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Computational fluid dynamics simulations of an orifice type pulse tube refrigerator: Effects of operating frequency

Dion Savio Antao, Bakhtier Farouk *

Department of Mechanical Engineering and Mechanics, Drexel University, Philadelphia, PA 19104, USA

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ABSTRACT

A numerical study is reported here for the investigation of the fundamental flow and heat transfer processes found in an orifice type pulse tube refrigerator (OPTR). The OPTR is driven by a cyclically moving piston at one end of the system with helium as the working fluid. The regenerator and the various heat exchangers are modeled as porous media and a thermal non-equilibrium model is applied in these regions. The system is studied for different operating frequencies of the driver piston. The simulations reveal interesting steady-periodic flow patterns that develop in the pulse tube due to the fluctuations caused by the piston and the presence of the inertance tube. The predicted secondary-flow recirculation patterns in the pulse tube are found to affect the OPTR performance. When the secondary-flow patterns are well-developed, they help isolate the cold and hot ends of the pulse tube and create a thermal buffer zone at the center of the pulse tube, enhancing the performance of the OPTR.

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1. Introduction

Thermoacoustic refrigerators have attracted renewed attention lately as they run on gases like helium argon, nitrogen or air and are able to attain cryogenic temperatures without the use of expensive cryogens like liquid helium or liquid nitrogen [1]. The small number of moving parts required makes them simple and hence potentially reliable. The pulse tube refrigerator is an interesting thermoacoustic device and was first discovered and reported by Gifford and Longsworth in 1964 [2,3]. They named the device 'pulse tube refrigerator' as the displacer (found in Stirling and Gifford–McMahon type refrigerators) was replaced by a hollow tube. This initial pulse tube refrigerator design is now called a Basic Pulse Tube refrigerator (BPTR) [4]. Initial experiments using Helium as the working gas resulted in cold temperatures of around 169 K for a single stage and 123 K for a double-stage cryocooler [5]. In 1984, Mikulin [6] demonstrated that the phase and amplitude relation between velocity and temperature can be managed by controlling the boundary conditions at the end of the pulse tube. This was done by placing an orifice and a reservoir (compliance/surge) volume at the end of the BPTR thus allowing a finite gas flow. The presence of the orifice changes the phase angle between the velocity and temperature at the cold end and increases enthalpy flow at the hot end. Mikulin's initial experiments attained 105 K with air as the working gas. This was the first OPTR. The OPTR was shown to result in lower temperatures, increased cooling and higher efficiencies than the BPTR.

Radebaugh at NIST (Boulder, CO) worked on the OPTR as it is known today – changing the location of the orifice compared to Mikulin's system. His initial work recorded temperatures around 60 K [7]. Recent variations of the OPTR include the addition of an extra orifice creating a double-inlet pulse tube and the addition of an inertance tube. The double-inlet allows the gas to be compressed from the hot end of the tube also, thus increasing the phase angle and reducing the flow through the regenerator (which reduces enthalpy flow losses). The inertance tube can be used instead of an orifice or in addition to one. Marquardt and Radebaugh reported the highest efficiency for the pulse tube refrigerator when using a combination of the orifice and the inertance tube [8].

In this paper, we report time-dependent axisymmetric CFD simulations where the transient as well as the cycle-averaged operation of an OPTR is studied for the processes occurring in an in-line orifice type pulse tube refrigerator. The compressible form of the Navier–Stokes equations is considered for the flow simulations. For the porous media regions (i.e. the regenerator and the heat exchangers), we employ the thermal non-equilibrium porous media model [9,10]. In the non-equilibrium model, the gas and the solid temperatures are different in the porous media. The effect of heat transfer between the gas and the solid phases are considered in the energy equations of the gas and solid phase regions. We investigated the effects of varying the operating frequency of the driver on the flow and heat transfer processes in the OPTR considered. The results show for the *first time* the existence of

* Corresponding author. Fax: +1 215 895 1478.
E-mail address: bfarouk@coe.drexel.edu (B. Farouk).

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cycle-averaged secondary flows in the pulse tube, their dependence on the operating frequency and how they affect the performance of the system.

2. Background

One-dimensional computational models have been widely used for modeling thermoacoustic devices. Swift et al. [11–13] developed a 1-D code for the entire PTR system (and other thermoacoustic engines and refrigerators) based on Rott's [14] linear acoustic equations. While, the 1-D codes provide relatively good estimations of various operating parameters of the PTR (dimensions, operating frequencies, etc.); they use idealistic assumptions and do not reflect the multi-dimensional nature of the flow and transport inside the PTR systems.

Lee [15,16] developed a set of 2-D differential equations for use in describing the steady secondary flows generated by the periodic compression and expansion of an ideal gas in pulse tubes. The equations were used to obtain an insight into the physics of the pulse tube in a basic pulse tube (BPTR) and an orifice pulse tube refrigerator (OPTR) for what is known as the thermally strong case. More recently, Flake and Razani [17] carried out an axisymmetric analysis of a BPTR and a PTR and showed cycle-averaged flow fields in the pulse tube. Cha et al. [18] studied two IPTR systems based on the geometry of the pulse tube (for two values of L/D ratio). They showed the formation of instantaneous vortical structures in the pulse tube for the small L/D case which had a negative effect on the cooling performance of the IPTR due to the mixing of flow in the pulse tube. Ashwin et al. [9] used a thermal non-equilibrium model in the porous media (heat exchangers and regenerator) and considered a finite wall thickness for the various components of the IPTR. The effect of a finite wall thickness was found to increase the steady-state temperature at the cold end of the pulse tube due to the heat conduction along the walls of the pulse tube from the hot end to the cold end.

More recently, there has been a push to build PTRs that work efficiently at higher frequencies [9,19–21]. The use of high frequency oscillations allows the system to be comparatively small in size. These smaller sized systems have niche applications in the space industry where localized low power (<1 W) cooling systems with extremely fast cool-down times are required.

In the pulse tube cryocooler considered in the present study, the displacer (present in the Stirling and GM systems) is replaced by the pulse tube [22]. The main function of the pulse tube is to insulate the processes at its two ends. The pulse tube must be large enough that gas flowing from the warm end traverses only part way through the pulse tube before flow is reversed. Similarly, the flow entering from the cold end never reaches the warm end. Gas in the middle portion of the pulse tube never leaves the pulse tube and forms a temperature gradient that insulates the two ends. The overall function of the pulse tube is to transmit hydrodynamic or acoustic power in an oscillating gas system from one end to the other across a temperature gradient with a minimum of power dissipation and entropy generation.

3. Problem geometry

Fig. 1 depicts the geometry studied (i.e. an in-line OPTR system). Only half the geometry shown in Fig. 1 is simulated (the axisymmetric assumption) to save on computation time. The OPTR consists of a compression chamber (which includes the moving piston), the transfer tube, aftercooler (the first red hatched region), regenerator (blue cross-hatched region), pulse tube with two heat exchangers at its ends (the other two red hatched regions), an orifice (a simple obstruction to the flow), an inertance tube and the compliance volume.

Table 1 summarizes the various dimensions of the problem geometry and the time-invariant boundary conditions in the simulations. The mathematical boundary conditions at the various components' surfaces will be explained in detail in a following section.

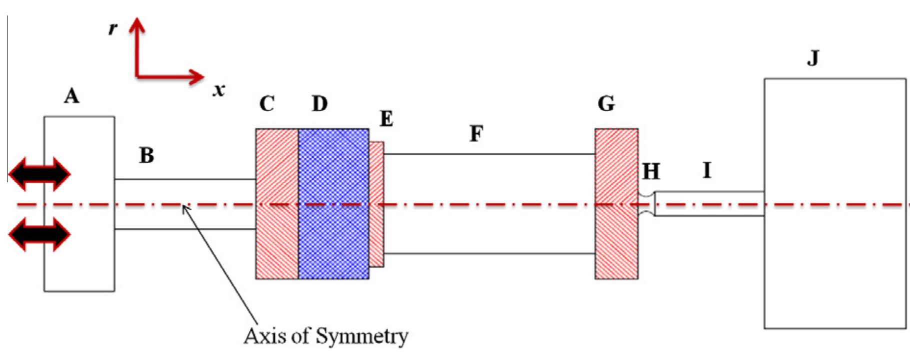

Fig. 1. Schematic of the OPTR considered.

Table 1
Dimensions and boundary conditions of the simulated system.

No.	Component	Radius (mm)	Length (mm)	Boundary condition along the outer wall
A	Compression Chamber	19.04	7.5	Adiabatic
B	Transfer tube	3.1	101.0	h_c = 20 W/m K
C	Aftercooler	8.0	20.0	T_w = 293 K
D	Regenerator	8.0	58.0	Adiabatic
E	Cold heat-exchanger	6.0	5.7	Adiabatic
F	Pulse tube	5.0	60.0	Adiabatic
G	Hot heat-exchanger	8.0	10.0	T_w = 293 K
H	Orifice valve	0.425	3.0	Adiabatic
I	Inertance tube	0.85	681.0	Adiabatic
J	Compliance volume	26.0	130.0	Adiabatic

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