Study on the design of heat pump cycle utilizing waste heat from nuclear power plants

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1. Introduction

With the global commitment to carbon neutrality, there's a growing emphasis on technological advancements aimed at reducing carbon emissions and enhancing the efficiency of various systems. Among these technologies, heat pumps have emerged as a key focus due to their ability to efficiently generate high-temperature water using low-temperature heat sources.

Conventionally, in Pressurized Water Reactors (PWRs), steam is condensed by exchanging heat with seawater in the steam condenser. However, this process leads to an increase in seawater temperature, impacting the surrounding natural environment adversely. The resultant rise in seawater temperature poses a significant threat to ecosystems[1,2]. Therefore, deploying a heat pump system presents itself as a viable solution for achieving carbon neutrality, as it can provide high-temperature water to surrounding industries at a minimal environmental cost while minimizing seawater temperature changes. In this study, we focus on proposing a heat pump cycle for integration into the steam condenser of a nuclear power plant.

2. Result and discussion

In the present study, we propose the introduction of heat pump systems as a method to utilize waste heat generated from nuclear power plants in surrounding industries. To compare the characteristics of heat pump cycles, three different configurations of heat pump cycles, as shown in Fig. 1, were established.

The first cycle consists of four components: a compressor, condenser, expansion valve, and evaporator, forming a single-stage compression heat pump cycle. The second cycle is configured as a two-stage compression cycle, incorporating a flash tank to partially cool the discharge from the first-stage compressor. The third cycle is designed to achieve higher superheat by exchanging heat between the superheated refrigerant from the first-stage compressor and the high-temperature refrigerant from the condenser. The significant difference between the second and third cycles lies in the cooling or superheating of the refrigerant through the flash tank or internal heat exchanger.

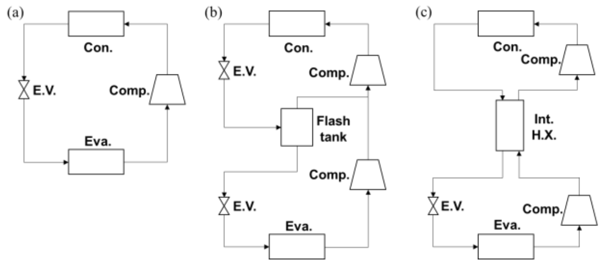

Fig. 1. Three different heat pump cycles. (a) Single-stage compression cycle(SC), (b) two-stage compression cycle with a flash tank(TSF), (c) two-stage compression cycle with an internal heat exchanger(TSI).

In order to exclude the influence of high and low temperatures, the condenser temperature was set at 120℃ and the evaporator temperature at 25℃. Additionally, the compressor efficiency was fixed at 75%.

Based on the results of previous research[3], R1233zd(E) refrigerant was utilized. Considering that the performance of the heat pump cycle may vary based on intermediate pressures, the intermediate pressures at TSF and TSI were treated as variables.

Firstly, we compared the COP of the cycles under the basic conditions. For TSF and TSI, the analysis was conducted under conditions where the compression ratios of the first and second stages were equal, and the results were illustrated in Fig. 2. Among the three cycles, SC exhibited the lowest COP at around 2.1. In contrast, TSF and TSI showed relatively higher COPs at 2.7 and 2.6, respectively, compared to SC.

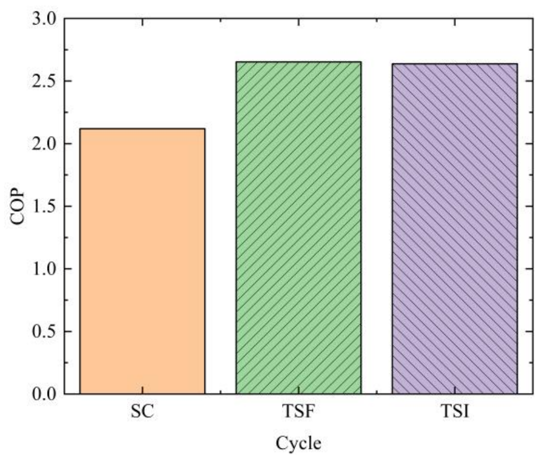

Fig. 2. COP of the three heat pump cycles.

When comparing the compression ratios of the compressors, as shown in Fig. 3, the single-stage compression cycle reaches a compression ratio of 12, whereas the two-stage compression cycle only achieves a compression ratio of around 3.5. Therefore, it is evident that SC faces constraints in compressor selection due to its high compression ratio.

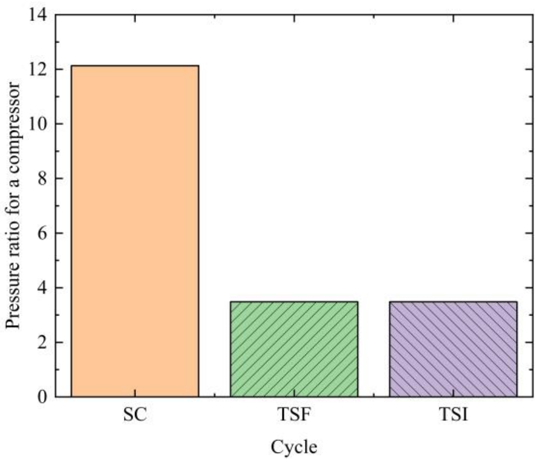

Fig. 3. Pressure ratio for a compressor at each cycle.

Furthermore, as mentioned earlier, TSF and TSI exhibit different performance characteristics depending on the intermediate pressure. Fig. 4 illustrates the trend of COP variation as the intermediate pressure ranges from 350 kPa to 450 kPa. The change in COP with increasing intermediate pressure shows contrasting tendencies for TSF and TSI. TSF demonstrates an increase in COP as the intermediate pressure rises, attributed to the larger flow rate of the second-stage compressor compared to the first-stage compressor due to the flash tank. This reduction in power consumption of the second-stage compressor leads to higher COP values. On the contrary, in the case of TSI, where the mass flow rates of the first and second-stage compressors are equal and the degrees of superheat differ, it was observed that unlike TSF, the overall power consumption of the compressors gradually increases as the intermediate pressure rises.

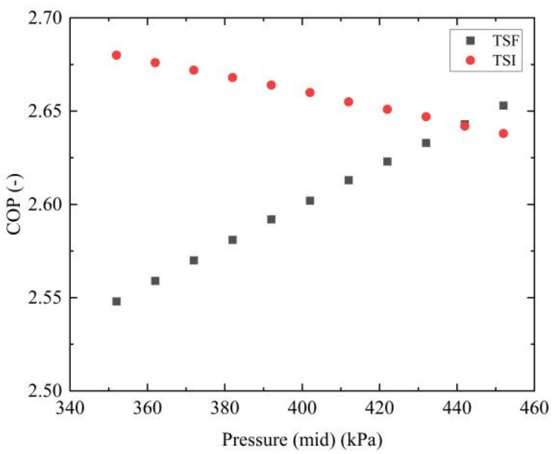

Fig. 4. The trend of COP variation according to the increase of intermediate pressure at TSF and TSI.

For a more quantitative analysis, Fig. 5 presents the power consumption of the compressors for TSF and TSI as the intermediate pressure varies. Initially, both TSF and TSI show an increase in power consumption for the first-stage compressor and a gradual decrease for the second-stage compressor as the intermediate pressure increases. However, the trend in total power consumption differs between the two. In the case of TSF, the total power consumption decreases gradually as the intermediate pressure rises. Conversely, for TSI, the total power consumption increases concurrently with the increase in intermediate pressure. This indicates that in terms of total power consumption, the increase in power consumption of the first-stage compressor dominates in the case of TSI.

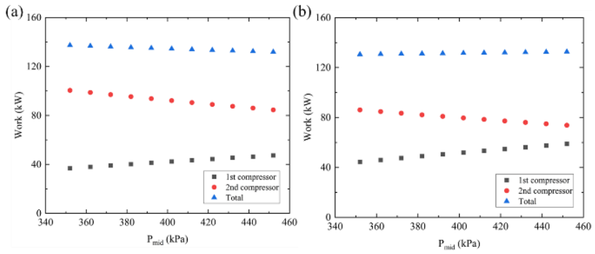

Fig. 5. Power consumption of the compressors at TSF and TSI.

Lastly, analyzing the outlet temperature of the second-stage compressor, as shown in Fig. 6, reveals that for TSF, due to partial cooling by the vapor from the flash tank, the maximum outlet temperature of the second-stage compressor reaches approximately 124℃. In contrast, for TSI, the internal heat exchanger leads to higher superheat, resulting in the outlet temperature of the second-stage compressor reaching up to 179℃. Increasing the temperature of the refrigerant entering the condenser can reduce the refrigerant flow rate, thereby decreasing the power consumption of the compressor. However, in this case, the possibility of thermal degradation of the refrigerant due to its temperature must be considered. According to prior studies [4,5], R1233zd(E) remains stable only up to approximately 175℃, beyond which thermal degradation may occur, posing long-term usage issues. Therefore, based on temperature analysis, TSF appears to be the most suitable cycle configuration for this system.

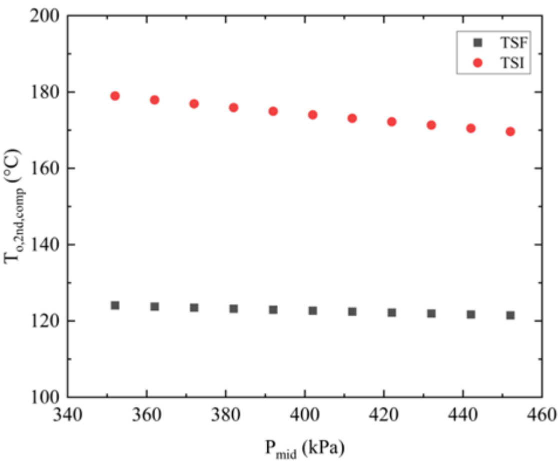

Fig. 6. Temperature of the refrigerant at the exit of the second compressor at TSF and TSI.

3. Conclusion

As part of exploring methods to utilize waste heat generated from nuclear power plants in surrounding industries, this study considered heat pump systems using R1233zd(E) as the refrigerant. We analyzed the characteristics of the cycle based on its configuration and pressure conditions through thermodynamic calculations. Our conclusions are as follows:

1) When the evaporator temperature is 25℃ and the condenser temperature is 120℃, it is reasonable to configure the heat pump system with a two-stage compression cycle due to the high-pressure ratio.

2) The two-stage compression cycle exhibits a higher COP compared to the single-stage compression cycle.

3) Considering the temperature of the refrigerant, it is advisable to configure the heat pump system with a flash tank type of two-stage compression cycle.

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