Numerical investigation of the potential efficiency of a cryogen-free VM type pulse tube cryocooler

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Abstract. The Vuilleumier (VM)-type pulse tube cryocooler is a novel kind of cryocooler used to obtain liquid helium temperature, which has been experimentally verified. However, the overall efficiency is not satisfying. Based on previous work on a low pressure ratio system, numerical investigations that explore the effect of high pressure ratios on cooling performance are presented in this paper. The research system is cryogen-free in which a Stirling type pulse tube cryocooler is used to provide the required cooling power for the thermal compressor and makes it adjustable for pre-cooling temperature to achieve optimum efficiency. First, after increasing the displacer swept volume to increase the pressure ratio, the dimensions of the main components were optimized with the lowest no-load temperature as the optimization target. Then the dependence of system performance on average pressure, displacer swept volume, frequency and cold HX temperature were studied considering the heat transfer temperature difference of the thermal bridge. Compared with the previous low efficiency under low pressure ratio, when the heat transfer temperature difference of the thermal bridge is 0 K, a higher relative Carrot efficiency of 1.29\% and a cooling power of 250 mW at 4.2 K were predicted with an average pressure of 3 MPa, a frequency of 3 Hz and a pressure ratio of 1.67. Further optimization is underway.

1. Introduction

The need for of small-scale 4 K cryocoolers with high efficiency is increasing due to the extensive application in space exploration, low-temperature superconduction and medical service, etc. Besides, these cryocoolers can be used as a pre-cooling source for milli-Kelvin refrigerators [1]. A two-stage Gifford-McMahon (G-M) cryocoolers or a G-M type pulse tube
cryocoolers (PTC) are traditional commercial 4 K cryocoolers, which can achieve a relatively high cooling power at liquid helium temperature [2,3]. However, the efficiency is reduced due to the work loss of the rotary valve and the suction and exhaust valves of the G-M compressor. Oil lubrication also limits their application in special circumstance such as aerospace. Multi-stage Stirling type pulse tube cryocoolers (SPTC) are another type of 4 K cryocooler which is developing rapidly due to their potential for long lifetime, compactness and low vibrations [4,5]. The SPTC normally works at frequencies above 20 Hz, which leads to high pressure drop loss across porous regenerator normally filled with small diameter spheres.

VM type pulse tube cryocoolers are driven by a thermal compressor and have potential advantages of compactness, long lifetime, oil-less operation and high efficiency at liquid helium temperature. Matsubara Y et al. [6] proposed the idea of using a thermal compressor to drive a 4 K pulse tube cryocooler. Based on the concept, Dai et al. [7] built a V-M type pulse tube cryocooler setup and obtained a lowest temperature below 4 K with a 20 K pre-cooler for the first time. Wang et al. [8] improved a two-stage gas-coupled VM-PT cryocooler based on the previous work by Pan and Zhang et al. [9,10], and the no-load temperature of 2.17 K was achieved. Wang et al. [11] developed a single stage VM type pulse tube cryocooler to obtain a lowest temperature of 7.58 K. In addition to the experimental research, a highest relative Carrot efficiency of 0.82% at 5 K was predicted by theoretical calculation [12].

Following the work introduced Wang et al. [12], it was found that the relatively low efficiency was due to the low pressure ratio so we have made some major modifications on the system. Compared with previous system performance, a much higher pressure ratio was used by changing the displacer swept volume. The dimensions of the main components were optimized with the lowest no-load temperature as the optimization target. Then considering the location of the cold heat exchanger of the thermal compressor and the cold-head of the pre-cooler and the structure design of the thermal bridge, the different heat transfer temperature difference between the two sides of the thermal bridge exists. The dependence of system performance on average pressure, displacer swept volume, frequency and cold HX temperature were studied under different heat transfer temperature difference on the thermal bridge.

2. System configuration

Fig.1 shows the schematic of the VM type pulse tube cryocooler. The system includes three subsystems: pre-cooler, thermal compressor, and low temperature stage pulse tube cryocooler. The pre-cooler provides the cooling power required for the thermal compressor. The thermal compressor consists of ambient heat exchanger, regenerator, cold heat exchanger and displacer driven by a motor, which operates between ambient temperature and the cold HX temperature to generate an oscillating pressure wave to drive the low temperature stage pulse tube cryocooler. The two subsystems are thermally coupled through a thermal bridge. The heat released by the thermal compressor is transferred to the cold-head of the pre-cooler through a thermal bridge. A heat transfer temperature difference exists across of the thermal bridge. The low temperature stage pulse tube cryocooler is mainly composed of regenerator, cold-head, U shape connecting tube, pulse tube and phase shifter. The orifice-reservoir and double-inlet are used to adjust the phase difference between the pressure wave and volume
flow rate. Compared with the previous system [12], some modifications were carried out and major component dimensions are listed in Table 1.

**Table 1.** Main structure parameters of VM type pulse tube cryocooler.

| Subsystem                        | Components     | Parameters                                                                 |
|----------------------------------|----------------|---------------------------------------------------------------------------|
| Thermal compressor               | Thermal HX     | Diameter 40 mm, length 52 mm, finned type HX                             |
|                                  | Regenerator I  | Diameter 27 mm, length 150 mm, 80# stainless steel screens               |
|                                  | Displacer      | Diameter 100 mm                                                           |
| Low temperature stage pulse tube | Regenerator I  | Inner diameter 23 mm, length 148 mm, 0.2 mm sphere diameter, with a porosity of 0.36. |
| cryocooler                       | (S304)         |                                                                           |
|                                  | Regenerator II | Inner diameter 23 mm, length 23 mm, 0.15 mm sphere diameter, with a porosity of 0.36. |
|                                  | (Er<sub>3</sub>Ni) |                                  |
|                                  | Pulse tube     | Inner diameter 14 mm, length 150 mm, wall thickness 0.15 mm              |
|                                  | Reservoir      | Volume 1 L                                                                |

**Figure 1.** Schematic of the VM type pulse tube cryocooler.

1. Connecting rod with linear motor  2. Ambient HX I  3. Displacer  4. Regenerator I (S304)  5. Cold HX  6. Regenerator II (Er<sub>3</sub>Ni)  7. Regenerator II (HoCu<sub>2</sub>)  8. Cold-head  9. U shape connecting tube  10. Flow straightener  11. Pulse tube  12. Ambient HX II  13. Double-inlet valve  14. Orifice valve  15. Reservoir

**3. Numerical simulation and analysis**
3.1. Simulation tool
The numerical simulation and optimization are carried out using Sage software. The helium gas property is obtained from Refprop. The baseline operation conditions are: 3 Hz in frequency, 2.5 MPa in average pressure, 7 mm in displacer amplitude, 300 K and 77 K for ambient HX and cold HX temperature of thermal compressor, respectively. For each case, openings of orifice and double-inlet value are used as the optimized variables, which means that the opening of each valve is different in each case.

3.2. Influence of the operating parameter
The overall relative Carnot efficiency as an important criterion for cooling performance is defined as

$$\eta = \frac{Q_{c}}{W_{in}} / \frac{T_{c}}{T_{h} - T_{c}}$$  \hspace{1cm} (1)

Where $T_{c}$ is 4.2 K as a low temperature heat source and $T_{h}$ is 300 K as an ambient temperature heat source; $Q_{c}$ is the cooling power absorbed by the cold-head at 4.2 K; $W_{in}$ is the total input power, which includes the electric power $W_{1}$ to drive the displacer of thermal compressor and another electric power $W_{2}$ consumed by the pre-cooler. In the total input power, the value of $W_{1}$ is much smaller than $W_{2}$ with about two orders of magnitudes difference, so it can be ignored in calculation. $W_{2}$ is calculated with

$$W_{2} = \frac{Q_{pre}}{\eta_{pre}} / \frac{T_{pre}}{T_{h} - T_{pre}}$$  \hspace{1cm} (2)

Where $T_{pre}$ and $Q_{pre}$ represent the temperature of cold-head of pre-cooler and cooling power of pre-cooler required for the thermal compressor, respectively. When the temperature difference of heat transfer on the thermal bridge is 0 K, $T_{pre}$ is equal to the cold HX temperature of thermal compressor. $\eta_{pre}$ is relative Carnot efficiency of pre-cooler at $T_{pre}$, which is an estimate based on the state of the art of cryocooler technology. By assuming that relative Carnot efficiency at 40 K and 77 K is 14.8% and 25.9% respectively [13], the efficiency value at each $T_{pre}$ can be roughly obtained through linear interpolation. In general, combining equation 1 and 2, the overall relative Carnot efficiency can be obtained

$$\eta = \eta_{pre} \cdot \frac{Q_{c}}{Q_{pre}} \cdot \frac{T_{pre}}{T_{h} - T_{pre}} \cdot \frac{T_{h} - T_{c}}{T_{c}}$$  \hspace{1cm} (3)

Where, $\eta_{pre}$ is expressed as

$$\eta_{pre} = 0.148 + \frac{0.259 - 0.148}{77 - 40} \cdot (T_{pre} - 40)$$  \hspace{1cm} (4)

Figure 2(a) shows the pressure ratio and acoustic power at the regenerator II cold end with various average pressures. A larger average pressure leads to a larger pressure ratio and acoustic power. The influence of average pressure on cooling performance at 4.2 K is shown in Figure 2(b). With the increasing of average pressure from 1.5 MPa to 3 MPa, the cooling power
increases from 123 mW to 250 mW and relative Carnot efficiency increases from 0.83% to 1.29% when the temperature difference of heat transfer on the thermal bridge is 0 K. Under the temperature difference of 5 K and 10 K, the relative Carnot efficiency is 1.11% and 0.95% with an average pressure of 3 MPa, respectively.

Figure 2. System performance at 4.2 K with various average pressure and heat transfer temperature difference on the thermal bridge. (a) Pressure ratio and acoustic power at cold end of regenerator II, (b) Cooling power and relative Carnot efficiency. 3 Hz, 7 mm displacement, 77 K cold HX temperature.

The displacement amplitude of the displacer can be increased to increase the amplitude of output pressure wave of the thermal compressor, which means both pressure ratio and acoustic power provided to the low temperature stage pulse tube cryocooler will be increased. Figure 3(a) shows the dependence of the pressure ratio and acoustic power at regenerator II cold end on displacer displacement amplitude. Figure 3(b) shows cooling power and relative Carnot efficiency at 4.2 K with various displacer amplitudes, both of which increase with the displacer amplitude. When the amplitudes increases from 5 mm to 9 mm, the cooling power increases by more than 3 times.

Figure 3. System performance at 4.2 K with various displacer amplitudes and heat transfer temperature difference on the thermal bridge. (a) Pressure ratio and acoustic power at cold end of regenerator II, (b) Cooling power and relative Carnot efficiency. 2.5 MPa, 3 Hz, 77 K cold HX temperature.
Figure 4(a) shows the pressure ratio and acoustic power at the cold end of regenerator II as a function of frequency. As the frequency increases, the pressure ratio increases slightly and then decreases. The frequencies has a great influence on acoustic power. As the frequency increases from 3 Hz to 7 Hz, the volume flow rate amplitude can increase sharply, which will lead to a significant increase in acoustic power. Figure 4(b) shows cooling power and relative Carnot efficiency at 4.2 K as a function of frequency, presenting the optimal cooling performance. Under the heat transfer temperature difference of 0 K, 5 K and 10 K, the optimum cooling power is 376 mW at 6.5 Hz and the optimum relative Carnot efficiency is 1.11%, 0.96% and 0.82% at 4 Hz, respectively.

**Figure 4.** System performance at 4.2 K versus frequency and heat transfer temperature difference on the thermal bridge. (a) Pressure ratio and acoustic power at the cold end of regenerator II, (b) Cooling power and relative Carnot efficiency. 2.5 MPa, 7 mm displacement, 77 K cold HX temperature.

Figure 5(a) shows pressure ratio and acoustic power at the cold end of regenerator II as a function cold HX temperatures. As the thermal compressor cold HX temperature changes from 85 K to 65 K, pressure ratio at the cold end of regenerator II increases from 1.53 to 1.82 and there is a slight change in acoustic power, not exceeding 0.5 W. Figure 5(b) shows cooling power and relative Carnot efficiency at 4.2 K as a function of cold HX temperature. The cooling power decreases as the cold HX temperature increases. There is an optimal cold HX temperature for thermal compressor to achieve the maximum efficiency, and the optimum relative Carnot efficiency is 1.11%, 0.95% and 0.81% under different heat transfer temperature difference with a cold HX temperature of 75 K, respectively.
Figure 5. System performance at 4.2 K under different heat transfer temperature difference on the thermal bridge versus cold HX temperature. (a) Pressure ratio and acoustic power at cold end of regenerator II, (b) Cooling power and relative Carnot efficiency. 2.5 MPa, 3 Hz, 7 mm displacement.

4. Conclusion
This paper numerically investigates the potential efficiency of a 4.2 K cryogen-free VM type pulse tube cryocooler considering the different heat transfer temperature difference of the thermal bridge at 77 K. The analysis shows that average pressure and displacer displacement amplitude have significant effect on the cooling performance. With an increasing average pressure and displacement, both cooling power and relative Carnot efficiency increase. There is an optimal frequency and cold HX temperature for the whole system to achieve the maximum efficiency. Under the influence of various losses, the optimal frequency value for cooling power and efficiency exists at different working condition. In addition, the heat transfer temperature difference of the thermal bridge has a certain effect on the relative Carnot efficiency, and the efficiency difference between 0 K and 10 K temperature difference does not exceed 0.35% under the conditions studied. By optimizing the system design, the temperature difference can be reduced to improve efficiency. Throughout the numerical investigation, when the heat transfer temperature difference is 0 K, a higher relative Carnot efficiency of 1.29% at 4.2 K is reached with an average pressure of 3 MPa, a frequency of 3 Hz and a pressure ratio of 1.67, which is comparable to that of a small-scale G-M type cryocooler working at liquid helium temperature. Meanwhile, a higher cooling power of 376 mW is obtained with an average pressure of 2.5 MPa, a frequency of 6.5 Hz and a pressure ratio of 1.58.

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