Effects of cold-end temperature and heat load on the cooling characteristics of a pulse tube refrigerator

Shaoshuai Liu1 | Zhenhua Jiang1,2 | Lei Ding1 | Haifeng Zhu1 | Xiaoping Qu1 | Yinong Wu1,2

1Shanghai Institute of Technical Physics, Chinese Academy of Science, Shanghai, China
2University of Chinese Academy of Sciences, Beijing, China

Correspondence
Shaoshuai Liu and Yinong Wu, Shanghai Institute of Technical Physics, Chinese Academy of Science, 500 Yutian Road, Shanghai 200083, China. Emails: liushaoshuai@mail.sitp.ac.cn (SL); wyn@mail.sitp.ac.cn (YW).

Funding information
China Postdoctoral Science Foundation, Grant/Award Number: 2018M630476; National Key R&D Program of China, Grant/Award Number: 2016YFB0500600; National Natural Science Foundation of China, Grant/Award Number: 51806231; Natural Science Foundation of Shanghai, Grant/Award Number: 18ZR1445600; Aeronautical Science Foundation of China, Grant/Award Number: 20172490002

Abstract
A pulse tube refrigerator (PTR) was developed to operate at different cooling capacities to meet the requirements of various applications. Changes in the thermal loads or the long operating time of the PTR will influence the cold conditions (cold-end temperature and heat load), leading to a deviation of the cooling performance from the optimal design conditions. The basic principles of the mass flow characteristics of the PTR are constructed based on enthalpy phase modulation, which is helpful for comprehending the relationships between cold-end parameters and other parameters. A REGEN model is introduced in which different mass flows at the cold-end are considered to simulate the effects of the cold-end parameters on the performance of the regenerator. The influences of the cold-end temperature and heat load on the cooling performance are investigated by using a coupling model of a DeltaEC model and a REGEN model. In addition, some experiments are performed on a PTR working at different cold-end temperatures and heat load. The experimental results are in good agreement with the simulation results. The cooling performance of the PTR is influenced by the average pressure and operating frequency of different cooling states. Relative Carnot efficiencies of 12.2% for 4 W@60 K with a specific power of 33 W/W and 16.1% for 15 W @120 K with a specific power of 9 W/W can be achieved by the PTR using different operating parameters.

KEYWORDS
cold-end temperature, DeltaEC, heat load, pulse tube refrigerator, REGEN

1 | INTRODUCTION

Pulse tube refrigerators (PTRs) are attractive for a wide variety of important applications, such as superconducting devices and the aerospace and military defense fields. The free low-temperature mechanical moving components offer the space-qualified PTR the great advantages of high reliability, low noise, low cost, and low vibration. After development for several decades, PTRs are readily available for providing the cooling requirements of devices. However, due to changes in the thermal load or the long operating time of the refrigerator, the cold-end temperature and heat load of the PTR deviate from the optimal design conditions.

A PTR is designed to operate at a fixed cooling capacity with high efficiency, but it can also obtain a different heat load at different cold-end temperatures because of its unique refrigeration mechanism. Radebaugh published an article about the development of PTRs, in which the operating principles...
for different types of PTRs were described. Enthalpy and entropy flow models were proposed, which can be utilized to analyze the relationships of the operating parameters between the cold end and the other parts. Hu et al. developed a PTR that produces 300 W of heat load at 80 K with a relative Carnot efficiency of 18.2%. For an electric power of 3.6 kW, the heat load is 285 W, corresponding to a relative Carnot efficiency of 20.9%. Cold-end temperatures of 60 K to 120 K were presented, and the maximum relative Carnot efficiency of 21.6% was reached at a cold-end temperature of 100 K. Their study showed that the maximum relative Carnot efficiency of the PTR could be obtained at a fixed cold-end temperature and that this efficiency would decrease with increasing input electric power. Hu et al. developed an efficient PTR for gas liquefaction. The electrical power dependence of the heat load and overall relative Carnot efficiency at 100 K and 120 K were presented in the study. The maximum heat load exceeded 1270 W at a cold-end temperature of 120 K with a relative Carnot efficiency of approximately 18.6%. However, for a heat load of less than 900 W, the relative Carnot efficiency exceeded 20%. The same trend occurred at a cold-end temperature of 100 K. To meet the requirements of varying heat load at different cold-end temperatures, many PTRs were developed for different capacities.

For changing parameters at the cold end (cold-end temperature and heat load), the operational behavior of a PTR varies from the optimal working conditions. Based on enthalpy phase modulation theory, Radebaugh discussed the effects of the mass flow, pressure wave, and phase angles at the cold end on the cooling performance of PTRs. Ladner’s research showed that there is a coupling relationship among the mass flow, the cold-end temperature, and the cooling capacity. de Boer used the cold-end parameters as one of the most important references while optimizing the cooling performance of regenerative cryocoolers. In all thermodynamic analyses of PTRs, the parameters at the cold end (mass flow, pressure wave, and phase angle) are indispensable and important components. As discussed in the above survey, the effects of the cold-end parameters on the performance of the regenerator and the cooling capacity need to be investigated in detail to design an efficient PTR.

In this paper, the relationships between the parameters at the cold end and the other parameters are presented theoretically, and it is demonstrated that the parameters at the cold end have an impact on the cooling performance. A REGEN model is introduced to calculate the effects of the mass flow and temperature at the cold end on the working characteristics of the regenerator. Therefore, the main losses in refrigeration can be obtained and the cooling performance can be analyzed accurately. Another simulation program (DeltaEC) is employed to simulate the cooling performance of a PTR with different cold-end temperatures and heat load. An experimental apparatus is briefly described, and some comparative studies of simulations and experiments are carried out.

2 | THEORETICAL STUDY

2.1 | Physical model and analytical method

A PTR consists of a compressor, a cold finger, a connecting tube, and a phase shifter (inertance tube and reservoir). A schematic of a PTR is illustrated in Figure 1, in which each component is expressed. The PTR is filled with helium as the working fluid, which is driven by the compressor (B). The oscillating flow generated by the compressor (B) enters the cold finger (C) through the connecting tube (G). Heat exchangers (warm heat exchanger and cold heat exchanger) transfer the heat between the oscillating flow and the cooling medium. The regenerator is filled with high heat capacity porous material in good thermal contact, which is capable of transferring the heat in a period with high regenerative efficiency. The phase shifter (inertance tube [H] and reservoir [D]) is utilized to provide an opportune phase relationship between the mass flow and pressure waves in the PTR. The cold end (red circle) of the PTR provides the required cooling capacity for the heat load. The required temperature and heat dissipation of the heat load affect the cooling performance of the PTR, which should be studied in detail.

Based on the initial parameters of the PTR (including the structural parameters and operating parameters; see Table 1), a one-dimensional PTR model is built by DeltaEC. The parameters at the cold end are calculated to be the input data of the regenerator model, which is accomplished by REGEN. The effects of the mass flow $m_c$ and cold-end temperature $T_c$ on the losses and phase shift ability in the regenerator are investigated by simulation. In addition, conditions in which a PTR is working at different $T_c$ with a same $Q_c$ and with different $Q_c$ at a same $T_c$ are both analyzed (see Figure 2).
TABLE 1  The initial parameters of the PTR

| Parameters       | Values      |
|------------------|-------------|
| Regenerator      | 23.5 × 75   |
| Pulse tube       | 12.5 × 85   |
| Inertance tube I | 3 × 1.3     |
| Inertance tube II| 4.5 × 3.1   |
| Reservoir        | 65 × 100    |

Note: The listed dimensions are: ID × length (all in mm).

### 2.2  Phase characteristics of the PTR

An instantaneous process in the components of the PTR can be analyzed by investigating the behavior within one thermodynamic cycle. According to the first law of thermodynamics and the conservation of mass as well as momentum, ignoring the work crossing the boundary, the equations can be described as:\[13\]:

\[
\frac{h_i}{r_h} (T - T) = \frac{\partial}{\partial x} \left[ \left( \frac{m}{A_g} \right) h \right] - \frac{\partial}{\partial x} \left[ k_s \frac{\partial T}{\partial x} \right] + \frac{\partial}{\partial t} (\rho u) \quad \text{(Energy, gas)}
\]

\[
\frac{h_i}{r_h} (T_m - T_m) = - \left[ 1 - \frac{n_k}{n_g} \right] \frac{\partial}{\partial x} \left[ k_m \frac{\partial T_m}{\partial x} \right] + \left[ 1 - \frac{n_k}{n_g} \right] \rho_m c_m \frac{\partial T_m}{\partial t} \quad \text{(Energy, matrix)}
\]

\[
\frac{\partial}{\partial x} \left[ \frac{m}{A_g} \right] = - \frac{\partial \rho}{\partial t} \quad \text{(Mass)}
\]

\[
\frac{-\partial P}{\partial x} = \left[ \frac{m}{A_g} \right] \left[ \frac{m}{A_g} \right] f_r \frac{1}{2p_0 \rho} + \frac{\partial}{\partial x} \left[ \frac{m}{A_g} \right]^2 + \frac{\partial}{\partial x} \left[ \frac{m}{A_g} \right] \quad \text{(Momentum)}
\]

A one-dimensional solution is adequate in a PTR. As shown in Equation (4), a pressure gradient can be caused in phase with the acceleration of the working fluid mass, which is made use of in the phase shifter of the PTR. The equation of conservation of mass, Equation (3), is usually integrated over the length of a particular component to relate the mass flows at both ends of the component. For an isothermal component, the integrated equation relates the mass from the hot end to that of the cold end by

\[
\dot{m}_h = \dot{m}_c + \frac{\dot{P} V}{R T_a}
\]

where \(V\) is the gas volume of the component, \(R\) is the gas constant, \(T_a\) is the average temperature, and \(m\) and \(p\) are time-varying parameters with arbitrary phase relationships. As the regenerator spans a large temperature gradient, the phase relationship needs to be controlled to reduce the losses in the regenerator. A typical phase diagram is presented in Figure 3, relating the pressure wave to the geometry of the PTR. For relatively small pressure amplitudes in the PTR, with the assumption of isothermal expansion, the mass conservation applied to the cold end of the PTR is

\[
\dot{m}_h = \frac{P_m V_c}{R T_c} + \frac{\dot{P} V_E}{2 R T_c}
\]

While the structural parameters of the PTR are determined, the impedance (phase angle and amplitude) at the warm end of the pulse tube is limited by the phase shifter. The pulse tube can be considered an adiabatic component, which means that no work or heat transfer occurs in the pulse tube. Changes in the cold-end temperature and heat load lead to influences on the regenerator, thus the cooling performance of the PTR is affected. For example, assuming an increase in the heat load that causes an increase in the mass flow \(m_c\), the phase relationship in the regenerator will worsen, resulting in a decrease in cooling performance. The effects need to be studied in detail for better applications.

### 3  SIMULATION

#### 3.1  Simulation process

As the performance of the regenerator has a significant impact on the PTR, a National Institute of Standards and Technology (NIST) code, REGEN 3.3,\[20\] is employed to investigate the effects of \(m_c\) and \(T_c\) on the cooling performance of the PTR. Based on the initial parameters of the PTR, the main input parameters of the REGEN model are given by DeltaEC (a thermoacoustic program)[21] and are listed in Table 2. The structural parameters and operating parameters in the REGEN model are the same as those in the DeltaEC model. The specific losses (regenerator losses and pressure drop losses) can be calculated in the REGEN model but not in the DeltaEC model. For the fixed cooling state of the PTR, the acoustic power is transferred from CHX to WXH, and the phase angle in the pulse tube is unchanged. Therefore, the

![Figure 2](image-url)  Study flowchart
mass flow at the cold end can be used to evaluate the heat load of the PTR.

Combining the DeltaEC simulation results and REGEN simulation results, the mass flow, phase angle, regenerator losses, and PV power are calculated for different $m_c$ values. To study and compare the influences of different $T_c$ and $Q_c$ in detail, three cooling states are introduced, in which the phase relationship in the regenerator is simulated. The maximum COP of the PTR would occur with the coupling of matched cold-end temperature and heat load, which are the various perspectives on the PTR’s refrigeration mechanism.

### 3.2 Effects of $m_c$ and $T_c$ on the PTR

Based on the above theoretical analysis, the fundamental parameters of the regenerator are simulated. The mass flow and phase angle at the warm end of the regenerator for different $m_c$ values are shown in Figure 4. In the calculations, the cold end phase angle $\theta_c$ and pressure ratio $P_r$ are fixed at $-23^\circ$ and 1.15, respectively. For with the same structure of the regenerator, the maximum mass flow at the warm end increases with an increase in $m_c$, while the phase angle at the warm end decreases with at increase in $m_c$.

The maximum mass flow at the warm end is 2.99 g/s when $m_c$ is 2.5 g/s but is 5.74 g/s when $m_c$ is 6.0 g/s. The phase angle should be 24.4° when $m_c$ is 4 g/s, which is an opposite phase relationship to that of the minimum of regenerator losses. According to Figure 3, the equilibrium position of the phase angle in the regenerator gradually moves from the cold end to the warm end. Increasing values of $m_c$ and $m_a$, combined with the phase angle, cause large losses in the regenerator. Figure 5 shows the relationship between PV power and mass flow at the cold end. The phase angle between the mass flow and pressure wave is fixed at $-23^\circ$ so that the PV diagrams at the cold end can be used to reflect the difference in heat load for different $m_c$ values. As shown in Figure 5, the area of the PV diagrams increases with increasing $m_c$, which means that a larger mass flow enhances the cooling capacity.

Figure 6 shows the regenerative losses, pressure drop losses, and total PV power of the PTR for different mass flows at the cold end. The difference in regenerative losses
is less than 5 W when $m_c$ changes from 2.5 g/s to 6.0 g/s because the heat recovery capacity of the regenerator filler is large enough to accommodate the mass flows. In contrast, the pressure drop loss is 16.52 W at a mass flow of 2.5 g/s but is 93.4 W at a mass flow of 6.0 g/s. As the energy lost by the pressure drop is proportional to the friction resistance (increasing with increasing mass flow), increasing the heat load required by a large $m_c$ would result in a large amount of total power input to the PTR. Figure 7 shows the PV diagrams at the warm end, presenting the total input PV power for the PTR for different mass flows at the cold end. Slight changes occur in the amplitude of the pressure wave at the warm end. Combined with the phase angle relationship shown in Figure 4, the PV power shows a rising trend with increasing $m_c$.

The heat load, PV power at the cold end, and COP could be used to intuitively evaluate the effects of $m_c$ on the cooling performance of the PTR, as shown in Figure 8. An increase in $m_c$ results in an increase in the PV power at the cold end, thus improving the heat load. However, there is an optimal $m_c$ for the maximum COP of the PTR (see point A), indicating that the PTR is designed to work at a specific cooling state for efficient operation.

Considering that the phase angle and the pressure ratio at the cold end both change with the cold-end temperature and the heat load, the DeltaEC model is used to simulate the influence of the phase relationship and the mass flow in the regenerator, which affect the cold-end temperature and the heat load, on the cooling performance under the real conditions. Figure 9 shows different phase relationships of the mass flow relative to the pressure in the regenerator of the PTR operating at different cold-end temperatures and heat load. Cases 1 and 3 are for the same cooling power but different cold-end temperatures, while cases 2 and 3 are for the same cold-end temperature but different heat load. The figure shows that the phase angles of case 1 at both the cold end and warm end are larger than those of case 3. Combined with the amplitudes of $m_h$ and $m_c$ in cases 1 and 3, a larger average mass flow yields more input PV power. The contrast between cases 2 and 3.
shows that increasing the heat load would widen the phase span in the regenerator, which coincides well with the trend of the REGEN results (see Figure 4).

4 | EXPERIMENTAL VERIFICATION

4.1 | Experimental setup

Experiments were conducted to verify the simulation results. The cooling performance of the PTR was investigated at different cold-end temperatures and heat load. The experimental PTR has the same structural parameters as the simulation model as listed in Table 1. A dual-piston linear compressor is driven by AC power, for which the output of the frequency and the input power can be regulated. The maximum input electric power of the linear compressor is approximately 230 W. The regenerator is fully packed with 350-mesh and 500-mesh stainless steel screens. The AC, WHX, and CHX are designed and fabricated in a slit configuration for high heat exchange efficiency. The length and width of the slits are calculated and optimized by CFD software, which is not shown in this paper. The structure of the phase shifter is designed and optimized for a heat load of 10 W at 90 K. In the design and optimization process of the phase shifter, the structural parameters, operating parameters, coil type, and temperature are all considered. The research methods can be found in the authors’ previous articles.10,22,23 A photograph of the experimental system is shown in Figure 10. The regenerator, CHX, and pulse tube are enclosed in a vacuum chamber to reduce the convective heat transfer losses. A ceramic heater is firmly attached to the CHX to determine the heat load and is powered by a DC power supply. A PT-100 resistance thermometer with an accuracy of ±0.1 K is fixed on the CHX to measure the cold-end temperature. The aftercooler is cooled by a water cooler to maintain the temperature at 293 K, while the compressor and the phase shifter are cooled by a cooling fan. A linear variable differential transformer (LVDT) is used to monitor the displacement of the piston, which can help to evaluate the PV power (acoustic power) in combination with a pressure sensor installed at the outlet of the compressor according to Equation (7):

\[
W_{pv} = P A \pi f \sin \theta_{p-x}
\]

4.2 | Results and discussions

Experiments in which the PTR was operating at different frequencies were carried out. Comparisons of the simulation results and experimental results of the PTR operating at different frequencies with a heat load of 4 W at 60 K are shown in Figure 11. The green line connecting the test data points is used to make a better comparison with the simulation results (the purple line). The PV power of the simulation results and the test results shows the same trend as the frequency increases, and the two results show good agreement. The difference between the simulation results and the test results is less than 4%. The small errors may be caused by nonlinear effects not considered in the 1-D simulation model. Figure 12 shows the experimental results for the input electric power dependence on the operating frequency with different cooling capacities and average pressures. Each cooling state has an optimal operating frequency for the minimum input electric power. The optimal frequency increases from 63 Hz to 64 Hz as the pressure increases because increasing the average frequency subjoins the stiffness of gas spring and coefficient of the gas damping. Increasing the average pressure could also introduce a better cooling performance. By transforming the cooling state from 15 W @ 110 K to 4 W @ 60 K and keeping the pressure constant, the optimal frequency changes from 64 Hz to 65 Hz. It can be concluded that by choosing a...
preferable operating frequency for different cold-end temperatures and heat load, a PTR could be used in a wide range of areas for refrigeration with high efficiency.

To investigate the effects of the cold-end temperature and heat load, the test results are illustrated below (see Figures 13 and 14). As the cold-end temperature changes from 90 K to 120 K, the input electric power decreases from 139 W to 90 W for a heat load of 10 W at 3.5 MPa, and the input electric power decreases from 174 W to 109 W for a heat load of 12 W at 3.5 MPa. The cooling performance of the PTR at 3.2 MPa is greater than that at 3.5 MPa when the cold-end temperature is higher than 100 K. In contrast, the cooling performance of the PTR at 3.2 MPa is lower than that at 3.5 MPa for a cold-end temperature below 90 K, which was also observed at 60 K (see Figure 14). This can be interpreted to mean that the thermal penetration depth and viscous penetration depth of the helium decrease with decreasing temperature. A large average pressure would enhance the heat transfer capability and improve the cooling performance. The specific power is also a parameter used to reflect the cooling performance of the PTR and is defined as the input electric power divided by the heat load. The specific power is approximately 33 W/W at 60 K but is 9 W/W at 120 K. According to thermodynamic principles, a large temperature difference between the hot end and the cold end needs a large required input power for the same heat load. Therefore, the specific power at 60 K is larger than that at 120 K. This phenomenon is also reflected in the different slopes shown in Figure 14 and Ref.25. The relative Carnot efficiencies of 12.2% for 4 W @ 60 K and 16.1% for 15 W @ 120 K can be achieved by the PTR with different operating parameters.

5 | CONCLUSIONS

The cooling conditions for the operating characteristics of a PTR were numerically and experimentally investigated. According to the REGEN simulation model, increasing the heat load could cause an increase in $m_c$, and worsen the phase relationship in the regenerator. Increasing $m_c$ and $m_h$ combined with the phase angle led to large losses in the regenerator, which would cause an increase in the required PV power. Experiments were carried out on the PTR operating at different cold-end temperatures and heat load to verify the simulation results. Comparisons of the operating frequencies show good agreement, with an error difference of less than 4%. The cooling capacity of the PTR at 3.2 MPa is greater than that at 3.5 MPa when the cooling temperature is higher than 100 K, while, the cooling performance of the PTR at 3.2 MPa is lower than that at 3.5 MPa for cooling temperatures below 90 K. Relative Carnot efficiencies of 12.2% for 4 W @ 60 K with specific power of 33 W/W and 16.1% for 15 W @ 120 K a specific power of 9 W/W can be achieved by the PTR with different operating parameters. Based on the analysis, there is an optimal combination of the cold-end temperature and
cold load for maximizing the COP in designing a PTR. This work quantitatively gives the effects of different cold-end temperatures and heat load on the cooling capacity and cooling efficiency, which are instructive in practical engineering applications.

ACKNOWLEDGMENTS
This work is supported by the National Natural Science Foundation Projects (No. 51806231), the Natural Science Foundation of Shanghai (No. 18ZR1445600), the Aeronautical Science Foundation of China (20172490002), National Key R&D Program of China (2016YFB0500600), and the China Postdoctoral Science Foundation (2018M630476).

NOMENCLATURE

SYMBOLS

\( A_g \)  
gas cross-sectional area, \( \text{m}^2 \)

\( f_r \)  
fanning friction factor

\( h \)  
specific enthalpy, \( \text{J/kg} \)

\( k \)  
thermal conductivity, \( \text{W/m K} \)

\( m \)  
mass flow, \( \text{kg/s} \)

\( n \)  
porosity

\( P \)  
pressure, \( \text{Pa} \)

\( R \)  
gas constant, \( \text{J K kg}^{-1} \)

\( r_h \)  
hydraulic radius

\( T \)  
temperature, \( \text{K} \)

\( u \)  
velocity, \( \text{m/s} \)

\( W \)  
power, \( \text{W} \)

\( x \)  
axial position

GREEK/MATH

\( \partial \)  
partial derivative

\( \rho \)  
density, \( \text{kg m}^{-3} \)

SUBSCRIPTS

\( a \)  
average

\( c \)  
cold end

\( e \)  
instantaneous expander

\( E \)  
total expander

\( it \)  
 inertance tube

\( g \)  
gas

\( m \)  
matrix

\( h \)  
warm end

REFERENCES

1. Radebaugh R. Pulse tube cryocoolers for cooling infrared sensors. In: Andresen BF, ed. Proceedings of SPIE; 2000:363.
2. Cha JS, Ghiaasiaan SM, Kirkconnell CS. Oscillatory flow in microporous media applied in pulse-tube and Stirling-cycle cryocooler regenerators. Exp Therm Fluid Sci. 2008;32:1264-1278.
3. Kushino A, Sugita H, Matsubara Y. Performance of Japanese pulse tube coolers for space applications. In: Cryocoolers 13. Boston, MA: Springer; 2005:101-107.
4. Radebaugh R. A review of pulse tube refrigeration. In: Advances in Cryogenic Engineering. Boston, MA: Springer; 1990:191-1205.
5. Gifford WE. Pulse tube refrigeration progress. Adv Cryog Eng. 1965;10:69-79.
6. Radebaugh R. Development of the pulse tube refrigerator as an efficient and reliable cryocooler. Proceedings of Institute of Refrigeration. 2000;1999-2000.
7. Hu JY, Zhang LM, Zhu J, et al. A high-efficiency coaxial pulse tube cryocooler with 500 W cooling capacity at 80 K. Cryogenics. 2014;62:7-11.
8. Hu JY, Chen S, Zhu J, et al. An efficient pulse tube cryocooler for boil-off gas reliquefaction in liquid natural gas tanks. Appl Energy. 2016;164:1012-1018.
9. Zia JH. A pulse tube cryocooler with 300 W refrigeration at 80 k and an operating efficiency of 19% Carnot. Cryocoolers. 2007;14:141-147.
10. Liu SS, Chen X, Zhang AK, et al. Investigation on phase shifter of a 10 W/70 K inertance pulse tube refrigerator. Int J Refrig. 2017;74:448-455.
11. Li ZP, Jiang YL, Gan ZH, Qiu LM, Chen J. Performance of a pre-cooled 4 K Stirling type high frequency pulse tube cryocooler with Gd2O2S. J Zhejiang Univ-Sci A. 2014;7:508-516.
12. Qiu LH, Zhi XQ, Dietrich M, Dietrich M, Gan ZH, Thummes G. Investigation on phase shifting for a 4 K Stirling pulse tube cryocooler with He-3 as working fluid. Cryogenics. 2015;69:44-49.
13. Radebaugh R. Thermodynamics of regenerative refrigerators. In: Generation of Low Temperature and It's Applications; 2003:1-20.
14. Radebaugh R, Herrmann S. Refrigeration efficiency of pulse tube refrigerator. Proceedings of the 4th International Cryocooler Conference; 1988:119-133.
15. Storch PJ, Radebaugh R. Development and experimental test of an analytical model of the orifice pulse tube refrigerator. In: Advances in cryogenic engineering 1988;851-859.
16. Ladner DR. Performance and mass vs. operating temperature for pulse tube and Stirling cryocoolers. In: International Cryocooler Conference. Boulder; Cryocoolers 16; 2011.
17. Boer PC. Optimal performance of regenerative cryocoolers. Cryogenics. 2011;51:105-113.
18. Razani A, Dodson C, Roberts T. A model for exergy analysis and thermodynamic bounds of Stirling refrigerators. Cryogenics. 2010;50:231-238.
19. Zhang XB, Zhang KH, Qiu LM, Gan ZH, Shen X, Xiang SJ. A pulse tube cryocooler with a cold reservoir. Cryogenics. 2013;54:30-36.
20. Gary JOGA, Radebaugh R, Huang YH, Marquardt E. REGEN 3.3: User Manual. NIST; 2008.
21. Ward B, Clark J, Swift GW. Design environment for low-amplitude thermoacoustic energy conversion. DeltaEC Version 6.4 b2 Users Guide. Los Alamos National Laboratory Network. 2016.

ORCID

Shaoshuai Liu https://orcid.org/0000-0002-4746-4852
22. Liu SS, Chen X, Zhang AK, et al. Investigation on the inertance tube of pulse tube refrigerator operating at high temperature. *Energy*. 2017;123:378-385.
23. Liu SS, Chen X, Zhang AK, Kan A, Zhang H, Wu Y. Impact of coiled type inertance tube on performance of pulse tube refrigerator. *Appl Therm Eng*. 2016;107:63-69.
24. Zhang AK, Wu YN, Liu SS, et al. Effect of impedance on a compressor driving pulse tube refrigerator. *Appl Therm Eng*. 2017;124:688-694.
25. Zhang AK, Wu YN, Liu SS, et al. High-efficiency 3 W/40 K single-stage pulse tube cryocooler for space application. *Cryogenics*. 2018;90:41-46.

**How to cite this article:** Liu S, Jiang Z, Ding L, Zhu H, Qu X, Wu Y. Effects of cold-end temperature and heat load on the cooling characteristics of a pulse tube refrigerator. *Energy Sci Eng*. 2020;8:731–739. [https://doi.org/10.1002/ese3.545](https://doi.org/10.1002/ese3.545)