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**Research Article**

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# Structural optimization of the rotary valve in a two-stage G-M-type pulse-tube cryocooler working at liquid helium temperatures

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**Abstract:** G-M-type pulse-tube cryocoolers (GM-PTC) working at liquid helium temperatures are promising in quantum technology and cryogenic physics for their high reliability and minimal vibration. These features stem from the fact that there are no extra moving parts introduced into the system. The rotary valve is a key component in GM-PTCs that transfers the output exergy from the compressor to the cold head. Because a low Carnot efficiency of 1.58% is achieved at liquid helium temperatures, optimizing the rotary valve is crucial for improving the efficiency of GM-PTCs. In this regard, an exergy-loss analysis method is proposed in this paper to quantitatively obtain the leakage loss and viscosity loss of a rotary valve by experimental measurements. The results show that viscosity loss accounts for more than 97.5% of the total exergy loss in the rotary valve, and that it is possible to improve the structure of the rotary valve by expanding the flow area by 1.5 times. To verify the method, the cooling temperature and power of a remote two-stage GM-PTC was monitored, with original or optimized rotary valves installed. The experimental results show that compared to the original rotary valve, the optimized rotary valve can improve the cooling efficiency of a GM-PTC by 16.4%, with a cooling power of 0.78 W at 4.2 K.

**Key words:** Rotary valve; Exergy analysis; Liquid helium temperature; G-M type pulse-tube cryocooler; High efficiency

## 1 Introduction

Compared with Gifford-McMahon (G-M) cryocoolers, G-M-type pulse-tube cryocoolers (GM-PTCs) have no moving parts at the cold end, and have inherently high reliability, long life, low vibration, and no electromagnetic interference (Kasai et al., 2022; Ji et al. 2022), making them a promising cooling source for applications such as the superconductive quantum interference device (SQUID), superconducting quantum chip cooling (Choi, 2023; Matthew et al., 2023), and particle physics (Alduino et al., 2019; The CUORE Collaboration, 2022). Although GM-PTCs have become mature commercial products (SHI Cryogenics Group, 2023; Cryomech Products, 2023), they still have limitations in terms of low cooling capacity and efficiency (de Waele, 2011; de Waele, 2012). The PT425 GM-PTC manufactured by Cryomech, for example, can provide a cooling capacity of 2.7 W at 4.2 K, but its relative Carnot efficiency is as low as 1.58% (Hao et al., 2022; Cryomech Products, 2023). Therefore, improving efficiency at liquid helium temperature is vital for large-scale application of GM-PTCs (Wang and Gan, 2013; Lei and Xu, 2022).

The GM-PTC system consists of a compressor, distribution valve, and cold head. A series of theoretical studies have been carried out on the side with the cold head, including attempts to optimize the regenerator (Gan et al., 2009), pulse tube (Liang and de Waele, 2007), and phase shifters (Zhu et al., 2007) to improve efficiency. With regard to the compressor and distribution valves, many numerical investigations have been conducted with the help of Sage

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(Gedeon, 2014) and computational fluid dynamics (Karpenko and Bogdevičius, 2020), and a very limited number of the studies have examined ways to measure the significant exergy losses suffered by the compressor and distribution valves of GM-PTCs (Liu et al., 2017).

The exergy loss of the compressor in a Stirling-type pulse-tube cryocooler is about 20% to 30% (Radebaugh et al., 2002; Tanaeva et al., 2006), but the exergy loss of the compressor in a GM-PTC uses up about 50% of the input electrical power (Liu et al., 2017). The compressors applied in GM-PTCs are commercial products, making it very difficult to improve exergy efficiency and keep costs low (Hitachi, 2023). In addition, there is still a 30% exergy loss in the distribution valves of the GM-PTC, according to the simulation conducted by Liu et al. (2017). Therefore, optimization of the distribution valves is one of the most essential research avenues for improving the cooling efficiency of GM-PTCs (Zhao et al., 2023; Karpenko et al., 2023).

The rotary valve and solenoid valve are two typical distribution valves for GM-PTCs. Since the empirical coefficients of each distribution valve need to be calibrated via experiments, the effect of the structure and timing of these two valves on the cooling performance of GM-PTCs is studied mostly by experimental investigations, as summarized below.

Many studies have been conducted to investigate the effect of the structure of distribution valves on GM-PTCs. Thummes et al. (1995) coupled a GM-PTC with a rotary valve and a series of needle valves to change the pressure-wave form into the cold head and investigated the effect of the wave forms on cooling performance. The results showed that a pressure-wave form with long dwell times at peak pressures resulted in a lower cooling temperature of 61 K compared to the 69 K obtained by a sine pressure-wave form. Wang et al. (2010) adopted three solenoid valves and a rotary valve as the distribution valves for a GM-PTC. When the number of solenoid valves increased from one to three, the hot-end pressure ratio rose from 1.71 to 1.94, achieving a lower no-load cooling temperature between 15.7 K and 14.4 K. Since the flow area of the solenoid valves was much smaller than that of the rotary valve, a higher hot-end pressure ratio of 2.19 was obtained when replacing the three solenoid valves with a rotary valve, achieving a lower no-load cooling temperature of 12.4 K.

With regard to the effect of the timing of distribution valves on cryocooler performance, Zhu et al. (1998) investigated the effect of the length of the waiting time (i.e., the period when the high-pressure valve and low-pressure valve are both closed) of solenoid valves on an orifice pulse-tube cryocooler by conducting numerical simulations and experiments. The numerical results showed that when the waiting time increased from 0 to 1/4 cycle, the no-load cooling temperature of the cryocooler decreased. Meanwhile, the experiments showed that when the waiting times were 1/360 cycle and 1/4 cycle, the no-load cooling temperature of the cryocooler was 45.1 K and 40.3 K, respectively. Qiu et al. (Qiu and Thummes, 2002) conducted an experimental investigation on the ratio of exhaust time to intake time in the range of 1.05 to 1.41 for a two-stage GM-PTC working at liquid helium temperatures. The optimum ratio depended on the input power of the compressor. When the input power of the compressor was low (1.7 kW), a smaller ratio (1.05) achieved a higher mass flow rate to the cold head and allowed the GM-PTC to reach a maximum cooling power of 202 mW at 4.2 K. When the input electric power of the compressor was high enough (6 kW), a larger ratio of 1.3 achieved a maximum cooling power of 500 mW at 4.2 K.

From the above literature survey, it is clear that most studies have concentrated on studying the feasibility of distribution valves by comparing the effects of different structures and valve timing on the performance of cryocoolers, but very few studies have focused on analyzing exergy loss in distribution valves. In this paper, we propose a test method for assessing the exergy loss, including the leakage loss and viscosity loss, in a rotary valve. With the help of this method, the distribution of leakage loss and viscosity loss in a rotary valve under different operating conditions can be obtained and analyzed, providing guidance to optimize the rotary valve by expanding its flow area by 1.5 times. To verify the method, we monitored the cooling temperature and power of a remote two-stage GM-PTC with original or optimized

rotary valves installed. The experimental results show that compared to the original rotary valve, the optimized rotary valve can improve cooling efficiency by 16.4%, with a cooling power of 0.78 W at 4.2 K.

## 2 Experimental test of exergy loss from a rotary valve

### 2.1 The tested rotary valve

The rotary valve, consisting of a suction valve and discharge valve, is a distribution valve that opens and closes periodically according to specific timing. As shown in Fig. 1, when the discharge valve is opened, the cold head is connected to the high-pressure gas source for pressurization (the suction period). When the suction valve is opened, the cold head is connected to the low-pressure gas source for depressurization (the discharge period). Normally, the suction and discharge valves are not open at the same time. When they are closed at the same time, it is called the quiet period.

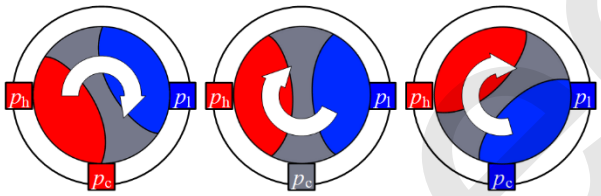


**Fig. 1 Schematic diagram of three steps for a working rotary valve: ((a). Discharge; (b). Quiet; (c). Suction.)**

Fig. 2 displays the basic structure of the tested rotary valve. Inside the rotary valve, there are a stator and a rotor. The valve spool has a symmetrical arrangement to eliminate the unbalanced differential pressure exerted on the rotor, and there is a dead space between the rotor and bushing; therefore, there is no friction in this rotary valve caused by the asymmetrical differential pressure. The stator has two orifices with an equivalent flow area of 16 mm2, to let the helium gas in and out of the cold head. The rotor has a groove connecting the high-pressure and low-pressure chambers of the compressor. By means of the matching relationship between the stator and rotor, the rotary valve changes the direct current flow (DC flow) into low-frequency alternating flow to drive the cold head of a GM-PTC. Thus, to improve the efficiency of GM-PTCs, clarification of exergy loss in the rotary valve is of great value.

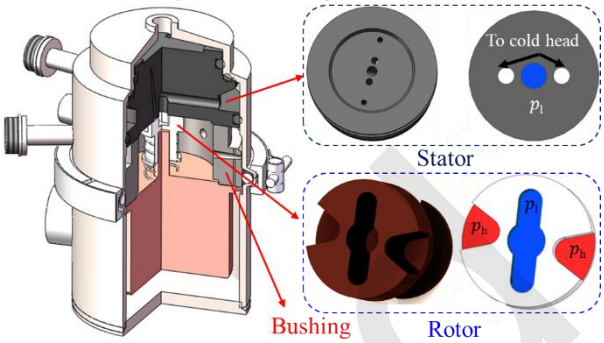


**Fig. 2 The basic structure of the tested rotary valve**

### 2.2 The tested rotary valve

The exergy loss generated in a rotary valve consists of leakage loss and viscosity loss (Zhao et al., 2023):

$$\dot{E} = \dot{E}_{\mathrm{leak}} + \dot{E}_{\mathrm{fric}}, \tag{1}$$

where $\dot{E}$ is the total exergy loss flow, $\dot{E}_{\mathrm{leak}}$ is the exergy loss caused by gas leakage, and $\dot{E}_{\mathrm{fric}}$ is the exergy loss caused by flow resistance.

#### 2.2.1 Leakage loss

The exergy loss caused by gas leakage between the high-pressure and low-pressure gas chambers is called leakage loss, and is expressed as follows:

$$\dot{E}_{\mathrm{leak}} = \frac{1}{\tau}\int_0^{\tau}(g_{\mathrm{h}} - g_{\mathrm{l}})\dot{m}_{\mathrm{leak}}\mathrm{d}t, \tag{2}$$

where subscript h and l denote the high pressure and low pressure corresponding to the suction and exhaust processes, respectively. $g$ is the specific Gibbs free energy, $\tau$ represents the period of one cycle, and $\dot{m}_{\mathrm{leak}}$ denotes the mass flow rate of the leaked gas.

The mass flow rate of gas leakage in Eq.(2), expressed by Eq.(3) (Zhu et al., 1997), is dependent on the pressure difference ( $\Delta p$ ) between the high-pressure ($p_{\mathrm{h}}$) and low-pressure ($p_{\mathrm{l}}$) gas sources and on the leakage coefficient:

$$\dot{m}_{\mathrm{leak}} = K\sqrt{\rho_{\mathrm{l}}(p_{\mathrm{h}} - p_{\mathrm{l}})} = K\sqrt{\rho_{\mathrm{l}}\Delta p}, \tag{3}$$

where $\rho_{\mathrm{l}}$ is the density of the gas at low pressure and

Uncorrected

$K$ is the leakage coefficient. The leakage coefficient is a characteristic factor of a rotary valve that changes with pressure differences and operating frequencies. It should be noted that the leakage mass flow rate $\dot{m}_{\mathrm{leak}}$ cannot be directly measured when the rotary valve is connected to a compressor and a cold head. Therefore, $\dot{m}_{\mathrm{leak}}$ should be calibrated by the leakage coefficient $K$ and pressure difference $\Delta p$ in the calculation of $\dot{E}_{\mathrm{leak}}$.

To calibrate the leakage coefficient of the rotary valve under various pressure differences and operating conditions, we established a test platform with the positions of measuring mass flow rate and pressure, as shown in Fig. 3.

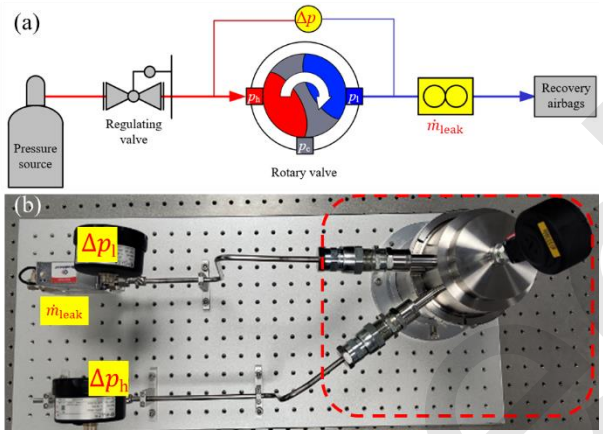


**Fig. 3 Rotary-valve leakage-coefficient test platform: (a). Schematic diagram; (b). Photograph**

In the experimental system, two Keller LEO3 digital pressure gauges were utilized to test the pressure difference in Eq.(3), and a Bronkhorst F-201A flowmeter was applied to monitor the leakage mass flow rate $\dot{m}_{\mathrm{leak}}$. Detailed information on the measured parameters is listed in Table 1.

**Table 1 Information on the parameters measured on the rotary-valve leakage-coefficient test platform**

| Parameter | Symbol | Sensor | Range | Uncertainty |
|---|---|---|---|---|
| Differential pressure | $\Delta p_{\mathrm{h}}$, $\Delta p_{\mathrm{l}}$ | Keller LEO3 | 0~3.0 MPa | ±3 kPa |
| Flowmeter for leakage test | $\dot{m}_{\mathrm{leak}}$ | Bronkhorst F-201A | 0~80 L/min | ±0.08 L/min |

It should be noted that the measured pressure differences take into account the pressure drops along the way of valve measuring connectors. Compared to the large pressure difference inside the rotary valve, the pressure drops caused by the connectors can be neglected in the testing process. During testing of the leakage coefficients, the load side of the rotary valve was blocked by a self-sealing connector to force all the leaked gas to go through the flowmeter. When the rotary valve worked at a fixed operating frequency, the high-pressure gas was adjusted to the desired pressure with the regulating valve. The gas leaked from the high-pressure side of the rotary valve to the low-pressure side and then entered the recovery airbags.

In addition to recording the pressure differences and leakage mass flow rate with different frequencies, we monitored ambient temperatures to calculate the $\rho_{\mathrm{l}}$ in Eq.(3), with the help of the NIST database. After obtaining $\dot{m}_{\mathrm{leak}}$, $\Delta p$, and $\rho_{\mathrm{l}}$ at a fixed operating frequency, we calibrated the leakage coefficient $K$. Fig. 4a presents the changes in the leakage coefficient with pressure difference, with different operating frequencies in the range of 1.2 Hz to 2.4 Hz.

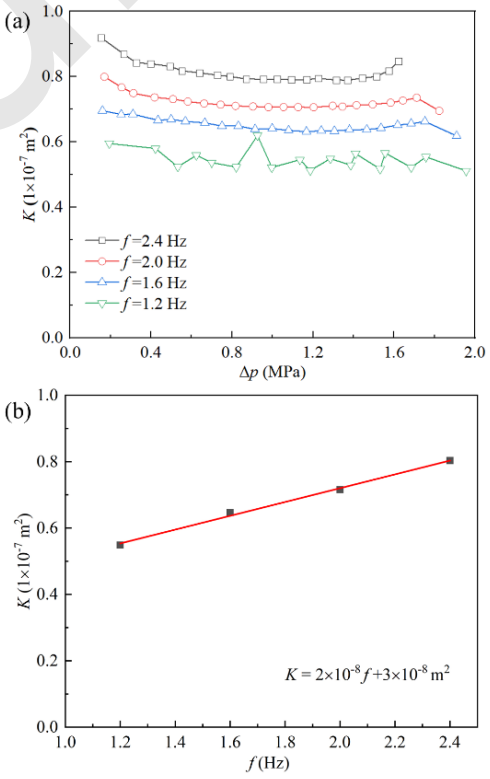


**Fig. 4 Experimental results of the leakage coefficient for the rotary valve: (a). relationship between leakage coefficient and pressure difference with different operating frequencies; (b). Linear fitting of leakage coefficient with operating frequency**

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When $\Delta p$ changed from 0.6 MPa to 1.6 MPa, the relative change of $K$ was less than 2%, indicating that $K$ was only related to the operating frequency. The averaged leakage coefficient, after calculating an integral average of $K$ from 0.6 MPa to 1.6 MPa at each point of operating frequency, is displayed in Fig. 4b. The analytical correlation between the leakage coefficient and operating frequency for the rotary valve can be obtained through the linear fitting.

### 2.2.2 Viscosity loss

Exergy losses caused by non-linear flow resistance of the nozzle in the discharge valve and suction valve are defined as follows:

$$\dot{E}_{\mathrm{fric,d}} = \frac{1}{\tau}\int_0^{\tau}(g_{\mathrm{h}} - g_{\mathrm{c}})\dot{m}\mathrm{d}t \ \ (\dot{m} > 0), \tag{4}$$

$$\dot{E}_{\mathrm{fric,s}} = \frac{1}{\tau}\int_0^{\tau}(g_{\mathrm{c}} - g_{\mathrm{l}})|\dot{m}|\mathrm{d}t \ \ (\dot{m} < 0), \tag{5}$$

where subscript c represents the load (cold head or reservoir), subscripts d and s are exergy loss from the discharge valve and suction valve, and $\dot{m}$ is the mass flow rate.

To assess the viscosity loss from the rotary valve under different operating conditions, we established the experimental platform shown in Fig. 5.

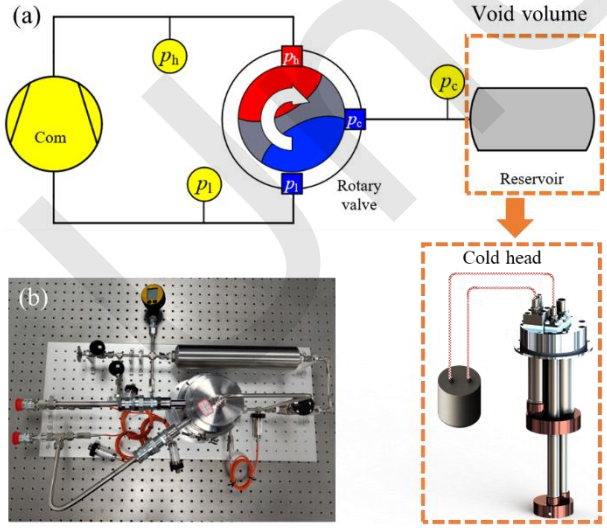


**Fig. 5 The experimental platform used to assess viscosity loss in the rotary valve:**
**(a). Schematic diagram; (b). Photograph**

The suction and discharge valves in the tested rotary valve were connected to the high-pressure and low-pressure chambers in a scroll compressor (S603DH) manufactured by Hitachi, with a steady-state input electrical power of 6~7 kW. The load connected to the rotary valve was two constant-volume gas reservoirs (0.5 L and 2 L) maintained at ambient temperature by water cooling. These reservoirs represented the total void volume of the cold head of the GM-PTC rather than the void volume of a specific component such as the pulse tube or reservoir. Table 2 lists the information on the parameters measured on the experimental platform, presented in Fig. 5.

**Table 2 Information on the parameters measured on the experimental platform shown in Fig. 5**

| Parameter | Symbol | Sensor | Range | Uncertainty |
|---|---|---|---|---|
| Electric power | - | DTSU5886 | 0~13.2 kW | ±0.1 kW |
| Pressure oscillation | $p_{\mathrm{h}}$, $p_{\mathrm{l}}$, $p_{\mathrm{c}}$ | Keller M5HB | 0~5.0 MPa | ±5 kPa |

Since the compressibility factor of helium at ambient temperature (about 300 K) is close to unity according to the NIST database, we next applied the ideal gas equation of state to describe the helium. It should be noted that the gas reservoir shown in Fig. 5 was utilized to measure the alternating mass flow rate through the rotary valve with the RC load approach (Swift, 2002). Normally, the temperature of the gas reservoir is assumed to be constant in the RC load approach (Zhou and Matsubara, 1998). To keep the temperature of the gas reservoir as constant as possible during the compression and expansion process, the reservoir was placed in water which was controlled at 300 K by a thermostat with an uncertainty of 0.03 K. Assuming the isothermal process in the gas reservoir, the alternating mass flow rate can be expressed by:

$$\dot{m} = \frac{V_{\mathrm{res}}}{RT_0}\frac{\mathrm{d}p_{\mathrm{c}}}{\mathrm{d}t}, \tag{6}$$

where $p_{\mathrm{c}}$ is the pressure in the reservoir, $V_{\mathrm{res}}$ is the volume of the reservoir, $R$ is the gas constant of helium, and $T_0$ is the ambient temperature. In addition to accurately measuring the alternating mass flow rate,

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monitoring the high-pressure and low-pressure is essential in calculating the Gibbs free energy in Eqs. (4)-(5). Therefore, we used three Keller M5HB high-frequency pressure transducers (with an uncertainty of ±5 kPa), to monitor the high pressure ($p_h$) and low pressure ($p_l$), as well as the transient pressure in the gas reservoir ($p_c$) (see Fig. 4).

Combined with Eqs. (4)-(6), the exergy loss of viscosity can be expressed as follows:

$$\dot{E}_{\mathrm{fric,d}} = \frac{V_{\mathrm{res}}}{RT_0}\int_{\min p_c}^{\max p_c}(g_h - g_c)\mathrm{d}p_c, \quad (7)$$

$$\dot{E}_{\mathrm{fric,s}} = \frac{V_{\mathrm{res}}}{RT_0}\int_{\min p_c}^{\max p_c}(g_c - g_l)\mathrm{d}p_c, \quad (8)$$

where $\min p_c$ and $\max p_c$ refer to the minimum and maximum pressure in the reservoir ($p_c$). Generally, the $\min p_c$ and $\max p_c$ are very close to the suction and discharge pressure, respectively.

The Gibbs free energy of helium gas is a function of state with temperature and pressure. Fig. 6 shows the relationship between the specific Gibbs free energy of helium and its pressure at $T_0$ (300 K), where the specific Gibbs free energy of helium is a monotonic and convex function of pressure. When the mass flow rate is fixed, the Gibbs free energy changes more at a lower pressure, indicating that a higher suction pressure of the rotary valve helps to decrease the viscosity exergy loss, based on Eqs. (7)-(8).

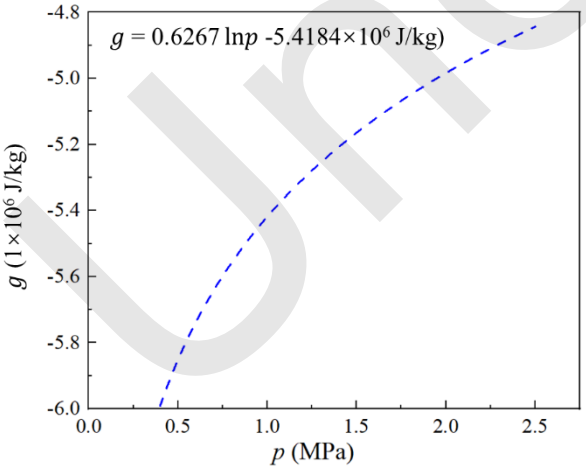


**Fig. 6 Relationship between specific Gibbs free energy of helium and pressure at 300 K**

Fig. 7 displays the ratio of $\dot{E}_{\mathrm{fric,d}}$ and $\dot{E}_{\mathrm{fric,s}}$ ($\Psi$) with different suction and discharge pressures. For a fixed discharge pressure of 2 MPa, as the suction pressure increases from 0.4 MPa to 1.8 MPa, $\Psi$ increases from 0.6 to 0.95, leading to a lower $\dot{E}_{\mathrm{fric,s}}$ at a higher suction pressure. Therefore, increasing suction pressure can reduce exergy loss at the suction valve.

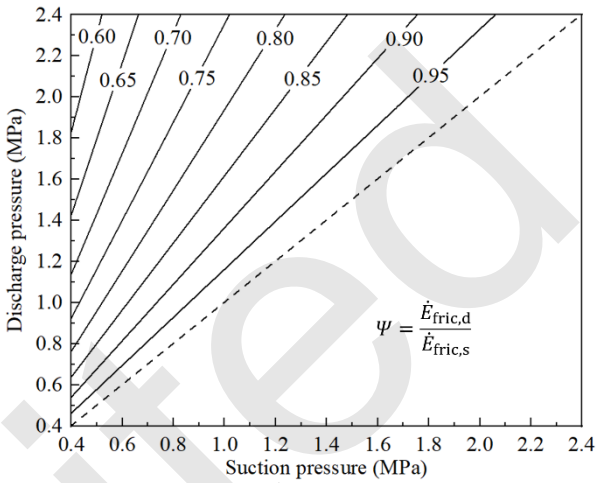


**Fig. 7 Ratios of $\dot{E}_{\mathrm{fric,d}}$ and $\dot{E}_{\mathrm{fric,s}}$ with different suction and discharge pressures**

## 2.3 Assessment of exergy loss generated in the rotary valve

After testing the leakage loss and viscosity loss in the rotary valve, we tested the distribution of $\dot{E}_{\mathrm{fric,d}}$, $\dot{E}_{\mathrm{fric,s}}$, and $\dot{E}_{\mathrm{leak}}$ generated in the rotary valve when two reservoirs with volumes of 0.5 L and 2 L were installed (see Fig. 8). The mean pressure ($p_m$) during the test was constant at 1.7 MPa, and the operating frequencies were between 0.8 Hz and 2.6 Hz.

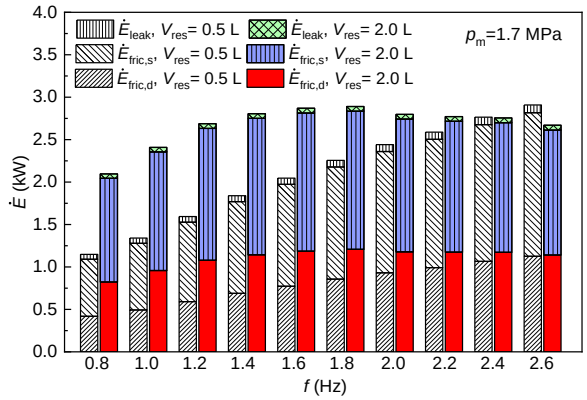


**Fig. 8 The tested distribution of $\dot{E}_{\mathrm{fric,d}}$, $\dot{E}_{\mathrm{fric,s}}$, and $\dot{E}_{\mathrm{leak}}$ in the rotary valve, using two reservoirs with volumes of 0.5 L and 2 L**

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As indicated in Fig. 8, the leakage loss was about 2.5% of the viscosity loss under all testing conditions, proving viscosity loss to be dominant in exergy loss at the rotary valve. Applying Eqs.(7)-(8) revealed that viscosity loss can be affected by the volume of the gas reservoir and the Gibbs free energy with different pressures. When the rotary valve is connected to a 0.5 L reservoir, the mass flow rate is small and increases linearly with the operating frequency, leading to a positive correlation of total exergy loss with operating frequency. When the rotary valve is connected to a 2 L reservoir, the mass flow rate is high and is limited by the flow area of the suction and discharge valves in the rotary valve. Thus, exergy loss does not consistently increase with operating frequency. Compared to $\dot{E}_{\mathrm{fric,d}}$, $\dot{E}_{\mathrm{fric,s}}$ takes up a slightly larger portion of the exergy loss. Therefore, minimizing the flow resistance of the suction valve and discharge valve is essential to reduce exergy loss.

## 3 Structural optimization to reduce friction loss in rotary valves

Based on assessments of the exergy loss at the rotary valve and work done by (Xu et al., 2019; Panda et al., 2019), we recommend reducing the flow resistance of the suction valve and discharge valve to optimize the rotary valve. The original rotary valve (RV1) has a flow area of 40 mm$^2$. With the help of Sage, the changes in the cooling power of a two-stage GM-PTC with different rotary-valve flow areas are presented in Fig. 9. More information about the two-stage GM-PTC can be found in section 4.

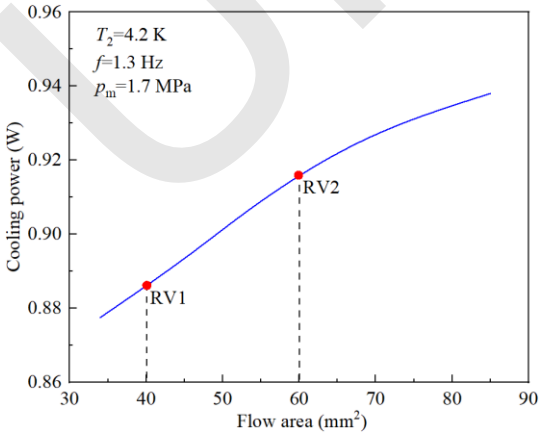


**Fig. 9 Relationship between the flow area of the rotary valve and the cooling power of the GM-PTC at 4.2 K, as plotted by Sage**

The figure shows that the cooling power at 4.2 K increases with the flow area of the rotary valve, indicating better performance with a larger flow area. However, because it is limited by the motor torque and three-dimensional structure of the rotary valve, the flow area can only be expanded to 60 mm$^2$. Hence, we fabricated and tested an optimized rotary valve (RV2) that had a larger equivalent flow area than the original one, to validate the exergy loss assessment described in Section 2. RV1 is shown in Fig. 2 (RV1), and RV2 in Fig. 10. Compared to RV1, the equivalent flow area of RV2 was about 1.5 times larger, with the same valve timing

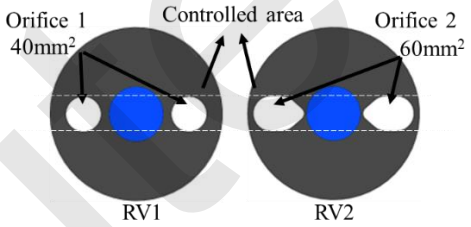


**Fig.10 Structure of the stators in both rotary valves (RV1 and RV2)**

Based on the two experimental platforms illustrated in Section 2, we tested the suction pressure, discharge pressure, and mass flow rate of RV1 and RV2 with different mean pressures, when the rotary valve was connected to a gas reservoir (0.5 L or 2 L). Fig. 11 shows the suction pressure and discharge pressure results for RV1 and RV2. The dashed and solid lines in Fig. 11 present the suction and discharge pressures on both sides of the RV1 and RV2, respectively. It is worth noting in Fig. 11 that for a fixed frequency and mean pressure, RV2 (solid line) obtained a lower high pressure ($p_{\mathrm{h}}$) and a higher low pressure ($p_{\mathrm{l}}$) than RV1 (dashed line). As a result, the pressure difference produced by RV2 was smaller than that produced by RV1, proving that RV2 had a lower flow resistance.

Fig. 12 shows the test results for the mass flow rates and leakage flow rates of RV1 and RV2. One can see that the leakage mass flow rates of the two rotary valves are very close. The maximum leakage mass flow rate was about 3% of the mass flow rate for both rotary valves. Thus, leakage loss can be neglected when assessing exergy loss in RV1 and RV2. When these values are used in Eq.(6), it becomes apparent that both the pressure-change rate and volume of the gas reservoir are linked to the tested mass flow rate.

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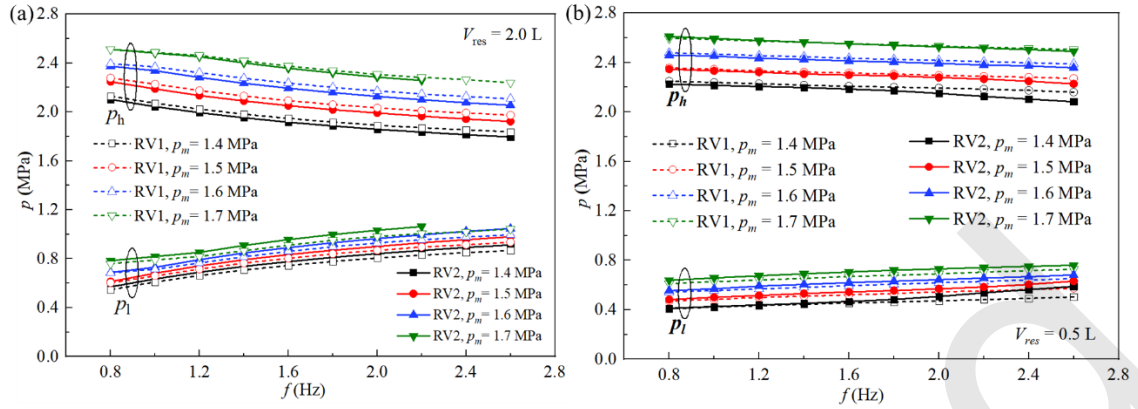


**Fig. 11 Tested suction and discharge pressures of RV1 and RV2 with different mean pressures and frequencies and two gas reservoir capacities: (a). 0.5 L reservoir; (b). 2 L reservoir. Dashed lines: RV1; solid lines: RV2**

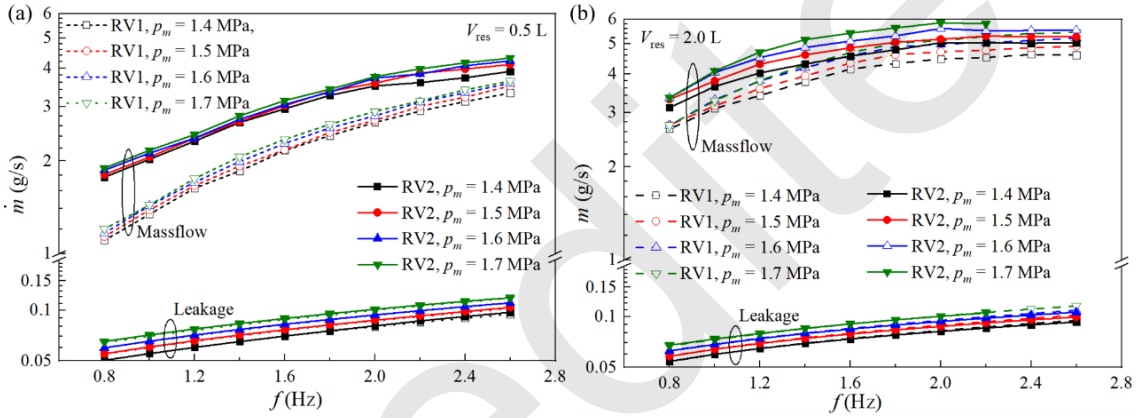


**Fig. 12 Tested mass flow rate and leakage mass flow rate for RV1 and RV2 with two reservoirs:(a). 0.5 L reservoir; (b). 2 L reservoir. Dashed lines: RV1; solid lines: RV2**

As shown in Fig. 12a, when the 0.5 L reservoir was connected to the rotary valves, the mass flow rate increased by up to 58% (from 1.2 g/s to 1.9 g/s at 0.8 Hz) after we replaced RV1 with RV2. For the 2 L reservoir (see Fig. 12b) the mass flow rate increased by up to 26% (from 2.7 g/s to 3.4 g/s at 0.8 Hz) after RV2 was swapped in. Therefore, the mass flow rate increased for both gas reservoirs with RV2. The assessment of exergy loss in RV1 and RV2 is displayed in Fig. 13.

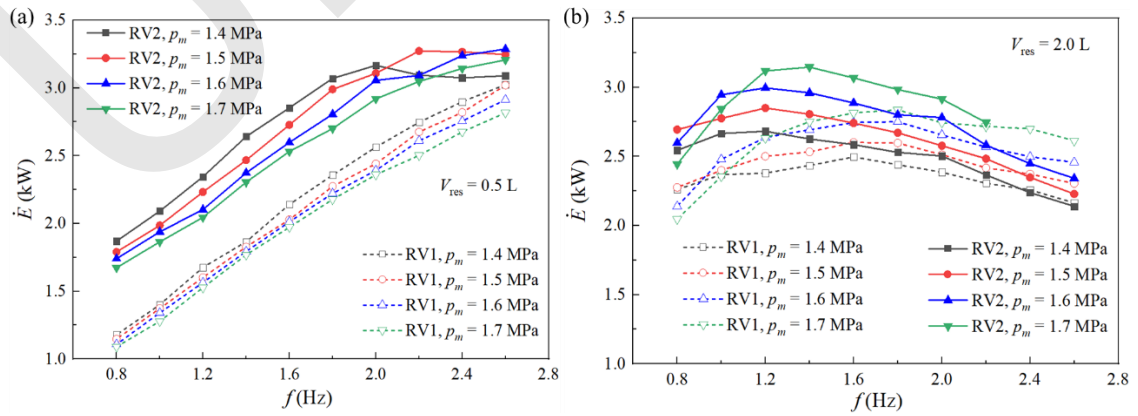


**Fig. 13 Exergy loss in RV1 and RV2 with two gas reservoirs:**
**(a). 0.5 L reservoir; (b). 2 L reservoir. Dashed lines: RV1; solid lines: RV2**

It should be noted that during the RC load testing we regarded the gas reservoir as a pure capacitive

component without consuming work (Swift, 2002). Therefore, the exergy transferred by the compressor was dissipated in RV1 and RV2. Fig. 13a shows that RV2 dissipated up to 1.5 times more exergy than RV1 with the 0.5 L reservoir.

When the 2 L reservoir was connected to the rotary valves, the maximum exergy dissipated by RV2 was up to 1.2 times that dissipated by RV1, as shown in Fig. 13b. Therefore, the flow resistance in the rotary valve can be reduced by increasing the flow area, and thus allowing more exergy to be delivered efficiently to the cold head by the rotary valve.

## 4 Experimental verification of exergy analysis upon a whole GM-PTC

Since we carried out the exergy-loss assessment of the rotary valves at ambient temperatures, we also verified the feasibility of the structure optimization of RV2 in a complete GM-PTC. To validate the improvement brought by RV2 to the cooling performance of GM-PTCs, we connected a remote two-stage GM-PTC to the test platform shown in Fig. 5. The main geometric parameters of the GM-PTC are listed in Table 3, and a schematic diagram of the testing system is shown in Fig. 14.

**Table 3 Geometrical parameters of the two-stage pulse-tube cryocooler**

| Components | Dimensions |
|---|---|
| 1st Regenerator | Φ60.2×200.5 mm,300＃SS304 |
| 1st Pulse tube | Φ46.0×172.0 mm |
| 2nd Regenerator | Φ35.0×185.0mm, 50% Pb+50% HoCu2 |
| 2nd Pulse tube | Φ27.4×379.0 mm |
| 1st /2nd Reservoir | 3.0 L |

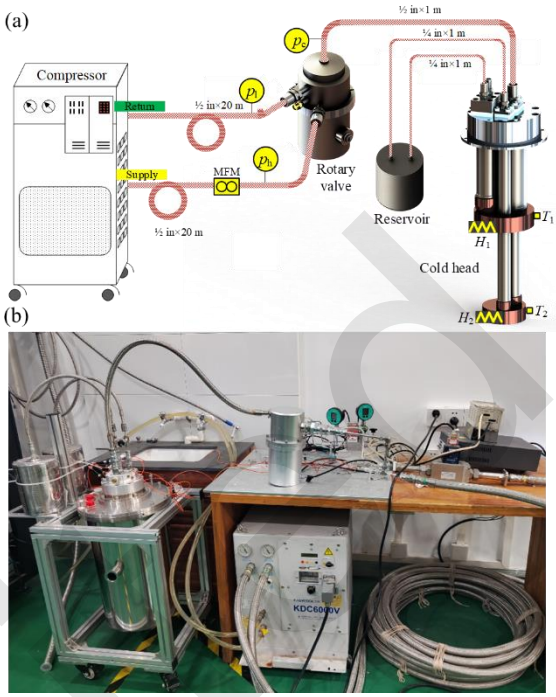


**Fig. 14 Test platform of a two-stage GM-PTC: (a). schematic diagram; (b). Photograph**

The testing system monitored the compressor suction and discharge pressures (namely $p_h$ and $p_l$), and the pressure of the load at the inlet of the cold head ($p_c$), with the same pressure gauges shown in Fig. 5. To calculate the output exergy of the compressor, we used a flowmeter to monitor its DC mass flow rate (Liu et al., 2017; Zhao et al., 2023). The performance of the GM-PTC was reflected by the cooling temperature and cooling power. Two Cernox thermometers and a Lakeshore 224 were used to monitor the cooling temperatures of the two cold heads of the GM-PTC. Two heating resistors were attached to the first- and second-stage cold heat exchangers to obtain the cooling power with a Keithley 2700 multimeter. Table 4 summarizes the information on the parameters measured on the test platform for a two-stage GM-PTC.

**Table 4 Information on the parameters measured on the test platform of a two-stage GM-PTC**

| Parameter | Symbol | Sensor | Range | Uncertainty |
|---|---|---|---|---|
| Electric power | - | DTSU5886 | 0~13.2 kW | ±0.1 kW |
| Pressure oscillation | $p_h$, $p_l$, $p_c$ | Keller M5HB | 0~5.0 MPa | ±5 kPa |
| Mass flow of compressor | - | TELEDYNE HFM305 | 0~8 g/s | ±0.08 g/s |
| Cooling temperature | $T_2$ | Cernox CX-1050-CU-HT-1.4L | 1.4~325 K | ±7.1 mK at 4.2 K |
| Cooling power | - | Keithley 2700 with a 50 Ω resistance | 0~18 W | ±5.5 mW |

We tested the cooling performance of the two-stage GM-PTC connected to RV1 or RV2 rotary valves by using the same S603DH compressor. The mean pressure in the GM-PTC was 1.7 MPa, the operating frequency of RV1 and RV2 was fixed at 1.3 Hz, and the input electrical power of the compressor was maintained at 6 kW. Fig. 15a shows the temperature changes in the second-stage cold heat exchanger during the cool-down process of the GM-PTC when connected to a RV1 or RV2 rotary valve.

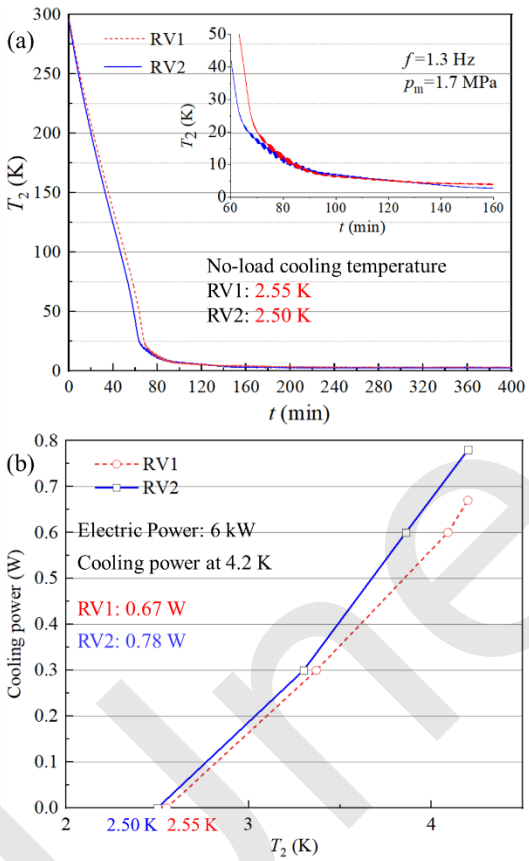


**Fig. 15 Comparison of the cooling performance of a two-stage GM-PTC with a RV1 or RV2 rotary valve installed: (a). Cool-down curve; (b). Cooling power**

During the entire cool-down period, the second-stage cold heat exchanger was cooled to 2.55 K and 2.50 K with RV1 and RV2 installed, respectively. The cooling speed of the GM-PTC from ambient temperature to about 20 K was 7.2% faster with RV2 (4.18 K/min) than with RV1 (3.9 K/min). The cool-down period for the GM-PTC with RV1 from 20 K to 4.2 K is very close to the period for RV2.

Fig. 15b shows that the cooling power changes with temperature when using RV1 or RV2 with the GM-PTC. The cooling power of the GM-PTC at 4.2 K was 0.67 W and 0.78 W with RV1 and RV2, respectively, indicating an improvement of 16.2% in cooling efficiency when using RV2 instead of RV1.

Fig. 16 shows the suction pressure, discharge pressure, and DC mass flow rate of the compressor with different cooling temperatures, using either RV1 or RV2. It can be seen from the figure that installation of the RV2 rotary valve significantly reduced the difference between suction and discharge pressures and raised the suction pressure, leading to an increased DC mass flow rate of the compressor.

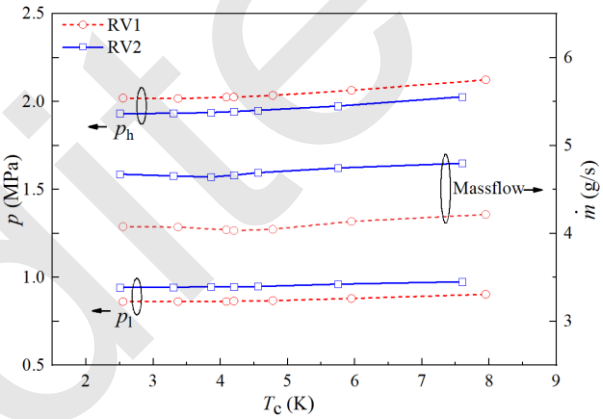


**Fig. 16 Mass flow rate versus suction and discharge pressures with different cooling temperatures**

To reveal the alternating flow characteristics in RV1 and RV2, the pressure variations in the high-pressure chamber ( $p_h$ ) and low-pressure chamber ($p_l$) and at the cold head ($p_c$) in one cycle, with a cooling temperature of 4.2 K, are displayed in Fig. 17. Compared to RV1, there was a smaller difference between the $p_h$ and $\max p_c$, as well as between $\min p_c$ and $p_l$, during one cycle when RV2 was used. The reduction of the pressure difference indicates lower flow resistance in the rotary valve, which corresponds to higher efficiency (by 16%) of the GM-PTC. As a result, increasing the flow area helps to lower the exergy loss generated in the rotary valve and produces better performance for the GM-PTC at liquid helium temperatures. However, it was difficult to assess the exergy loss test in the cold head since the amplitude and phase relationship between the alternating mass flow rate and pressure were hard to monitor without the addition of numerical simulation.

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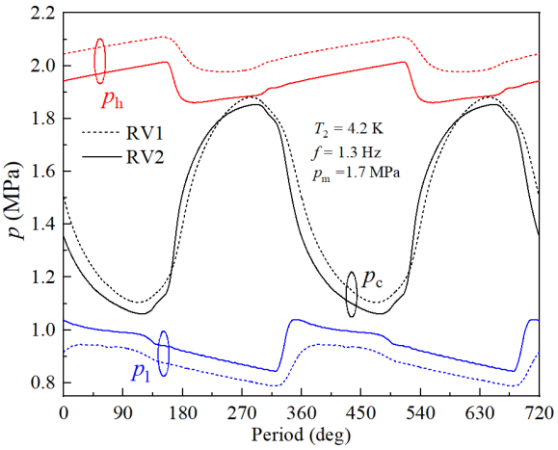


**Fig. 17 The pressure variation in both rotary valves (RV1 and RV2) at a cooling temperature of 4.2 K**

## 5 Conclusions

To improve the cooling efficiency of GM-PTCs at 4.2 K, we conducted structural optimization of the rotary valve by assessing the exergy loss, and verified the optimization by experimental testing. We used an experimental test method to assess the exergy loss (consisting of leakage loss and viscosity loss) in the rotary valve and conducted it on two experimental platforms. The results show that the leakage loss is less than 2.5% of the total exergy loss and the viscosity loss accounts for more than 97.5% of the total exergy loss, indicating that reduction of flow resistance in the rotary valve is essential to lower exergy loss. With the help of the exergy-loss assessment results, we chose an optimized rotary valve (RV2) with 1.5 times the flow area of the original one (RV1) to analyze the exergy loss. The maximum exergy transferred from the compressor to the load side by RV2 is 1.5 times that transferred by RV1. Finally, we conducted a test of the cooling performance of a remote two-stage GM-PTC which had either the optimized rotary valve (RV2) or the original one (RV1) installed, to further validate the exergy-loss assessment of the rotary valves. The results show that when the input power of the compressor is 6 kW, the GM-PTC provides a cooling power of 0.78 W at 4.2 K when replacing RV1 with RV2, which leads to an improvement of 16.4% in the cooling efficiency of the GM-PTC.

## Acknowledgments

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## Author contributions

Qinyu ZHAO and Ruize LI designed the research. Yanrui ZHANG and Jun CHENG processed the corresponding data. Qinyu ZHAO and Haoren WANG wrote the first draft of the manuscript. Qinyu ZHAO, Haoren WANG and Bo WANG helped to organize the manuscript. Bo WANG, Hua ZHANG, and Zhihua GAN revised and edited the final version.

## Conflict of interest

Qinyu ZHAO, Jun CHENG, Yanrui ZHANG, Haoren WANG, Bo WANG, Ruize LI, Hua ZHANG, and Zhihua GAN, declare that they have no conflict of interest.

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## 中文概要

**题　目：液氦温区 G-M 脉管制冷机的旋转阀结构优化**

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**目　的：**液氦温区 G-M 脉管制冷机以其长寿命和低振动等优势逐渐成为量子科技和低温物理等尖端科学领域的理想制冷源，但现有旋转阀部件中的不可逆损失严重限制了其在液氦温区的制冷效率。本文旨在提升液氦温区 G-M 脉管制冷机的制冷性能，针对 G-M 脉管制冷机的关键部件旋转阀，开发了以泄露损失和流阻损失为指标的㶲损评估方法，指导了旋转阀的 结构优化并基于实验进行验证，对液氦温区高效大冷量的 G-M 脉管制冷机的开发具有重要意义。

**创新点：**1. 通过交变流动旋转阀㶲损失方程，构建以泄露损失和流阻损失为指标的 G-M 脉管制冷机旋转

阀㶲损评估方法；2.搭建实验测试平台，形成对旋转阀烟损失的定量测算。3.完整开展从㶲损失理论分析到旋转阀关键部件优化再到实际制冷机性能提升的闭环研究路径。

**方　法：** 1. 通过理论分析，推导获得旋转阀内㶲损失特性与操作参数的定量关系(公式(2)-(6))；2.分别搭建了旋转阀泄漏流量测试平台(图 3)和㶲损失评估平台(图 5)，根据实验平台定量分析了不同运行频率、充气压力和负载对旋转阀㶲损失分布的影响情况；3. 通过 Sage 软件计算了旋转阀流阻损失对脉管制冷机制冷性能的影响情况(图 9)，结合实际工艺扩大了现有旋转阀流通面积；4.通过旋转阀㶲损失分析和脉管制冷机性能测试的方法验证了所述理论。

**结　论：** 1. 在液氦温区 G-M 脉管制冷机内的旋转阀中，流阻损失占 97.5%以上，泄露损失只占 2.5%，因此流阻损失是主要的㶲损失；2. 旋转阀流通面积扩大 1.5 倍后其内流阻显著降低，压缩机输出㶲可提高 1.2~1.5 倍；3. 使用低流阻旋转阀驱动同一台制冷机冷头时，脉管制冷机可获得 0.78 W@4.2 K 制冷量，制冷性能提高约 16.2%。基于实验测量与理论计算对精密冲压过程中工艺参数需要根据不同的环境变量进行调节。

**关键词：** 旋转阀；㶲分析；液氦温区；G-M 脉管制冷机；高效制冷

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