Dynamic simulation of the cool down process of double-pressure helium liquefaction cycle

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Abstract. A double-pressure helium liquefaction cycle, which consists of two compressors, six heat exchangers, two expanders and several valves, was proposed in this paper. The thermodynamic parameters of the process were calculated and analyzed. The mathematical models of the above-mentioned key equipment are established, and the cool down process of the helium refrigerator is dynamically simulated. By adjusting the parameters of each valve in the process, the working process of the compressor was simulated, and the cool-down process of heat exchangers and the expanders was obtained and analyzed. Finally the stable refrigeration capacity was obtained. The simulation of the dynamic process has a great guiding role for the actual experimental process afterwards.

1. Introduction

With the development of large scientific devices such as particle accelerator and nuclear fusion experiment, there are higher requirements for large cryogenic refrigeration system to provide a more stable low-temperature environment. Therefore, the importance of the stability and efficiency of the helium refrigeration system must be taken seriously. In recent years, dynamic simulation of large-scale equipment has been widely used in the petrochemical industry [1].

The thermodynamic properties of working substances depend to a large extent on temperature, so the properties of cryogenic working substances in cryogenic systems vary widely. At low temperatures, both the working substance of the heat exchanger and the thermodynamic properties of the material can be properly described by nonlinear relations, so that modern large cryogenic systems are characterized by transient processes. In order to adapt the operating parameters of the system to real-time changes, the control system and automation are put forward higher requirements [2-4].

The first dynamic cryogenic simulation systems were proposed in the 1980s for cryogenic systems of large superconducting complexes. There are two methods to solve the dynamic simulation problem. One is to use FORTRAN and C language to develop special numerical simulation program. The second is to use SPEEDUP, FLOWTRAN and other general tools for dynamic simulation of the system. The first method is more widely used. Dynamic simulation of large-scale helium cryogenic refrigeration system can effectively guarantee the stability of the system, and explore the influence of dynamic characteristics of cooling process and experimental conditions changes on the system.
A real-time process simulator C-PREST has been developed by National Institute for Fusion Science (NIFS) for dynamic simulation of a 10kw class helium refrigerator/liquefier cryogenic system in Large Helical Device (LHD) [5]. The effectiveness of the simulation model is verified according to the design values and experimental results. The cooling process of steam turbine operation is introduced. The CERN development team developed a process simulator PROCOS (Process and Control Simulator) based on Ecosimpro and successfully applied PROCOS to a 1.8K superfluid helium system [6-7]. CERN has designed a commercial cryogenic component library, CRYOLIB, in collaboration with Spanish company EAI. Dynamic simulation of different operating modes of a simple Brayton refrigerator was carried out by Dymola/Modelica at the Technical University of Dresden, Germany [8-9].

Aspen Hysys is also widely used as process simulation software in large scale cryogenic systems. Based on Hysys, a dynamic simulation of 250W@4.5K double-pressure helium liquefaction process was carried out.

2. Methodology

2.1. The double-pressure helium liquefaction cycle

In the dynamic simulation process, screw compressor (C), plate-fin heat exchanger (HX1-HX6), turbine expander (E1, E2), liquid helium Dewar, heater (H), valves and so on are included. Its flow chart is shown in Figure 1. The high purity helium gas in the buffer tank enters the high pressure compressor (C) and is compressed to high pressure. The high and low pressure stability is controlled by the pneumatic valve V10. Due to the limited experimental conditions, the high-pressure helium outlet is connected with the pressure reducing valve. After entering the pressure reducing valve, part of the helium is decompressed to the medium pressure. The stability of the medium and low pressure helium is controlled by the pneumatic valve group V11.

When the high pressure, medium pressure and low pressure of helium gas are stable, open the inlet valve (V1, V2) and outlet valve (V3) of the cold box to start the cooling process. In the first heat exchanger, helium gas is pre-cooled by liquid nitrogen, and the high-pressure helium gas enters the turbine after passing through the primary heat exchanger. The speed of the turbine expander is controlled by the valve V5. The medium-pressure helium gas does not pass through the turbine. After six heat exchangers, it enters the throttle valve V7, which produces liquid helium.

![Figure 1. The scheme of the cycle](image)

2.2. Heat exchanger

There are six plate-fin multi-flow heat exchangers in the cold box. When designing the model of the heat exchanger, the size, number of flows, distributions of path and the geometric parameters of fins should be set. Heat exchanger is a key component in helium liquefaction process, and its modeling quality will directly affect the final simulation result, so the parameter setting of heat exchanger should be as detailed as possible.
The aluminum fin of heat exchanger adopts NIST standard Al3003F physical property library. The fitting formula of thermal conductivity and specific heat capacity $C_p$ is as follows:

$$\log_{10} y = a + b \log_{10} T + c (\log_{10} T)^2 + d (\log_{10} T)^3 + e (\log_{10} T)^4 + f (\log_{10} T)^5 + g (\log_{10} T)^6 + h(\log_{10} T)^7 + i(\log_{10} T)^8$$  \hspace{1cm} (1)

In the formula, $T$ represents temperature in K; $y$ represents the thermal conductivity K or the specific heat capacity $C_p$ in W/(m·K) and J/(kg·K), respectively. According to the temperature variation in the flow process, the thermal conductivity and specific heat capacity of the fluid in the heat exchanger at different times can be obtained.

The convective heat transfer coefficient $h$ of the heat exchanger fluid is defined as:

$$h = \alpha \text{Re}^\beta \text{Pr}^\gamma$$ \hspace{1cm} (2)

In the formula, the coefficients $\alpha$, $\beta$ and $\gamma$ are related to the fin parameters of the heat exchanger, and the fin factor of the plate-fin heat exchanger is related to the Re number. The definition formula of the heat transfer coefficient $h$ can be rewritten as:

$$h = j \cdot (\text{Re}) \cdot C_p \cdot g / (\text{Pr})^{2/3}$$ \hspace{1cm} (3)

Wherein, $j$ is the fin factor, $\text{Re}$ is the Reynolds number, and $g$ is the mass flow rate in kg/(m$^3$·s).

2.3. Turbine expander

The helium turbine expander is a key component in the helium refrigeration system, but the property of the helium at the inlet of turbine is not constant. During the cooling process, temperature and pressure of helium change with time, and the isentropic efficiency and volume flow rate of the turbine also change accordingly. In Aspen Hysys, both the turbine efficiency and the volume flow are fixed values, so the efficiency and the turbine flow are defined as the equations about the inlet parameters. In the simulation process, its volume flow equation is defined as:

$$Q_v = K_1 \frac{P_{in}}{\rho_{in} \sqrt{Z_{in} T_{in}}}$$ \hspace{1cm} (4)

The efficiency of turbine expander is defined as:

$$\eta = \eta_s \left(1.9 \frac{v}{v_d} - \left(\frac{v}{v_d}\right)^2 + 0.1\right)$$ \hspace{1cm} (5)

$$v = \pi DN / \sqrt{2\Delta h}$$ \hspace{1cm} (6)

Wherein, $\eta$ is the practical efficiency of the expander, $\eta_s$ is the ideal efficiency of the expander. $v$ is the characteristic ratio.

2.4. Helium screw compressor

Helium screw compressor is a positive displacement compressor, so it is necessary to ensure the same volume of imported helium gas in the simulation process. Therefore, the compression process of the compressor is regarded as isothermal compression, which can be calculated according to the following formula:

$$V_C = C_n G \eta_v D_l^2 L_n$$ \hspace{1cm} (7)

$$W_e = \frac{mRT_0 \ln(P_{out}/P_{in})}{\eta_T}$$ \hspace{1cm} (8)

Where, volumetric efficiency is $\eta_v = 0.95 - 0.012 \frac{P_{out}}{P_{in}}$, the adiabatic efficiency is $\eta_T = \frac{h_{out} - h_{in}}{h_{out} - h_{in}}$. $T_0$, $P_{out}$ and $P_{in}$ are ambient temperature (300K), pressure of helium at the inlet and outlet of the compressor respectively. $h_{out}$ and $h_{in}$ are the enthalpy of helium at the inlet and outlet of the compressor respectively, and $h_{out}$ is the ideal outlet enthalpy after the adiabatic process.
2.5. Valve

There are different types of valves in HYSYS, and the valves can be opened in different ways, including linear, fast opening, and equal percentage. The actual curve can also be drawn based on the relationship between the flow rate and the flow coefficient in the actual experiment. In addition, the valve flow coefficient CV value should be provided to determine the valve size.

2.6. Controller

Feedback regulation control can be represented using a built-in proportional integral derivative (PID) controller. In addition to the action type (direct or reverse), there are three parameters that must be specified: Kc, Ti, and Td. Use the event scheduler tool to enforce program control policies in the model. The latter allows the definition of a sequence of tasks triggered by a predetermined condition, such as simulation time, a logical expression becoming true, and stabilizing a given variable at a given time.

3. Result and discussion

3.1. The simulation result of simulation

The TS diagram of the process that reaches a stable state after dynamic simulation and the TS diagram of the design value of the process are shown in Figure 2. The TS diagram of design value is calculated under ideal conditions. There is little difference between the two results. It can be seen from the TS diagram that there are high-pressure helium gas flow and low-pressure helium gas flow entering the cold box, and their mass flow rates are both 20g/s. During the experiment, it is also necessary to consider controlling the flow ratio of the two helium gas flows into the cold box.

The cooling capacity of design is 320W, while the cooling capacity of the dynamic simulation is 330W, which is not much different. The reason why the cooling capacity value is larger after the dynamic simulation is that the temperature before throttling is lower than the design value, while the flow rate remains unchanged.

![Figure 2.T-S diagram of simulation and design](image)

3.2. The cool down process

In the process, the environment temperature is around 300K. When the compressor runs steadily, open the inlet and outlet valves of the cold box. Because the high and low pressure helium flows into the cold box in two ways, it is also necessary to open the turbine valve when opening the high-pressure helium inlet valve.

The cooling process of each heat exchanger is shown in Figure 3. During the cooling process, after 4 hours, the outlet temperature of HX1 reaches about 80K and no longer changes, while the outlet temperature of the remaining several heat exchangers reaches a stable temperature after about 6 hours.
After about 7 hours of cooling, liquid helium begins to be produced. From the perspective of the cooling trend and rate, the simulation model can predict the cooling process of the refrigerator, which has a certain contrast effect to the experimental process.

![Figure 3. The cooldown process of exchangers in the cold box](image)

3.3. The comparison with the result of modified-Claude cycle

It takes about 7 hours to complete the cooling process of the cold box in the dynamic simulation of the double-pressure helium liquefaction cycle. It can be seen that the temperature decreasing rate of all heat exchangers is roughly the same. The experimental result of a large scale helium cryogenic plant of 250W@4.5K based on the modified-Claude cycle is showed in Figure 4. Compared with the result in double-pressure helium liquefaction cycle, the outlet temperature of HX1 and HX3-HX6 drops more slowly, because the expander is in operation at the beginning of this process, and in the modified-Claude cycle, the expander is turned on when the inlet temperature of helium of expander drops to about 200K. The cooling time has also decreased.

When the expander is started sooner, the cold box will cool down faster, which can save cooling time and liquid nitrogen consumption.

![Figure 4. The experiment result of modified-Claude cycle](image)

4. Conclusion

The double-pressure helium liquefaction process was introduced and its dynamic cooling process was simulated, including helium screw compressor, heat exchanger, turbine expander, valve and other key equipment. The cool-down process of cold box is compared with the experimental results of previous literature, and its difference is analyzed.

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The model can be used to analyze the influence of different cooling schemes on cooling rate. Further optimization of the cooling process control scheme, especially the automatic control capability of high helium turbine expander cooling process. The factors affecting the efficiency of helium refrigerator can also be obtained by changing the efficiency of the dynamic process of each part of the refrigerator.

The simulation results can be used as a guide for the actual dynamic simulation. It can be compared with the experimental process to verify the correctness of the dynamic simulation.

5. References

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