

# Theoretical and Experimental Study on a Pulse Tube Cryocooler Driven with a Linear Compressor

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## ABSTRACT

A single-stage Stirling-type pulse tube cryocooler (PTC) has recently been designed using the REGEN modeling software developed at NIST Boulder. REGEN 3.2. The REGEN-based approach is based on a finite difference method to solve the mass, energy, and momentum conservation equations, used to optimize the regenerator. The regenerator operates at a charge pressure of 2.0 - 2.5 MPa and a frequency of 60 Hz. According to the REGEN3.2 calculation, a regenerator with a length of 45 mm and a diameter of 15 mm is capable of providing a cooling power above 5 W. An inertance tube comprised of two sections is used to regulate the phase between the pressure and mass flow rate at the hot end of the pulse tube. Its geometry is based on a transmission line model, named INERTANCE, developed by Radebaugh of NIST.

The single-stage pulse tube cryocooler built according to the above calculation achieved the lowest temperature of 57.0 K, and a cooling capacity of 5.0 W at 79.1 K. This performance agrees well with the design prediction.

Furthermore, the object-oriented software Sage has been used to simulate the pulse tube cryocooler performance after running the experiment. The results compare well with the experiment and provide the opportunity for further optimization of the inertance tube.

## INTRODUCTION

Pulse tube cryocoolers have attracted the enthusiastic attention of a large number of researchers and have been used in space technology and high-tech application because of their lack of moving parts at the cold end, simple construction, long operating life and low vibration.

Due to the development of the linear compressor with flexure bearings and clearance seals, research on high frequency pulse tube cryocoolers driven by a linear compressor have also received considerable attention. The space cryocoolers technology with cooling temperatures above 55 K has matured in the USA.<sup>1</sup> In order to meet the further requirement of miniaturization for space applications, it is worthwhile to pursue the development of pulse tube cryocoolers operating at higher frequencies (>60 Hz).<sup>2,3</sup> The development of a 4 K space pulse tube cryocooler is also important.<sup>4,5</sup> Both directions require basic research on the performance of pulse tube cryocoolers at

80 K with a frequency of 60 Hz. This represents our starting point for research at lower temperatures and higher frequencies.

Through the use of the REGEN 3.2 and INERTANCE codes developed at NIST, we have designed and fabricated a single-stage Stirling-type pulse tube cryocooler operating at 60 Hz. The experimental performance measurements agree quite well with the design predictions.

The modeling software, Sage, developed by Gedeon Associates is also used to simulate the pulse tube cryocooler. The simulation results compare well with the experiment. Furthermore, Sage has been used to identify possible performance improvements that can be realized through modifications to the inertance tube. The associated phase analysis for the optimization appears reasonable.

## DESIGN OF PTC

Beginning with an input power of 280 W for a linear compressor, a pulse tube cryocooler is designed with the goal of producing 10 W of cooling power at 80 K. For this relatively small-size PTC, a large charge pressure (2.5 MPa) and high frequency (60 Hz) have been chosen. A brief review of the design procedure is given below.

### Regenerator

The regenerator is the key component of the PTC, as it directly effects the thermodynamic performance of the cryocooler. In this article, we use the analysis tool REGEN3.2<sup>6,7</sup> to design the regenerator.

The material of the regenerative matrix represents the first design choice. Since the thermal penetration depth of helium under conditions of 60 Hz, 2.5 MPa and 80K is 88 mm, we chose #400 mesh stainless steel screen whose hydraulic diameter is 55 mm as the matrix of the regenerator.

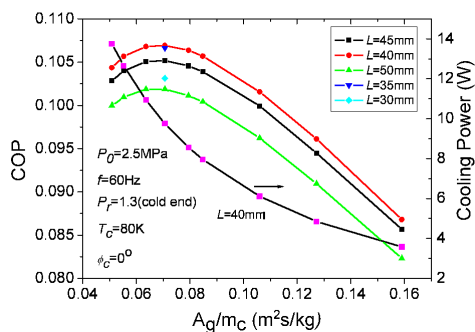
Secondly the optimum length was determined with respect to the highest COP. Figure 1 displays curves of the COP for various values of regenerator length, as a function of the gas cross-sectional area to the mass flow rate at the cold end.

A regenerator length of 45 mm is chosen after taking into account axial conduction losses and welding requirements. According to the design goal of 10 W of refrigeration capacity and considering the commercial availability of thin-walled tubes, a 15.4 mm regenerator diameter is selected.

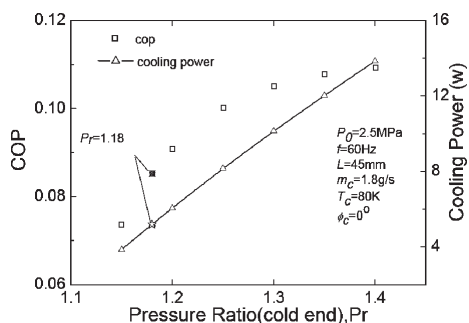
Figure 2 shows the COP and cooling power as a function of the cold-end pressure ratio with the optimal mass flow of 1.8 g/s at the cold end. Note that only 5.20 W of cooling power is predicted if the pressure ratio is 1.18 rather than 1.3.

### Pulse tube

For an ideal pulse tube: (1) the compression and expansion processes in the pulse tube must be adiabatic; (2) the swept volume of the fictitious gas piston within the pulse tube must be much smaller than that of the pulse tube.<sup>8</sup>



**Figure 1.** COP versus length of the regenerator and cooling power at 80 K for a specified  $A_g$  with a variable  $m_c$ .



**Figure 2.** COP versus pressure ratio at cold end

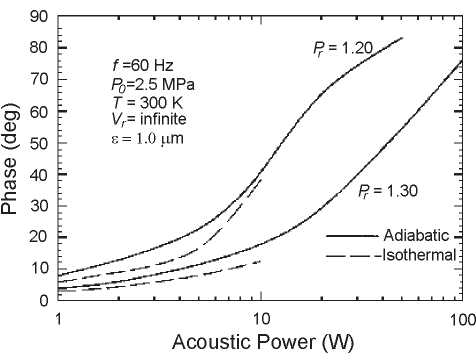
The thermal penetration depth of helium with a frequency of 60 Hz, an average pressure of 2.5 MPa and a temperature of 300 K is 0.185 mm. This value becomes smaller at lower temperature. Thus, the adiabatic condition is satisfied if the radius of pulse tube is 10 times greater than 0.185 mm. The second condition is met if the pulse tube volume is more than 3 times the swept volume of the gas piston at the cold end. According to the optimized case as determined by REGEN3.2, the swept volume of the gas piston is 0.675 cm<sup>3</sup>. Thus, the volume of pulse tube must be at least 2.03 cm<sup>3</sup>. A larger pulse tube volume leads to a larger phase angle that must be produced by the inertance tube. The length of the pulse tube and regenerator are identical due to the choice of a U-type configuration. A thin-walled stainless steel tube with an inner diameter of 9 mm was chosen for the pulse tube. With this dimension, the ratio of the pulse tube radius to the thermal penetration depth is 24.3, and the volume ratio  $V/V_e$  equals 4.3.

**Inertance tube**

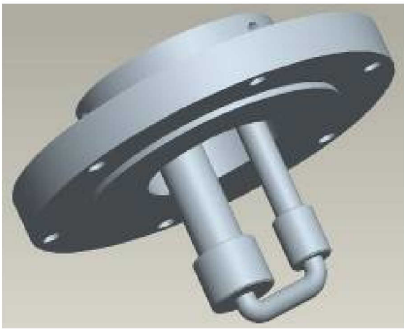
Because the performance of the PTC is strongly influenced by the phase relationship between pressure and flow oscillations, the inertance tube design is an important step in the overall design of the PTC.

Considering the limited acoustic power (for this design, REGEN3.2 predicts 20 W of acoustic power at the cold end), and the associated limitation on the possible phase shift given by inertance tube which is less than 30° with the pressure ratio at 1.3, as shown in Figure 3, we chose a phase angle of zero degree between the mass flow and pressure oscillations at the cold end of the regenerator. The corresponding phase shift required at the interface of the inertance tube and pulse tube is -38°. It is difficult to obtain even a -38° phase shift through the use of a single diameter tube<sup>9</sup>, but by using a two-sectional inertance tube with different diameters a -34° phase shift may be obtained, thereby closely satisfying the design requirement.

Due to limitations associated with the linear compressor used for the test, the pressure ratio at the cold end of the regenerator was only 1.18. In this situation, the required phase shift at the entrance of inertance tube is only -26.3°. Such an adjustment is easy to obtain with the inertance



**Figure 3.** Maximum phase shift of single-sectional inertance tube vs. acoustic power<sup>9</sup>



**Figure 4.** Assembled drawing of the main components of the PTC.

**Table 1.** Geometric parameter of the cryocooler.

		Length (mm)	Outer diameter (mm)	thickness (mm)
Regenerator		45	15.9	0.254
Pulse tube		45	9.53	0.254
Inertance tube	Small	595	3	0.6
	Large	695	4	0.75
#400 stainless steel screen		Wire diameter	Porosity	Hydraulic diameter
		25.4 $\mu$ m	0.6858	55.44 $\mu$ m
Reservoir volume		250 cc		

tube. According to the design method mentioned above, the calculated dimensions of the cryo-cooler are summarized and shown in Table 1.

EXPERIMENT RESULT

The fabricated single-stage Stirling-type pulse tube cryocooler consists of the pulse tube, re-generator, inertance tube, reservoir and linear compressor. The main parts are shown in Figure 4. The linear compressor used for the experiment is a modified K535 Stirling cooler made by Ricor, Israel. The compressor input power and frequency are modulated by a frequency transformer.

Figure 5 shows a typical cool-down process for the pulse tube cryocooler with the charge pressure of 2.5 MPa and frequency of 60 Hz. It takes about 8.5 minutes to cool down to 80 K and the lowest temperature obtained is 57.0 K. The measured pressure ratio is 1.25, at the inlet of aftercooler and 1.18 at the warm end of pulse tube (essentially equivalent to the value at the cold end of the pulse tube) rather than the design value of 1.30.

Figure 6 shows the measured curve of cooling capacity versus temperature. The cooling power is 5.0 W at 79.1 K with an input electric power of 290 W. The calculated result by REGN3.2 with pressure ratio of 1.18 at the cold end, is 5.20 W at 80K (as shown in Figure 2) and is quite close to the measured results. It is reasonable to expect that a cooling capacity of 10 W at 80 K could be obtained if the linear compressor were capable of providing a pressure ratio of 1.30 at cold end.

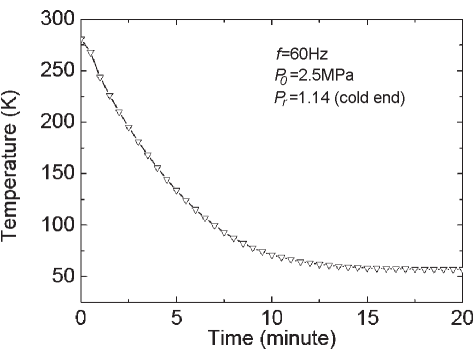


Figure 5. Cooling down process of the PTC.

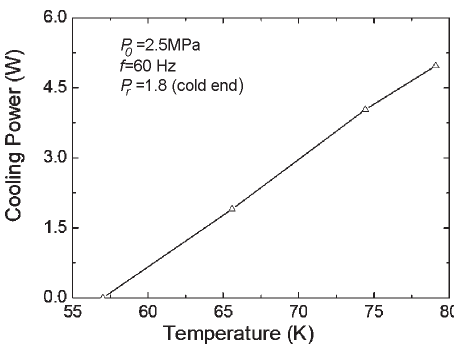


Figure 6. Cooling power vs. temperature.

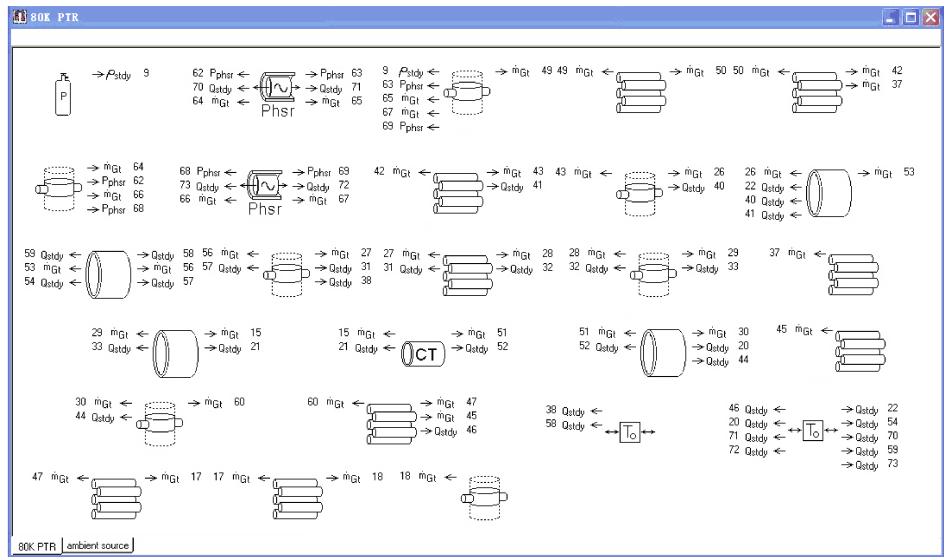


Figure 7. Sage model scheme of the single stage pulse tube cryocooler.

SAGE SIMULATION

Subsequent to the experimental measurements, Sage was used to simulate the performance of the same pulse tube cryocooler. The scheme of the single-stage PTC model is presented in Figure 7. The simulation results presented below are in good agreement with the experimental measurements.

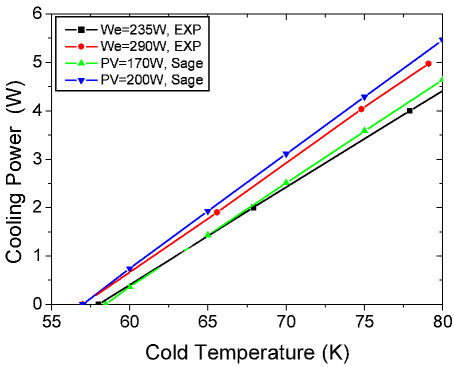
Figure 8 compares the simulation results with the corresponding experimental measurements under the design condition of 2.5 MPa charge pressure and 60 Hz frequency. Note that while the electric power consumed by the compressor was measured, the associated electric-to-acoustic conversion efficiency is not precisely known. However, a reasonable conversion efficiency of 70% results in an excellent agreement between the measured and simulation results. The corresponding amplitudes of the compressor piston for the 170 W and 200 W simulation results are 2.95 mm and 3.2 mm respectively.

The performance of the PTC was also investigated as a function of the charge pressure. Figure 9 compares the experimental results and the Sage simulation.

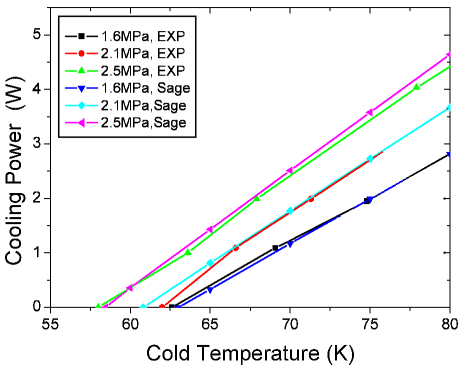
The input electric power of the linear compressor does not vary significantly as the charge pressure changes. The results shown in Figure 9 correspond to values of 235 W, 240 W and 230 W as the pressure is increased. The corresponding amplitudes of the compressor piston in the simulation are 3.8 mm, 3.25 mm and 2.95 mm, and the respective PV power in the compression space is 180 W, 175 W and 170 W. The calculated conversion efficiency of electric power to PV power for the compressor is 78%, 73% and 72% respectively.

As the charge pressure increases, the cooling power also increases. The same trend is evidenced in Sage by an increasing mass flow amplitude at the cold end of the regenerator. At 80 K, the corresponding cold end mass flow rates are 2.34 g/s, 2.80 g/s and 3.22 g/s respectively.

The rated charge pressure of the Ricor K535 Stirling Cryocooler is 1.6 MPa. This pressure is possibly a resonance condition for the compressor, since as the charge pressure is increased above this value, the efficiency decreases.



**Figure 8.** Tracking between Sage and experiment for cooling power with different input power.



**Figure 9.** Tracking between Sage and experiment for cooling power with different charging pressure.

**Table 2a.** Comparison of the experiment simulation and optimized case.

	Q@80K	$\dot{m}_{comp}$	$\dot{m}_c$	$\dot{m}_i$	Pr1	Pr2
EXP. SIMU.	5.46 W	3.62 g/s	3.41 g/s	0.99 g/s	1.36	1.19
OPT. INER.	7.98 W	3.53 g/s	3.08 g/s	0.95 g/s	1.40	1.24

Table 2b. Comparison of the experiment simulation and optimized case.

	P.Drop Amp	AEfric	AEQw	AEQx	TotalAE
EXP. SIMU.	0.118 MPa	50.98 W	15.0 W	2.77 W	68.73 W
OPT. INER.	0.107 MPa	43.51 W	14.5 W	2.62 W	60.66 W

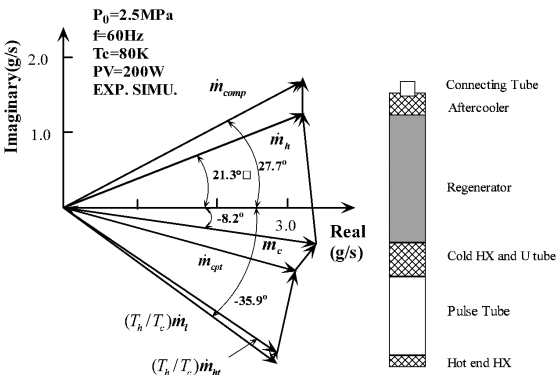


Figure 10. The simulated phase relationship of mass flow relative to pressure at cold end for experiment simulation.

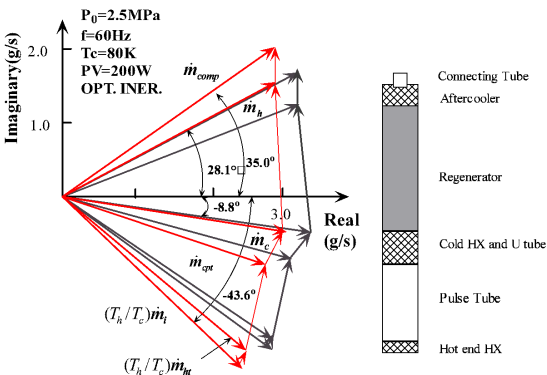


Figure 11. The simulated phase relationship (red line) of mass flow relative to pressure at cold end for an optimized case.

OPTIMIZATION OF INERTANCE TUBE

Since it is convenient to modify the inertance tube of a fabricated PTC, we have further investigated the influence of changes to the inertance tube geometry while keeping all the other components and parameters the same. The built-in optimization function in Sage is used with both the length and diameter of the two-sectional inertance tube as the optimization parameters. The calculated optimum case has a higher inertance tube impedance.

As Table 2a shows, the optimized cooling power exceeds the experimental case by nearly 50% at 80 K. Here Pr1 refers to the pressure ratio at the entrance of the aftercooler while Pr2 is the pressure ratio at the inlet of the inertance tube. The pressure ratio increases with the optimized case and the decreased mass flow results in a lower entropy generation, which should account for the improvement of performance.

As Table 2b shows, for the optimized case, a lower pressure drop in the regenerator, a lower available energy loss due to viscous flow friction (AEfric), non-ideal heat exchange with matrix solid (AEQw), and axial heat flow (AEQx) is obtained.

The phase diagrams generated by Sage for the same two cases are shown in Figure 10 and Figure 11. A close examination reveals that the average mass flow in the regenerator is almost in phase with pressure, a favorable condition for minimizing losses.

Thus, the design of the inertance tube is good. The optimized inertance tube provides a larger phase shift, as high as 43.6 degree rather than the 35.9 degree value associated with the experimental case.

The vertical lines in the phase diagram correspond to  $\dot{P}V/RT_a$  for the mass storage term in the regenerator and  $\dot{P}V/\gamma RT_a$  for the energy storage term in the pulse tube. Notice that as the pressure ratio increases and mass flow amplitude decreases, the phase difference between the mass flow at the compressor and that at the inertance tube will increase and a larger phase shift is required. If the inertance tube has a lower capability of phase shift, the phase angle of mass flow at the entrance of the inertance tube will be higher, the mass flow amplitude will become larger and the pressure ratio lower, assumed that the PV power provided by the compressor is fixed. Just notice the equation  $PV = P_{amp} \dot{m} \cos(\theta)/\rho_a$ . The model simulating experimental results given in Figure 10 shows that the inertance phase shift and impedance magnitude are less than that calculated by the optimized model for the same PV power, where the PV power is based on the equation:  $PV = 1/2 P_{amp} \dot{m} \cos(\theta)/\rho_a$ .

## CONCLUSION

A single-stage Stirling-type pulse tube cryocooler is designed and fabricated by means of REGEN 3.2 and INERTANCE. A cooling capacity of 5.0 W at 79.1 K has been measured, in good agreement with design predictions. In addition, a simulation developed with Sage also is in good agreement with the experimental measurements. Further performance improvements are suggested by optimizing the inertance tube, thereby providing guidance for future experiments. The associated phase analysis confirms the possibility for further performance improvements.

## ACKNOWLEDGMENT

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