Study of dynamic process of a large-scale helium refrigerator

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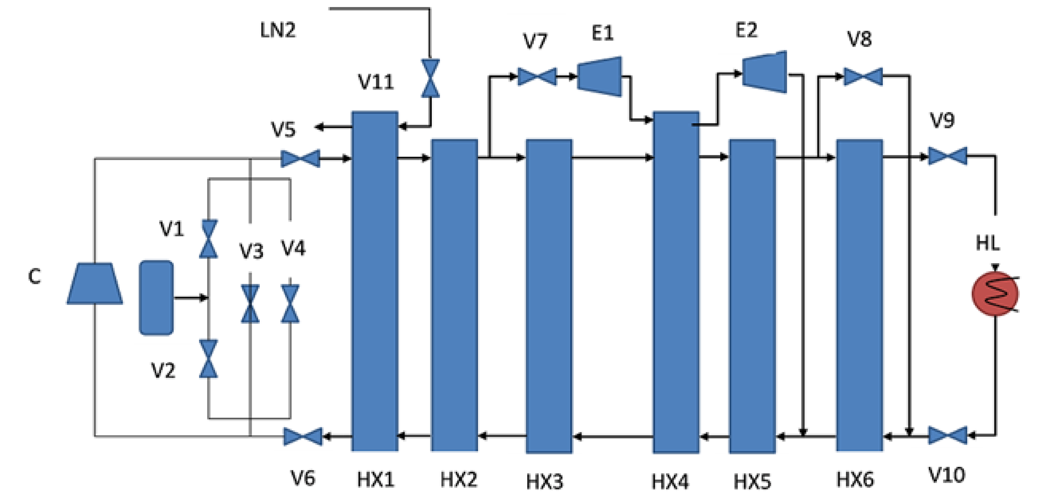
Background

With the development of cryogenic and superconducting technology, some large scientific devices have been established. Large scale cryogenic systems are important subsystems for cooling superconducting magnets and their stability directly affects the running of these large scientific devices. Dynamic simulation can improve the understanding of the process of large cryogenic system and helps us to reduce cool-down time. Thus, dynamic simulation is an important and promising tool for analyzing the system process.

Objectives

- In this paper, a numerical model of 250 W @ 4.5 K helium refrigerator was developed.
- Dynamic simulation of 250W@ 4.5K helium refrigerator is done and the deviations between the simulation and experiments are discussed.
- To improve the cold-down rate, the turbines start time is discussed.

Process description



The system consists of a compressor (C), six heat exchangers (HX1-6), two turboexpanders (E1, E2), a throttle (JT) and a Heat load (HL).

Process flow diagram of the large-scale helium cryogenic system

Conclusion

- Dynamic simulations of an existing helium cryogenic system have been performed for cool-down operation. The dynamic simulation data match the design data well, which verify the correctness of dynamic model.
- The effect of turbine opening time on cool-down is discussed. The results show that the sooner earlier the turbine starts, the faster the cool-down rate will be. In order to improve the cool-down rate, we should start the turbine earlier if the turbine allows.

Establish of critical equipment model

Compressor

Warm screw compressor is a volumetric machine, thus the inlet volumetric flow of the compressor is constant. Warm screw compressor is defined by displacement V and power.

V is the displacement of compressor and can be computed by :

$$V = C_n C_f \, D_1^2 L_n \eta_v$$

Where η_v is volumetric efficiency and can be computed by inlet and outlet pressure:

$$\eta_v = 0.95 - 0.012 \frac{P_{out}}{P_{in}}$$

And isothermal compression is considered here ($T_{in} = T_{out}$). Thus isothermal efficient is used to evaluate the compressor.

$$\eta_T = \frac{h_{out} - h_{in}}{h_{out,t} - h_{in}}$$

Where $h_{out,t}$ is the outlet enthalpy of the isothermal process.

Then the power of compressor can be computed by:

$$w_e = \frac{\dot{m} R T_0 \ln\left(\frac{P_{out}}{P_{in}}\right)}{\eta_T}$$

Heat Exchangers

Plate fin heat exchangers are the key equipment of cold box, the model of which directly affects the accuracy of the dynamic model. Heat exchanger model can be defined by heat transfer coefficient.

Convection heat transfer coefficient is computed by:

$$h = j(Re)Cp \cdot m/Pr^{2/3}$$

Where, j is the heat transfer factor and is the function of Re:

$$ln\, j = -10^{-2}(ln\, Re)^3 + 0.555843(ln\, Re)^2 - 4.09424\, ln\, Re + 6.21681$$

The variations of metal materials of heat exchangers are highly non-linear with temperature. Therefore, we have used the logarithmic equation for calculating the thermal conductivity and heat capacity of metal.

$$lg\, Cp = 46.6467 - 314.292\, lg\, T + 866.662(lgT)^2 - 1298.3(lgT)^3 + 1162.27(lgT)^4 - 637.795(lgT)^5 + 210.351(lgT)^6 - 38.3094(lgT)^7 + 2.96344\,(lgT)^8$$

$$lg\lambda = 0.63736 - 1.1437 lgT + 7.4624(lgT)^2 - 12.6905(lgT)^3 + 11.9165(lgT)^4 - 6.1872(lgT)^5 + 1.6394(lgT)^6 - 0.1727(lgT)^7$$

Turbine

In the cool-down process, the thermodynamic parameters of turbines change over time, lead to the capacity flow and isentropic efficiency changes. Turbines are defined by capacity flow and isentropic efficiency. Capacity flow can be computed by these formulas:

$$\text{When } \frac{P_{out}}{P_{in}} > \nu_{cr} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}},\ q_m = A_2\sqrt{2\frac{k}{k-1}\frac{P_{in}}{V_{in}}\left[\left(\frac{P_{out}}{P_{in}}\right)^{\frac{2}{k}} - \left(\frac{P_{ou}}{P_{in}}\right)^{\frac{k+1}{k}}\right]}$$

$$\text{When } \frac{P_{out}}{P_{in}} \le \nu_{cr} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}},\ q_m = A_2\sqrt{2\frac{k}{k+1}\frac{P_{in}}{V_{in}}\left(\frac{2}{k+1}\right)^{\frac{2}{k-1}}}$$

Where ν_{cr} is critical pressure ratio, q_m is mass flow.

Then, the capacity flow can be computed by:

$$V = \frac{q_m}{\rho}$$

The efficiency of the turbine is a function of the characteristic ratio （v=U/Cs）:

$$\eta = \eta_s\left[1.9\frac{v}{v_d} - \left(\frac{v}{v_d}\right)^2 + 0.1\right]$$

where U=πDN, N is the shaft speed of the turbine and D is the diameter of the blade, $c_s = \sqrt{2\Delta h}$.

Valve

Thermodynamic transformation through valves is considered isenthalpic ($h_{in} = h_{out}$) and the mass flow through valves is computed by Cv.

In the cryogenic system, we use Fisher model to design the valve:

$$f = 1.06\nu_{fracfac} C_g \sqrt{\rho P_{in}} sin\left(\frac{59.64Cp_{fac}}{C_1}\sqrt{1 - \frac{P_{out}}{P_{in}}}\right)$$

$$Cp_{fac} = \sqrt{\frac{0.4839}{1 - \left(\frac{2}{1+\gamma}\right)^{\frac{\gamma}{\gamma-1}}}}$$

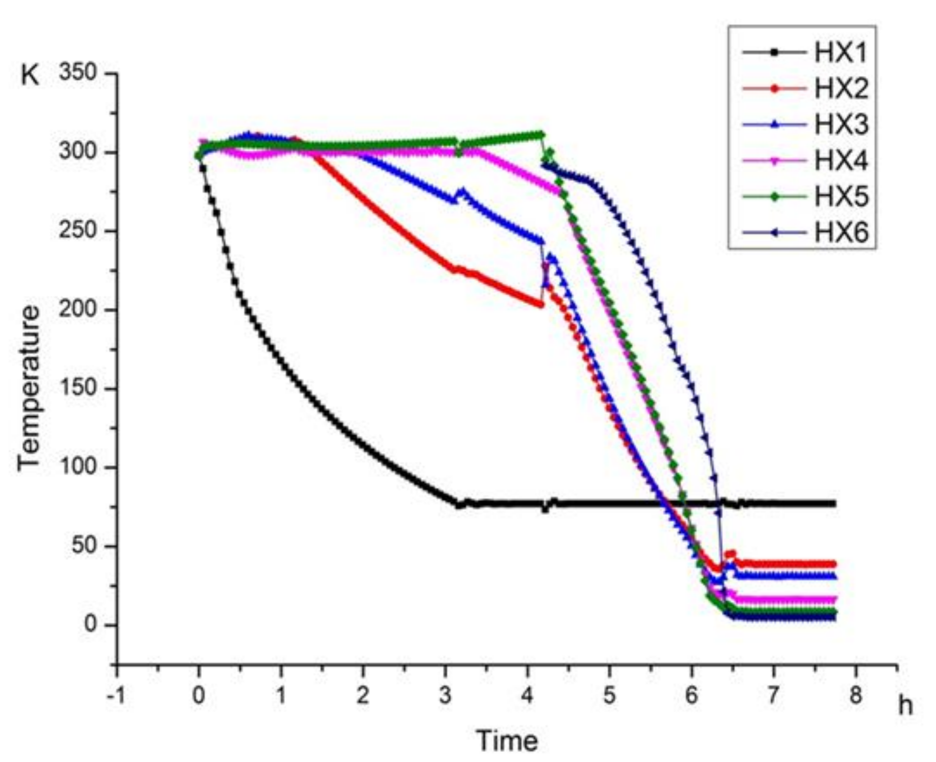
Where f is the capacity flow of valve and $C_1 = C_g/C_v$.

Cv of valves can be obtained in Table below.

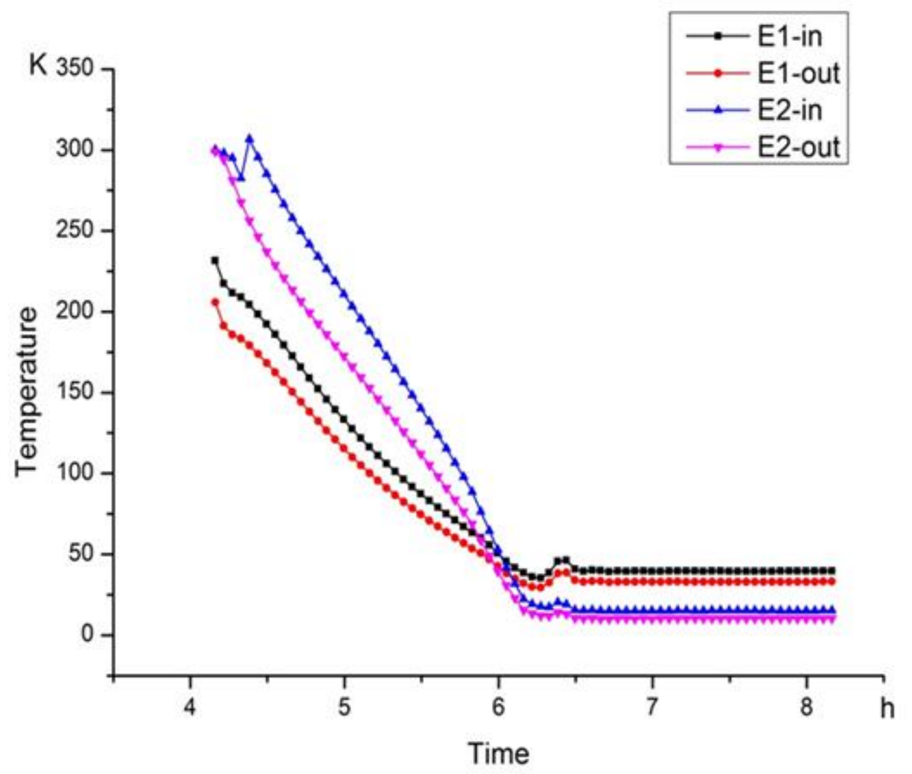
Valve	V1/V2	V3	V4	V7	V8	V9	V10	V11
Cv	3	6	0.8	2.45	0.294	0.15	7.77	0.625

Results

Cool-down of cold box



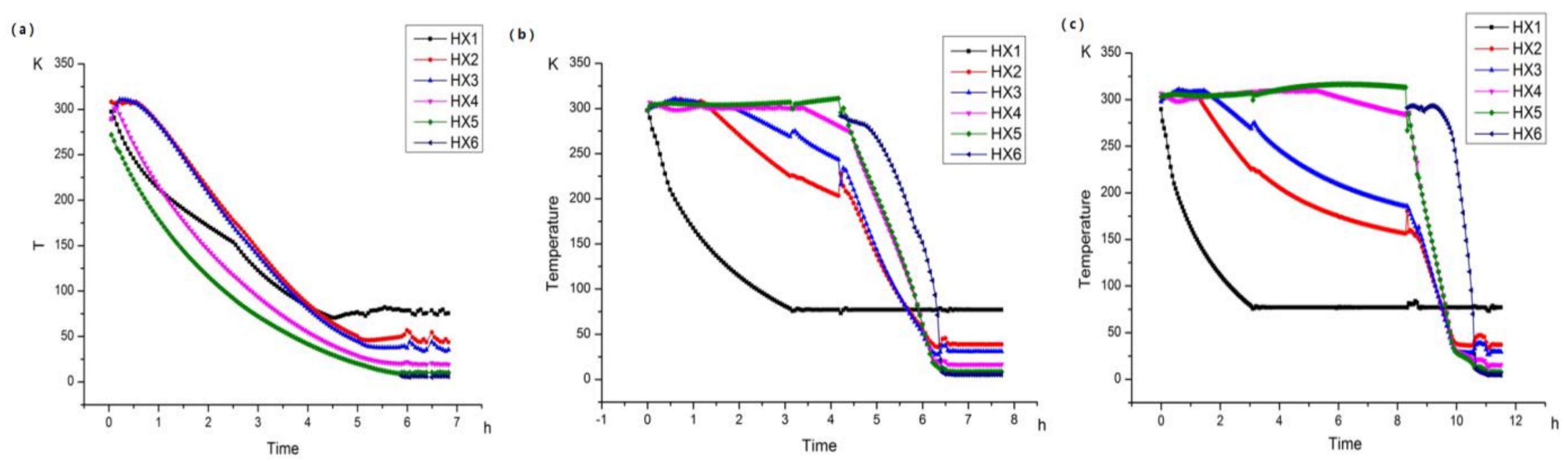
the heat flow outlet temperature curves of the heat exchangers



The inlet and outlet temperature curves of turbines

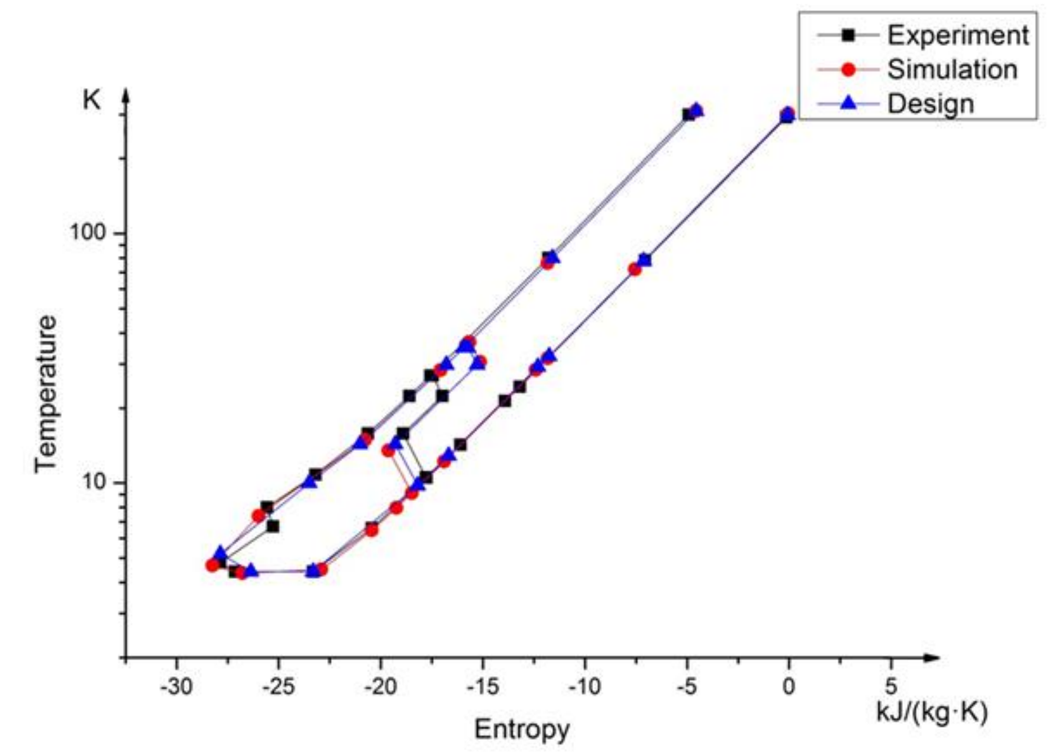
After 7 hours, the system achieves quasi-steady state. The cold box is cooled, and HX1 is the first to reach the predetermined temperature.

The effect of turbine opening time on cool-down



- The effect of the turbine opening time on the cool-down rate is discussed below. Turbine E1and E2 starts at 0, 250min and 500min, the outlet temperature curves of the heat exchangers with time are shown in (a), (b), (c).
- By contrast, we found that the earlier the turbine starts, the faster the cool-down rate will be. When turbine starts at 0 min, the cool-down time is 6h, which is 5h less than it starts at 500min. In order to improve the cool-down rate, we should start the turbine as earlier as possible.

Quasi-steady state analysis



T-S chart of simulation, design and experimental data in quasi-steady state.

The refrigeration capacity of dynamic simulation is 304W, which match design data (299W) well. But experimental data is 252W, which deviate from design data.

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