Thermodynamic analysis and economical evaluation of two 310-80 K pre-cooling stage configurations for helium refrigeration and liquefaction cycle

Z G Zhu[1*], M Zhuang[1], Q F Jiang[1,2], Q Y Zhang[1] and H S Feng[1]

[1] Institute of Plasma Physics, Chinese Academy of Sciences, Hefei 230031, P.R. China
[2] University of Science and Technology of China, Hefei 230026, P.R. China

Email: zzg@ipp.ac.cn

Abstract. In 310-80 K pre-cooling stage, the temperature of the HP helium stream reduces to about 80 K where nearly 73% of the enthalpy drop from room temperature to 4.5 K occurs. Apart from the most common liquid nitrogen pre-cooling, another 310-80 K pre-cooling configuration with turbine is employed in some helium cryoplants. In this paper, thermodynamic and economical performance of these two kinds of 310-80 K pre-cooling stage configurations has been studied at different operating conditions taking discharge pressure, isentropic efficiency of turbines and liquefaction rate as independent parameters. The exergy efficiency, total UA of heat exchangers and operating cost of two configurations are computed. This work will provide a reference for choosing 310-80 K pre-cooling stage configuration during design.

1. Introduction
To provide enough cooling capacity to mitigate heat loads from the superconducting magnets, thermal shields, current leads etc. at different temperatures, helium cryoplants are employed in tokomak, synchrotrons, particle accelerators and colliders. As a critical section of helium refrigeration and liquefaction cycle, the pre-cooling section reduces the temperature of HP stream to low temperature. 310-80 K pre-cooling stage is one of the most important pre-cooling stages, where nearly 73% of the enthalpy drop from room temperature to 4.5 K occurs. Thus, heat transfer area of the heat exchangers in the 310-80 K pre-cooling stage is maximal, which largely determines the cost and volume of cold box.

Among the cryogenic systems having been commissioned in the last ten years, SNS, EAST and JT-60SA cryogenic helium systems adopt liquid nitrogen to cool HP helium stream to about 80 K [1-3]. Pre-cooling with liquid nitrogen can accelerate system cooling rate, increase liquefaction rate and allow adaption to off-design conditions. However, KSTAR cryogenic system uses turbine expander-based cooling stage that replaces the liquid nitrogen pre-cooling stage [4]. Wendelstein-7X cryogenic system has liquid nitrogen pre-cooling as well as turbine expander-based pre-cooling. During steady operation, turbine expander-based pre-cooling is put into operation. Only during cool down or to increase liquefaction rate of helium, liquid nitrogen is supplied [5]. In the future, ITER will have its own nitrogen production facility on site to support the liquid nitrogen pre-cooling stage of all helium plant [6].

Though two kinds of 310-80 K pre-cooling methods are adopted in many cryogenic systems,

thermodynamic and economical analyses are rarely discussed in the literature. In this study, exergy efficiency, total UA of heat exchangers and operating cost of two 310-80 K pre-cooling configurations are computed and compared. The aim of this work is to provide a theoretical basis for choosing 310-80 K pre-cooling stage configuration.

2. Thermodynamic model of 310-80 K pre-cooling stage

Schematic diagrams for 310-80 K pre-cooling stages with liquid nitrogen and turbine pre-cooling options are presented in figure 1. In the pre-cooling stage with liquid nitrogen (figure 1.a), heat exchanger HX1 has three streams: HP stream, LP stream and N2 stream. The room temperature helium in the HP stream is pre-cooled to about 80 K by cold nitrogen and returning helium in the LP stream. In the pre-cooling stage with turbine (figure 1.b), both HX3 and HX4 have only two streams. HX3 is used for pre-cooling the helium before it is expanded in the turbine and returns as cold LP stream through HX4 to provide cooling to the incoming HP stream.
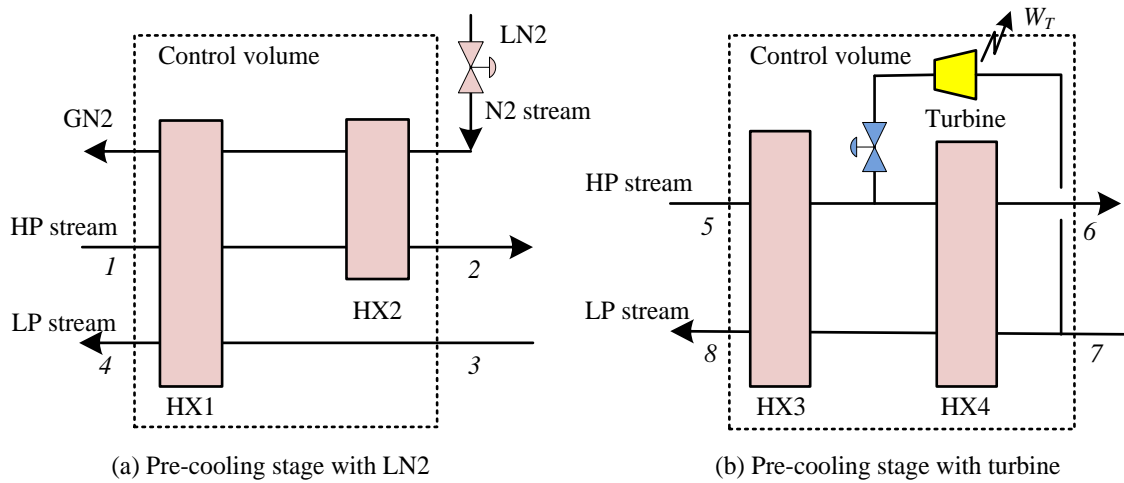

(a) Pre-cooling stage with LN2   (b) Pre-cooling stage with turbine

Figure 1. Two different arrangements of 310-80 K pre-cooling stages.

For a stream, specific exergy is defined as [7]:

$$ex = (h - h_0) - T_0(s - s_0) \tag{1}$$

where ex is the maximum work per unit mass that can be extracted out of the system while taking it to the reference state (T_0, P_0). Usually, atmospheric condition is taken as the reference state ($T_0 = 310$ K, $P_0 = 1.01$ bar). h is specific enthalpy, s is specific entropy of the fluid. Subscript '0' refers to the reference state. h_0 is specific enthalpy, s_0 is specific entropy of the reference state.

Exergy (kW) transfer associated with mass flow rate ($\dot{m}$) is given as:

$$Ex_{flow} = \dot{m} \times ex \tag{2}$$

2.1 310-80 K pre-cooling stage with liquid nitrogen

For steady state, exergy balance across the control volume as presented in figure 1.a gives

$$0 = (\dot{m}_1 \, ex_1 + \dot{m}_3 \, ex_3 + \dot{m}_{LN_2} \, ex_{LN_2}) - (\dot{m}_4 \, ex_4 + \dot{m}_2 \, ex_2 + \dot{m}_{GN_2} \, ex_{GN_2}) - Ex_{loss\_LN2} \tag{3}$$

Some part of exergy supplied to the control volume is lost due to irreversibility associated with the processes. Ex_{loss_LN2} denotes total exergy loss in the pre-cooling stage with liquid nitrogen.

Rearranging Eq. (3):

$$\dot{m}_1 \, ex_1 - \dot{m}_4 \, ex_4 + \dot{m}_{LN_2} \, ex_{LN_2} - \dot{m}_{GN_2} \, ex_{GN_2} = \dot{m}_2 \, ex_2 - \dot{m}_3 \, ex_3 + Ex_{loss\_LN2} \tag{4}$$

Terms on left hand side of Eq.(4) represent the exergy input to the pre-cooling stage with liquid nitrogen. The exergy transferring to the next pre-cooling stage below 80K is denoted by the first two terms on right hand side of Eq.(4).

$$\text{Exergy input to the control volume} = \dot{m}_1\, ex_1 - \dot{m}_4\, ex_4 + \dot{m}_{LN_2}\, ex_{LN_2} - \dot{m}_{GN_2}\, ex_{GN_2} \tag{5}$$

$$\text{Exergy output of the control volume} = \dot{m}_2\, ex_2 - \dot{m}_3\, ex_3 \tag{6}$$

Exergy efficiency for the 310-80 K pre-cooling stage with liquid nitrogen is:

$$\eta_{ex\_LN_2} = \frac{\dot{m}_2\, ex_2 - \dot{m}_3\, ex_3}{\dot{m}_1\, ex_1 - \dot{m}_4\, ex_4 + \dot{m}_{LN_2}\, ex_{LN_2} - \dot{m}_{GN_2}\, ex_{GN_2}} \tag{7}$$

2.2 310-80 K pre-cooling stage with turbine

Similarly, exergy balance across the control volume of the pre-cooling stage with turbine gives

$$0 = (\dot{m}_5\, ex_5 + \dot{m}_7\, ex_7) - (\dot{m}_6\, ex_6 + \dot{m}_8\, ex_8) - \dot{W}_T - Ex_{loss\_T} \tag{8}$$

where, $\dot{W}_T$ is expander work of the turbine (kW), Ex_{loss_T} represents total exergy loss in the pre-cooling stage with turbine.

Rearranging Eq. (8):

$$\dot{m}_5\, ex_5 - \dot{m}_8\, ex_8 = \dot{m}_6\, ex_6 - \dot{m}_7\, ex_7 + W_T + Ex_{loss\_T} \tag{9}$$

As the expansion work produced by turbine is not utilized, it is eventually converted to heat and taken away by cooling water. The total exergy leaving the system is

$$\text{Exergy input to the control volume} = \dot{m}_5\, ex_5 - \dot{m}_8\, ex_8 \tag{10}$$

$$\text{Exergy output of the control volume} = \dot{m}_6\, ex_6 - \dot{m}_7\, ex_7 \tag{11}$$

Exergy efficiency for the 310-80 K pre-cooling stage with turbine is:

$$\eta_{ex\_Turbine} = \frac{\dot{m}_6\, ex_6 - \dot{m}_7\, ex_7}{\dot{m}_5\, ex_5 - \dot{m}_8\, ex_8} \tag{12}$$

To compare their operation cost, the price ratio (*M*) is defined as the ratio of the local electricity price (per kW h) to liquid nitrogen acquisition cost (per liter).

$$M = \frac{price_{electricity}}{price_{liquid\_nitrogen}} \tag{13}$$

The operation cost ratio (*C*) is defined as the ratio of the operation cost of the pre-cooling stage with turbine to operation cost of the pre-cooling stage with liquid nitrogen.

$$C = \frac{P_{com}}{Q} \cdot M \tag{14}$$

where, P_{com} is the power consumed by compressors for the part of helium through turbine (kW h), Q is the liquid nitrogen consumption (Liter). When *C* is larger than 1, the operating cost of the pre-cooling stage with turbine is larger than with liquid nitrogen. If *C* is less than 1, the operation cost with liquid nitrogen is higher.

The thermodynamic study has been performed on the basis of the following assumptions:
1. The system is in steady state.
2. The isothermal efficiency of compressor is kept constant at 50%. The efficiency of turbine is constant.

3. Pressure loss in oil removal system, heat exchanger and pipes is negligible.
4. No heat leak of heat exchangers.
5. The temperature difference of hot end in HX1and HX3 is 10 K while HX2 and HX4 2 K.
6. Both HP inlet temperatures, 1 and 5, are 310 K. The HP outlet temperatures, 2 and 6, are 80 K. HP steam outlet mass flow rate is 100 g/s.

Thermo-physical properties of helium are obtained using HEPAK.

3. Results and discussions
Using the methodology described above, the effects of compressor discharge pressure and liquefaction rate on exergy efficiency, total UA of heat exchangers and operating cost of two 310-80 K pre-cooling stages in refrigeration mode and liquefaction mode are studied.

3.1 Refrigeration mode
In the refrigeration mode, the relationships between exergy efficiency, total UA of heat exchangers, operating cost and discharge pressure are presented in figures 2-4. Figure 2 depicts that the exergy efficiency of 310-80 K pre-cooling stage with liquid nitrogen is higher than with turbine. The result is expected, as expansion work of the turbine is not utilized. When discharge pressure of compressors increases from 14 bara to 22 bara, there is a slight increment in exergy efficiency for two pre-cooling stages. As shown in figure 3, total heat exchanger UA of pre-cooling stage with turbine is larger than with liquid nitrogen. For pre-cooling stage with turbine, higher compressor discharge pressure lead to an increment in total heat exchanger UA while compressor discharge pressure has no effect on total heat exchanger UA for the pre-cooling stage with liquid nitrogen.

Figure 4 shows in the more detail the significant effect of the price ratio M on their operation cost ratio C. When M is larger than 1.2, the operating cost of the pre-cooling stage with turbine is larger when compared with liquid nitrogen. Higher isentropic efficiency of the turbine lead to less helium through the turbine, thus the power consumed by compressors will drop, which cause a lower C. However, this figure indicates that compressor discharge pressure makes little difference on C.
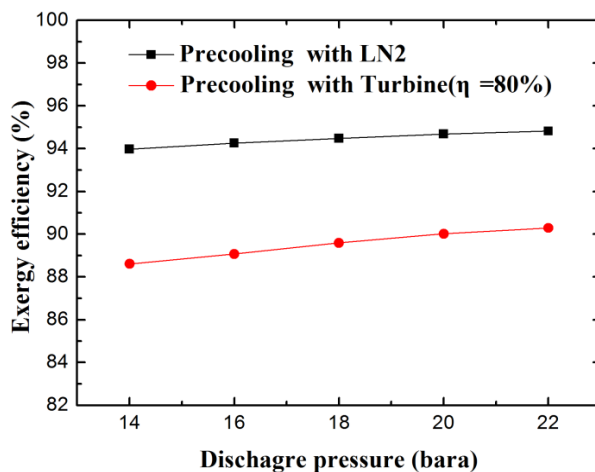

Figure 2. Effect of compressor discharge pressure on exergy efficiency for refrigeration mode.
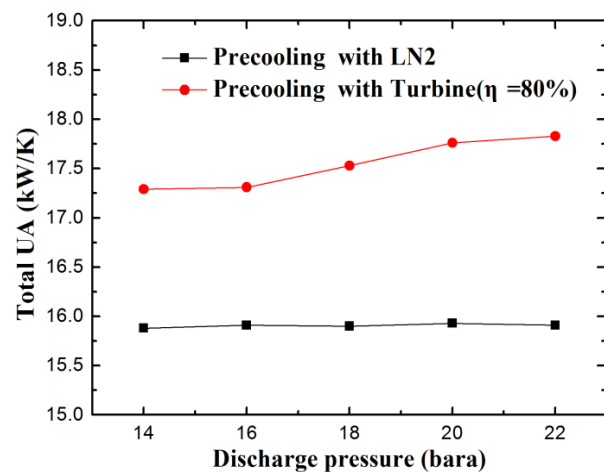
Figure 3. Effect of compressor discharge pressure on total UA for refrigeration mode.

3.2 Liquefaction mode
For liquefaction mode, the effects of liquefaction rate on exergy efficiency of the pre-cooling stage at 14 bara and 20 bara discharge pressure are respectively shown in figures 5-6. From figures 5 and 6, the exergy efficiency of pre-cooling stage with liquid nitrogen is much higher than with turbine, which is similar with refrigeration mode. With the increment of liquefaction rate, the two exergy efficiencies have a decremental trend. As the returning cold helium in heat exchangers decreases, the exergy

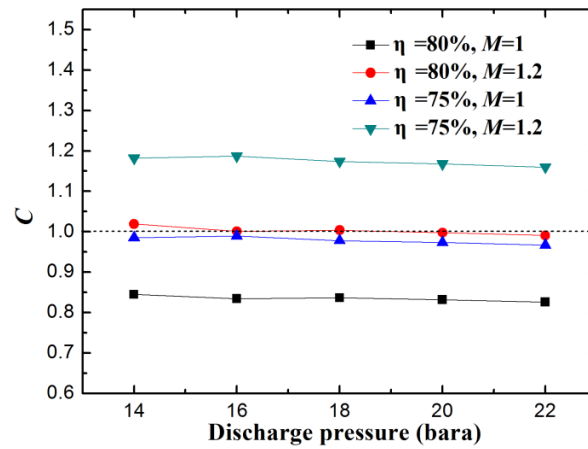efficiency of the pre-cooling stage with turbine is more susceptible.

Figure 4. Effects of discharge pressure, η and M on C for refrigeration mode.
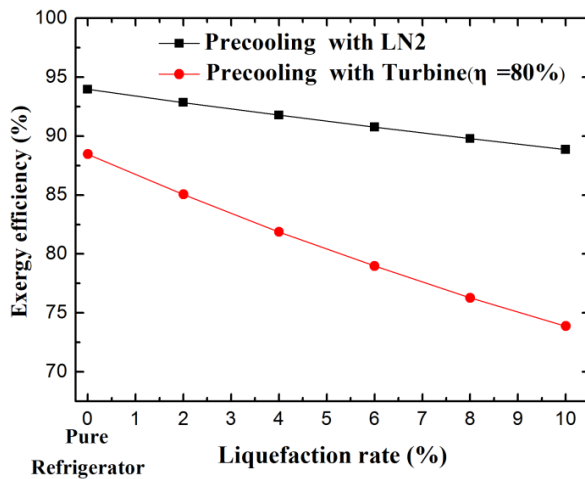

Figure 5. Effect of liquefaction rate on exergy efficiency for liquefaction mode ($P_d = 14$ bara).
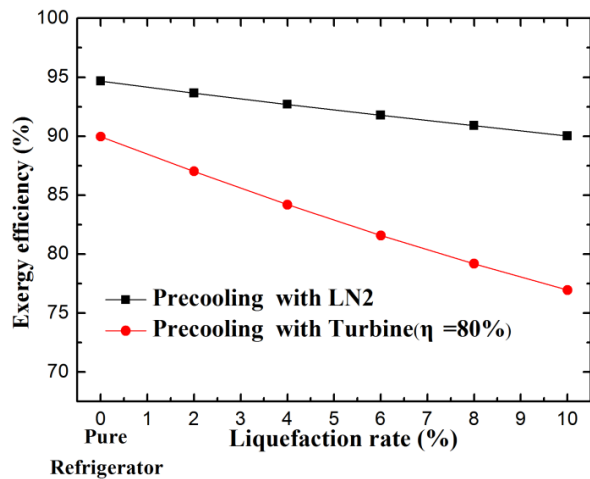

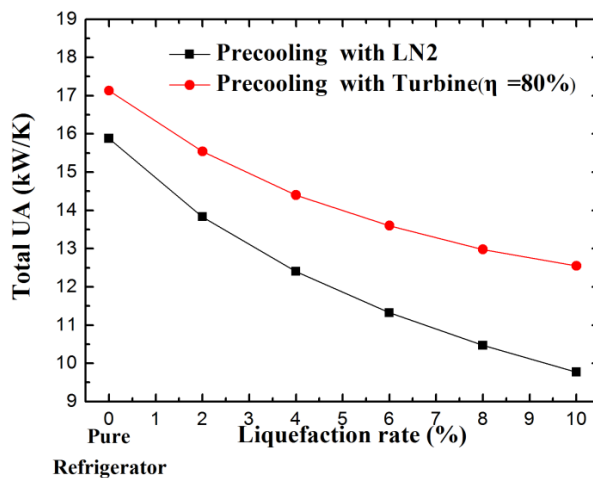**Figure 6.** Effect of liquefaction rate on exergy efficiency for liquefaction mode ($P_d = 20$ bara).

Figure 7. Effect of liquefaction rate on total UA for liquefaction mode ($P_d = 14$ bara).
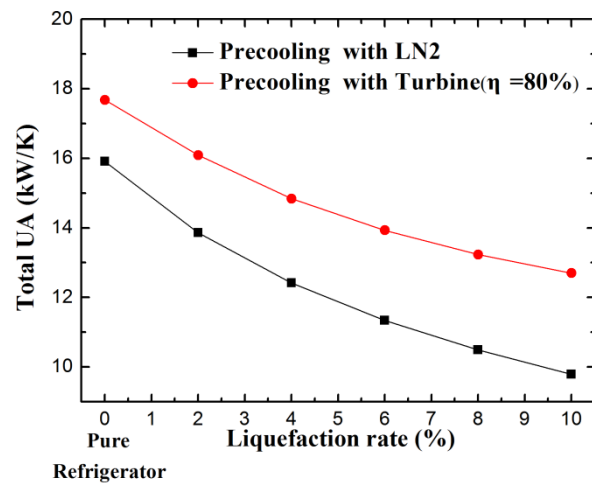

Figure 8. Effect of liquefaction rate on total UA for liquefaction mode ($P_d = 20$ bara).

Because of higher heat transfer temperature difference, total UA with liquid nitrogen pre-cooling is smaller than with turbine. As depicted in figures 7 and 8, the two total UAs decrease as liquefaction rate increases. This is due to less amount of heat exchange caused by less mass flow of the returning cold helium in the LP steam. The relationships between operating cost of the pre-cooling stage and liquefaction rate, isentropic efficiency of the turbine and the price ratio *M* are presented in figures 9 and 10. These findings are in agreement with the results of refrigeration mode.
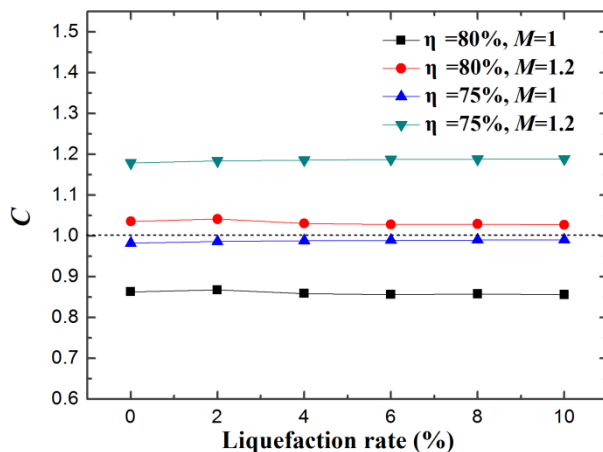

Figure 9. Effect of liquefaction rate on *C* for liquefaction mode (P_d = 14 bara).
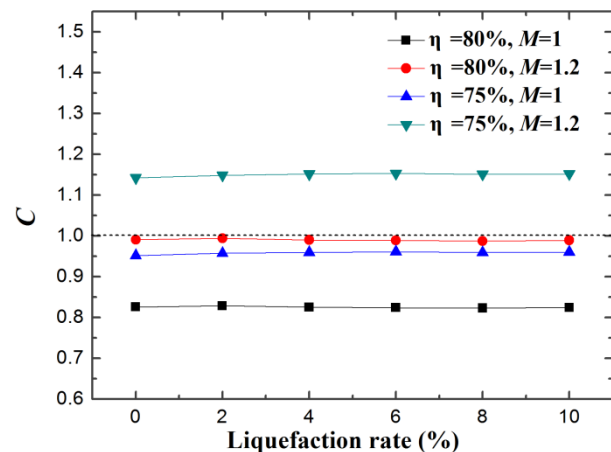
Figure 10. Effect of liquefaction rate on *C* for liquefaction mode (P_d = 20 bara).

Besides the turbine isentropic efficiency, acquisition cost of liquid nitrogen and local electricity price have a great influence on economical performance. According to [8], the specific power consumption is about 0.38 kW·h/m3 (oxygen) for China's large-scale air separation plants. The ratio of industrial electricity price to the production cost of liquid is about 13 in China. Therefore, it is more economical to set up liquid nitrogen cryoplant to provide liquid nitrogen for 310-80 K pre-cooling stage for large-scale cryogenic system, like ITER cryogenic system.

4. Conclusions

Both in refrigeration and liquefaction modes, exergy efficiency of 310-80 K pre-cooling stage with liquid nitrogen is higher than with turbine, while the demand of total heat exchangers UA is smaller due to larger heat transfer temperature difference. Apart from the turbine isentropic efficiency, acquisition cost of liquid nitrogen and price of the local electricity influence their economical performance. For large-scale helium cryoplant, it is more economical to set up a liquid nitrogen cryoplant to provide liquid nitrogen to cool the HP helium stream because of very low production cost of liquid nitrogen.

5. References

[1]   Hatfield D, Casagrande F, Campisi I, Gurd P, Howell M, Stout D, Strong H, Arenius D, Creel J and Dixon K 2016 *AIP Conf. Proc.* **823** 1436
[2]   Bai H, Bi Y, Zhu P, Zhang Q, Wu K, Zhuang M and Jin Y 2006 *Fusion Eng. Des.* **81** 2597
[3]   Lamaison V, Beauvisage J, Fejoz P, Girard S, Gonvalves R, Gondé R, Heloin V, Michel F, Hoa C and Kamiya, K 2014 *AIP Conf. Proc.* **1573** 337
[4]   Kim Y S, Park Y, Lee, Y, Kwag S, Chang Y, Song N, Woo I, Park H, Cho K and Chang H 2009 *IEEE Trans. Appl. Supercond.* **19** 1577
[5]   Kuendig A, Dhard C, Raatz S and Bau H 2008 *Proc. 22nd Int. Cryog. Eng. Conf.* Seoul Korea
[6]   Monneret E, Chalifour M, Bonneton M, Fauve E, Voigt T, Badgujar S, Chang H and Vincent G 2015 *Phys. Procedia* **67** 35

[7]   Kotas T J 1985 *The exergy method of thermal plant analysis*. (Stoneham MA: Butterworth Publishers) pp 288.

[8]   Zhang X B, Chen J Y, Yao L, Huang Y H, Zhang X J and Qiu L M 2014 *J Zhejiang Univ.-Sc. A* **15** 309