Table 1. During the operation of heat pump system, it is assumed that the refrigerant is evaporated in the evaporator and heated with a superheat temperature of 3 °C at the outlet of the evaporator. The pinch point temperature between the waste water and the refrigerant is set to be 5 °C. In the condenser, the refrigerant is condensed and subcooled by 5 °C at the outlet of the condenser. Note that for the two-stage heat pump with refrigerant injection and internal heat exchanger, it is assumed that the refrigerant is additionally subcooled by 3 °C through the internal heat exchanger.

To characterize the thermodynamic and economic performances of those high temperature heat pump systems, the mass flow rate of refrigerant, the average power consumption of the compressor, the mass flow rate of the hot water produced, the COP, exergy efficiency and payback period for each system were analyzed. Details are as follows.

2.1. Thermodynamic analysis

2.1.1. Mass flow rate of working fluid

In the evaporator, heat is transfer from the waste water to the refrigerant. Simply, the heat released by waste water can be calculated by

$$Q_{\mathrm{eva}} = m_{\mathrm{w1}} c_{\mathrm{p}} \left(t_{\mathrm{w1,eva,in}} - t_{\mathrm{w1,eva,out}} \right) \tag{1}$$

where m_{w1} is the mass flow rate of waste water, c_{p} is the specific heat of waste water, $t_{\mathrm{w1,eva,in}}$ and $t_{\mathrm{w1,eva,out}}$ represent the temperature of waste water at the inlet and outlet of the evaporator, respectively.

Based on the conservation of mass and energy, the mass flow rate of working fluid in the evaporator can be easily calculated. For the heat pump of system 1 and system 2, it can be given by

$$m_{\mathrm{R}} = Q_{\mathrm{eva}} / \left(h_{\mathrm{R,eva,out}} - h_{\mathrm{R,eva,in}} \right) \tag{2}$$

where $h_{\mathrm{R,eva,in}}$ and $h_{\mathrm{R,eva,out}}$ are the refrigerant enthalpy values at the inlet and outlet of evaporator.

For other two-stage heat pump systems, the mass flow rate of refrigerant in low-pressure stage can be calculated by

$$m_{\mathrm{R,low}} = Q_{\mathrm{eva}} / \left(h_{\mathrm{R,eva,out}} - h_{\mathrm{R,eva,in}} \right) \tag{3}$$

And, the total mass flow rate of refrigerant in high-pressure stage for system 3, system 4, system 5 and system 6 can be, respectively, calculated by

$$m_{\mathrm{R,high,system\ 3}} = m_{\mathrm{R,low,system\ 3}} (h_2 - h_7)/(h_3 - h_7) \tag{4}$$

$$m_{\mathrm{R,high,system\ 4}} = m_{\mathrm{R,low,system\ 4}} (h_2 - h_8)/(h_3 - h_5) \tag{5}$$

$$m_{\mathrm{R,high,system\ 5}} = m_{\mathrm{R,low,system\ 5}} (h_9 - h_7)/(h_9 - h_6) \tag{6}$$

$$m_{\mathrm{R,high,system\ 6}} = m_{\mathrm{R,low,system\ 6}} (h_2 - h_7)/(h_3 - h_6) \tag{7}$$

where h_{i} ($i = 2, 3, 5, 6, 7, 8, 9$) represents the enthalpy value of the refrigerant at the state point of i as marked in Fig. 1.