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A pulse tube cryocooler with a cold reservoir

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ABSTRACT

Phase difference between pressure wave and mass flow is decisive to the cooling capacity of regenerative cryocoolers. Unlike the direct phase shifting using a piston or displacer in conventional Stirling or GM cryocoolers, the pulse tube cyocooler (PTC) indirectly adjusts the cold phase due to the absence of moving parts at the cold end. The present paper proposed and validated theoretically and experimentally a novel configuration of PTC, termed cold reservoir PTC, in which a reservoir together with an adjustable orifice is connected to the cold end of the pulse tube. The impedance from the additional orifice to the cold end helps to increase the mass flow in phase with the pressure wave at the cold end. Theoretical analyses with the linear model for the orifice and double-inlet PTCs indicate that the cooling performance can be improved by introducing the cold reservoir. The preliminary experiments with a home-made single-stage GM PTC further validated the results on the premise of minor opening of the cold-end orifice.

1. Introduction

The phase difference between the pressure wave and mass flow at the cold end plays a decisive role in the cooling performance for the regenerative cryocoolers [11]. In Stirling or GM cryocoolers, the phase difference can be compulsively realized through controlling the relative movement between a displacer and compression piston. However, the phase shifting in a pulse tube cryocooler is indirect due to no moving part at the cold end, and it is adjusted by the phase shifters generally located at the hot end, through the movement of an imaginary gas piston, which is the fluid that never leaves the pulse tube during a cycle. A reservoir was added to the hot end of the pulse tube by Mikulin et al. to introduce the in-phase flow to the pressure at the cold end [6]. The added orifice significantly increased the PV power for an ideal thermodynamic process [12], while the pressure ratio decreased and the “gas piston” stroke got smaller. To effectively recover the dissipated power through the orifice, a moving plug was introduced at the hot end [7]. Although the plug can be controlled to produce an optimal phase difference, the merit of the simplicity of the PTC was decreased with the additional moving part. For a double-inlet type PTC [13], since the magnitude of the in-phase flow through the bypass valve was restricted by the phase between the hot and cold ends of pulse tube, the phase shifting capability was also limited.

The density of gas increases as temperature drops at the cold end, the indirect phase shifting capability at the hot end becomes limited. To compensate for decrease of the pressure ratio and “gas piston” stroke within the hot-end phase shifters, several configurations also have been developed to directly adjust the phase at low temperature. A capillary tube was added between the middle of the regenerator and pulse tube, termed multi-bypass type, to increase the pressure amplitude inside the pulse tube [5]. However, this configuration introduced additional DC flow, which may reduce the cooling performance. A separate three-stage Stirling PTC with a cold inertance tube reached a no-load temperature of 4.97 K with He-4 [9,10], while, since the density of gas increases while viscosity decreases as the temperature gets lower, a cold inertance tube, which is normally located at room temperature, lowers the sound speed of gas and increases the gas inertia, which is unfavorable to reach a larger phase shift [8,4].

To demonstrate the possibility of direct phase shifting in cold end for a PTC, this study proposed a new configuration in which a reservoir was added through an orifice directly to the cold end of pulse tube. The combination of a reservoir and an orifice can bring the in-phase component of mass flow relative to the pressure wave, and reach a large phase shift directly at the cold end of PTC [14]. The linear model analysis was applied for both the orifice PTC and the double-inlet PTC with a cold reservoir. The dependence of cooling performance on the cold orifice opening for both PTCs was presented. Preliminary experimental studies with a single stage GM PTC also confirmed the performance improvement over the conventional PTCs.

2. Theoretical model

2.1. Orifice PTC

The simplified model of analysis for the orifice PTC is shown in Fig. 1. A cold reservoir through a needle valve is connected to the

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Nomenclature

c	flow coefficient
C	flow conductance (MPa s m^{-1})
D	double-inlet valve
DC	flow conductance ratio of the bypass valve to regenerator
f	frequency (Hz)
F	dimensionless frequency
$\dot{H}$	enthalpy flow (W)
$\dot{m}$	mass flow rate (kg s^{-1})
OC	flow conductance ratio of orifice to regenerator
OV	volumetric ratio of reservoir to pulse tube
O_1	hot-end orifice
O_2	cold-end orifice
p	pressure (MPa)
p_0	average pressure (MPa)
$\dot{Q}_c$	cooling capacity (W)
$\dot{Q}_0$	dimensionless cooling capacity
$\dot{Q}_d$	compression power (W)
T	temperature (K)
V	volume (m^3)
$\dot{V}pt$	volumetric flow rate at the cold end of pulse tube (m^3 s^{-1})
ρ	density (kg m^{-3})
δ	phase difference between pressure and mass flow
κ	polytropic constant
π	pressure amplitude ratio of reservoir to pulse tube

Subscripts

b	bypass valve
c	cold end of PTC
cr	cold reservoir
h	hot end of PTC
hr	hot reservoir
p	pulse tube
r	regenerator

cold end of a conventional orifice PTC. The temperatures at the hot end and cold end of regenerator are T_h and T_c, respectively. The assumptions made in this theoretical analysis are

(1) Ideal gas and one-dimensional flow.
(2) The temporal pressure in the pulse tube, the hot end of the regenerator, and the two reservoirs are sinusoidal with different phase.
(3) The amplitude of the pressure is far smaller than the time-averaged pressure.
(4) The volumetric flow rate through the regenerator, the bypass valve and the two orifices are proportional to the pressure difference across them.
(5) The void volume in the regenerator is assumed zero.
(6) The hot and cold reservoir and pulse tube are adiabatic.

From the assumption (2), the pressure p_p in pulse tube as well as p_{hr} in the hot reservoir and p_{cr} in the cold reservoir is:

$$p_p(t) = \Delta p_p \sin(\omega t) + p_0 \tag{1}$$

$$p_{hr}(t) = \Delta p_{hr} \sin(\omega t + \delta_h) + p_0 \tag{2}$$

$$p_{cr}(t) = \Delta p_{cr} \sin(\omega t + \delta_c) + p_0 \tag{3}$$

Here, p_0 is the time-averaged pressure, Δp and δ represents the amplitude of pressure and phasing difference, respectively. The subscript p means pulse tube, hr and cr mean hot reservoir and cold reservoir, respectively. From the assumption (4), the volumetric flow $\dot{V}_{hr}$ and $\dot{V}_{cr}$ through the hot and cold-end orifice can be expressed as:

$$\dot{V}_{hr} = C_h[p_p(t) - p_{hr}(t)] \tag{4}$$

$$\dot{V}_{cr} = C_c[p_p(t) - p_{cr}(t)] \tag{5}$$

C_h and C_c is the flow conductance across the corresponding valves.

According to the assumption (6), the hot reservoir is adiabatic, then, the variation of the pressure in the hot reservoir can be given as [2]:

$$dp_{hr}(t) = (\kappa p_0/V_{hr})dV_{hr}(t) \tag{6}$$

$dV_{hr}(t)$ means the net volumetric flow rate entering and leaving the reservoir, substitute Eqs. (1) and (4) into equation:

$$\pi_h \cos(\omega t + \delta_h) = (F \cdot OV_h)^{-1} OC_h[\sin(\omega t) - \pi_h \sin(\omega t + \delta_h)] \tag{7}$$

where $\pi h = \Delta P_{hr}/\Delta P_p$ and $OV_h = V_{hr}/V_p$ are the pressure amplitude and volumetric flow rate ratio of the hot reservoir to the pulse tube. $OC_h = C_h/C_r$ is the flow conductance ratio of the hot-end orifice to the regenerator, C_r means the flow conductance across the regenerator. The dimensionless frequency F is defined as $F = \omega V_p/\kappa P_0 C_r$.

Equating the coefficients of terms $\cos(\omega t)$ and $\sin(\omega t)$ on the two sides of equal sign of Eq. (7):

$$\tan \delta_h = -F \cdot (OV_h/OC_h) \tag{8}$$

$$\pi_h = OC_h / \sqrt{OC_h^2 + (F \cdot OV_h)^2} \tag{9}$$

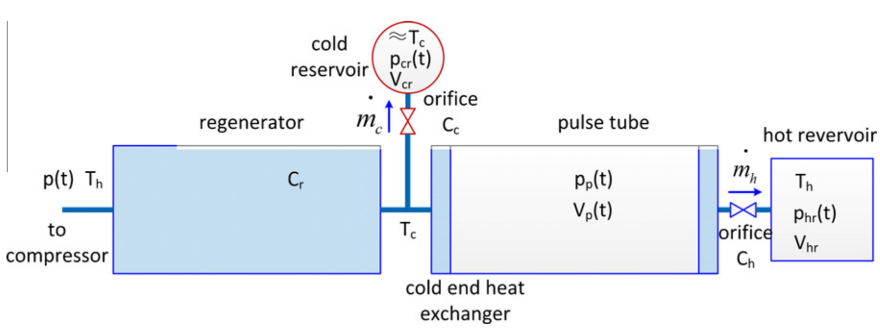

Fig. 1. Simplified model used for analyses of orifice PTC with cold reservoir.

In the same manner, the following relations for the cold reservoir are obtained:

$$\tan\delta_c = -F\cdot(OV_c/OC_c) \tag{10}$$

$$\pi_c = OC_c/\sqrt{OC_c^2 + (F\cdot OV_c)^2} \tag{11}$$

where $\pi_c = \Delta p_{cr}/\Delta p_p$ and $OV_c = V_{cr}/V_p$ and $OC_c = C_c/C_r$.

Substituting Eqs. (1)–(3) into Eqs. (4) and (5), one obtains the volumetric flow rates through the hot- and cold-end orifices as:

$$\dot V_{cr} = C_c[p_p(t) - p_{cr}(t)] = C_c\Delta p_p[\sin(\omega t) - OV_c\sin(\omega t + \delta_c)] \tag{12}$$

$$\dot V_{hr} = C_h[p_p(t) - p_{hr}(t)] = C_h\Delta p_p[\sin(\omega t) - OV_h\sin(\omega t + \delta_h)] \tag{13}$$

According to de Waele [1], the volumetric flow rate $\dot V_{pt}$ at the cold end of pulse tube can be expressed as the summation of the flow rates in pulse tube and two reservoirs:

$$\dot V_{pt} = (V_p/\kappa p_0)\cdot(dp_p(t)/dt) + \dot V_{hr} + \dot V_{cr} \tag{14}$$

Substituting Eqs. (12) and (13) into Eq. (14), leads to:

$$\begin{aligned}\dot V_{pt} = \Delta p_p C_r\{&F\cos(\omega t) + OC_h[(1-\pi_h^2)\sin(\omega t) - \pi_h\sin\delta_h\cos(\omega t)]\\ &+ OC_c[(1-\pi_c^2)\sin(\omega t) - \pi_c\sin\delta_c\cos(\omega t)]\}\end{aligned} \tag{15}$$

Eq. (15) can be considered as the sum of three parts, in which the first part on the right hand of Eq. (15) is the flow rate variance due to the movement of the gas piston in pulse tube, while the next two parts are the flow rate through the hot-end and the cold-end orifice, respectively. The first part comprises only the cosine component which is always orthogonal to the pressure wave, and does not contribute to the cooling power for PTC; while the other two both comprise the sinusoidal and cosine components, and only the sinusoidal component has the contributions to the cooling power. For the ideal thermodynamic processes, according to [2], the cooling power can be expressed as:

$$\dot Q_c = \langle\dot H\rangle = \langle p_p(t)\dot V_{pt}\rangle \tag{16}$$

Here, symbol $\langle\ \rangle$ means the time-averaged value in a cycle. Substituting Eqs. (1) and (14) and Carrying out the integral, leads to:

$$\dot Q_0 \equiv \frac{\langle\dot H\rangle}{C_r\Delta p_p^2/2} = OC_h(1-\pi_h^2) + OC_c(1-\pi_c^2) \tag{17}$$

According assumption (4), we have

$$\dot V_{pt} = C_r[p(t) - p_p(t)] \tag{18}$$

V_r is the volumetric flow rate through the regenerator, and $p(t)$ is the pressure at the hot end of the regenerator. Using Eqs. (1) and (15), $p(t)$ is calculated as:

$$\begin{aligned}p(t) = \Delta p_p\{&F\cos(\omega t) + OC_h[(1-\pi_h^2)\sin(\omega t) - \pi_h\sin\delta_h\\ &\times\cos(\omega t)] + OC_c[(1-\pi_c^2)\sin(\omega t) - \pi_c\sin\delta_c\\ &\times\cos(\omega t)]\} + \Delta p_p\sin(\omega t) + p_0\end{aligned} \tag{19}$$

Also, from the assumption (5), the volumetric flow rate through the regenerator is calculated as:

$$\dot V_r = (T_h/T_c)\cdot\dot V_{pt} \tag{20}$$

Combining Eqs. (19) and (20), the compressing power is obtained as:

$$\begin{aligned}\dot Q_d = \langle p(t)\dot V_r\rangle = \frac{C_r}{2}\cdot\frac{T_h}{T_c}\cdot\Delta p_p^2\{&[OC_h(1-\pi_h^2) + OC_c(1-\pi_h^2)]\\ &\times[1 + OC_h(1-\pi_h^2) + OC_c(1-\pi_h^2)]\\ &+ (F - OC_h\pi_h\sin\delta_h - OC_c\pi_c\sin\delta_c)^2\}\end{aligned} \tag{21}$$

So the COP of orifice PTC with a cold reservoir can be expressed as:

$$\begin{aligned}COP &= \langle\dot H\rangle/\dot Q_d\\ &= \frac{T_c}{T_h}\cdot\frac{OC_h(1-\pi_h^2) + OC_c(1-\pi_c^2)}{[OC_h(1-\pi_h^2) + OC_c(1-\pi_c^2)][1 + OC_h(1-\pi_h^2) + OC_c(1-\pi_c^2)] + (F - OC_h\pi_h\sin\delta_h - OC_c\pi_c\sin\delta_c)^2}\end{aligned} \tag{22}$$

For the ideal orifice type PTC without the cold reservoir, $OC_c = 0$ and $\pi_c = 0$. Neglecting the pressure fluctuation in the hot reservoir $\pi_h \to 0$, then Eq. (22) can be rewritten as:

$$COP = \frac{T_c}{T_h}\cdot\frac{OC_h}{OC_h(1+OC_h) + F^2} \tag{23}$$

which is the same to the theoretical calculations obtained by de Boer [2].

It was noticed here that the orifice at the cold end of the pulse tube may induce additional heat dissipation due to the irreversible flow through it. As the cold orifice is considered as adiabatic, the entropy production rate was estimated as $\dot S_o = \overline{R(\dot m_c/M_{He})\mathrm{In}P_p/P_o}$ [3], which was turned out to be much less than 1% of the cooling power. So the thermodynamic analyses are not further performed in the paper.

2.2. Double-inlet PTC

For the case of double-inlet type PTC, a bypass tube and a bypass valve connects the hot ends of the pulse tube and the regenerator, as shown in Fig. 2.

In the theoretical analyses, Eqs. (1)–(13) are still applicable. From the assumption (4), the mass flow rate across the bypass valve is expressed as:

$$\dot m_b = \rho_h C_b[p(t) - p_p(t)] \tag{24}$$

And C_b is the flow conductance of the bypass valve. The volumetric flow rate at the cold end of the pulse tube in double-inlet PTC with a cold reservoir is the summation of those through the hot-end orifice, bypass valve V_b, cold-end orifice and the movement of the gas piston, leads to:

$$\dot V'_{pt} = (V_p/\kappa p_0)(dp_p(t)/dt) + \dot V_{hr} + \dot V_{cr} - \dot V_b \tag{25}$$

Substituting Eqs. (12), (13), and (24) into Eq. (25), then the temporal pressure at the hot end of the regenerator can be calculated as:

$$\begin{aligned}p(t)' = \Delta p_p\{&F\cos(\omega t) + OC_h[(1-\pi_h^2)\sin(\omega t) - \pi_h\sin\delta_h\\ &\times\cos(\omega t)] + OC_c[(1-\pi_c^2)\sin(\omega t) - \pi_c\sin\delta_c\\ &\times\cos(\omega t)]\}/(1+DC) + \Delta p_p\sin(\omega t) + p_0\end{aligned} \tag{26}$$

and $DC = C_b/C_r$ is the flow conductance ratio of the bypass valve to the regenerator. Combining Eqs. (24)–(26), the volumetric flow rate at the cold end of the pulse tube can be obtained:

$$\dot V'_{pt} = \frac{\Delta p_p C_r\{F\cos(\omega t) + OC_h[(1-\pi_h^2)\sin(\omega t) - \pi_h\sin\delta_h\cos(\omega t)] + OC_c[(1-\pi_c^2)\sin(\omega t) - \pi_c\sin\delta_c\cos(\omega t)]\}}{1+DC} \tag{27}$$

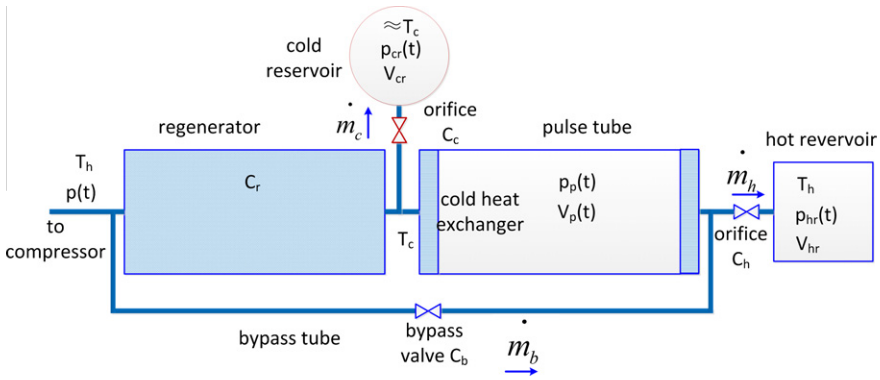

Fig. 2. Simplified model used for analysis of double-inlet PTC with cold reservoir.

Like the single orifice PTC, the dimensionless cooling power can be expressed from the equation:

$$\dot{Q}'_0 = \frac{\langle \dot{H} \rangle'}{C_r \Delta p_p^2/2} = [OC_h(1-\pi_h^2) + OC_c(1-\pi_c^2)]/(1+DC) \tag{28}$$

Similarly, the volumetric flow rate at the hot end of the regenerator can be calculated as:

$$\begin{aligned}\dot{V}'_r &= (T_h/T_c)\dot{V}'_{pt} + V_b \\ &= \frac{(T_h/T_c + DC)}{1+DC} \cdot \Delta p_p C_r \{F\cos(\omega t) + OC_h[(1-\pi_h^2)\sin(\omega t) \\ &\quad - \pi_h \sin\delta_h \cos(\omega t)] + OC_c[(1-\pi_c^2)\sin(\omega t) - \pi_c \sin\delta_c \\ &\quad \times \cos(\omega t)]\}\end{aligned} \tag{29}$$

Then the compression power for the double-inlet type PTC with a cold reservoir can be given as:

$$\begin{aligned}\dot{Q}'_d &= \langle p(t)' \dot{V}'_r \rangle \\ &= \frac{C_r \Delta p_p^2}{2(1+DC)^2} \cdot (T_h/T_c + DC) \cdot \{[OC_h(1-\pi_h^2) + OC_c(1 \\ &\quad - \pi_h^2)][1 + OC_h(1-\pi_h^2) + OC_c(1-\pi_h^2)] + (F - OC_h\pi_h \\ &\quad \times \sin\delta_h - OC_c\pi_c \sin\delta_c)^2\end{aligned} \tag{30}$$

The COP for the double-inlet PTC can be deduced by combining the Eqs. (28) and (30):

$$COP' = \langle \dot{H} \rangle' / \dot{Q}'_d = \frac{1}{(T_h/T_c + DC)} \cdot \frac{[OC_h(1-\pi_h^2) + OC_c(1-\pi_h^2)](1+DC)}{[OC_h(1-\pi_h^2) + OC_c(1-\pi_h^2)][1 + OC_h(1-\pi_h^2) + OC_c(1-\pi_h^2)] + (F - OC_h\pi_h \sin\delta_h - OC_c\pi_c \sin\delta_c)^2} \tag{31}$$

For the ideal double-inlet PTC without the cold reservoir, in which $OC_c = 0$ and $\pi_c = 0$, and neglecting the pressure fluctuation at the hot reservoir, Eq. (31) can be rewritten as:

$$COP' = \frac{OC_h(1+DC)}{(1 + DC + OC_h + F^2)(T_h/T_c + DC)} \tag{32}$$

which is the same to the theoretical calculations obtained by de Waele [1].

3. Computational results and analysis

The dimensionless cooling power $\dot{Q}_0$ from the Eq. (17) is plotted in Fig. 3 as a function of the volumetric ratio OV_c for the orifice PTC. Compared to the counterpart of a conventional orifice PTC ($OC_c = 0$), the cooling power of the PTC with a cold reservoir is

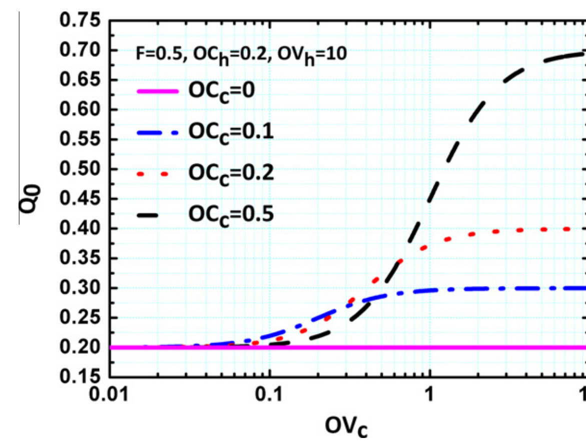

Fig. 3. Dependence of dimensionless cooling power on volumetric ratio of cold reservoir to the pulse tube for an orifice PTC with a cold reservoir.

larger, and it rises with the increase of OV_c. However, as OV_c increases, the increasing rate of $\dot{Q}_0$ decreases, and the cooling power reaches a maximum. Take T_c = 40 K, T_h = 300 K for an example, the dependence of COP on the volumetric ratio OV_c is displayed in Fig. 4. Since both the in-phase and the orthogonal component of the flow consume the input power, the COP depends on the combination of both parts. For a certain flow conductance ratio OC_c, when the OV_c is small, the orthogonal flow overwhelms the in-phase one, and the COP gets lower than the counterpart, however, as the OV_c rises, the in-phase flow becomes dominant and the COP increases and exceeds the basic case of a conventional PTC. Moreover, the rise of OV_c leads to a decrease of cold-end amplitude ratio of pressure fluctuation π_c, see Fig. 5. Take OC_c = 0.1 for an example, when OV_c < 0.2 the amplitude ratio π_c keeps above about 0.7, the high pressure fluctuation in an adiabatic system consumes compression power without contributing to the cooling capacity at the cold end of the pulse tube. Therefore, COP of the novel PTC with a cold reservoir increases cooling capacity with larger volumetric ratios OV_c, as shown in Figs. 3 and 4.

Like the orifice PTC, the dimensionless cooling power and COP for the double-inlet PTC are also plotted as the functions of the

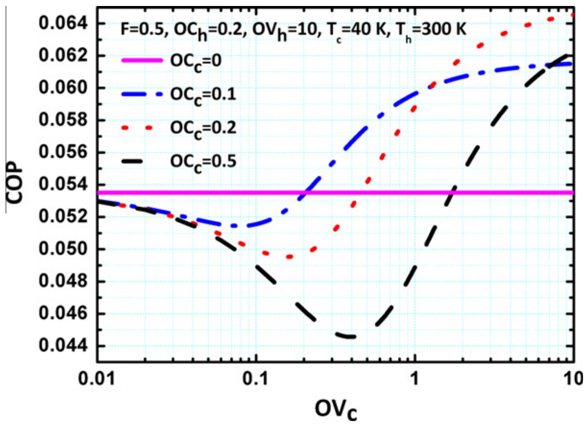

Fig. 4. Dependence of COP on volumetric ratio of cold reservoir to the pulse tube for an orifice PTC with a cold reservoir.

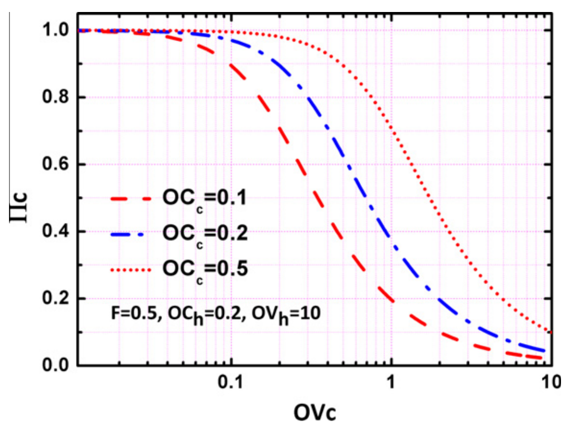

Fig. 5. Pressure amplitude ratio of the cold-end reservoir to the pulse tube.

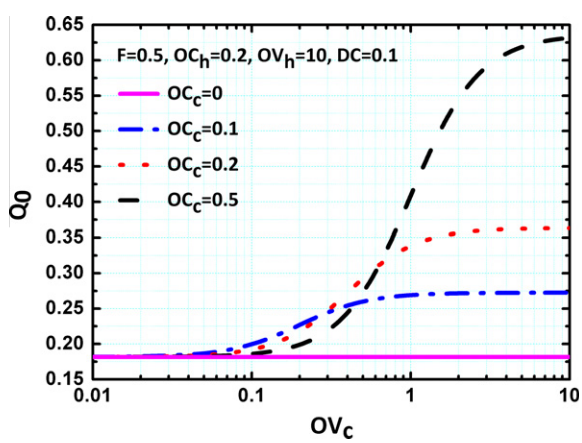

Fig. 6. Dependence of dimensionless cooling power on volumetric ratio of cold reservoir to the pulse tube for a double-inlet PTC with a cold reservoir.

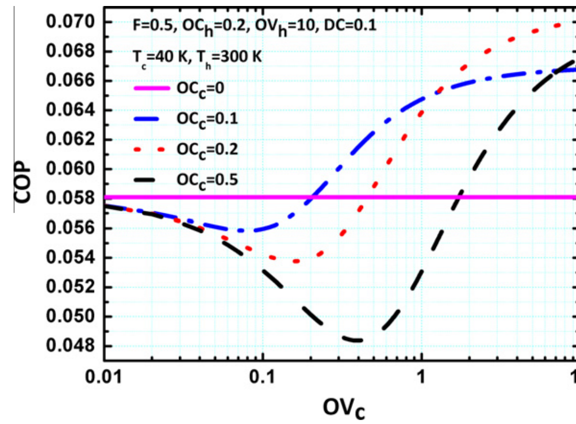

Fig. 7. Dependence of COP on volumetric ratio of cold reservoir to the pulse tube for a double-inlet PTC with a cold reservoir.

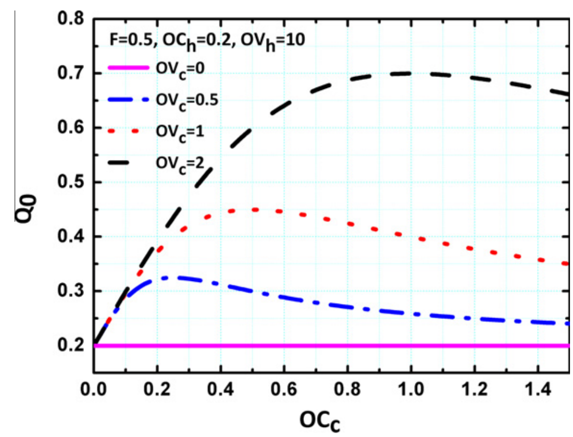

Fig. 8. Dependence of dimensionless cooling power on flow conductance ratio of cold-end orifice to the regenerator for an orifice PTC with a cold reservoir.

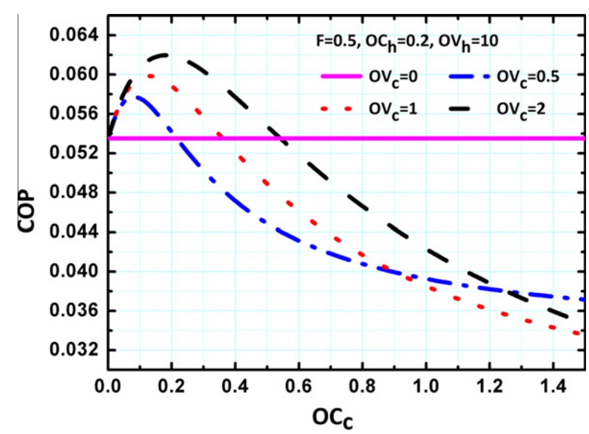

Fig. 9. Dependence of COP on flow conductance ratio of cold-end orifice to the regenerator for an orifice PTC with a cold reservoir.

volumetric ratio OV_c in Figs. 6 and 7. For the given conditions as $F = 0.5$, $OC_c = 0.2$, $OV_h = 10$, $DC = 0.1$ and large OV_c the double-inlet PTC with a cold reservoir also shows superiority over the conventional type in the cooling power and COP. Comparing Figs. 4 and 7, it reveals that for certain cold-end flow conductance ratio OC_c, the COP of double-inlet PTC with the cold reservoir is larger than that of the orifice type with the cold reservoir.

In Figs. 3 and 4, The optimal cooling capacity and COP closely depends on the cold-end flow conductance ratio OC_c for a given volumetric ratio OV_c. Figs. 8 and 9 show the dependence of the dimensionless cooling power and COP on OC_c for the orifice PTC with a given OV_c. There is an optimal OC_c for a fixing cold reservoir. However, since the orthogonal component of flow becomes overwhelming to the parallel one with the increase of OC_c, the COP will decrease when OC_c is larger. So there is a compromise between the cooling power and the COP for determination of OC_c if volumetric ratio OV_c is set.

4. Experimental validation

4.1. Experimental setup

To further validate the new configuration, the preliminary experiments have been implemented with a home-made single stage GM PTC. Fig. 10 shows the schematic and the photograph of a single stage GM type PTC with a cold reservoir connected through an orifice to the cold end of the pulse tube. The pulse tube and the regenerator are fabricated from stainless steel tubes with outer diameter, wall thickness and length as follows: pulse tube: ∅16 mm × 0.3 mm × 250 mm; regenerator: ∅20 mm × 0.3 mm × 210 mm. The regenerator is alternately packed with the 247 meshes of stainless steel and phosphor-bronze screens. A reservoir of 0.5 L was connected through an orifice to the hot end of the pulse tube. While the cold reservoir of 0.45 L was connected to the U shape tube between the cold ends of regenerator and pulse tube through an needle valve (Type: Swagelok ASS-ORS3MM, C_v = 0.09,). The flow conductance ratio OC_c can be enabled by the opening of the cold-end orifice (O_2 in Fig. 10) during the experiments.

The cryocooler operates in the double-inlet mode with the anti-parallel arrangement to adjust the DC flow. A water-cooled helium compressor with rated input power of 4 kW was used to generate the pressure oscillation through a rotary valve. The refrigeration temperature of the cold head was measured by a calibrated Rh–Fe resistance thermometer with an accuracy of 0.1 K, while calibrated Pt 100 resistance thermometers were installed on the outer wall of the pulse tube, regenerator and the cold reservoir to monitor the temperature distribution. The cooling power was determined by measuring the electrical power input to a heater firmly attached to the cold end of the pulse tube with the measurement accuracy of 1.0 mW.

The system was initially charged to 1.25 MPa with helium and operated with frequency of 2.0 Hz. The only distinct aspect for the PTC with a cold reservoir is the cold-end orifice O_2. Additionally, a by-pass valve was connected between the inlet and outlet of the helium compressor, and the pressure ratios in the hot end of regenerator were maintained on the same level during the tests.

4.2. Experimental results and discussion

In order to investigate the effects of the cold reservoir to cooling performance, and validate the tendencies in theoretical analysis,

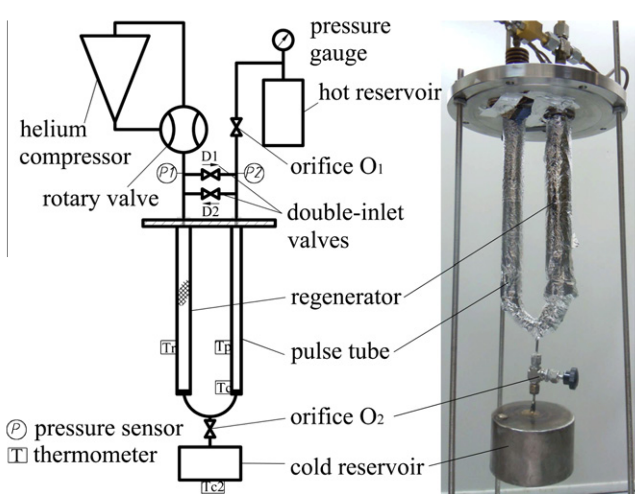

Fig. 10. Schematic and photograph of the single stage GM type PTC with a cold reservoir.

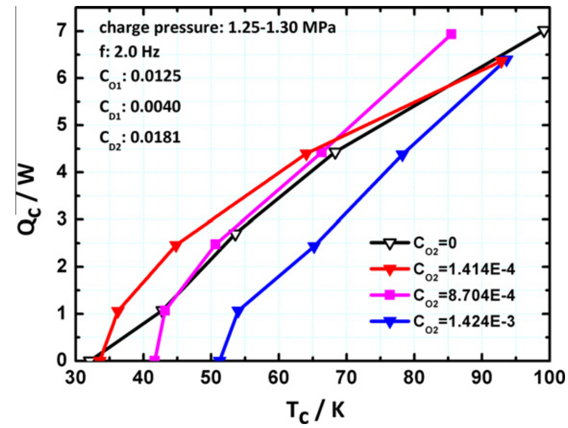

Fig. 11. Comparison of cooling capacities with different openings of the cold-end orifice.

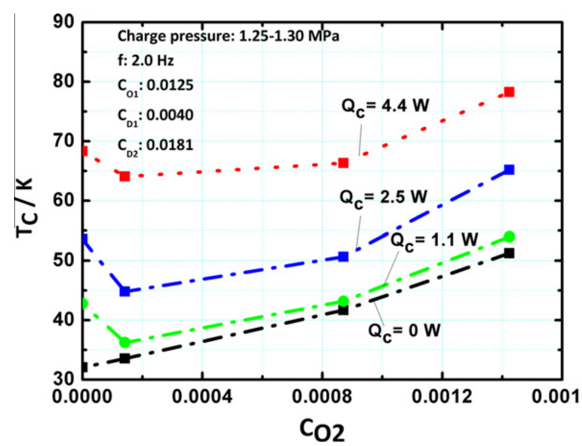

Fig. 12. Variation of cold head temperature of PTC with different openings of the cold-end orifice and cooling capacities.

cooling capacities Q_c of the PTC with different settings of cold-end orifice O_2 have been measured. As shown in Fig. 11, the case with flow coefficient of cold-end orifice c_{O2} = 0 is considered as a baseline. The experimental results show the following points:

When the opening of cold-end orifice is minor, cooling capacity of the novel PTC with the cold reservoir is improved to some extent in comparison to the counterpart, which agrees with the tendency of the thermodynamic analysis above. When the flow coefficient of the cold-end orifice $c_{O2} = 1.414 \times 10^{-4}$, the increase in cooling capacity at 40 K and 50 K can be up to 105.7% and 37.4%, respectively. However, as c_{O2} gets larger, the no-load temperature and cooling capacity deteriorate. The opening of cold-end orifice O_2 has significant effect on the cooling capacity. With the large c_{O2}, more mass flow is required during the adiabatic expansion to improve the cooling performance, whereas, the diameter of regenerator is inadequate in the experiments, leading to a rise of the regenerator losses and decreases the cooling capacity. The smaller values of c_{O2} contributed to the improvement of cooling capacity.

Fig. 12 displays the dependence of the cold head temperature with different cooling capacities on c_{O2}. Agreeing with the theoretical analysis, there is an optimal OC_c, which is embodied by flow coefficient c_{O2} of cold-end orifice in the experiments for the cooling performance. When $c_{O2} = 1.414 \times 10^{-4}$, the cold head temperature is lowest with 1.1, 2.5 and 4.4 W, while T_c increases with the rise of c_{O2}.

Introducing a cold reservoir to the cold head of PTC also increases the cooling load, and the heat from adiabatic compression should be taken away through regenerator and pulse tube. Additional heat load significantly affects the cooldown time of PTC in the experiments. The cooldown time of cold head of PTC with a large c_{O2} is significantly longer than those with a minor c_{O2}. However, with a minor c_{O2}, the cooling down of regenerator and cold reservoir is much slower than those with large c_{O2}, due to small mass flow rate required for the adiabatic expansion in the cold reservoir.

5. Conclusions

To investigate the possibility of direct phase shifting, and increase of cooling capacity for a pulse tube cryocooler, a cold reservoir and an orifice were introduced to the cold end of pulse tube. With the additional in-phase flow brought directly to the cold end, the phase shifting with a cold reservoir becomes direct and enhanced, with the cooling power per unit flow rate increased.

Strictly speaking, the new configuration of PTC with a cold reservoir is at the cost of complexity, however, it was confirmed that the cooling capacity of PTC with a cold reservoir can be improved notably on the basis of both theoretical analyses and our experimental validations. There exists an optimal flow conductance ratio OC_c, which can be embodied by the opening of the cold-end orifice in experiments to improve the cooling performance over PTCs with conventional phase shifter. 105.7% increase in cooling capacity at 40 K and 37.4% at 50 K were achieved in the preliminary experiments when flow coefficient of the cold-end orifice $c_{O2} = 1.414 \times 10^{-4}$.

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