

Application of plate-fin heat exchangers in mitigating the effects of pulsed heat load

Cui Lv ^{1,3}, Ji Hao Wu ^{1,2}, Qing Li ^{1,2}, Xiu Juan Xie ^{1,2}, Lai Feng Li ^{1,2} and Hui Ming Liu ¹

¹Technical Institute of Physics and Chemistry Chinese Academy of Sciences, Beijing 100190, China

²State Key Laboratory of Technologies in Space Cryogenic Propellants, Technical Institute of Physics and Chemistry Chinese Academy of Sciences, Beijing 100190, China

³University of Chinese Academy of Sciences, Beijing 100049, China

E-mail address: wujihao@mail.ipc.ac.cn

Abstract. The heat loads of the superconducting magnets in the international fusion devices are pulsating with time, which will cause the variation of the return mass flow rate of the refrigerator. It is necessary to mitigate the fluctuations to avoid the performance degradation of turbines and compressors which are designed to operate in a narrow range of mass flow rate. In this paper, a new technic was proposed after thermodynamic analysis, where a single heat exchanger in traditional style was replaced by two different UAs plate-fin heat exchangers in the coldest part. The mass flow fluctuation could be decreased by adjusting the flow distribution between heat exchangers to change the actual UAs. The results of the dynamic simulation revealed that the mitigation of mass flow rate was 81.3% and 93.98% during high and low heat loads.

1. Introduction

The large scale cryogenic helium liquefiers are widely used in scientific apparatus, which suffered the periodic pulsed heat load. The distinct effect of the pulsed heat load is the fluctuation of the mass flow rate. However, the design range of the mass flow rate of the compressors and turbines is commonly narrow. When the device doesn't work under the rated condition, the performance and stability will largely decrease. In some extreme conditions, it will shut down the cryogenic liquefiers. Therefore, it's necessary to improve the coldest part to mitigate the effects of pulsed heat load.

In recent years, how to mitigating the effects of pulsed heat load becomes one of the hot topics. The simplest way was based on the high heat load value to design the helium liquefiers, and consumed the redundant cooling capacity in low heat load. In this condition, it will cause the plant geometries oversized, and the energy wasted. Kuendig.A[1] put forward the idea of using warm gas injection and floating the MP-return pressure to handle the mitigations of pulsed heat load. SATOH et al [2] calculated the needed amount of turbines and heat transfer area to adapt the variation of the heat load.

¹Corresponding author name: Ji Hao Wu

Address: ¹ Technical Institute of Physics and Chemistry Chinese Academy of Sciences, Beijing 100190, China
Tel.: +86-010-82543409; Fax: +86-010-82543662.

However, the biggest challenge of the above solutions is that the process is required to debug the whole plants. Hence, the stability and operability of the refrigerators will be largely affected. Then, mitigating the fluctuations through improving the cold end configuration received more and more attentions. Rohan dutta et al[3-5] based on the above idea modified the cold end structure ,such as adding one plate-fin heat exchanger with the same UA in the usual cycle, adding the cold compressors, but these methods only can be used in high or low heat load. The supercritical helium tank was also proposed to use in the cold end, and dynamic simulation revealed that using 40 m³ supercritical tank can mitigate about 78%, 74% of fluctuations in return stream during high and low heat load conditions, respectively. In order to completely mitigate the mass flow rate fluctuation, Rohan dutta et al [6] combined the plate-fin heat exchangers, the cold compressors and the supercritical helium tank as a new cold end configuration. Results of the work revealed that this method can control the fluctuation of the return mass flow in the order of $\pm 4\%$, but the required technology for the cold compressors and fabrication of large supercritical buffers were a few technological challenges.

Though analysis, this paper revealed that when the design state points before the JT valve were variable, the optional cold end configurations of mitigating the fluctuations were different. The detailed discussion was shown in section 2.2. The cold end was modified by two different UAs plate-fin heat exchangers. Verification of the concept through dynamic simulation revealed that mitigation of about 81.3% and 93.98% of fluctuations in return stream mass low rate could be achieved during high and low heat load conditions.

2. The description and analysis of the process and the heat load.

2.1.1. The description of the usual cycle configuration

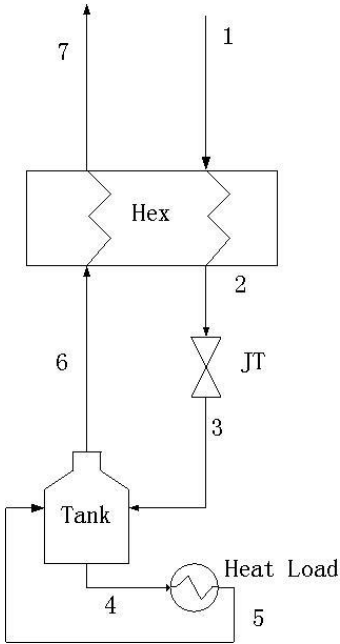


Figure 1 The usual cycle configuration

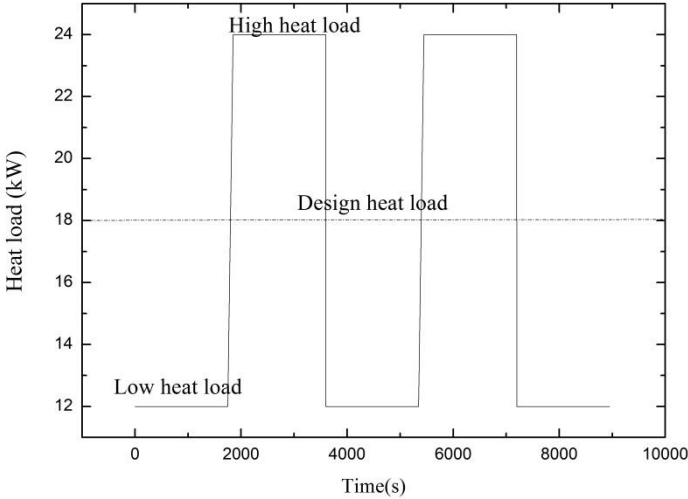


Figure 2 Pulsed heat load profile applied to the helium refrigerator [1]

The usual cold end configuration of the refrigerators is shown in Figure 1. It comprises of one heat exchanger, one JT valve, and a tank together with the user heat load loop. The parameters of the state points which shown in Figure 1 were listed in table 1.

Table 1 the parameters of state points

State points	T (K)	P (bar)
1	12	18.4
2	5.6	18
3	4.5	1.25
4	4.5	1.25
5	4.5	1.24
6	4.5	1.25
7	10.29	1.2

The cooling capacity of the large scale helium liquefiers are about a few or tens of kilowatts. Table 2 summarized the pulsed heat load of the typical cryogenic plants. It's concluded from table 2 [3] that the fluctuation amplitude of the Kuendig proposed heat load is biggest, and the pulse duration is shortest, so the effect on the refrigerators is most serious. Therefore, the heat load curves of the Kuendig et al [1] proposed were used in this paper, which were shown in Figure 2.

Table 2 the pulsed heat load of the typical helium liquefiers

Organization	High heat load (kW)	Hold time (s)	Duration time when switched heat loads (s)	Low heat load (kW)	Hold time (s)	Duration time when switched heat loads (s)
ITER	30	1500	1950	24	1	149
KSTAR	5.5	4050	450	3.3	6300	450
Kuendig	24	1750	1	12	1749	100

2.2. Process analysis

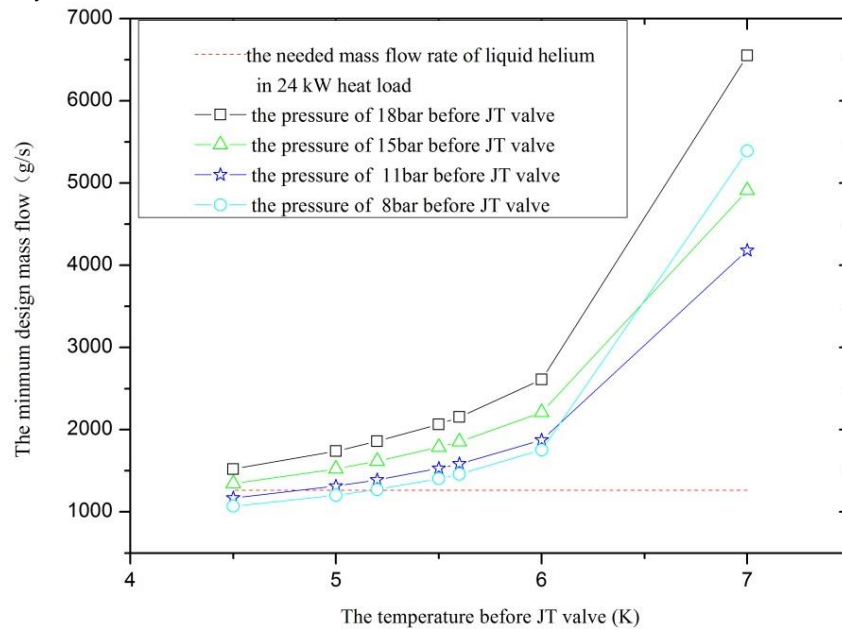


Figure 3 The relationship between the temperature and pressure before the JT valve and the return mass flow

Generally, the design heat load of refrigerator is the time-average heat load. The design mass flow rate can be calculated by following equations:

The vapor fraction x after JT valve can be calculated by:

$$x = \frac{h_{in} - h_{sl}}{h_{sv} - h_{sl}} \quad (1)$$

The needed liquid helium mass flow rate is calculated as follows:

$$\dot{m}_L = \frac{\dot{Q}_{design}}{h_{sl}} \quad (2)$$

The design mass flow rate of the refrigerator is:

$$\dot{m}_{design} = \frac{\dot{m}_L}{1-x} \quad (3)$$

The Figure 3 was depicted the relationships of the temperature, pressure before the JT valve and the design mass flow rate. In the condition of 24kW heat load, the needed minimum liquid helium mass flow rate is 1250g/s which showed in the red dotted line in Figure 3.

The return mass flow rate of the refrigerator (\dot{m}_{return}) includes two parts: the vapor generated after expansion in JT valve (\dot{m}_{JT}) and vapour generated by the heat load ($\dot{m}_{heat\ load}$)

$$\dot{m}_{return} = \dot{m}_{heat\ load} + \dot{m}'_{JT} = \frac{\dot{Q}}{h_l} + x_{JT} \dot{m}_{JT} \quad (4)$$

In steady state, the return mass flow rate (\dot{m}_{return}) is equal to the design mass flow rate (\dot{m}_{design}). From equation 4, it can be concluded that when the design mass flow rate of the refrigerator (\dot{m}_{design}) is less than evaporation of liquid helium by high heat load ($\frac{\dot{Q}_{high}}{h_l}$), it's not possible to completely mitigate the flow rate fluctuation by changing the vapor fraction after JT valve, as shown under the red dotted line in the Figure 3. In this condition, combining two or more methods is necessary to mitigate the fluctuation of the mass flow rate. When \dot{m}_{design} is more than $\frac{\dot{Q}_{high}}{h_l}$, changing the state before the JT valve is one useful method, which was shown above the red dotted line in Figure 3. The results provide a guideline in choosing the suitable cold end to mitigation of the fluctuation of the mass flow rate.

3. Modification of cycle configuration for mitigation of the effects of pulsed heat load

3.1. Cycle configuration

Based on the information in Figure 3 and the parameters in table1, changing the inlet conditions before the JT valve can be used to mitigate the fluctuation of the mass flow rate. The coldest part was modified by incorporating the two different UAs heat exchangers. Through adjusting the UA of the heat exchanger, the temperature before the JT valve varied. The flow process of the modified configuration was shown in Figure 4. The UA of the heat exchanger can be derived by the following equations (5-13).

The dimensionless fluid heat transfer coefficient j [9] can be calculated as following equation (5).

$$j = 0.377 R_e^{-0.5485} \left(\frac{s}{n}\right)^{-0.4544} \left(\frac{t}{l}\right)^{0.2074} \left(\frac{t}{s}\right)^{-0.3448} \quad (5)$$

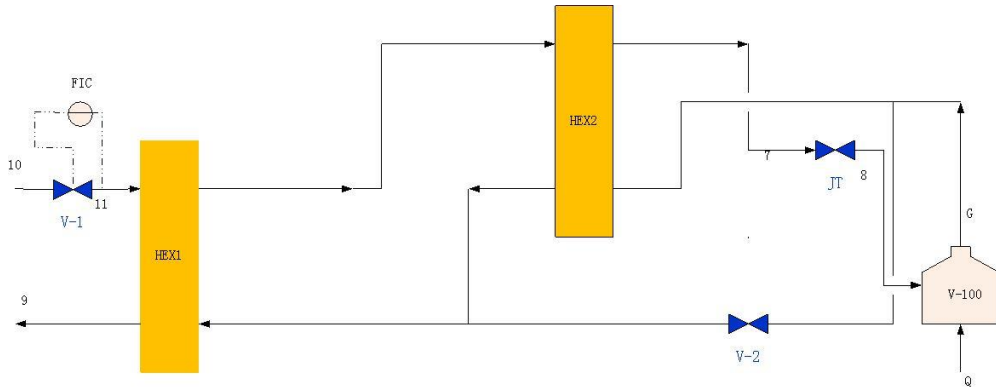


Figure 4 The flow process of the modified cycle

$$j = \frac{Nu}{RePr} \quad (6)$$

$$\dot{m} = \rho u A_c \quad (7)$$

$$Re = \frac{ud}{\mu} \quad (8)$$

$$Nu = \frac{UL}{\gamma} \quad (9)$$

Where, the s , n , t , l denote the space, height, thickness, length of fins, respectively. γ , μ are the thermal conductivity and the viscosity of the fluid.

Without considering the temperature effects on the fluid properties, Substituting equation (5-8) into equation (9), we obtain the fluid heat transfer coefficient U as follows:

$$U = k_1 Nu = k_1 j Re Pr = k_1 k_2 Re^{0.4515} = k_1 k_2 k_3 \dot{m}^{0.4515} = k \dot{m}^{0.4515} \quad (10)$$

So the UAs of the hot and cold fluid can be written as:

$$(UA)_h = (k \dot{m}^{0.4515})_h \quad (11)$$

$$(UA)_c = (k \dot{m}^{0.4515})_c \quad (12)$$

The UA of heat exchangers is calculated as following [7]:

$$\frac{1}{UA} = \frac{1}{(UA)_h} + \frac{1}{(UA)_c} \quad (13)$$

The total UA was decided by the high heat load in steady state. Through calculating the needed UA in low heat load, we can distribute the total UA to each heat exchanger. Therefore, changing the total UA can be achieved by adjusting the mass flow rate of the heat exchangers. In the cycle, the valves operation of the modified cycle were as following: during high heat load, the valve v-2 was closed; during the low heat load, the valve v-2 was controlled to adjust the temperature before JT valve. The valve v-1 was used to control the actual mass flow rate.

3.2. Solution method

The dynamic simulation was used to verify the applicability of the modified coldest part. The following assumptions have to be performed:

1. The temperature and pressure of the point 10 are constant (12K, 18.4bar). The pressure of outlet (point 9) is constant (1.2bar)
2. The vapor leaved the user's load heat exchanger is in the saturated condition.

3. No heat in-leak to the system

Before starting the simulation, all the streams and equipment are to be specified in accordance to the process condition. When the system is running, the heat load has been applied to the \dot{Q} according to the pulsed heat load curves in section 2.1

4. Results and discussions

The comparison between the usual and the modified coldest part has been shown in Figure 5-6 (the '1,2' represents the usual and the modified cold end, respectively). In order to estimate the advantages and disadvantages of the modified system, the term %-mitigation has been cited as following [6]:

$$\% \text{-mitigation|HHL} = 1 - \left(\frac{|\dot{m}_{\text{HHL,Withmitigation}} - \dot{m}_{\text{ss}}|}{\dot{m}_{\text{HHL,Withoutmitigation}} - \dot{m}_{\text{ss}}} \right) \quad (14)$$

$$\% \text{-mitigation|LHL} = 1 - \left(\frac{|\dot{m}_{\text{LHL,Withmitigation}} - \dot{m}_{\text{ss}}|}{\dot{m}_{\text{ss}} - \dot{m}_{\text{LHL,Withoutmitigation}}} \right) \quad (15)$$

Where \dot{m} is the return stream mass flow rate; and the subscripts HHL, LHL and SS represent values during high heat load, low heat load, and steady state, respectively.

4.1. Mitigation of return stream mass flow rate

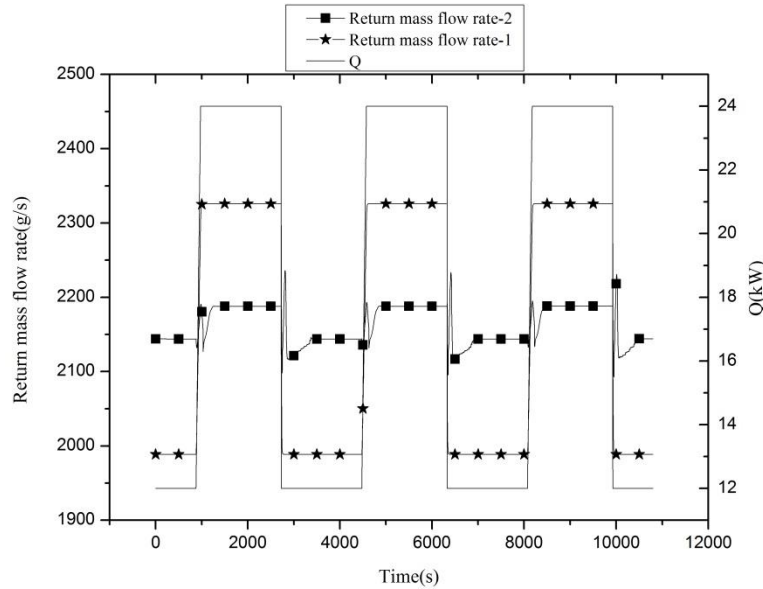


Figure 5 the comparison of return mass flow rate between the usual and modified coldest part during the operation in pulsed heat load.

In the design condition, the return mass flow rate was 2155g/s. When the refrigerator suffered the pulsed heat load, the return mass flow rate increased during the high heat load owing to the increase in evaporation of liquid helium and vice versa. The return mass flow rate of the usual cycle with one heat exchanger in coldest part in high and low heat load are 2326.4g/s, 1989g/s, respectively. In modified cold end, the values of the return mass flow rate are 2187g/s, 2145g/s during high and low heat load. The 81.3%, 93.98%-mitigation had been achieved in modified cycle during the high and low conditions. There still existed some fluctuations when the load was switched from one load condition to the other. The reasons were listed as following: 1) Numerical calculation was instable during the variation of the valve. 2) The thermal inertia of HEX2 resulted in the mismatch between the cooling capacity and the heat load.

4.2. Variation of the liquid percent level of dewar

Another effect of the pulsed heat load is the variation of the helium inventory in the V-100. At the low heat load condition, there is an increase in the liquid percent level. The opposite effect takes place at the high heat load condition. The variation of the liquid percent level in 3 periods was depicted in Figure 6. In the general coldest part, the liquid percent level decreased about 0.54% in one period. However, it is about 1.78% in one period in modified cold end. One main reason may be the time delay of re-establishing the new balance condition of the heat exchangers when the heat load is switched. Therefore, it's necessary to improve the design capacity to make up the variation of liquid inventory.

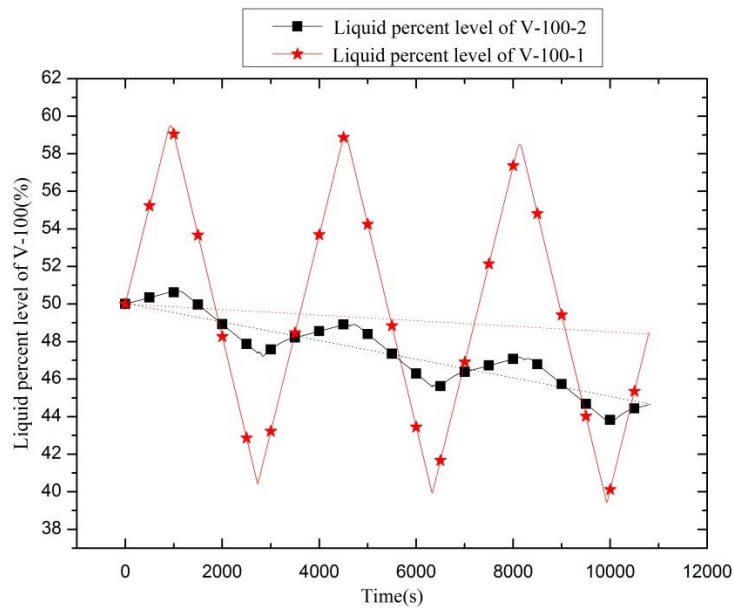


Figure 6 Variation of dewar liquid percent level between the usual cycle and the modified cold end configuraion

5. Conclusion

When the design mass flow rate is more than the vapor generation of the heat load, in theoretically, the fluctuation of mass flow rate can be mitigated completely through changing the flow distribution of the two different UAs heat exchangers during high and low heat load. However, owing to thermal inertia of the heat exchangers, the dynamic simulation demonstrated that fluctuations can be mitigated 81.3%, 93.98% in the high and low heat load conditions. Therefore, the control strategies have to be optimized to minimize the fluctuations in sudden variation of the heat loads. Comparing to other methods, this coldest part doesn't have any moving parts, and have better operability and usability. When the design mass flow rate is less than the vapor generation of the heat load, it can't possible to complete mitigation of the return mass flow rate only by changing the vapor fraction after the JT valve. At the moment, combining other methods in the coldest part such as adding supercritical tanks are the applicable methods.

Reference

- [1] Kuendigand A and Schoenfeld H 1995 Helium refrigerator design for pulsed heat load in tokamaks *Adv. Cryog. Eng.* **51 B** 1995-2001
- [2] Satoh S, Mito T, Satow T, Yamada,S, Yamamoto J and Motojima,O 1993 Conceptual design of cryogenic refrigeration for the large helical device *Fusion Eng. Des* **20** 129-136
- [3] Dutta R, Ghosh P and Chowdhury K 2011 Application of parallel heat exchangers in helium refrigerators for mitigating effects of pulsed load from fusion devices *Fusion Eng. Des.*,**86(4-5)**

- 296-306.
- [4] Dutta R, Ghosh P and Chowdhury K 2012 Mitigation of effects of pulsed heat loads in helium refrigerators for fusion devices using supercritical helium storage *IEEE Trans. Appl. Supercond.* **22(6)** 4203-712.
 - [5] Dutta R, Ghosh P and Chowdhury K 2013 Mitigation of effects of pulsed heat load from fusion devices on helium refrigerator: A novel technique using vapor compression cycle *Int. J. Refrig* **36(6)**:1776-89.
 - [6] Dutta R, Ghosh P and Chowdhury K 2013 A cycle configuration for large-scale helium refrigerator for fusion devices towards complete mitigation of the effects of pulsed heat load *Fusion Eng. Des* **88(11)** 2972-82.
 - [7] Deschildre C, Barraud A, Bonnay P, Briend,P, Girard A, Poncet, J M, Roussela P and Sequeirac S E 2008 Dynamic Simulation of an Helium Refrigerator *Adv. Cryog. Eng: Transactions of the Cryogenic Engineering Conference-CEC(Chattanooga)*, **Vol 53** 475-82.
 - [8] Dutta R, Ghosh P and Chowdhury K 2011 Customization and validation of a commercial process simulator for dynamic simulation of Helium liquefier *Energy* **36(5)** 3204-14.
 - [9] Lv c, Wu J H, Li Q, Xiu X J and Pan W 2015 CFD studies on heat transfer and pressure drop characteristics of an offset strip-fin heat exchanger in helium systems. *CEC-ICMC2015 (Tucson)*