Performance analysis of a new double stage compressor helium liquefaction cycle

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Abstract. Large scale helium liquefaction/refrigeration plant plays a vital role in advanced large equipment such as fusion reactors and accelerators. A new helium liquefaction cycle with double stage compressor is proposed in this paper, which is different from the Collins cycle. The helium gas from the compressor is divided into two parts. The medium-pressure helium gas as one part flows through the heat exchangers and is liquefied by throttling in the valve. The other part, high-pressure helium gas flows into the turbine for adiabatic expansion with decrease of its temperature and pressure. Both high-temperature helium gas streams become low-temperature and low-pressure. In this paper, the new cycle is parametrically analysed to demonstrate the effect of each parameter on the performance of the cycle. The results of the liquefaction capacity are obtained in different working conditions. Exergy analysis, which is an effective way to analyse thermodynamic cycles, is performed on the liquefaction process and the result is significant for the appropriate design of helium liquefaction system.

1. Introduction

In recent years, the application of superconducting technology in scientific research has become more and more extensive, especially in large-scale scientific engineering such as nuclear fusion equipment and high-energy accelerators. The stability and efficiency of cryogenic refrigeration systems will affect the normal operation of the entire scientific device. It is precisely because of the development of these large scientific devices that low temperature superconducting technology has developed rapidly. Various worldwide high-energy accelerators, such as ITER, LHC, HERA, and EAST, are also equipped with large-scale cryogenic refrigeration systems to provide a cryogenic environment.

In helium liquefaction cycle, power requirement is approximately 70 W for transporting 1W heat from a low temperature (4.2K) to the ambient temperature (300K) in an ideal cycle. Therefore, it is very important to improve the cycle efficiency, which is closely related to the cycle parameters and the performance of the components in the system. A lot of work has been done before. Thomas et al. found that the 80% helium mass flowing through the compressor is equally divided into two parts into two expanders respectively to obtain the maximum amount of liquefaction [1]. Cammarata et al. optimized the Collins cycle by GA (Genetic Algorithm) method [2]. Thomas et al. found that the increase of liquefaction rate is linear with the effectiveness of the heat exchanger by analyzing the UA of the heat exchanger [3].

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Exergy analysis is a practical method to evaluate system efficiency from the perspective of quantity and quality of energy. It can reveal the distribution law of exergy destruction, which indicates the direction of system performance improvement. Thomas et al uses the method of exergy analysis to obtain the exergy destruction distribution at different stages of the cycle, and formulates the optimal cycle scheme [4]. Matsubara et al proposed an optimization method based on the exergy analysis, which defined the cycle performance coefficient as a function of the temperature of the throttle valve and the hot end of the heat exchanger [5].

2. Methodology

2.1. Cycle

Collins cycle consists of precooling refrigeration stages and liquefaction stage. In precooling refrigeration stages, two expanders are adopted to achieve the liquid helium temperature. One part of the steam flowing in expanders is used to precool the other part of the steam. The final liquefaction stage works on the Linde-Hampson cycle. The modified Claude cycle consists of modified Brayton stages (shown in Fig 1.) is considered as the basic cycle in some liquefers.

The efficiency of the liquefaction process is closely related to the arrangement of components, the efficiency of components, such as heat exchanger and expander, and the optimization of the cycle. Therefore, from the point of view of process optimization, a new helium liquefaction cycle with double stage compressor is proposed in this paper.

Low pressure (LP) helium gas is compressed to medium pressure (MP) helium gas by screw compressor. Part of the MP helium gas is compressed to high pressure (HP) helium gas and enters the expanders which precool stream with the aid of isentropic expansion and finally joins into the returned LP stream. The other part of the MP helium gas enters the cold box precooled in heat exchangers by liquid nitrogen and LP helium gas in counter flow. Then the MP helium is throttled to LP in the valve before the two-phase helium stream leaves the cold box into Dewar. The gas from the Dewar returns to the cold box and is warmed by the HP and MP stream in heat exchangers.

2.2. Exergy analysis

The exergy balance equation across control volume is written as:

![Figure 1. The scheme of the modified Claude cycle](image1)

![Figure 2. The scheme of the new cycle](image2)
\[
\sum_j Q_j \left(1 - \frac{T_0}{T_j}\right) - W + \sum_{in} \dot{E}_{x_{Mass}} - \sum_{out} \dot{E}_{x_{Mass}} - \dot{E}_{x_{Dest}} = 0
\]  

(1)

\[\sum_j Q_j \left(1 - \frac{T_0}{T_j}\right)\] represents the exergy transfer associated with heat, where \(Q_j\) represents the rate of heat transfer across the boundary. \(\sum_{in} \dot{E}_{x_{Mass}}\) and \(\sum_{out} \dot{E}_{x_{Mass}}\) represents the exergy across the inlet and outlet of control volume. \(W\) represents the work accompanying the exergy transfer. \(\dot{E}_{x_{Dest}}\) represents the exergy destruction due to the thermodynamic irreversibility associated with the process.

\[\dot{E}_{x_{Dest}} = m \times ex\]

(2)

\[\dot{E}_{x_{Dest}} = T_0 \Delta S_{gen}\]

(3)

\(m\) and \(ex\) represents the mass flow rate and exergy per unit mass. \(\Delta S_{gen}\) represents the rate of exergy generation in the process.

Equation (4) is obtained by applying the energy balance equation to the liquefaction process and considering the cold box as the control volume, which is shown as follows:

\[m_1(ex_1 - ex_9) + m_2(ex_{16} - ex_9) - m_L(ex_L - ex_9) + m_2(ex_{20} - ex_{19}) + m_2(ex_{22} - ex_{21}) + m_W2(ex_{23} - ex_{24}) - \dot{E}_{x_{Dest\text{-}coldbox}} = 0\]

(4)

The liquefaction rate is expressed as follow:

\[y = \frac{m_L}{m} = \frac{m_1}{m(ex_L - ex_9)}(ex_1 - ex_9) + \frac{m_2}{m(ex_L - ex_9)}(ex_{16} - ex_9) + \frac{m_2}{m(ex_L - ex_9)}(ex_{20} - ex_{19} + ex_{22} - ex_{21}) - \frac{m_W2}{m(ex_L - ex_9)}(ex_{23} - ex_{24})\]

(5)

The exergy equilibrium equation of whole system considered as a control volume is shown as follows:

\[W_{COMP} = m_L(ex_L - ex_{IN}) + W_{EXP1} + W_{EXP2} + \dot{E}_{x_{Dest\text{-}Cycle}}\]

(6)

In equation (6), \(W_{COMP}\) is the sum of the work of the two compressors. \(W_{EXP1}\) and \(W_{EXP2}\) are the work done by the turbo expander, and in the actual cycle, the work of expander is negligible compared to the compressor.

The exergetic efficiency of the cycle and the heat exchanger is defined as:

\[\eta_{Ex\text{-}Cycle} = \frac{m_L(ex_L - ex_{IN})}{W_{COMP}} \times 100\]

(7)

\[\eta_{Ex\text{-}Hx} = \frac{m_{hp}(ex_{HOUT} - ex_{HIN})}{m_{hp}(ex_{CIN} - ex_{COUT})} \times 100\]

(8)

The purpose of system exergy analysis is to improve exergy output and maximize exergetic efficiency. Therefore, the exergy destruction of each component should be reduced as much as possible. When exergetic efficiency of the system is increased, the economic cost of the system will be increased. Therefore, it’s necessary to balance the relationship between system efficiency and economic cost when designing the system.

2.3 Assumptions
The assumptions in the simulation are shown below: (1) Mass flow of the compressor is 33g/s. (2) The pressure drop at the hot end of the heat exchanger is 1kPa and the cold end is 2 kPa. The pressure drop
at the rest of the equipment is negligible. (3) The inlet temperature and pressure of the COMP1 are 310K and 1.05bar respectively. (4) The outlet temperature and pressure of the COMP2 are 310K and 13bar respectively. (5) The isentropic efficiency of the expander is 70%. (6) System is at steady conditions.

2.4. The analysis of the parameters in the process
In some medium-sized helium refrigeration or liquefier, the high pressure is at around 10~15bar. So the high pressure is determined to 13bar and the medium pressure is a key parameter in the process. The medium-pressure of the process is exactly the pressure of the JT refrigeration stage, and there is an optimal throttle point for throttling at different temperature. Therefore, an appropriate medium pressure should be selected to ensure the optimal efficiency of the system. In the following sections, we chose a pressure between 5-9 bar as the medium pressure.

The isentropic efficiency of expander is highest when the expansion ratio is between 2.4 and 4.5, and both the expansion ratio and efficiency of the expander affect the efficiency of the cycle. The outlet pressure of EXP1 is an important parameter in the process which determines the effect of pre-cooling.

3. Result and discussion

3.1. The comparison of the two cycles
The new cycle (cycle (b) in Fig.3) and the modified Claude cycle (cycle (a) in Fig.3) with liquid nitrogen are simulated, and the T-S diagram is shown in Fig.3 which consists of detailed thermodynamic values. The assumed parameters are the same for both of the cycle such as the mass flow rate and the expansion ratio of the expander.

Table1 gives the parameters in the two cycles based on the T-S diagram. The parameters Wc, y, FOM and MLN2 represents the work consumed in the cycle, liquefaction rate, the specific work (W) needed to get liquid helium (g/s) and the liquid nitrogen consumed in the cycle, respectively.

The medium pressure of the new cycle is chosen as 6 bar. Liquefaction rate of the cycle (a) is higher than that of cycle (b), but the liquefaction rate is higher when the same work is consumed, so the cycle (a) is a better choice. It is more economical in large-scale liquefaction systems.

![Figure 3. T-S diagram](image-url)
3.2. Effect of medium pressure on performance of the cycle

For the new cycle, the choice of medium pressure is very important, so the medium pressure is changed during the simulation process, and the liquefaction rate under the optimal conditions is obtained. The exergy destruction of each component in the cold box under different medium pressures is showed in Fig.4. It can be seen that EXP 2 has the largest exergy destruction. The exergy destruction of hx4 and hx1 is relatively large, while hx3 is the smallest. The exergy destruction of hx4 decreases gradually with the increase of the medium pressure, while the other heat exchangers increase as the medium pressure increases.

The heat transfer of heat exchanger across finite temperature difference leads to the exergy destruction. The greater the temperature differences are between the hot and cold fluids, the greater is the exergy destruction. The imbalance of the mass flow rate between hot and cold fluids cause the temperature difference. Exergetic efficiency changes with the medium pressure as shown in the Fig.5. As the medium pressure increases, exergetic efficiency decreases, but exergetic efficiency of hx3 and hx6 is smaller than that of other heat exchangers. In order to improve the efficiency of the entire cycle, it is necessary to improve the efficiency of each component in the system, so when designing the system, attention should be paid to the optimization design of hx3 and hx6.

3.3. Effect of the outlet pressure of EXP1 on performance of the cycle

Under the condition that the medium pressure is kept unchanged at 7 bar, the outlet pressure of EXP1 is changed, and the flow liquefaction rate is maximized by changing the flow distribution. The exergy
destruction distribution of different components is obtained as shown in Fig. 5. It can be seen that the exergy destruction of the EXP 2 is the largest, and as the outlet pressure of EXP1 increases, the exergy destruction of EXP1 decreases significantly, and the exergy destruction of EXP2 increases remarkably.

In the heat exchanger group, exergy destruction of hx4 and hx5 increased as the pressure increased, while exergy destruction of other heat exchangers decreased as well. Exergetic efficiency of hx1 and hx2 increased as the pressure increased, while exergy efficiency of other heat exchangers decreased.

Exergetic efficiency of hx3 and hx5 is lower than that of other heat exchangers. Exergy destruction of heat exchanger is related to the temperature difference between hot and cold ends to some extent. When the temperature difference of heat exchanger is reduced, exergy destruction decreases while exergetic efficiency increases, but it also means that the area of the exchanger increases. So the relationship between exergy destruction and area should be taken into consideration when choosing the temperature difference of heat exchanger.

![Figure 6](image1.png)

**Figure 6.** Variation of exergy destruction in the cold box components with outlet pressure of EXP1

![Figure 7](image2.png)

**Figure 7.** Variation of exergetic efficiency of heat exchangers with outlet pressure of EXP1

3.4. The combined effect of outlet pressure of EXP1 and medium pressure on the performance of the cycle

Fig. 8 depicts the combined effect of outlet pressure of EXP1 and medium pressure on the exergetic efficiency of the cycle. The outlet pressure of EXP1 is varied from 5 to 7bar while the medium pressure is from 5 to 9bar. For the given configuration and operations, the cycle efficiency is maximized when outlet pressure of EXP1 is 6bar. The exergetic efficiency is increased with the medium pressure.

When the medium pressure is increased, that is, the work done by COM 1 is reduced, while that of COM 2 is increased, the sum of the work done by the compressor will increase to some extent. So the liquefaction rate is increased as the medium pressure increases.

When the outlet pressure of EXP1 is increased, the cooling capacity provided by EXP1 is reduced, but the cooling capacity of EXP2 is also increased. When the pressure is larger than 6 bar, the total cooling capacity of the expander decreases as the pressure increases, and the exergetic efficiency is also relatively reduced. When the pressure is less than 6 bar, the outlet temperature of EXP1 decreased which is also the inlet temperature of hx4. The temperature of the cold flow of hx4 is close to the inlet temperature of the hot flow. So in order to ensure the normal operation of hx4, the mass flow rate of
MP helium gas should be reduced which leads to the reduction of the liquid production and exergetic efficiency.

![Figure 8. The exergetic efficiency of the cycle](image)

4. Conclusion
In this paper, a new type of helium liquefaction process is proposed. The equilibrium equation is applied to the system and the exergy equilibrium equation is obtained of the cycle. The exergy destruction of each component is obtained under different work condition. The new cycle is parametrically analyzed to demonstrate the effect of each parameter on the performance of the cycle. The following conclusions were obtained.

Comparing the new cycle with the modified Claude cycle, the liquefaction rate of the new cycle is larger when the same work is consumed in the process. The exergy destruction of EXP2 is the largest compared with other components in the cold box. The exergy destruction of hx4 is the largest in all heat exchangers and the exergetic efficiency of hx3 and hx6 is pretty low, so it is necessary to optimize them. As the medium pressure increases, the efficiency of the helium of the cycle increases. When the outlet pressure of EXP1 is changed, the exergy efficiency reaches the maximum at 6 bar.

The result of the study may be helpful to design large-scale helium liquefaction cycles.

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