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Performance Analysis of High Temperature Heat Pump Cycle for Industrial Process

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ABSTRACT

High-temperature heat pumps (HTHP) that can supply heat at temperatures above 200 °C have a very large potential to enhance the energy efficiency of the industrial sector and reduce its CO_2 emissions. In the current work, the thermodynamic performance of three different vapor compression cycles, which use R-718 (water) as a working medium, have been evaluated using a commercial process simulation tool (EBSILON Professional 15). All considered cycles use two-stage vapor compression with different intercooling strategies between the stages and the thermodynamic performances of those cycles have been investigated.

The comparison studies on the coefficient of performance (*COP*) and heat supply temperatures have been conducted for all cycle architectures. With increasing temperature difference between a heat source and heat sink, ΔT, the *COP* values decreased as expected. The highest *COP* value was found for the cycle configurations where both compressors have the same pressure ratio (PR).

The investigation on the HTHP capacities and exergy efficiency, η_{exergy}, with optimized PR has also been carried out. Both *COP* and η_{exergy} values increased with decreasing ΔT and the intercooler cycles with heat exchanger showed higher *COP* and η_{exergy} than the spray injection cycle.

Keywords: High temperature heat pump, Industrial process, Vapor compression cycle, R-718 (water), Thermodynamic analysis.

1. INTRODUCTION

As the impact of climate change becomes more prevalent in communities, the combustion of fossil fuels for the supply of process heat is becoming unattractive due to increasing fuel costs and CO_2 emissions. CO_2 emissions account for over 72% of the greenhouse gas that causes global warming and the largest final CO_2 emitting sector is the industrial sector, ≈ 40% [1,2].

An efficient way to recover waste heat from industrial processes and reduce greenhouse gas emissions is to use highly efficient heat pumps. However, heat pumps are not widely used in industrial processes unlike domestic applications that use commercial heat pumps actively. Electrically driven heat pumps have proven to be a suitable method for supplying process heat effectively while improving the overall process energy efficiency [3-6]. However, most heat pumps currently produce heat at a maximum temperature of 150 °C because of lower efficiency and component limitations [7]. In addition, most investigations into the multistage cycle focused on conventional refrigerants with low critical points [8]. This still leaves a large industrial heat demand at higher temperatures uncovered [9].

In this context, this research work proposed three different two-stage water vapor compression cycles with heat sinks at temperatures above 200 °C, and the thermodynamic performance has been characterized under a wide parameter variation using the 0D steady-state cycle simulation that can assess the system performance based on the enthalpy change of working fluid.

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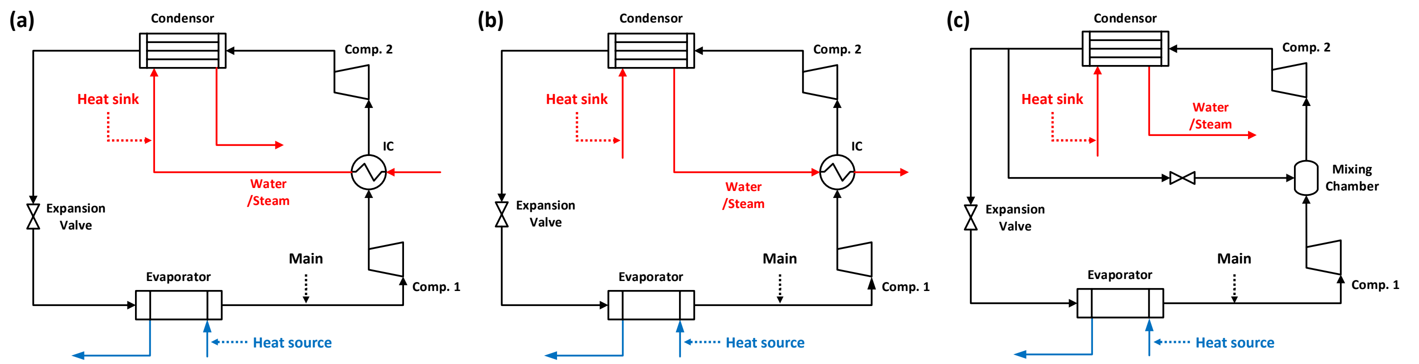

Fig. 1. Schematic diagram of heat pump cycles. (a) IC-in, (b) IC-out, (b) Spray injection

2. HTHP MODELING AND ANALYSIS METHODS

2.1 Multistage water vapor compression cycles

Two-stage vapor compression cycles with different intercooling concepts are proposed for high-temperature heat pumps as follows. The working fluid is first compressed to an intermediate pressure then it is cooled close to its condensation temperature. The intermediate pressure is usually determined by the geometric mean of the pressures in the evaporator and condenser as shown in Fig. 1. Two cycles have an intercooler heat exchanger to reduce the working medium temperatures, Fig. 1(a) and (b), and spray injection has been introduced to the third cycle, Fig. 1(c). Especially, the intercooler (IC) cycles can be divided into IC-in and IC-out according to the direction of heat sink flow.

2.2 Simulation processes

The cycle modeling and the investigation of thermodynamic performance have been carried out with EBSILON Professional 15, a commercial process simulation software. This steady-state simulation tool with a graphical user interface is widely used in industry and research [10]. As boundary conditions for simulation, the total power consumption of two compressors was fixed to 120 kW, and the superheating and subcooling degrees at the evaporator and condenser outlet were assumed to be 10 K. Additionally, the pinch point temperatures between the main cycle and heat sink were chosen equal to 10 K.

In the heat pump simulation, the evaporation temperatures varied from 60 °C to 110 °C with condensation temperatures ranging from 150 °C to 250 °C.

2.3 Thermodynamic evaluation

To evaluate the cycle efficiency, the *COP* values were calculated as the ratio of taken heat by heat sink through heat exchangers, $Q_{\text{sink out}}$ - $Q_{\text{sink in}}$, to the power consumption of the compressor, $W_{\text{comp.}}$, as expressed in Eq. 1.

$$COP = \frac{Q_{\text{sink,out}} - Q_{\text{sink,in}}}{W_{\text{comp.}}} \tag{1}$$

Compared to conventional energy analysis, exergy analysis describes all thermodynamic losses in the system components and the whole system quantitatively. The difference in the flow availability of the stream and that of the same stream at dead state is called flow exergy, ε, Eq. 2.

$$\varepsilon = (h - T_0 s) - (h_0 - T_0 s_0) \tag{2}$$

where h_0 and s_0 are the enthalpy and entropy values of the refrigerant at reference pressure and temperature (P_0 = 101.325 kPa, T_0 = 293.15 K).

The thermal exergy associated with heat exchanger capacity, Q, is defined as Eqs. 3-5 [11]. The E_{in} is the total exergy input of the heat pump system, and $E_{\text{IC out}}$ and $E_{\text{Inj out}}$ are the total exergy output for the intercooler and spray injection cycle respectively.

$$E_{\text{in}} = W + \left(1 - T_0/T_{\text{Evap.in}}\right) \cdot Q_{\text{Evap.}} \tag{3}$$

$$E_{\text{IC out}} = \left(1 - T_0/T_{\text{Cond.out}}\right) \cdot Q_{\text{Cond.}} + \left(1 - T_0/T_{\text{m.IC}}\right) \cdot Q_{\text{IC}} \tag{4}$$

$$E_{\text{Inj out}} = \left(1 - T_0/T_{\text{Cond.out}}\right) \cdot Q_{\text{Cond.}} \tag{5}$$

Particularly for calculating the corresponding exergy in the intercooler, the logarithmic mean temperature was used.

$$T_{\mathrm{m\,IC}} = (T_{\mathrm{IC\,in}} - T_{\mathrm{IC\,out}})/\ln\left({T_{\mathrm{IC\,in}}}/{T_{\mathrm{IC\,out}}}\right) \quad (6)$$

From the above equations, the total exergy efficiency, η_{exergy}, was defined as the ratio of total exergy output, E_{out}, to total exergy input, E_{in}, of the heat pump system and evaluated regarding the main flow to maintain thermodynamic consistency.

$$\eta_{\mathrm{exergy}} = E_{\mathrm{out}}/E_{\mathrm{in}} \quad (7)$$

3. RESULTS AND DISCUSSION

3.1 Simulation results of multistage compression cycles

As mentioned above, the thermal characteristics of heat sink flow are an important parameter to evaluate the actual performance of the cycles. To compare the performance of the three different cycles, a parametric study with the capacity and supply temperatures was performed under various operating conditions ($T_{\mathrm{Evap.}}$, $T_{\mathrm{Cond.}}$, P_{int}). Fig. 2 shows the *COP* and $T_{\mathrm{sink\,out}}$ of the heat pump cycles by changing the pressure ratio (PR) between 1st, π_{12}, and 2nd, π_{34}, compressors. Each symbol represents a different cycle configuration; the filled rectangle is IC-in, Fig. 1(a), the blank rectangle is IC-out, Fig. 1(b), and the blank triangle represents spray injection, Fig. 1(c). Additionally, the different line colors have been used to denote the different working temperatures.

The variation of the *COP* values under different operating conditions is shown in Fig. 2(a). The *COP* values decreased with the increasing temperature difference between the heat source and sink, Δ*T*. The IC-in and IC-out cycles showed the same results but the spray injection cycle showed lower *COP* than IC cycles, due to its lower mass flow rate of the heat sink, m_{sink}. The overall *COP* values increase rapidly when 1st stage has a low-pressure ratio, $\pi_{12}/\pi_{34} < 0.5$, and the maximum *COP* values are confirmed in the case of both compressors have the same pressure ratio, $\pi_{12}/\pi_{34} = 1.0$, under all temperature conditions.

In the supply temperatures, $T_{\mathrm{sink\,out}}$, the general values increased with increasing Δ*T*, Fig. 2(b). Because IC-in and spray injection cycles have the same temperature constraints for heat sink, they showed the same results for $T_{\mathrm{sink\,out}}$, and those values decreased as PR increased. However, since $T_{\mathrm{sink\,out}}$ values for the IC-out cycle are directly related to the intermediate pressure, the values increased with increasing PR.

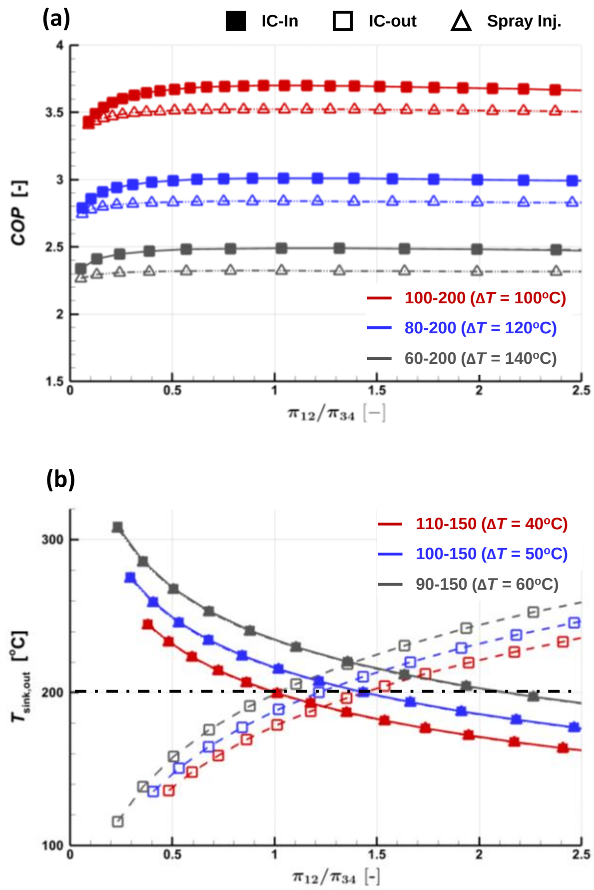

Fig. 2. Comparison between IC and injection cycles according to pressure ratio (a) *COP* and (b) $T_{\mathrm{sink,out}}$.

3.2 Thermodynamic performance at optimized PR

In order to compare the overall thermodynamic performances and quality of proposed HTHP cycles, the *COP* and exergy efficiency, η_{exergy}, at optimized PR, $\pi_{12}/\pi_{34} = 1.0$, were evaluated. Fig. 3 shows the *COP* and exergy efficiency as a function of the evaporator and condenser temperatures at optimized PR ($T_{\mathrm{Evap.}}$ = 60-110 °C, $T_{\mathrm{Cond.}}$ = 150-250 °C, Δ*T* = 5 K).

With increasing Δ*T*, the *COP* values decreased to around 2.7 from 8.5, Fig. 3(a). Both IC cycles show the same *COP* values and those values are 2-9% higher than that of the spray injection cycle. Because IC cycles can utilize two heat exchangers, condenser and intercooler, they have higher mass flow rates for heat sinks, m_{sink}, to satisfy the cycle constraints and higher m_{sink} induced higher COP.

The general result of exergy analysis also shows a similar tendency to *COP*, Fig. 3(b); the η_{exergy} decreased with increasing Δ*T*. IC and spray injection cycles show similar highest efficiency value, ≈ 0.94, when the cycles have low ΔT but the difference of efficiency increased as increasing ΔT; 0.78 and 0.73 for IC and spray injection cycles respectively. Especially, although the spray injection cycle has 1-7% higher $Q_{Cond.}$ values, the IC cycles show higher η_{exergy} values after considering Q_{IC} eventually.

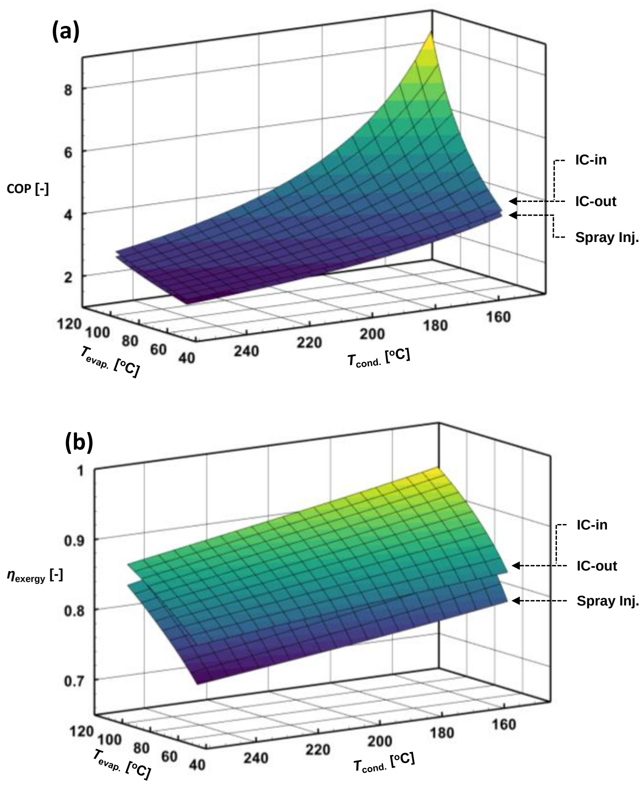

Fig. 3. Simulation results of (a) COP and (b) η_{Exergy} at π_{12}/π_{34} = 1.0.

4. CONCLUSIONS

Three different multistage water vapor compression cycles that can supply over 200 °C were proposed. Each cycle is designed to have different intercooling architectures and the thermal performances were estimated under various conditions.

As the increasing temperature lifts, ΔT, the *COP* has decreased, however, on the contrary, the supply temperatures, $T_{sink\ out}$, have increased. In particular, all cycles show the highest *COP* value when 1st and 2nd compressors have the same pressure ratio, π_{12}/π_{34} = 1.0. The thermodynamic performance of HTHP cycles at the optimized pressure ratio was also investigated. The overall *COP* and exergy efficiency values increased with decreasing ΔT. IC-in and IC-out cycles show the same *COP* and η_{exergy} results, and those values are slightly higher than that of the spray injection cycle because IC cycles utilize two heat exchangers, condenser and intercooler.

In conclusion, the proposed high-temperature heat pump cycles showed a high potential to enhance energy efficiency and electrify industrial process heat delivery.

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