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Study on a 5.0 W/80 K single stage Stirling type pulse tube cryocooler*

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Abstract: A single stage Stirling pulse tube cryocooler was designed based on REGEN 3.2 and fabricated for testing. The experimental results show that the cooler can provide a cooling capacity of 5.0 W at 79.1 K, and produce a no-load temperature of 57.0 K, operating with an average pressure of 2.50 MPa and a frequency of 60 Hz, performance results that are very close to the calculated values. The cryocooler can be cooled from room temperature to 80 K in 8.5 min. The fast cooldown time is a result of the small regenerator.

Key words: Cryocooler, Regenerator, High frequency, Pulse tube
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INTRODUCTION

The pulse tube cryocooler has attracted the enthusiastic attention of a large number of research groups and is being used in space technology and high-tech applications because it includes no moving parts in the cold region, is simple to construct, and provides a long operating life and minimal vibration.

There are two types of pulse tube cryocoolers: the Gifford-McMahon (GM) type and the Stirling type. The former uses a rotary valve between the compressor and the regenerator to distribute the working gas. It is usually operated at a low frequency (1~2 Hz) and is large in size and weight in order to produce the larger refrigeration capacities typically required for ground-based applications. However, its efficiency is relatively low due to the irreversible loss associated with the valve. The Stirling type pulse tube refrigerator has no valve between the compressor and the regenerator. Thus, the gas flow in the pulse tube oscillates with the same frequency as the compressor, which is typically in the 30~60 Hz range. The Stirling type pulse tube is therefore also frequently referred to as a high-frequency pulse tube cryocooler. By eliminating the irreversible loss of the valve, the Stirling type pulse tube cryocooler has a higher thermodynamic efficiency and a compact configuration. It has been a hot research subject in recent decades.

Thanks to the development of the linear compressor with flexure bearings and clearance seals, research on the high-frequency pulse tube cryocooler driven by a linear compressor has received considerable attention at home and abroad. The technology of space cryocoolers operating at temperatures above 55 K has matured in the USA (Ross and Boyle, 2007). In order to meet further miniaturization requirements for space applications, pulse tube cryocoolers operating at even higher frequencies (>60 Hz) are being

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developed (Vanapalli *et al*., 2007; Dai *et al*., 2007) and are expected to be of significant interest for the future.

Through the use of REGEN 3.2, a software package developed at the National Institute of Standards and Technology (NIST) (Gary and Radebaugh, 1991; Radebaugh, 2003), we have designed and fabricated a single stage Stirling type pulse tube cryocooler to operate at 60 Hz. The experimental performance of the cooler is in agreement with the design predictions. A net refrigeration capacity of 5.0 W at 79.1 K has been obtained with an electrical input power to the compressor of 290 W. Also, a no-load temperature of 57.0 K was achieved with a charging pressure of 2.50 MPa and a frequency of 60 Hz.

REGENERATOR

The regenerator is an important component of the cryocooler, as it directly affects the thermodynamic performance of the cryocooler. REGEN 3.2, used as the design optimization tool for the regenerator, is based on a set of finite difference equations for the conservation of energy and mass as follows:

Conservation of energy (gas)

$$\frac{h_\mathrm{t}}{r_\mathrm{h}}(T_\mathrm{m}-T)=\frac{\partial}{\partial x}\left(\frac{\dot{\boldsymbol{m}}}{A_\mathrm{g}}h\right)-\frac{\partial}{\partial x}\left(\lambda_\mathrm{g}\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial t}(\rho u). \quad (1)$$

Conservation of energy (matrix)

$$\begin{aligned}\frac{h_\mathrm{t}}{r_\mathrm{h}}(T-T_\mathrm{m})=&\left(\frac{1-n_\mathrm{g}}{n_\mathrm{g}}\right)\frac{\partial}{\partial x}\left(\lambda_\mathrm{m}\frac{\partial T_\mathrm{m}}{\partial x}\right)\\&+\left(\frac{1-n_\mathrm{g}}{n_\mathrm{g}}\right)(\rho_\mathrm{m}u_\mathrm{m})\frac{\partial T_\mathrm{m}}{\partial t}.\end{aligned} \quad (2)$$

Conservation of mass

$$\frac{\partial}{\partial x}\left(\frac{\dot{\boldsymbol{m}}}{A_\mathrm{g}}\right)=\frac{\partial \rho}{\partial t}. \quad (3)$$

In Eqs.(1)~(3), h_t is the heat transfer coefficient, r_h is the hydraulic radius, T_m is the average temperature, h is the specific enthalpy, $\dot{\boldsymbol{m}}$ is the mass flow rate, A_g is the gas cross-sectional area, x is the axial position, λ_g is the thermal conductivity of the gas, ρu is the volumetric heat capacity of gas, λ_m is the axial thermal conductivity of the matrix, n_g is the porosity of the matrix, and $\rho_\mathrm{m}u_\mathrm{m}$ is the volumetric heat capacity of the matrix.

Along with the initial input parameters of mass flow at the cold end and its phase with respect to the pressure at the cold end, average pressure, pressure ratio, operating frequency and the geometry of the matrix, the conservation equations may be solved by REGEN 3.2.

The model can also be used to calculate the inefficiency of the regenerator and the losses associated with conduction through the matrix and the pressure drop through the regenerator. The conduction loss through the thin wall of the regenerator should be calculated alone. Finally, REGEN 3.2 enables a calculation of the coefficient of performance (COP, the ratio of the net refrigeration capacity to the PV power at the entrance of the regenerator) for the regenerator as a function of a wide array of parameters, including frequency and geometry.

The material and configuration of the regenerative matrix is typically fixed at the outset, along with the average charge pressure, pressure ratio, and operating frequency. For fixed values of the regenerator diameter and the mass flow at the cold end, an optimal length for the regenerator can be obtained by maximizing the COP as a function of the length. Subsequently the mass flux at the cold end can also be optimized by investigating the dependence of the regenerator COP as a function of that parameter. To obtain the desired cooling power at the specified operating conditions, we linearly adjusted the diameter of the regenerator while maintaining the same optimized mass flux.

The thermal penetration depth for oscillating heat transfer is given by

$$\delta_\mathrm{t}=\sqrt{\frac{2\lambda}{\omega\rho c_\mathrm{p}}}, \quad (4)$$

where λ is the gas thermal conductivity, ω is the angular frequency, ρ is the gas density and c_p is the specific heat of the gas. The thermal penetration depth of helium under the conditions of 60 Hz, 2.50 MPa and 80 K is 88 μm. We have therefore chosen a #400 mesh stainless steel screen, whose hydraulic diameter is 55 μm, as the matrix for our regenerator. With an operating frequency of 60 Hz, a charge pressure of 2.50 MPa, and a pressure ratio of 1.3 at the cold end of regenerator, REGEN 3.2 predicts a refrigeration power of 10.0 W at 80 K. In addition, REGEN 3.2 predicts a refrigeration capacity of 5.20 W at 80 K for the pressure ratio of 1.18 rather than that of 1.30 (all other conditions being the same). The second value is of interest due to the fact that the linear compressor used in our experiment provided a maximum pressure ratio of only 1.18 during preliminary tests. Additionally, the performance of our pulse tube refrigerator is limited by the ability of the inertance tube to provide an optimal phase angle adjustment. Due to the small acoustic power of the pulse tube system, the phase angle of the mass flow with respect to the pressure at the cold end is limited to 0° (Optimal regenerator performance is obtained if the mass flow at the cold end lags the pressure by 30°). Fig.1 displays the calculated values of the regenerator COP vs the ratios of the gas cross-sectional area to the mass flow rate at the cold end for lengths of 30 mm, 35 mm, 40 mm, 45 mm and 50 mm, respectively. The associated cooling power for the 40 mm length is also provided.

As may be observed, the regenerator length of 40 mm provides the highest COP, and the optimal gas cross-sectional area per unit mass flow rate at the cold end is 0.07066 m^2·s/kg. For a mass flow amplitude of 1.8 g/s in the pulse tube, the corresponding refrigeration capacity is 9.75 W at 80 K for pressure ratio of 1.3. To accommodate axial conduction losses and welding clearance, a regenerator length of 45 mm has been selected for the experiment. In keeping with the design goal of 10.0 W refrigeration capacity and the availability of thin-walled tubes from commercial sources, the regenerator diameter is chosen as 15.4 mm.

Fig.2 displays the COP and cooling power at various pressure ratios with an optimal mass flow amplitude of 1.8 g/s at the cold end. Note that only 5.20 W may be obtained if the pressure ratio is 1.18 rather than 1.3 (see the open star in the bottom-left region of Fig.2).

PULSE TUBE

The working gas precooled by the regenerator expands in the pulse tube and provides the refrigeration. A large temperature gradient is developed axially along the pulse tube. For an ideal pulse tube, the acoustic power is constant from the cold end to the hot end and no entropy is generated along its length because of the adiabatic condition across the walls. The following conditions should be satisfied in order to estimate length of an ideal pulse tube: (1) the compression process and expansion process in the pulse tube must be adiabatic; (2) the swept volume of the virtual gas piston within the pulse tube must be significantly smaller than that of the pulse tube.

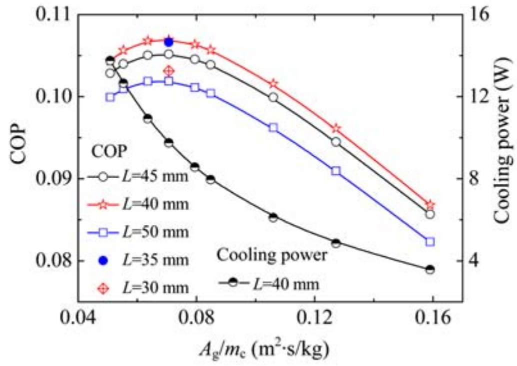

Fig.1 COP vs length of the regenerator
P_0=2.5 MPa; f=60 Hz; P_r=1.3 (cold end); T_c=80 K; ϕ_c=0°

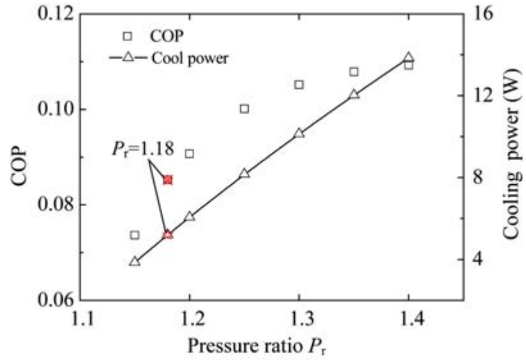

Fig.2 COP vs pressure ratio at the cold end of the regenerator
P_0=2.5 MPa; f=60 Hz; L=45 mm; m_c=1.8 g/s; T_c=80 K; ϕ_c=0°

The thermal penetration depth of helium is 0.185 mm for an oscillating flow with a frequency of 60 Hz, an average pressure of 2.5 MPa and a temperature of 300 K. This value becomes smaller as the temperature decreases. Heat transfer between the wall and the bulk flow within the pulse tube is negligible if the radius of pulse tube is 10 times larger than the thermal penetration depth of 0.185 mm. In this case the adiabatic condition is met. The second condition may be satisfied by a pulse tube volume that is 3~5 times larger than the swept volume of the gas piston at the cold end. According to the optimized results obtained with

REGEN 3.2, the swept volume of the gas piston, V_e, is 0.675 cm^3. Thus, the volume of pulse tube, V, must be at least 2.03 cm^3 to meet the adiabatic condition at both ends if the ratio of the pulse tube volume to the gas piston volume is 3.0. A larger pulse tube volume leads to the necessity for a larger phase shift by the inertance tube. The lengths of the pulse tube and the regenerator are held equal in the design, because a U-type configuration is adopted. A thin-walled stainless steel tube with inner diameter of 9.0 mm is used for the pulse tube. Finally, the ratio of the pulse tube diameter to the thermal penetration depth is set at 24.3, and the volume ratio V/V_e=4.3.

PHASOR DIAGRAM

Pressure, temperature and mass flow in the cryocooler are assumed to vary sinusoidally with time. By using an average temperature for the regenerator and adiabatic conditions in the pulse tube, the mass and energy conservation equations applied to the regenerator and the pulse tube respectively yield (Radebaugh, 2003):

$$\dot{\boldsymbol{m}}_{\mathrm{h}} = \dot{\boldsymbol{m}}_{\mathrm{c}} + \frac{\dot{\boldsymbol{P}}V}{RT_{\mathrm{a}}}, \tag{5}$$

$$\dot{\boldsymbol{m}}_{\mathrm{c}} = (T_{\mathrm{ht}}/T_{\mathrm{c}})\dot{\boldsymbol{m}}_{\mathrm{ht}} + \frac{\dot{\boldsymbol{P}}V_{\mathrm{t}}}{\gamma RT_{\mathrm{c}}}, \tag{6}$$

where $\dot{\boldsymbol{m}}_{\mathrm{h}}$, $\dot{\boldsymbol{m}}_{\mathrm{c}}$ and $\dot{\boldsymbol{m}}_{\mathrm{ht}}$ are the mass flows at the hot end of the regenerator, the cold and hot ends of the pulse tube, respectively. $\dot{\boldsymbol{P}}$ is the rate of change of pressure in the regenerator. V is the gas volume of the component, R is the mass-specific gas constant, T_{a} is the average temperature in the regenerator, T_{c} is the temperature of the cold end, T_{ht} is the temperature of the hot end of the pulse tube, and γ is the ratio of specific heats.

All the phasor quantities of the system are shown in Fig.3. The relationships between the various phasors (magnitudes and angles) are determined by the mass conservation equations for all the isothermal components (aftercooler, regenerator and two heat exchangers) and the energy conservation equation for the adiabatic pulse tube. By setting the phase angle of the pressure oscillation at the cold end of pulse tube to 0° and assuming that the pressure is essentially uniform throughout the system at any moment in time, it can be observed that the mass flow lags the pressure wave at the entrance of the inertance tube by 38°.

We can see from Fig.3a, that if the mass flow at the cold end of the regenerator is to be in phase with the pressure oscillation, a phase angle of −38° is required at the inertance tube entrance. Calculations (Radebaugh *et al*., 2006) indicate that it is difficult to obtain a −38° phase shift through the use of a single-diameter inertance tube. However, a double-diameter inertance tube may provide a −34° phase shift thereby closely meeting the design requirement.

As previously mentioned, the linear compressor used for the test limited the attainable pressure ratio at the cold end of the regenerator to 1.18. In this situation, the required phase angle at the entrance of inertance tube is −26.3°, a value which is easily produced as shown in Fig.3b.

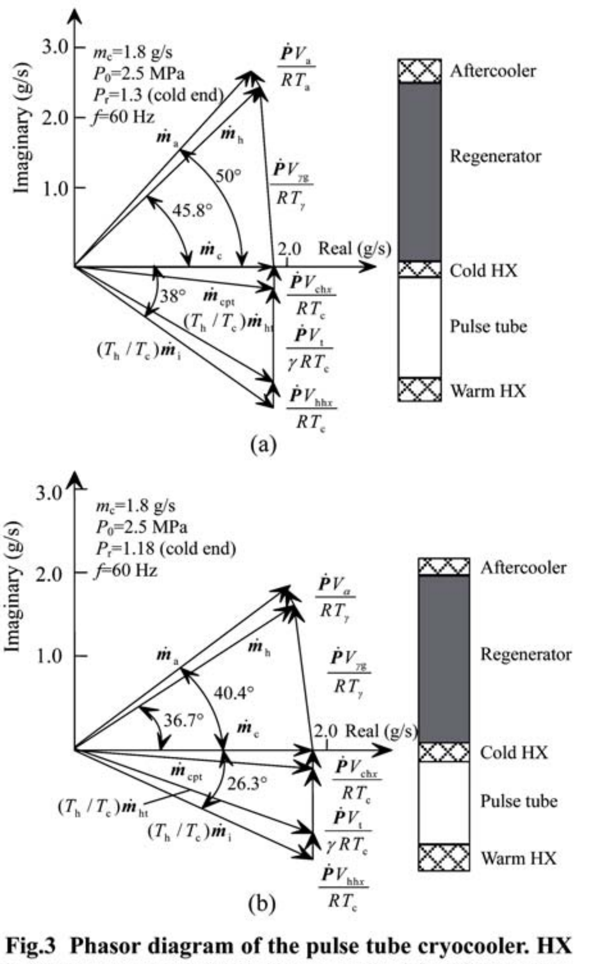

Fig.3 Phasor diagram of the pulse tube cryocooler. HX is heat exchanger. (a) Pressure ratio=1.3; (b) Pressure ratio=1.18

According to the design method mentioned above, the calculated dimensions of the cryocooler are summarized and shown in Table 1.

Table 1 Calculated geometry of the cryocooler

Component	Length (mm)	Outer diameter (mm)	Thickness (mm)
Regenerator	45	15.90	0.254
Pulse tube	45	9.53	0.254
Inertance tube			
Small	595	3	0.600
Large	695	4	0.750
Component	**Wire diameter (μm)**	**Porosity**	**Hydraulic diameter (μm)**
Stainless steel (#400) screen	25.4	0.6858	55.44

Note: the reservoir volume is 250 cm^3

EXPERIMENTAL RESULTS

The fabricated single stage Stirling type pulse tube cryocooler consists of a pulse tube, a regenerator, an inertance tube, a reservoir and a linear compressor, as shown in Fig.4. The linear compressor used for the experiment is a modified K535 Stirling cooler made by Ricor Inc., Israel. The compressor output power and frequency are modulated by a frequency inverter.

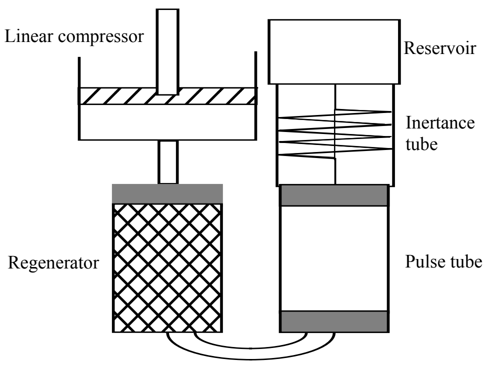

Fig.4 Schematic of the cryocooler

The heat exchangers at the hot ends of the regenerator and the pulse tube are both cooled by a water jacket. The vacuum surrounding the cryocooler is maintained at 10^{-4} Pa by a turbo-molecular pump. The temperature at the cold end of the pulse tube is measured by a calibrated platinum resistance thermometer with an accuracy of ±0.1 K. The pressure sensors are mounted at the entrance of the aftercooler, the hot end of the pulse tube, and at the reservoir. Because the pulse tube is simply an empty tube, the pressure at the cold end is approximately equal to the measured value at the hot end of the pulse tube. The pressure and temperature signals are collected by a computer (Kong *et al*., 2003).

Fig.5 displays the measured cool down process of the pulse tube cryocooler with a charge pressure of 2.50 MPa and a working frequency of 60 Hz. Because the output pressure ratio of the linear compressor used in the test is 1.18 (less than the design value of 1.30), the lowest temperature obtained is 57.0 K.

Fig.6 shows the measured cooling capacity vs temperature. The net refrigeration power is 5.0 W at 79.1 K with an electrical power input of 290 W. The calculated result by REGN 3.2 indicates that with a pressure ratio of 1.18 at the cold end, a refrigeration power of 5.20 W at 80 K could be obtained—a value that is quite close to the measured results. One may therefore expect that a cooling capacity of 10 W at 80 K could be obtained through the use of a linear compressor providing a pressure ratio of 1.3.

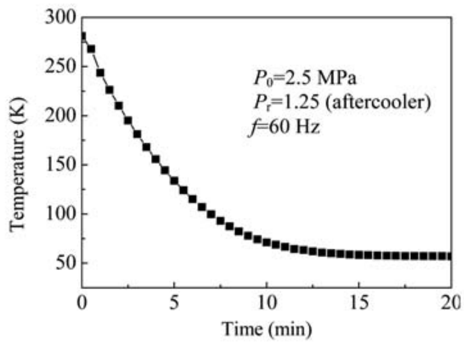

Fig.5 Cool down process of the single stage Stirling type pulse tube cryocooler

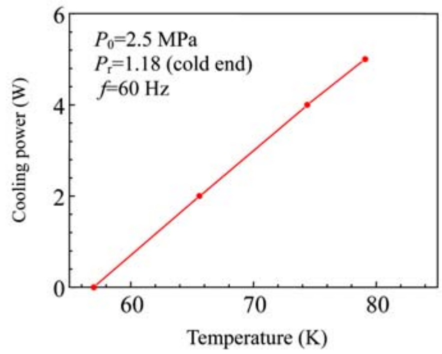

Fig.6 Cooling power vs temperature

CONCLUSION

A single stage Stirling type pulse tube cryocooler has been designed and fabricated based on calculations via REGEN 3.2. Preliminary experimental results indicate that with a pressure ratio 1.18, a cooling capacity of 5.0 W at 79.1 K and a no-load cooling temperature of 57.0 K may be obtained, in agreement with the design predictions.

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References

Dai, W., Yu, G.Y., Zhu, S.L., Luo, E.C., 2007. 300 Hz thermoacoustically driven pulse tube cooler for temperature below 100 K. *Applied Physics Letters*, **90**(2):24-104. [doi:10.1063/1.2405882]

Gary, J., Radebaugh, R., 1991. An Improved Model for the Calculation of Regenerator Performance (REGEN 3.1). Proceedings of Fourth Interagency Meeting on Cryocoolers, David Taylor Research Center Technical Report DTRC 91/003, p.165-176.

Kong, B., Gan, Z.H., Cheng, Z.Z., Tang, K., Qiu, L.M., Chen, G.B., 2003. Data Acquisition System for Pulse Tube Refrigerators. Proceedings of the Sixth National Cryogenic and Refrigeration Engineering Conference. Xi'an Jiaotong Univerxity Press, Xi'an, China, p.142-146 (in Chinese).

Radebaugh, R., 2003. Thermodynamics of Regenerative Refrigerators. *In*: Ohtsuka, T., Ishizaki, Y. (Eds.), Generation of Low Temperature and It's Applications. Shonan Technical Center, Kamakura, Japan, p.1-20.

Radebaugh, R., Lewis, M., Luo, E., Pfotenhauer, J.M., Nellis, G.F., Schunk, L.A., 2006. Inertance Tube Optimization for Pulse Tube Refrigerators. Advances in Cryogenic Engineering 51, AIP Conference Proceedings, p.59-67.

Ross, R.G.Jr., Boyle, R.F., 2007. An Overview of NASA Space Cryocoolers Programs-2006. *In*: Miller, S.D., Ross, R.G.Jr. (Eds.), Cryocoolers 14, p.1-10.

Vanapalli, S., Lewis, M., Gan, Z.H., Radebaugh, R., 2007. 120 Hz pulse tube cryocooler for fast cooldown to 50 K. *Applied Physics Letters*, **90**(7):072504. [doi:10.1063/1.2643073]