Performance analysis of multi-refrigerant multi-variable environment refrigeration system based on marine cold chamber

Jie Zhu1,2,3,4#, Dazhang Yang1,2,3,4#, Qing Zhang1,2,3,4 and Jing Xie1,2,3,4*

1College of Food Science and Technology, Shanghai Ocean University, Shanghai, 201306, China
2Shanghai Professional Technology Service Platform on Cold Chain Equipment Performance and Energy Saving Evaluation, Shanghai, 201306, China
3Quality Supervision, Inspection and Testing Center for Cold Storage and Refrigeration Equipment, Ministry of Agriculture, Shanghai 201306, China
4National Experimental Teaching Demonstration Center for Food Science and Engineering (Shanghai Ocean University), Shanghai, 201306, China

Jie Zhu and Dazhang Yang contributed equally to this study

Corresponding author’s Email: jxie@shou.edu.cn

Abstract. This paper is based on the engineering equation solver to calculate the performance of refrigeration system under multi-refrigerant multivariable environment, which can simulate the operating performance of the system under different working conditions and the view of each state point, and provide help for the design and optimization of trans-critical CO2 refrigeration cycle refrigeration units. Using thermodynamic analysis, the system performance is observed by using the equations of conservation of energy, conservation of momentum and conservation of mass as the basis for the change of evaporator pressure, gas cooler pressure and intermediate pressure in the system; the state of each position in the ejector part of the system is calculated; after the calculation is completed, the pressure-enthalpy diagram of the system can be plotted; according to the different data parameters selected, the COP of the system can be plotted with the change of certain data parameters for different parameters. The results showed that the calculation method used is suitable for different refrigerants, compared with the traditional calculation and selection, the advantage of the EES software is that it can use the laws of thermodynamics for calculation and iteration.

1. Introduction
Ship borne transportation has played an important role in the development of the world economy since its development. International shipping traffic, especially pelagic fishery, accounts for a large proportion of the total traffic[1, 2]. Due to people's increasing attention to the environment and economy, the concept of cold chain transportation has been put forward, and the consumption of marine cold chamber has increased sharply. With the prominence of environmental problems, the demand for environmental protection and energy-saving marine cold chamber is very urgent[3]. Generally speaking, from the perspective of energy operation, there are two types of energy during ship operation: mechanical energy and thermal energy. That is, the mechanical energy in the compressor and the
thermal energy in the evaporator and condenser\textsuperscript{[4]}. For marine refrigeration, the working environment and energy utilization are a very special research field. Compared with civil and household refrigeration, the marine cold cabin generally needs lower evaporation temperature and endless seawater as cooling water. At the same time, there are particularly strict requirements for safety, weight and size in industrial equipment. Marine cold tanks are mostly used when Freon refrigerants are prosperous. Since many international documents and the damage of high GWP refrigerants to the environment is becoming more and more obvious. The world organization restricts the use of high GWP refrigerants and stipulates the final service life of some refrigerants, which limits the subsequent operation of the refrigeration system of the refrigerant and has a great impact on the existing marine cold chain transportation\textsuperscript{[5]}, according to the provisions of international law, this aspect has been limited\textsuperscript{[6]}. Therefore, a refrigerant that can analyse the performance of multiple refrigerants in new systems is important. When the evaporator pressure is too low, the refrigerant volume increases, and when the theoretical pressure volume remains unchanged, the mass flow of the system decreases relatively; When the evaporation temperature is too low, the inlet and outlet pressure difference of the compressor is large, the compressor efficiency is reduced, which is unfavourable to the operating life of the compressor and leads to the reduction of the overall performance of the system\textsuperscript{[7, 8]}. In addition, when the system's condensing temperature rises, the condensing pressure will rise with the condensing temperature, the exhaust temperature of the refrigeration compressor also rises; the unit refrigeration capacity of the refrigerator is reduced, the suction ratio volume remains unchanged, the unit volume refrigeration capacity decreases; the theoretical work of the refrigeration compressor increases; if you ignore the change in the volumetric efficiency of the compressor does not count, The refrigerant mass flow remains unchanged, which leads to an increase in the theoretical power of the compressor\textsuperscript{[9-12]}

CO\textsubscript{2} is selected as refrigerant in this paper. The environmental protection characteristics of CO\textsubscript{2} - zero oxygen consumption potential and negligible GWP have attracted more and more attention\textsuperscript{[13, 14]}. In addition, as the working medium in the refrigeration system, carbon dioxide also has excellent properties compared with Freon refrigerants in use\textsuperscript{[15]}. However, the main reason why CO\textsubscript{2} refrigeration system cannot be widely used is its inefficiency. The main reason is that the huge pressure difference between HP and LP stage results in serious throttle loss before and after throttle valve\textsuperscript{[16]}. Therefore, in order to improve the performance of the trans-critical CO\textsubscript{2} refrigeration cycle, it is necessary to retrofit the traditional trans-critical CO\textsubscript{2} refrigeration cycle. For example, in order to reduce the pressure difference between the front and rear of the compressor, a two-stage compression system is adopted to reduce the working load of the compressor, or in order to recover the expansion work consumed by throttling loss, an expander or ejector is used in the system to recover the expansion work\textsuperscript{[12, 13, 17]}.

In this paper, the performance of a trans-critical CO\textsubscript{2} refrigeration system is studied based on the characteristics of a refrigeration system with multiple working fluids and variable environment in an offshore environment. Because of the changeable marine environment, the ambient temperature of ships will change with the change of latitude and longitude, and the refrigerants used by different ships will be different. This makes it extremely inconvenient to use a common manual calculation method, which makes it impossible to update the calculation of the cooling capacity of the condenser and the power consumption of the compressor in operation in offshore systems from time to time. In addition, intermediate pressure is particularly important in refrigeration systems containing two-stage compressors, each with its own optimum intermediate pressure value. From the variation of evaporation pressure (P\textsubscript{e}), gas cooler pressure (P\textsubscript{gc}) and intermediate pressure (P\textsubscript{i}), this paper analyses the variation rule of system performance due to the variation of main parameters in operation. This includes the COP of the system, cooling capacity and compressor power consumption.

2. Cycle description and modelling
As shown in Figure 1 and Figure 2, this paper deals with the calculation of a multi-refrigerant multivariable environment refrigeration system for marine use, which includes an evaporator (EVA), a low-pressure compressor (LPC), an ejector (EJE), a high-pressure compressor (HPC), a gas cooler
(GC), an expansion valve (EV), and an intercooler (ITC). Compared with the conventional same, the expansion valve used for throttling is replaced by an injector, and the refrigerant from the intercooler to the compressor is injected by the injector. The flow process of refrigerant in the system cycle is as follows: the hot refrigerant first passes from the outlet of the HPC through the GC (state point 2), then is cooled by the GC (state point 3), then draws the saturated vapor CO2 refrigerant directly through the EJE from the ITC (state point 1), and then enters the ITC in a mixed two-phase fluid state (state point 4) after working through the EJE. The refrigerant leaving the ITC is divided into two parts. For saturated vapor refrigerant (state point 1), one part passes through the HPC and the other part is directed to the EJE. The saturated liquid refrigerant coming out of the ITC (state point 5) passes through the EVA (state point 6) after being throttled by EV (state point 7) and then enters the LPC. Finally, the vapor refrigerant is compressed by the LPC (state point 8) and mixed with the two-phase fluid refrigerant in the ITC. As mentioned above, the fundamental reason for using an EJE in the cycle is to replace the EV in a conventional system and to draw some of the saturated vapor refrigerant from the ITC, unlike directly into the HPC. Another part of the refrigerant from the new ITC is in saturated liquid under medium pressure, rather than supercooled liquid under high pressure under the traditional ITC. In this new cycle, the refrigeration capacity of EVA always increases and the compression work of HPC decreases. As a result, the refrigerant mass of the LPC is reduced and the corresponding compression work of the LPC is reduced. Therefore, when two systems have the same refrigeration requirement, the mass flow rate of the LPC in the system is reduced and thus the LPC power is reduced, thus improving the cycle performance. The refrigerant entering the HPC can also be controlled through the EJE inlet, making the system more adjustable.

Figure 1. Schematic diagram of marine cold chamber system

Figure 2. Pressure enthalpy diagram of marine cold chamber system

3. Theoretical modelling

In order to simplify the system, a thermodynamic model is established according to the laws of thermodynamics, and the following assumptions are made:

1. The system is in steady-state and steady-state flow process during operation.
2. The pressure loss and kinetic energy loss in the connecting pipe are ignored.
3. The refrigerant has no enthalpy difference before and after working in the throttle valve.
4. The pressure drops and heat loss of refrigerant in pipes and equipment are ignored.

In the above assumptions, the thermodynamic modelling of each component and process of the system is carried out according to the laws of thermodynamics.

The power consumption of the compressor is calculated by the following formula:

\[ W_{\text{coml}} = m_i (h_2 - h_1) = m_i (h_{i,\text{in}} - h_{i,\text{out}}) / \eta_{\text{coml}} \]  \hspace{1cm} (1)

\[ W_{\text{comh}} = m_i (h_8 - h_4) = m_i (h_{i,\text{in}} - h_{i,\text{out}}) / \eta_{\text{comh}} \]  \hspace{1cm} (2)
where $\eta_{\text{coml}}$ and $\eta_{\text{comh}}$ are the isentropic efficiency of LPC and HPC. It is known from the article (Chen et al., 2017) [10] that this value is related to the relationship between the inlet and outlet pressure of the compressor:

$$\eta_{\text{coml}} = 0.815 + 0.022(P_0^0 / P_0) - 0.0041(P_0^0 / P_0)^2 + 0.0001(P_0^0 / P_0)^3$$

(3)

$$\eta_{\text{comh}} = 0.815 + 0.022(P_0^0 / P_0) - 0.0041(P_0^0 / P_0)^2 + 0.0001(P_0^0 / P_0)^3$$

(4)

For ejector, according to the conservation of energy:

$$h_4 \cdot (1 + w) = h_3 + w \cdot h_1$$

(5)

For the intercooler, the mass relationship can be calculated by the conservation law of mass and energy:

$$m_4 = m_h + m_1 = m_h + w \cdot m_b$$

(6)

$$m_4 \cdot h_4 + m_1 \cdot h_1 = m_4 \cdot h_1 + m_1 \cdot h_1$$

(7)

For evaporator, the refrigeration capacity is calculated using the following equations:

$$Q_e = m_1 \cdot (h_1 - h_6)$$

(8)

The coefficient of performance is:

$$COP = \frac{Q_e}{W_{\text{coml}} + W_{\text{comb}}}$$

(9)

4. Results and discussion

Calculate the refrigeration system according to the basic state shown in Table 1. On this basis, the main influencing parameters in the system are changed to observe the performance of the system. Variation range [19]: the pressure $P_{gc}$ of the gas cooler is in the range of 8.3 - 10 MPa, the evaporation pressure of the LP evaporator ($P_e$) is from 1.2 - 1.5 MPa, the outlet temperature ($T_{gc}$) of the gas cooler is in the range of 35 to 44. It is assumed that the refrigerant mass flow at the inlet of the low-pressure compressor is 1.0 kg · s$^{-1}$. In addition, each section of the ejector part of the system is composed of main flow nozzles (the main body of the ejector enters through the nozzle, resulting in pressure drop and large kinetic energy vortex), the pilot jet chamber, the mixing chamber where two fluids are mixed, and the diffuser section where the mixed fluid passes through. The kinetic energy vortex generated after the main body passes through the nozzle can introduce the fluid from the low-pressure section. In the assumption of Table 1, the ejector efficiency at the nozzle, the mixing chamber and the diffuser section is on this basis, the thermodynamic analysis and calculation of trans-critical CO2 refrigeration system have been carried out.

Table 1. Parameters of the system in the assumed basic case.

| Parameters | Values |
|------------|--------|
| $\eta_{ld}$ | 0.8    |
| $\eta_{sm}$ | 0.9    |
| $\eta_{se}$ | 0.9    |
| $P_e$/MPa | 1.2    |
| $P_{gc}$/MPa | 9.3 |
| $P_1$/MPa | 4.4 |

According to the parameters in the basic state, it is calculated in the EES software. The parameters of each state point after calculation are shown in Table 2. The state parameters include pressure, enthalpy, temperature and entropy. According to the data in the table, thermodynamic analysis of the trans-critical CO2 refrigeration system is carried out based on the thermodynamic model constructed to observe the changes of the system's performance. It should be noted that the parameter is invoked from the EES database.
Table 2. The thermodynamic parameters of all state point

| Point | Pressure/MPa | Enthalpy/ kJ·kg⁻¹ | Temperature/°C | Entropy/ kJ/(kg·K)² |
|-------|--------------|------------------|----------------|---------------------|
| 1     | 4.4          | -82.9            | 9              | -0.948              |
| 2     | 9.3          | -48.6            | 69             | -0.928              |
| 3     | 9.3          | -211.6           | 35             | -1.437              |
| 4     | 4.4          | -166.0           | 9              | 0.586               |
| 5     | 4.4          | -283.5           | 9              | -1.659              |
| 6     | 1.2          | -283.5           | -35            | -1.610              |
| 7     | 1.2          | -70.6            | -35            | -0.716              |
| 8     | 4.4          | 1.8              | 67             | -0.672              |

Figure 3 can observe the variation curve of COP of NERC with $P_{gc}$ at different $P_e$. As expected, each evaporator temperature has an optimal HP and a COP maximum value. As the $P_{gc}$ increases, the specific work of the compressor increases. However, it can be seen from the pressure enthalpy diagram in Figure 2 that when the $P_{gc}$ is low, the refrigeration effect ($h_1$-$h_4$) of the two-phase liquid in the ITC is low, which requires increasing the mass flow of the refrigerant to achieve the desired refrigeration effect. In addition, under any $P_{gc}$, the COP value increases with the increase of the $P_e$. From the results, the optimal pressure in the COP change curve of NERC system with the change of $P_{gc}$ under different $P_e$ shows that the curve trend before and after the optimal pressure decreases and is different. This phenomenon exists under evaporation pressure. Therefore, it can be inferred that there is an optimal $P_{gc}$ under any condition to make the system performance reach the best state, and the existence of this value should be emphatically measured in the experiment, so as to make the system in the best state during actual operation.

![Figure 3. Effect of $P_{gc}$ on COP of different $P_e$.](image)

The influence of system COP with $P_i$ under different $P_{gc}$ is shown in Figure 4. From the figure that there is an optimal intermediate pressure $P_i$ for any $P_{gc}$, and according to the figure, it can be observed that the optimal intermediate pressure gradually decreases with the increase of gas cooler pressure $P_{gc}$, and the COP of the system decreases with the increase of gas cooler pressure $P_{gc}$.
The influence of system COP with $P_i$ under different $P_{gc}$ is shown in Figure 4. There is an optimal $P_i$ for any $P_{gc}$, and according to the figure, the optimal $P_i$ gradually decreases with the increase of $P_{gc}$, and the COP of the system decreases with the increase of $P_{gc}$. With the change of $P_{gc}$, the change of system COP under different $T_{gc}$ is shown in Figure 5. Each $T_{gc}$ has an optimal $P_{gc}$, and the system performance tends to decline before and after the optimal value. For any $P_{gc}$, the COP decreases with the increase of $T_{gc}$ due to the high heat loss in the gas cooler. As shown in the figure, under different $T_{gc}$, the COP of the system changes obviously with the change of $P_{gc}$, and there is obviously an optimal $P_{gc}$ to make the system reach the optimal state. This conclusion also verifies that the system has an optimal $P_{gc}$ under any state, and the system has a downward trend before and after the optimal $P_{gc}$. Different from the above, under different $T_{gc}$, the system COP varies greatly at low pressure. When the temperature reaches a high temperature, the system COP drops to a very low level, which indicates that the system should pay attention to maintaining the system $T_{gc}$ at a low level during actual operation.

5. Conclusions
In this paper, based on EES software, the thermodynamic modelling of the new trans-critical CO2 refrigeration system designed by the system is carried out to analyse the performance of the system. CO2 is used as circulating refrigerant. Through this study, it can also be applied to different refrigerants. The effects of the main parameters of the refrigeration cycle on the system performance are studied. The main results are as follows:

1. The calculation method used is suitable for different refrigerants. The system cycle can be analyzed by selecting different refrigerants in the software.
2. Compared with the traditional calculation and selection, the advantage of the EES software mentioned in this paper is that it can be directly imported into the software in the form of equations by using the laws of thermodynamics for calculation and iteration, and different trend change diagrams can be derived according to the changes of different parameters.
3. According to the method provided in this paper, the system loss can also be calculated, and the system loss can also be calculated and analyzed under different parameter changes.
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