APPLICATION OF INDUSTRIAL WASTE HEAT IN SOLAR REFRIGERANT SYSTEM
An Example of a Textile Factory in Jinjiang

by

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A waste heat recovery solar ejector-compression refrigeration system was designed, and a thermodynamic model was established to deal with the waste heat produced by a textile factory in China. The influence of the outlet temperature of the heat recovery device on the coefficient of performance and the generator heat load were researched under different refrigerants, the energy saving efficiency between waste heat recovery solar ejector-compression refrigeration system and its conventional solar ejector refrigeration system was compared. This paper showed that a higher outlet temperature resulted in a lower heat load of generator and a higher coefficient of performance. When the temperature increment was 30 °C, the heat load of the generator could be saved by 31.98%, and the coefficient of performance could be increased by 42.94%.

Key words: *waste heat recovery, solar ejector-compression refrigeration system, thermodynamic analysis*

Introduction

According to statistics, fabric productions will increase by 1.61 times in 2021 than that in 2016. However, waste water and exhaust gas generated by textile factory will increase by more than 1.61 times [1]. The energy loss of waste water from printing and dyeing process is more than 29% in textile industry [2]. If the waste heat is released into the atmosphere directly, it will cause serious energy waste and environmental pollution. Therefore, finding potential applications of waste heat from textile factory can greatly reduce the shortage of high grade energy and improve the serious environmental pollution effectively.

The solar ejector refrigeration system (SERS) has a broad development prospect due to its utilization of simple structure and low investment. At present, many researches about the utilization of waste heat in SERS have been carried out. Korolija and Greenough [3] designed and manufactured a small scale steam ejector experimental device driven by waste heat and solar energy. Kumar and Khalig [4] studied the feasibility of SERS using waste heat as driving source in the experimental data center environment. Zegenhagen and Ziegler [5] analyzed the practicability of an exhaust gas driven ejector refrigerant system which uses the engine waste heat as driving source. Noghrehabadi *et al.* [6] and Zheng *et al.* [7] investigated the performance

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of SERS driven by solar energy. Huang *et al.* [8] applied ejector to a conventional absorption refrigeration system and conducted a simulation analysis. Zhang *et al.* [9] focused on the flow and transportation process in an ejector, not only considered the change of ejectors geometry design, but also studied how the external conditions changing affect its performance. Zhang *et al.* [10] proposed a compound cycle system for automotive waste heat recovery and air conditioning refrigeration. Those results showed that the system had highly energy saving efficiency and environmental benefits, and could be used for the recovery of low-grade waste heat.

In summary, it is confirmed that the SERS has a bright prospect in waste heat recovery. The extensive works about waste heat application from the textile factory have been carried out widely. However, there are few researches about the application of the textile factory waste heat recovery in SERS. Therefore, this paper presented a waste heat recovery solar ejector-compression refrigeration system (WHRSERS), and studied the influence of the heat recovery device outlet temperature on the coefficient of performance (COP) and generator heat load.

Thermodynamic analysis of waste heat recovery system

Based on the investigation about the textile factory in Jinjiang, the waste water temperature was between 50-80 °C and daily average amount about 2500 tons were discharged from dyeing process of a high temperature and high pressure machine. In addition, the exhaust gas was emitted *via* five sets of heat setting machines, the rated exhaust capacity of a single setting machine could achieve 20000 m^3/h, and the average exhaust gas temperature could reach 120 °C. In order to effectively utilize waste heat, two application styles were developed according to the temperature range. The waste heat from exhaust gas was treated as the auxiliary driving source of WHRSERS, the waste heat of waste water was used to preheat the refrigerant.

Schematic diagram of waste heat recovery system

The schematic diagram of WHRSERS is shown in fig. 1. The system consists of an ejector subsystem, a compression subsystem, and they are connected by an inter-cooler – k. The ejector subsystem utilizes waste heat from exhaust gas as the auxiliary driving source, and the waste heat from waste water are used to preheat the refrigerant before entering the generator – c, to realize the recycle of waste heat. The working principle of the WHRSERS is as follows: the refrigerant absorb heat from solar collector – a, and waste heat recovery device – e, it is vaporized and pressurized with heat exchanging in generator, c, to produce high-temperature and high pressure refrigerant steam. This steam (firstly fluid) enters the ejector – g, and then flows through the convergent-divergent nozzle. As it enters the mixing section, a low pressure region is caused by the adiabatically expansion of the flow, which induces vapor (secondly fluid) from the inter-cooler – k, of evaporation side. The firstly and secondly fluids are then mixed in the ejector – g, and the combined stream enters the condenser – h, where it condenses to liquid with heat exchanging. The condensate liquid is then divided into two parts. One part is pressurized via the refrigerant pump – i, and preheated by waste water heat from the recovery device – f, then flows into the generator – c. The other part is decompressed and expanded through the throttle valve – j, and then enters the inter-cooler – k, of evaporation side where the refrigerant is absorbed heat and evaporated to produce low-temperature and low-pressure steam. This steam enters the ejector – g, thus completing the ejector subcycle. In the compression subsystem, the compressed refrigerant vapor coming from the compressor – n, is condensed and release heat in the inter-cooler – k of compression side. The condensate undergoes a pressure reduction via the throttling valve – j, and then enters the evaporator – m, where it is evaporated to absorb heat from air-conditioned area. The vapor is compressed by the compressor – n, and then enters the inter-cooler to complete the compression sub-cycle.

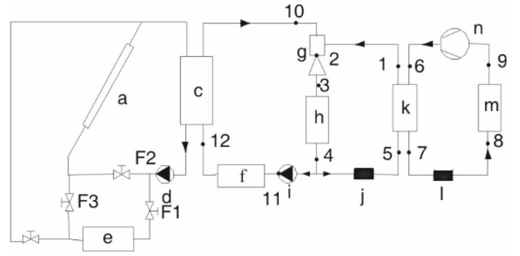

Figure 1. The schematic diagram of WHRSERS; *(a) solar collector, (c) generator, (d) pump, (e) exhaust heat recovery device, (f) waste water heat recovery device, (g) ejector, (h) condenser, (i) refrigerant pump, (j) throttle valve, (k) inter-cooler, (l) expansion valve, (m) evaporator, (n) compressor*

Thermodynamic model of heat recovery system

Figure 2 shows the *T-S* diagram of WHRSERS. The state of each point corresponds to the system cycle in fig. 1, P_g and T_g represent the generator pressure and temperature respectively, P_c and T_c – the condenser pressure and temperature, respectively, P_{int} and T_{int} – the inter-cooler pressure and temperature, respectively, P_e and T_e – the evaporator pressure and temperature, respectively.

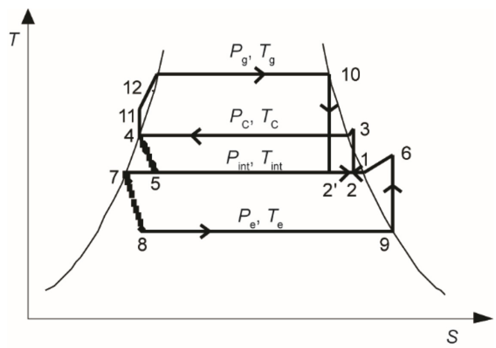

Figure 2. The *T-S* diagram of WHRSERS

According to the *T-S* diagram of WHRSERS in fig. 2, the thermodynamic model was established:

The cooling capacity, Q_e, can be expressed as:

$$Q_e = m_e(h_9 - h_8) \tag{1}$$

The compressor power, W_c, is given by:

$$W_c = \frac{m_e(h_6 - h_9)}{\eta_c} \tag{2}$$

The inter-cooler heat capacity of evaporator side, $Q_{e,int}$, is:

$$Q_{e,int} = m_{e,int}(h_1 - h_5) \tag{3}$$

The inter-cooler heat capacity of condensation side, $Q_{c,int}$, is:

$$Q_{c,int} = m_e(h_6 - h_7) \tag{4}$$

The heat balance equation of inter-cooler can be expressed:

$$Q_{c,int} = \gamma Q_{e,int} \tag{5}$$

The enthalpy value keeps constant after throttling:

$$h_4 = h_5, \quad h_7 = h_8 \tag{6}$$

The heat capacity absorption of the generator, Q_g, can be described:

$$Q_g = m_g(h_{10} - h_{12}) \tag{7}$$

The entrainment ratio, μ, is:

$$\mu = \frac{m_{\text{e,int}}}{m_{\text{g}}} \tag{8}$$

The mass balance equation of the conservation can be expressed as:

$$m_{\text{c}} = m_{\text{g}} + m_{\text{e,int}} \tag{9}$$

The heat balance equation of ejector is:

$$h_{10} + \mu h_1 = h_3(1+\mu) \tag{10}$$

The heat capacity of condenser, Q_{c}, is defined:

$$Q_{\text{c}} = m_{\text{c}}(h_3 - h_4) \tag{11}$$

The heat capacity of the waste heat recovery device $Q_{\text{h,r}}$ can be described:

$$Q_{\text{h,r}} = m_{\text{g}}(h_{12} - h_{11}) \tag{12}$$

The outlet temperature of the waste heat recovery device, T_{12}, is given by:

$$T_{12} = T_{11} + \Delta t \tag{13}$$

The power by the refrigerant pump, W_{p}, can be expressed:

$$W_{\text{p}} = \frac{m_{\text{g}}(h_{11} - h_4)}{\eta_{\text{p}}} \tag{14}$$

The COP can be calculated:

$$COP = \frac{Q_{\text{e}}}{Q_{\text{g}} + W_{\text{c}} + W_{\text{p}}} \tag{15}$$

The entrainment ratio of system can be obtained from eqs. (9) and (10), it can be written $\mu = m_{\text{e,int}}/m_{\text{g}} = (h_{10} - h_3)/(h_3 - h_1)$ according to eqs. (3)-(5) and eq. (14), $m_{\text{e}}/m_{\text{g}} = \gamma[\mu(h_1 - h_5)]/(h_6 - h_7)$ can be obtained. To simplify the calculation, order $l = m_{\text{e,int}}/m_{\text{g}}$. Equations (1), (2), (7), and (14) are substituted into eq. (15), the COP can be shown:

$$COP = \frac{l(h_9 - h_8)}{(h_{10} - h_{12}) + \dfrac{l(h_6 - h_9)}{\eta_{\text{c}}} + \dfrac{h_{11} - h_4}{\eta_{\text{p}}}} \tag{16}$$

Based on the previous thermodynamic theory, the thermodynamic model of WHRSERS was established using EES software. The input parameters include generator temperature, T_{g}, evaporator temperature, T_{e}, intermediate temperature, T_{int}, condenser temperature, T_{c}, and system cooling capacity, Q_{e}. The output parameters are COP and generator heat load, Q_{g}.

Results and discussion

Based on the previous thermodynamic model of WHRSERS, the following conditions are selected as: generator temperature is 80 °C, intermediate temperature is 10 °C, evaporator temperature is –10 °C, condenser temperature is 38 °C, and the system cooling capacity is 51 kW. The refrigerants are R134a, R1234ze, and R1234yf, respectively.

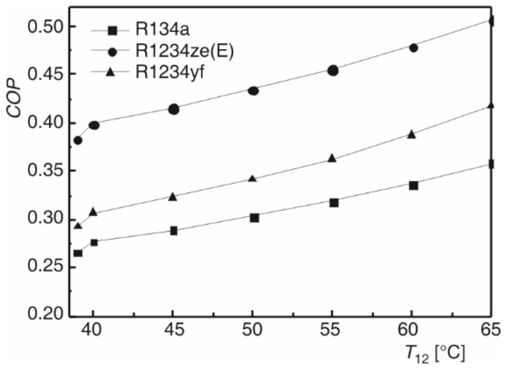

Figure 3. The change of COP with T_{12}

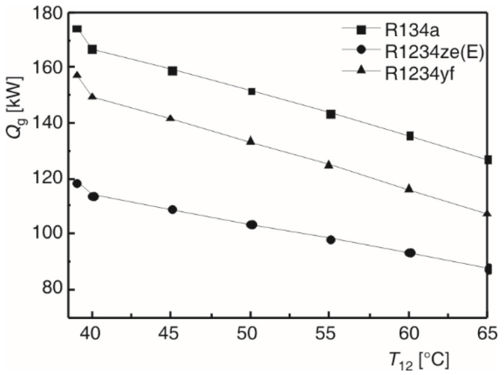

Figure 4. The change of generator heat load with T_{12}

Figures 3 and 4 show that the variation of COP and generator heat load with T_{12}. It can be seen that the COP is directly proportional to T_{12}, while the generator heat load has the opposite trend. The COP of each refrigerant system increases and the generator heat load decreases with the T_{12} increasing. When the T_{12} is 65 °C with R1234ze(E), the minimum generator heat load is 87.76 kW, and the maximum COP can reach 0.51. This is because that the higher temperature of T_{12}, the higher preheated refrigerant temperature when entering the generator, the less heat absorbed in the generator, the smaller heat load needed by the generator, and the higher COP can be obtained. Furthermore, it also can be observed that the sequence of COP is: R1234ze(E) > R1234yf > R134a, and the sequence of generator heat load is: R134a > R1234yf > R1234ze(E).

In order to further study the energy saving efficiency of the system by adding the waste heat recovery device, the preheating temperature increment was taken as 10 °C, 20 °C, and 30 °C and the thermal calculations were performed for the two systems, respectively.

According to figs. 5 and 6, it is evident that compared with the traditional SERS, the WHRSERS generator heat load is lower and the COP is higher at the same conditions. The energy saving efficiency increases with the preheating temperature increasing. The results showed that the condition of temperature increment is 30 °C, the maximum COP and minimum generator heat load of the WHRSERS were obtained. Compared with the traditional SERS, the WHRSERS using R134a, R1234yf and R1234ze(E) as refrigeration fluid, respec-

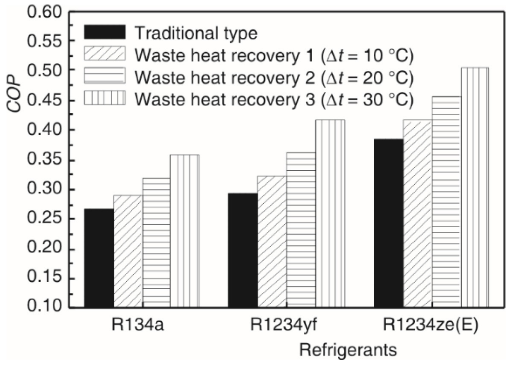

Figure 5. The COP of WHRSERS and traditional type

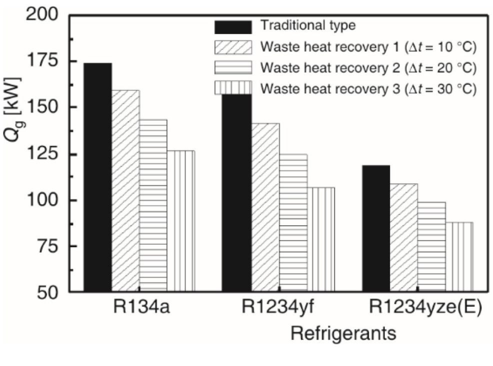

Figure 6. The generator heat load of WHRSERS and traditional type

tively, the generator heat load can save 27.14%, 31.98% and 26.13%, respectively, the COP can increase 34.51%, 42.94% and 32.07%, respectively.

Conclusions

In this paper, the WHRSERS was designed and the thermodynamic model was established based on EES. The influence of the heat recovery device outlet temperature on the COP and generator heat load were researched under different refrigerants, the energy saving efficiency between the WHRSERS and conventional SERS was compared. The main conclusions are as follows.

- The generator heat load decreases with decrease of outlet temperature of the heat recovery device for R134a, R1234yf, and R1234ze(E) as refrigerants under the same conditions. The sequence of *COP* is: R1234ze(E) > R1234yf > R134a, the sequence of generator heat load is: R134a > R1234yf > R1234ze(E).
- Compared with the traditional SERS, the generator heat load of the WHRSERS is lower and the COP is larger under the same conditions. The energy saving efficiency increases obviously with the increase of the preheating temperature, and when the temperature increment is 30 °C, the maximum heat load of generator can be saved by 31.98%, and the maximum COP can be increased by 42.94%.

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