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Numerical and experimental characterization of the inertance effect on pulse tube refrigerator performance ☆

Dion Savio Antao [1], Bakhtier Farouk *

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

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ABSTRACT

A synergistic investigation of numerical and experimental studies is reported to elucidate the effects of the inertance tube length and diameter on the flow physics and the performance of a single stage inertance pulse tube refrigerator (IPTR). A time-dependent axisymmetric compressible computational fluid dynamic (CFD) model of the IPTR is used to predict its performance. The phase relationships between the pressure and the mass flow in the regenerator, pulse tube and inertance tube regions and the performance of the system are determined from the numerical simulations. The predictions from the computational model show that the phase angle difference between the pressure and the velocity at the center of the regenerator is the minimum and the pressure amplitude is the maximum at the optimum inertance value that leads to the largest acoustic power. In the experimental studies, the effect of inertance is studied for tubes of two different diameters and various lengths. The effect of input power supplied to the PTR on its performance was also studied. The experimental results complement the observations from the numerical simulations.

1. Introduction

The pulse tube refrigerator (PTR) is a type of regenerative thermoacoustic gas-cycle refrigerator that runs on gases like helium, argon, nitrogen or air. The PTR runs on the reverse Stirling cycle and is capable of achieving cryogenic temperatures. The small number of moving parts required to build a PTR makes it a simple and hence potentially reliable cryocooler. Being a regenerative refrigerator, it operates with oscillating pressures and mass flows. The oscillating flow is obtained from a pressure wave generator which may be a piston driver (or linear motor) [1–3], an electrodynamic loudspeaker [4] or a thermoacoustic engine [5–9]. The latter driver mechanism makes the entire engine-PTR system purely thermoacoustic with no moving parts [10–14]. The advantage of no-moving parts and using inert gases is that the resultant thermoacoustic energy conversion system is sustainable and environmentally friendly.

In the PTR, the displacer piston (as found in the Stirling and Gifford McMahon (GM) regenerative systems) is eliminated [15]. This displacer piston is replaced by a hollow tube (called the pulse tube) and a phase shifting mechanism. Due to the absence of moving parts in the low temperature part of the device, the PTR is suitable for a wide variety of applications and is highly reliable [2]. PTRs have been used in industrial applications such as liquefaction of gases [14] and in military applications such as for the cooling of infrared sensors [15]. It has also been suggested that PTRs may be used to liquefy oxygen on Mars [16].

The phase shifting mechanism performs the function of maintaining the appropriate phase relationships between the pressure wave and the flow rate (or velocity). Proper phasing must exist in the regenerator for the PTR to operate optimally (see Refs. [17–19] for phasor diagrams of the PTR). Various phasing mechanisms have been used in the past [15,20]. The first method used was the orifice valve and compliance volume which was used to convert the standing wave PTR into a pseudo traveling wave system [21,22]. In an orifice, due to the purely resistive nature of the flow impedance, the flow rate and the pressure are in phase [15]. This causes the phase angle in the regenerator to be extremely large (∼40–60°) and hence the flow rate through the regenerator is very high (to achieve a constant acoustic power). To reduce this flow rate, a different phasing shifting mechanism was developed. An extra orifice and a by-pass line that connects the warm end of the pulse tube and the warm end of the regenerator were added to the system, creating a double-inlet pulse tube [23]. The flow rate

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* Corresponding author. Fax: +1 215 895 1478.
E-mail address: bfarouk@coe.drexel.edu (B. Farouk).
[1] Address: Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, MA 02139, USA.

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Nomenclature

A	area
A_0	piston amplitude
c	specific heat capacity
C_F	drag factor (Porous Media)
f	frequency
h_0	enthalpy
h	heat transfer coefficient
i	internal energy
k	thermal conductivity
p	pressure
r_h	hydraulic radius (Porous Media)
S	source term
T	temperature
t	time
$\mathbf{u}$	velocity vector

Greek symbols

ε	porosity
k	permeability
μ	dynamic viscosity
ρ	density
τ	stress tensor
ω	angular frequency

Subscripts

f	gas/fluid
p	solid/porous media
r	ratio
s	surface

through the regenerator is reduced using the by-pass, but DC flow is generated through the pulse tube from its warm end to its cold end. This DC flow leads to a decrease in the performance of the PTR. Another phasing mechanism used in PTRs is the inertance tube. The inertance tube is a tube placed between the warm end of the pulse tube and the compliance volume. Using the inertance tube, the losses in the regenerator are minimized when the flow and pressure are out of phase at the entrance to the inertance tube by about 50–60° [24]. Consequently, the flow in the cold end of the regenerator lags the pressure and the flow in the warm end leads the pressure (~30° in both cases) which results in the flow and pressure being in phase at the center of the regenerator [24]. The inertance tube can be used to replace the orifice phase shifting mechanism or to supplement its effects. Marquardt and Radebaugh reported the highest efficiency for the pulse tube refrigerator when using a combination of the orifice and the inertance tube [16].

In this study, the effects of inertance on the transport phenomena and the performance of an inertance type PTR (IPTR) are investigated rigorously with a combination of numerical and experimental studies. The inertance tube is the sole phase shifting mechanism used in the PTR considered. For the experimental study, a single-stage inertance PTR was designed and fabricated (that closely matches the geometry used in the numerical study) to reach liquid N_2 temperature. A detailed time-dependent axisymmetric CFD model of an IPTR is used to predict its performance. The CFD code is used to predict the effects of the inertance on the phase relationships (between the pressure and velocity) and the system performance.

2. Mathematical model for the IPTR

The flow and heat transfer simulation model incorporates the fluid dynamic equations of conservation of mass (continuity equation), momentum (Navier–Stokes equation) and energy in the fluid and porous domains. The non-equilibrium nature of the gas and solid (porous media) temperature is taken into account by solving a separate energy equation for the solid phase.

2.1. Problem geometry

Fig. 1 below shows the geometry studied (*i.e.*, an inline IPTR system). The motion of the piston is captured by a moving grid scheme near the piston wall in the compression space (component '*A*' in Fig. 1) and the re-meshing scheme used is the Transfinite Interpolation scheme [25].

The IPTR simulated has the same components (see Table 1 below) as the experimental system reported in the next section. These include a compression chamber with a moving piston (*A*), a transfer tube (*B*), an aftercooler (*C*, the first red hatched region), a regenerator (*D*, blue cross-hatched region), a pulse tube (*F*) with a heat exchanger at each end (*E* and *G*, the other two red hatched regions), a connecter tube/diffuser cone (*H*), an inertance tube (*I*) and a compliance volume (*J*).

2.2. Governing equations

The conservation equations for the gas (helium) within the system undergoing periodic compression and expansion are given as follows:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{u}) = 0 \tag{1}$$

$$\frac{\partial (\rho \vec{u})}{\partial t} + \nabla \cdot (\rho \vec{u}\vec{u}) = -\nabla p + \nabla \cdot \tau_{ij} \tag{2}$$

$$\frac{\partial (\varepsilon \rho h_0)}{\partial t} + \nabla \cdot (\varepsilon \rho \vec{u} h_0) = \nabla \cdot (k_f \nabla T_f) + \nabla (\varepsilon \vec{u}\, \tau_{ij}) + \varepsilon \frac{\partial p}{\partial t} + S_f \tag{3}$$

$$p = \rho R T_f \tag{4}$$

where, ρ is the density, u is the (r – x) velocity vector, p is the pressure, τ_{ij} is the viscous stress tensor, T_f is the gas/fluid temperature, R is the ideal gas constant, k_f is the thermal conductivity of the fluid, ε is the porosity of the porous zones and the total enthalpy is h_0 given by:

$$\begin{aligned} h_0 &= i + \tfrac{p}{\rho} + \tfrac{1}{2}(\vec{u})^2 \\ i &= cT_f \\ T_f &= \tfrac{1}{c}\left[h_0 - \tfrac{p}{\rho} - \tfrac{1}{2}(\vec{u})^2\right] \end{aligned} \tag{5}$$

where, i is the internal energy and c is the specific heat capacity of the fluid.

The source term S_f in the gas-phase energy equation (Eq. (3)) is only applicable for the porous media zones (where the porosity ε is <1.0) and is used to couple the gas and solid phase energy equations:

$$S_f = \frac{h_{fp} A (T_p - T_f)}{\varepsilon} \tag{6}$$

For the present simulations, ε is dependent on the component and in the rest of the domain (gas phase) ε has a value of 1.0. The value