Performance analysis of EES-based transcritical carbon dioxide two-stage compression/ejector refrigeration system for shipboard cold chambers

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Abstract. Carbon dioxide (CO₂) as a natural refrigerant has a broad application prospect in the refrigeration system of shipboard cold chambers. In this work, a novel transcritical CO₂ two-stage compression refrigeration system equipped with a two-phase ejector for shipboard cold chambers is proposed. At the same time, simulation of the two-phase ejector and the system was carried out by Engineering Equation Solver (EES). Additionally, The effects of high pressure stage exhaust pressure, intermediate pressure and evaporating temperature on system performance were also investigated. The results showed that the system COP increased with the evaporating temperature increased from -35°C to -25°C. The system COP decreased with the increase in exhaust pressure, when the evaporating temperature is maintained at -35°C. The optimal intermediate pressure is 4.75 MPa, when the evaporating temperature is fixed at -35°C.

1. Introduction

At present, ships are responsible for 90% of the world’s cargo trade transportation, and the annual emission of greenhouse gases during the transportation of ships is about 1 billion tons, accounting for about 3% of the total global CO₂ equivalent emissions [1]. The shipping industry is growing rapidly, bringing prosperity to the world economy, but also exacerbating air pollution. In September 1987, the Montreal Protocol proposed ozone-depleting substances and a phase-out schedule. With the increasing global emphasis on environmental protection, the international community has proposed further restrictions on the greenhouse effect of refrigerants in ships [2]. In April 2018, the International Maritime Organization (IMO) determined the greenhouse gas emission reduction strategy for the world’s first shipping industry. The main content is that by 2050, the annual total greenhouse gas emissions will be reduced by at least 50% compared with 2008. The Kigali Amendment to the Montreal Protocol, which entered into force on January 1, 2019, regulates and reduces hydrofluorocarbons (HFCs) and their refrigerant blends with...
a high global warming potential (GWP). In May 2019, the 74th session of the IMO Marine Environment Protection Committee (MEPC) developed a process for assessing the impact of greenhouse gas emission reductions from ships and a follow-up consideration program for emission reductions to further promote the low-carbon process [3-5]. What’s more, as a natural refrigerant, CO₂ has no environmental problems and has broad application prospects in ship refrigeration system [6-8]. CO₂ as a refrigerant has many advantages: ①Cheap and good for large refrigeration equipment; ②Non-combustible, non-toxic, adaptable to various lubricants and mechanical equipment materials; ③High pressure and high density, CO₂ system can be more compact and lighter under the same cooling and heating requirements; ④Good heat transfer performance.

Generally, the commonly used shipboard refrigeration system mainly consists of four main components: compressor, condenser, throttling valve and evaporator, which is an important auxiliary equipment of the ship and a guarantee for the normal operation of the equipment, which can improve people’s living and working environment and ensure the storage conditions of food [9]. However, this shipboard refrigeration system is a transcritical CO₂ two-stage compression system equipped with a two-phase ejector. The purpose of introducing the two-phase ejector in the shipboard cold chamber refrigeration system is to reduce the throttling loss and enhance the performance of the system. The structure of the two-phase ejector is mainly composed of four parts: working nozzle, receiving chamber, mixing chamber and diffuser [10]. The working fluid enters the working nozzle and then decreases the pressure and increases the speed to a higher flow rate from the nozzle and enters the receiving chamber, and then the low-pressure fluid from the evaporator is induced, and the two enter the mixing chamber, where the working fluid and the induced fluid mix with each other and exchange momentum and energy to equalize the speed, while along with the increase in pressure, the mixed fluid enters the diffuser and the pressure continues to increase, finally forming a mixed fluid with a pressure between the working fluid pressure and the induced fluid pressure. The final formation of a mixed fluid with a pressure between the working fluid pressure and the intermediate pressure of the pilot fluid pressure [11, 12]. Throughout the process, the ejector is able to increase the pressure of the induced fluid without directly consuming mechanical energy, which is the main feature of the ejector [13].

In this work, a novel transcritical two-stage compression refrigeration system equipped with a two-phase ejector for shipboard cold chambers is proposed. At the same time, simulation of the ejector was also carried out through the EES. In addition, the effects of high pressure stage exhaust pressure, intermediate pressure and evaporative temperature on the system coefficient of performance (COP) were explored.

2. System description
The transcritical CO₂ two-stage compression/ejector refrigeration system in the shipboard cold chamber and its pressure-enthalpy diagram are illustrated in Figure 1. The system is composed of an evaporator (EVA), a low-pressure expansion valve (EXP II), a gas-liquid separator (SEP), a two-phase ejector (EJE), a low-pressure compressor (LCM), a high-pressure compressor (HCM), an intercooler (ICL), a high-pressure expansion valve (EXP I) and a gas cooler (GC).
The working principle of the two-stage compression/ejector refrigeration system for shipboard cold chambers is as follows: the saturated steam (state point 9) enters the low-pressure compressor and is compressed into medium temperature superheated steam (state point 10). After that, it enters the intercooler and is cooled to saturated steam (state point 4) by heat conduction with two-phase CO₂ fluid from the high-pressure expansion valve. Subsequently, it enters the high-pressure compressor and is compressed into a supercritical fluid with high temperature and high pressure (state point 1), and is cooled under constant pressure through a gas cooler (state point 2) to release sensible heat to the surrounding environment. The medium-temperature supercritical fluid from the gas cooler enters the high-pressure expansion valve, and becomes two-phase CO₂ fluid (state point 3) through throttling and decompression. Then it enters the intercooler and superheated steam (state point 10) for heat exchange. The saturated steam (state point 4) enters the high-pressure compressor, and the saturated liquid (state point 5) enters the working nozzle of the two-phase ejector as the main fluid. The ejector leads the saturated steam from the evaporator outlet (state point 13) to the suction chamber of the ejector (state point 1). The two fluids are mixed at medium pressure in the mixing chamber (state point 7), and then enter the diffuser for pressure. Finally, the main fluid leaves the ejector in two-phase state (state point 8) and is separated into saturated steam (state point 9) and saturated liquid (state point 11) in the gas-liquid separator. The saturated steam enters the low-pressure compressor for recompression, while the saturated liquid is decompressed through the low-pressure expansion valve (state point 12), and then enters the evaporator to produce refrigeration effect. This completes a refrigeration cycle.

3. Mathematical modeling and simulation

3.1. Two-phase ejector model

To simplify the modeling process for the two-phase ejector, the assumptions are as follows:

- The mixing process in the mixing chamber of the two-phase ejector is carried out at constant pressure.
- The two-phase ejector is insulated from the surrounding environment.
- The two-phase ejector uses a one-dimensional steady-state model.
- Neglect the variation of kinetic energy at the inlet and outlet of the injector.
- The efficiencies of the working nozzle, suction chamber, mixing chamber and diffuser remain unchanged.

The entrainment ratio µ can reflect the performance of the ejector, given as:

\[ \mu = \frac{\dot{m}_{13}}{\dot{m}_{s}} \]  

where \( \dot{m}_{13} \) and \( \dot{m}_{s} \) stands for the mass flow rates of the secondary and primary flow of the two-phase ejector, respectively.

The quality of the mixed flow x exiting the ejector should satisfy the following relationship:
Additionally, Figure 2 illustrates the flow chart of ejector simulation calculation. A simulation model of the two-phase ejector was built in EES according to the equations in Figure 2, as shown in Figure 3.

3.2. Energetic model

The energy analysis model is based on the first law of thermodynamics. To simplify the model, assumptions are as follows:

- System maintains steady-state operation.
- Ignore the pressure drop and heat loss of pipes and heat exchangers such as gas cooler, evaporator.
- The compression process of the compressor is adiabatic and non-isentropic.
- The leaving fluid at the separator and evaporator are saturated.

The COP of this transcritical CO₂ two-stage compressor/ejector refrigeration system can be calculated as:

$$\text{COP} = \frac{Q_2}{W_{CM}} = \frac{Q_2}{W_{HCM} + \frac{W_{HCM}}{\mu} (h_3 - h_2)}$$

(3)
4. Results and discussion

This section discusses the effects of high pressure stage exhaust pressure, intermediate pressure and evaporating temperature on system performance.

Figure 4 illustrates the variation of the system COP with different exhaust pressure at different evaporation temperatures. It can be found that the system COP decreases significantly with the increase of the exhaust temperature. At an evaporation temperature of -35°C, the COP of the system decreases from 1.241 to 1.074 (with a reduction of 13.46%) as the exhaust pressure changes from 8 MPa to 10 MPa. In addition, when the evaporation temperature increases, the system COP does not change significantly.

![Figure 4. The variation of COP at different exhaust pressures.](image)

Figure 5 illustrates the variation of the system COP with different intermediate pressure at different evaporating temperatures. It can be found that there are two peaks in the system COP when the intermediate pressure changes from 3.0 MPa to 5.0 MPa. At an evaporation temperature of -35°C, the two peaks are 1.126 and 1.130, corresponding to intermediate pressures of 3.75 MPa and 4.75 MPa, respectively. It follows that the optimal intermediate pressure that maximizes the system COP in the above range is 4.75 MPa. At the same time, the COP increases with the augment of evaporating temperature, but the change is small.

![Figure 5. The variation of COP at different intermediate pressures.](image)

![Figure 6. The variation of COP at different evaporating temperature.](image)

Figure 6 illustrates the variation of the system COP with different intermediate pressure at different discharge pressures. It can be found that the system COP increases and then decreases with the increase of evaporation temperature. More importantly, as the exhaust pressure increases, the system COP decreases significantly. At an exhaust pressure of 8.5 MPa, the system COP raises from 1.191 to 1.195 when the evaporation temperature changes from -35°C to -25°C, and then decreases slowly as the evaporation temperature increases.

5. Conclusion

In this work, a novel transcritical CO₂ two-stage compression/ejector refrigeration system equipped with a two-phase ejector for shipboard cold chambers is proposed. At the same time, the simulation of a two-phase ejector was performed by EES. Finally, the effect of the system parameters: exhaust pressure,
intermediate pressure and evaporating temperature on the COP of this system was investigated. The main findings are summarized as follows:

- With the evaporating temperature increases from -35°C to -25°C, the system COP shows an increasing trend; after that, it slowly decreases with the augment of evaporating temperature.
- With the evaporating temperature is maintained at -35°C, the system COP decreases with the increase in exhaust pressure.
- There exists an optimal intermediate pressure of 4.75 MPa, which makes the COP of the system reach a maximum of 1.130, when the evaporating temperature is fixed at -35°C.

Acknowledgement

This paper is supported by the Science and technology innovation action plan of Shanghai Science and Technology Commission (19DZ1207503) and Shanghai Professional Technology Service Platform on Cold Chain Equipment Performance and Energy Saving Testing Evaluation(17DZ2293400).

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