120 Hz pulse tube cryocooler for fast cooldown to 50 K

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A pulse tube cryocooler operating at 120 Hz with 3.5 MPa average pressure achieved a no-load temperature of about 49.9 K and a cooldown time to 80 K of 5.5 min. The net refrigeration power at 80 K was 3.35 W with an efficiency of 19.7% of Carnot when referred to input pressure-volume (PV or acoustic) power. Such low temperatures have not been previously achieved for operating frequencies above 100 Hz. The high frequency operation leads to reduced cryocooler volume for a given refrigeration power, which is important to many applications and can enable development of microcryocoolers for microelectromechanical system applications.
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Cryocoolers are often used in applications where small size and mass are important, for example, space applications. The pressure oscillator dominates the size of cryocoolers, but for a given acoustic power the pressure oscillator size can be reduced by operating at higher frequencies. Until recently, efficient operation of regenerative cryocoolers at frequencies above about 60 Hz has not been possible because of poor heat transfer in the regenerative heat exchanger. A no-load temperature of 40 K was achieved previously at a frequency of 70 Hz,[1] but experiments at 350 Hz achieved only 147 K.[2] We have shown that, with the right combination of operating parameters and regenerator geometry, efficient operation at frequencies even up to 1 kHz may be possible.[3] This letter presents experimental results that verify this prediction of efficient operation at 120 Hz. This frequency is higher than the resonance frequency of commercially available pressure oscillators, so at this time we focus only on the efficiency of the cold head.

The layout drawing of the pulse tube cryocooler is shown in Fig. 1. The pressure oscillator generates an oscillating pressure in the system that causes an oscillating flow governed by the complex flow impedance of the inertance tube. The temperature gradient in the regenerator matrix (fine mesh screen) precools the helium working fluid when it travels toward the cold end by storing heat in the matrix heat capacity during the first half-cycle. Heat is returned to the helium gas when it flows toward the warm end during the second half-cycle. High surface area and heat capacity are required of the matrix to ensure efficient heat transfer and nearly isothermal processes. Processes in the pulse tube must be nearly adiabatic for efficient operation, which occurs when the tube radius is large compared with the thermal penetration depth δ_t in the helium gas. At a temperature of 300 K and a pressure of 3.5 MPa, δ_t is 0.12 mm for a frequency of 120 Hz, and it varies with the frequency f as $\delta_t \propto f^{-1/2}$. Thus, higher frequency operation can enable minia-

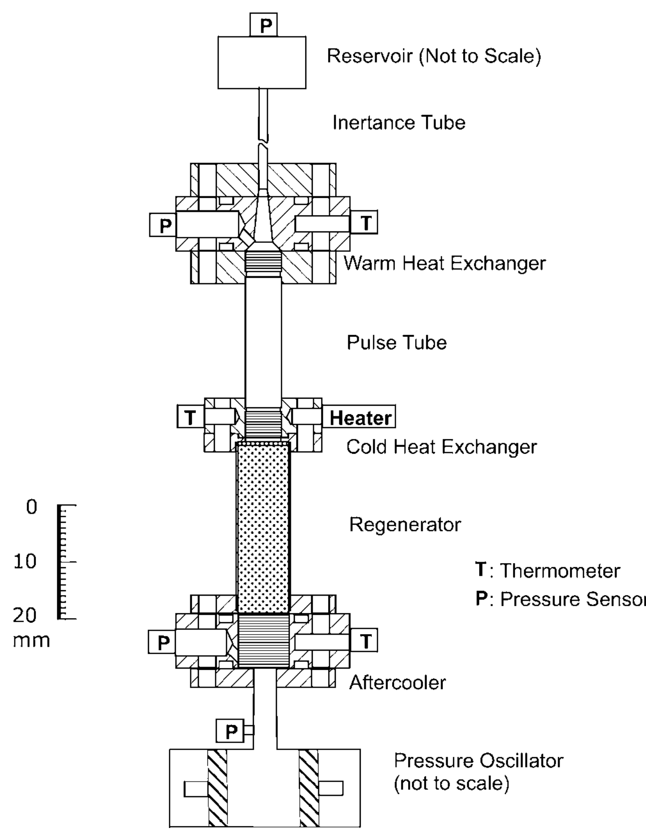

FIG. 1. Layout drawing of the in-line pulse tube cryocooler used in the experiments.

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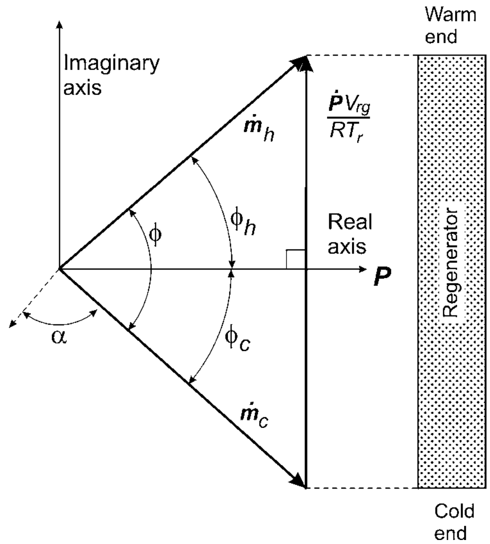

FIG. 2. Phasor diagram representing conservation of mass in the regenerator.

ture pulse tube cryocoolers fabricated with microelectromechanical system technologies.

Figure 2 shows a phasor representation of mass conservation within the regenerator of a typical inertance type of pulse tube cryocooler, where the pressure at the cold end is used as the reference phasor along the real axis. Because of the gas volume V_{rg} in the regenerator, the conservation of mass requires that the flow at the warm end will lead the flow at the cold end, and is given by

$$\dot{\boldsymbol{m}}_h = \dot{\boldsymbol{m}}_c + \frac{\dot{\boldsymbol{P}}V_{rg}}{RT_r}, \tag{1}$$

where the bold variables represent time varying or phasor quantities, $\dot{\boldsymbol{m}}_c$ is the flow rate at the cold end, R is the gas constant per unit mass, T_r is the mean temperature of the regenerator, and $\dot{\boldsymbol{P}}$ is the rate of change of pressure in the regenerator, given by

$$\dot{\boldsymbol{P}} = i2\pi f\boldsymbol{P}, \tag{2}$$

where i is the imaginary unit and $\boldsymbol{P}$ is the dynamic pressure. This phasor leads the pressure by 90°, as indicated by the vertical phasor along the imaginary axis. According to Eq. (2) $\dot{\boldsymbol{P}}$ and the vertical phasor in Fig. 2 increase with frequency. Thus, at very high frequencies the magnitude of the flow at the warm end becomes very large for a fixed flow at the cold end. In that case the regenerator losses become high. Because $\dot{\boldsymbol{P}}$ is proportional to frequency [see Eq. (2)] V_{rg} must be reduced as frequency increases in order to keep the vertical phasor from increasing significantly. The phase angle at the hot end is proportional to the ratio of the vertical phasor to the average flow rate $\dot{\boldsymbol{m}}_1$, which for the optimum hydraulic diameter can be written as[3]

$$\frac{\dot{P}V_{rg}}{\dot{m}_1 RT_r} \propto \frac{f}{P_0(P_1/P_0)^2}, \tag{3}$$

where P_0 is the average pressure and P_1 is the amplitude of the dynamic pressure. Note that an increase in P_0 or P_1/P_0 can compensate for an increase in f to help maintain a constant ratio in Eq. (3). As $\dot{\boldsymbol{P}}$ increases with frequency, V_{rg} must be reduced. To transfer the same amount of heat in the smaller gas, volume requires the surface area to remain constant, which can be achieved only by decreasing the hydraulic diameter.

The National Institute of Standards and Technology numerical model known as REGEN 3.2 (Refs. 4 and 5) was used for optimization of the regenerator. A filling pressure of 3.5 MPa, a pressure ratio of 1.3 at the cold end, and an operating frequency of 120 Hz were used in the calculations. The dimensions of the regenerator were varied to determine the maximum coefficient of performance, which is the ratio of net refrigeration power to the acoustic power at the warm end of the regenerator. The pulse tube volume was chosen to be three times the swept volume at the cold end to ensure thermal decoupling between the two ends. The inertance tube dimensions were determined from a transmission line model.[6] The optimized dimensions for the regenerator are 9.02 mm inner diameter, 30 mm length, filled with #635 mesh (finest commercially available), stainless steel screen of 20.3 μm wire diameter, and a porosity of 0.60. The pulse tube has a nominal inner diameter of 4.46 mm and a length of 30 mm. The dimensions of the inertance tubes are as follows: tube 1, connected to the pulse tube, is 1.36 mm nominal inner diameter by 0.86 m long; tube 2, connected to the reservoir, is 1.76 mm nominal inner diameter by 0.75 m long. The nominal volume of the reservoir is 50 cm^3.

Experiments were performed on the designed cryocooler powered by a pressure oscillator that could deliver a maximum pressure-volume (PV) power of 500 W. Piezoresistive pressure sensors were used to measure the average and oscillating pressures at the pressure oscillator, aftercooler and warm heat exchanger. The pressure difference between the pressure oscillator and the aftercooler was used to determine the mass flow and PV power into the aftercooler. A piezoelectric pressure sensor was used to measure the oscillating pressure at the reservoir to determine mass flow at the reservoir entrance. Diode thermometers with an accuracy of ±0.25 K were used to measure temperatures at the aftercooler, cold heat exchanger, and warm heat exchanger. Lock-in amplifiers were used to measure the oscillating pressure signals.

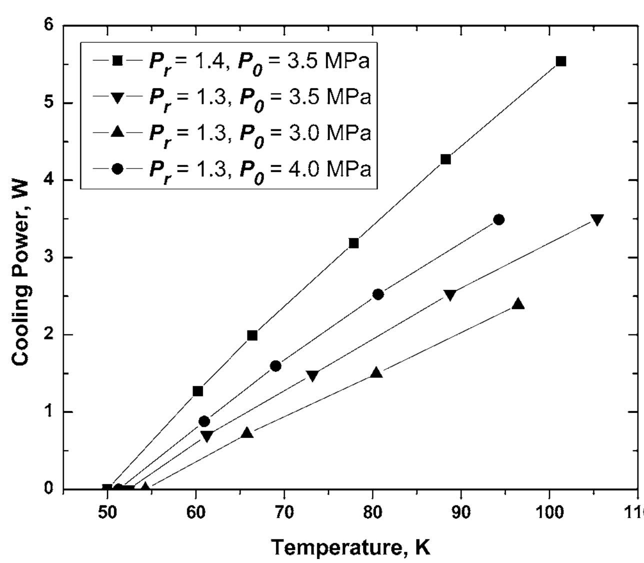

FIG. 3. Net refrigeration power of the pulse tube cryocooler with different warm-end pressure ratios and average pressure.

The no-load temperature of 49.9 K was reached with an average pressure of 3.5 MPa, a pressure ratio P_r of 1.4 at the aftercooler, and a frequency of 120 Hz. The net refrigeration power as a function of temperature is shown in Fig. 3, for the conditions given above and several other conditions. The net refrigeration power at 80 K is 3.35 W with an input PV power at the aftercooler of 46.4 W. For an aftercooler temperature of 298.7 K the efficiency is about 19.7% of Carnot. To compare the performance of the cryocooler with filling pressure, tests were conducted with filling pressures of 3.0, 3.5, and 4.0 MPa, keeping the pressure ratio and frequency fixed at 1.3 and 120 Hz, respectively. The slopes of the load curves increase with average pressure, which is expected because of the increase in pressure amplitude and acoustic power. The total background loss, which includes radiation loss and heat conduction through the regenerator matrix and the tube walls, was determined to be about 0.9 W from the slope of the heating curve at various loads. The cooldown times to 80 and 50 K were about 5.5 and 11 min, respectively. This very fast cooldown time is due to the high power density in the small regenerator volume used at high frequencies.

We have shown that a pulse tube cryocooler can be operated at a frequency as high as 120 Hz and still reach a low temperature of 50 K. The cooldown time to 80 K was 5.5 min and the net refrigeration power at 80 K was 3.35 W. The efficiency at 80 K was about 19.7% of Carnot, when referred to the input PV power, and nearly as high as the most efficient pulse tube cryocoolers operating around 60 Hz.[1] The high efficiency at such high frequencies was made possible by using a higher average pressure and a finer screen mesh than that normally used at 60 Hz. The high frequency operation allows for a smaller pressure oscillator and a faster cooldown time.

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