Numerical Study on the Performance Sensitivity of a Pulse Tube Cryocooler with Active Phase Shifter

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Numerical Study on the Performance Sensitivity of a Pulse Tube Cryocooler with Active Phase Shifter

K. Liang [1], **A. Abolghasemi** [2], **C.R. Stone** [2], **M.W. Dadd** [2], **P.B. Bailey** [2]

[1]Department of Engineering and Design, University of Sussex, Brighton, UK BN1 9QT
[2]Department of Engineering Science, University of Oxford, Oxford, UK OX1 3PJ

kun.liang@sussex.ac.uk

Abstract. Pulse tube cryocoolers have been widely used for cooling infrared sensors, superconducting and gas liquefaction. An active phase shifter allows the phase angle between the pressure and the mass flow rate to be varied, thus high efficiency and better performance can be achieved. In this work, a CFD technique using dynamic meshing was adopted to investigate the oscillating flow and performance sensitivity of an in-line pulse tube cryocooler with an active phase shifter. A high performance computer (HPC) was used for the 2D simulation. The effect of phase angle, operating frequency, pressure drop, and fill pressure were studied. The pulse tube cryocooler appears to have an optimal operating frequency (55 Hz) and displacement phase angle (40 deg) between compressor and displacer. A fill pressure of 28 bar leads to better performance than 24 bar and 20 bar. The CFD results are validated by the experimental results of a prototype in-line pulse tube cryocooler.

1. Introduction

Stirling pulse tube cryocoolers are used in industrial applications such as semiconductor fabrication and in military applications such as infrared detectors. The thermodynamics of Stirling pulse tube cryocoolers are complex and are not always intuitively understood. The displacer piston (as can be found in the Stirling and GM cryocoolers) is eliminated. A key requirement for pulse tube cryocooler is to ensure an optimal phase angle between mass flow and the pressure pulse at the cold end [1]. The adjustment of the phase angle can be achieved by using an orifice plate, an inertance tube or a warm end displacer. The orifice plate or inertance tube does not have any moving components in the cold head assembly. However, it is difficult to ensure a correct phase angle between mass flow and pressure at the cold end. A warm end displacer allows this angle to be controlled. Moreover, the expansion power at the warm end, which is otherwise dissipated as heat when using an orifice plate or an inertance tube, can be recovered via the displacer. This can further improve the efficiency of the pulse tube [2]. A number of studies on pulse tube cryocoolers using CFD can be found in [3, 4, and 5].

An in-line pulse tube cryocooler with an active displacer (phase shifter) has been developed at the Cryogenics Engineering Group at the University of Oxford in collaboration with Honeywell Hymatic. In this study a CFD study on the sensitivity of performance of the pulse tube is presented which is validated by experimental results on the prototype pulse tube. The verified CFD model can then be used to predict the performance with improved design.

2. CFD Model

A CFD software (FLUENT) was used to simulate the flow and heat transfer in the pulse tube cryocooler. The model is 2D axis-symmetric as shown in Figure 1. The pulse tube cryocooler with phase shifter consists of: the linear compressor, connecting tube 1, aftercooler, regenerator, cold end heat exchanger, pulse tube, warm end heat exchanger, connecting tube 2, and displacer. The parameters of the pulse tube cryocooler are shown in Table 1.

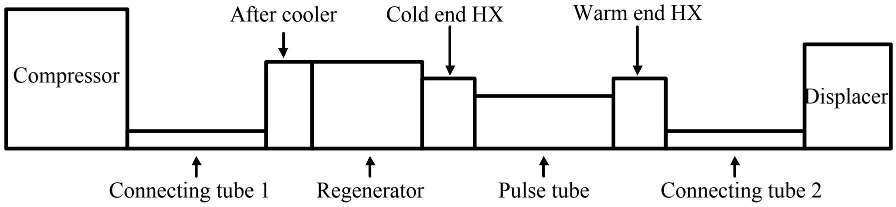

Figure 1. Schematic of the CFD model of the pulse tube cryocooler with an active displacer

Table 1. Parameters of the pulse tube cryocooler with active displacer

Items	Geometry/specification
Linear compressor	2 pistons (19 mm diameter), maximum stroke 9 mm each
Connecting tube 1	Length 45 mm with inner diameter 5 mm
After cooler (HX1)	Water cooling, length 13.7 mm, diameter 12.7 mm
Regenerator	Stainless steel, 400*0.025 mm mesh, diameter 15.5 mm, length 46 mm
Cold end HX (HX2)	Copper, 50*0.19 mm mesh, diameter 13 mm, length 5 mm
Pulse tube	Inner diameter 7.7 mm, length 85 mm
Warm end HX (HX3)	Copper, 50*0.19 mm mesh, diameter 12.7 mm, length 5 mm
Connecting tube 2	Total length 350 mm with inner diameter 2 mm
Displacer	Diameter 18 mm, maximum stroke 4 mm

In FLUENT, the conservation equations of mass, momentum and velocity are solved using a finite volume method. The flow was calculated by the SIMPLE method [6]. A second order upwind differential scheme was applied for the approximation of the convective terms. The number of cells for the study is about 75,000 after a mesh independence study. The regenerator and the heat exchangers were modelled as porous zones. In FLUENT, porous media are modelled by the addition of a momentum source term to the fluid flow equations. The source term consists of viscous loss term and an inertial loss term. Table 2 shows the boundary conditions of the CFD model. The compressor and the displacer were modelled using a dynamic mesh. A user defined function (UDF) was written to define the motion of the pistons with a phase difference. The working fluid is helium which was modelled as an ideal gas with temperature dependent viscosity and thermal conductivity. The thermal conductivity and specific heat capacity of the regenerator materials were also temperature dependent. The warm end heat exchanger and aftercooler were set at 300 K.

Table 2. Boundary conditions of the CFD model for the pulse tube cryocooler

Linear compressor	Moving piston with UDF for displacement
Aftercooler	Wall temperature of 300 K
Regenerator	Adiabatic
Cold end heat exchanger	Adiabatic / fixed temperatures (50 K -120 K)
Pulse tube	Adiabatic
Warm end heat exchanger	Wall temperature of 300 K
Displacer	Moving piston with UDF for displacement

Figure 2 plots the temperature contours of the pulse tube cryocooler without showing the compressor and displacer. The minimum temperature at the cold end is about 50 K with no load. The thermal load

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was calculated based on the heat flux across the cold end surface area when the cold end temperature was fixed. The CFD model was run for cold end temperatures of 5 – 120 K, and phase angles between two pistons of 35 – 50 deg. The frequency was fixed at 55 Hz.

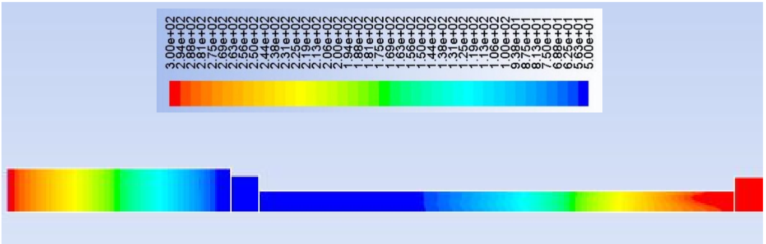

Figure 2. Temperature contours of the pulse tube from CFD model with no load (compressor and displacer not shown)

3. Measurement

Figure 3 shows the schematic of the pulse tube instrumentation. The data acquisition (DAQ) and control system was based on LabVIEW. A NI 6251 DAQ card was adopted for acquiring pressure, voltage, current and displacement signals as well as generating two waveforms with different phase angles to drive the compressor and displacer through an HH power amplifier. Two power meters were used to measure the electrical power into the compressor and displacer. A Lakeshore temperature PID controller was used to maintain a user defined cold end temperature by adjusting the heater power. The cold end temperature was measured by a platinum resistor thermometer (PRT) and a silicon diode cryogenic temperature sensor. Temperatures at other locations within the system were recorded using T-type thermocouples. Three linear variable differential transformer (LVDT) displacement transducers were used to measure the position of the pistons for both the compressor and the active displacer. A phase meter measured the phase difference between the displacements of the two pistons. Figure 4 shows the prototype pulse tube cryocooler in the laboratory.

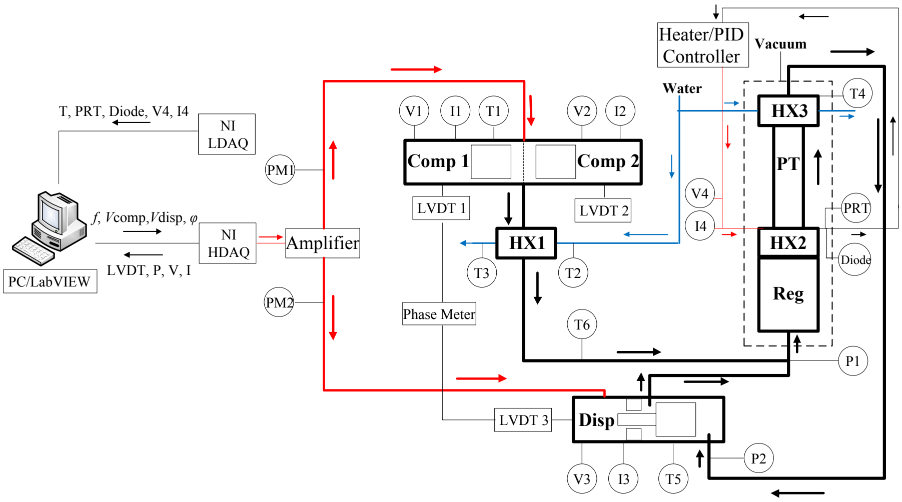

Figure 3. Schematic of the pulse tube instrumentation

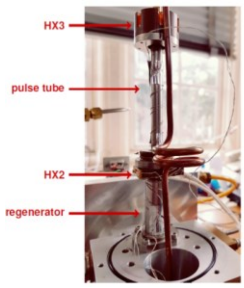

Figure 4. The prototype pulse tube cryocooler with active displacer

The measurements were conducted for fill pressures of 20 bar, 24 bar, and 28 bar, frequencies of 50-65 Hz, phase angles of 38-48 deg and cold end temperatures of 50-120 K.

4. Results and Discussions

Figure 5 shows an example of measured displacements for the compressor and displacer with a phase angle of 40 deg. The strokes for the compressor and displacer are 9 mm and 2.4 mm respectively. These two waveforms were transferred to the UDF in FLUENT to define the dynamic mesh.

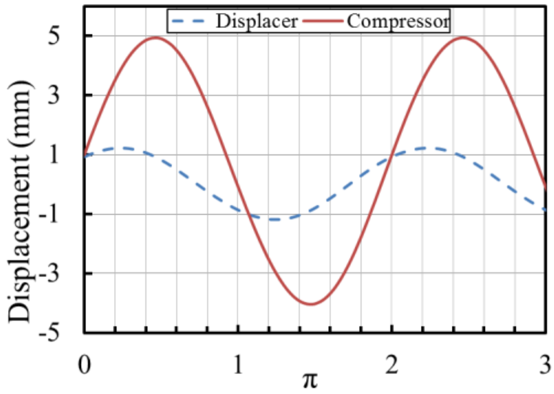

Figure 5. Displacement of the compressor and displacer

The pressure and mass flow at the inlet and outlet of the regenerator are shown in Figure 6 for a fill pressure of 28 bar, a frequency of 55 Hz, a cold end temperature of 80 K and a phase angle of 40 deg. The pressure drop across the regenerator is 0.6 bar which is the dominant loss in the pulse tube cryocooler. The mass flow is almost in phase with the pressure at the inlet of the cold end (outlet of the regenerator).

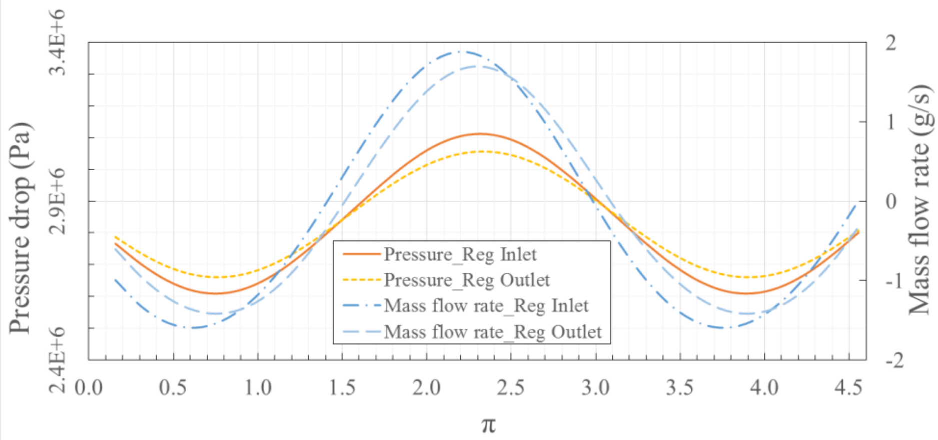

Figure 6. Pressure and mass flow rate at the inlet and outlet of the regenerator

The cooling capacity increases linearly with cold end temperature as shown in Figure 7 at 28 bar, 55 Hz and 40 deg. The CFD model agrees very well with the measurements. The cooling capacity at 80 K is 2.7 W. The relative Carnot efficiency from measurements in Figure 8 shows that 55 Hz appears to be the optimal frequency for 28 bar operation of the pulse tube cryocooler. Note that the resonant frequency for the linear compressor at 28 bar and 9 mm is 60 Hz. The relative Carnot efficiency is much higher at 28 bar than at 24 bar and 20 bar. The optimal frequency does not change noticeably with fill pressure.

The performance sensitivity to phase angle is shown in Figure 9. Both the CFD model and measurements indicate an optimal phase angle of 40 deg for 28 bar, 55 Hz and 80 K. When the phase angle increases, the shaft power required also increases as the expansion work increases. However, a higher phase angle does not vary the cooling capacity by much as the loss in the regenerator increases. A few degrees variation in phase angle is not crucial to the operation of the pulse tube cryocooler.

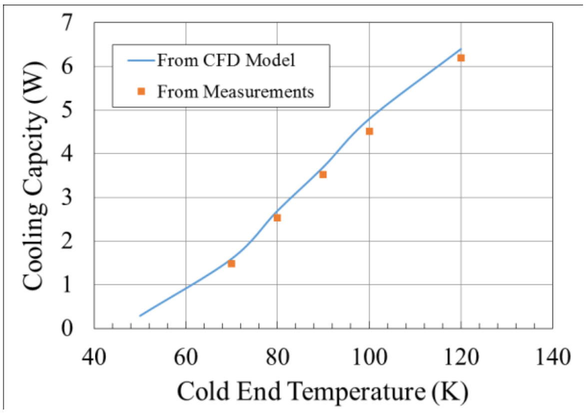

Figure 7. Cooling capacity vs. cold end temperature

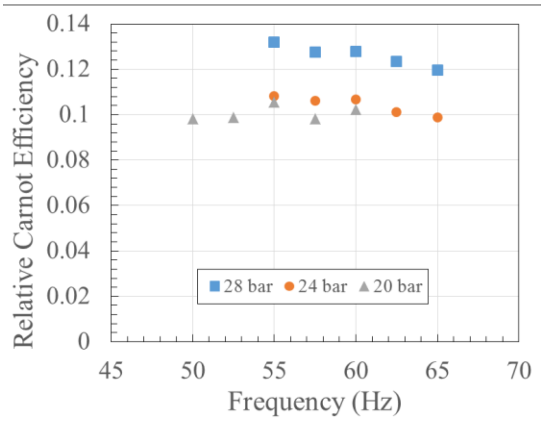

Figure 8. Relative Carnot efficiency vs. frequency

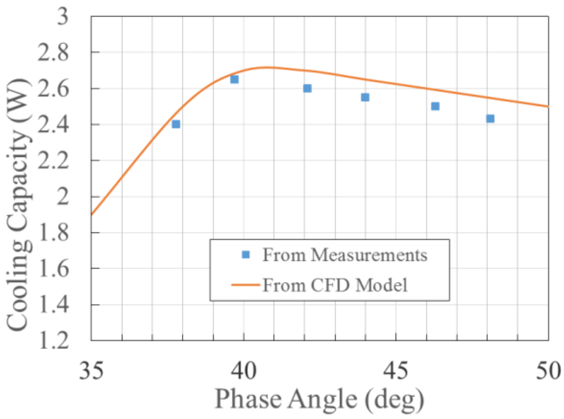

Figure 9. Cooling capacity against phase angle

5. Conclusions and Future Work

A 2D axis-symmetric CFD technique using a dynamic mesh was adopted to investigate the performance sensitivity of an in-line pulse tube cryocooler with an active phase shifter. The effect of phase angle, operating frequency, pressure drop, and fill pressure were studied. The pulse tube cryocooler appears to have an optimal operating frequency (55 Hz) and displacement phase angle (40 deg) between the compressor and displacer. A fill pressure of 28 bar leads to better performance than at 24 bar or 20 bar. The CFD results are validated by the experimental results of a prototype in-line pulse tube cryocooler. Future works includes CFD simulations with different frequency and fill pressures and optimization of the warm end geometry by using CFD.

6. References

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