Energy Conversion and Management 86 (2014) 194–203

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Investigation of a high frequency pulse tube cryocooler driven by a standing wave thermoacoustic engine

A.A. Boroujerdi *, M. Ziabasharhagh [1]

Faculty of Mechanical Engineering, K.N. Toosi University of Technology, P.O. Box: 19395-1999, Tehran, Iran

A R T I C L E I N F O

Article history:
Received 7 December 2013
Accepted 5 May 2014

Keywords:
Pulse tube cryocooler
Standing wave engine
Thermoacoustic
Numerical model
APAT
Parallel plate stack

A B S T R A C T

In this work, a typical thermoacoustically driven pulse tube cooler as a no-moving part device has been investigated by a numerical method. A standing wave thermoacoustic engine as a prime mover in coupled with an inertance tube pulse tube cryocooler has been modeled. Nonlinear equations of unsteady one-dimensional compressible flow have been solved by the finite volume method. The model presents an important step towards the development of nonlinear simulation tools for the high amplitude thermoacoustic systems that are needed for practical use. The results of the computations show that the self-excited oscillations are well accompanied by the increasing of the pressure amplitude. The necessity of implementation of a nonlinear model to investigate such devices has been proven. The effect of APAT length as an amplifier coupler on the performance of the cooler has been investigated. Furthermore, by using Lagrangian approach, thermodynamic cycle of gas parcels has been attained.

1. Introduction

Thermoacoustics have been meant as a combination of thermal effects and sound by Rott [1]. He developed the mathematics describing acoustic oscillations in a gas in a channel with an axial temperature gradient, with lateral channel dimensions of the order of the gas thermal penetration which are much shorter than the wavelength. According to the sound wave, thermoacoustic devices can be categorized as standing wave and traveling-wave systems. According to the energy conversion, there are two types of thermoacoustic effects: one is the acoustic oscillation powered by heat energy in thermoacoustic engine (prime mover), and the other is the heat flow driven by acoustic power in thermoacoustic cooler. In thermoacoustic engines, thermal energy from an external source is converted into kinetic energy in the form of acoustic waves. Energy conversion occurs in the stack (or regenerator) which smoothly spans the temperature difference between the hot heat exchanger and the ambient heat exchanger. During the last decade, numerous attempts have been proposed to investigate and to optimize the thermoacoustic engines [2–7]. Parametric studies have been performed for investigation of influences of working liquid [8,9], stack geometry and resonator length [10], resonator diameter [11], and resonance tube geometry shape [12] on performance of thermoacoustic systems. Furthermore, some general optimizations have been made [13–16].

Linear thermoacoustics has been expounded and summarized systematically by Swift in 1988 [17]. The theory supposes that the time dependence of all fluctuating quantities, such as pressure, velocity, temperature and so on, are purely sinusoidal. It is unable to display the thermodynamic cycle and the mechanism of thermoacoustic conversion clearly. Besides, the linear thermoacoustics is suitable only for stationary thermoacoustic system, and cannot reflect its varying process. It is incapable of dealing with the nonlinear thermoacoustic phenomena, such as amplitude saturation, and frequency shift, which have been observed in lots of experimental thermoacoustic apparatuses.

Nonlinear thermoacoustics as a branch of nonlinear acoustics began to be developed in the last decade. Karpov and Prosperetti from John Hopkins University have done a lot of outstanding work in this field [18–21]. They carried the expansion to fourth order in the perturbation parameter, and obtained explicit results for the initial growth, nonlinear evolution, and final saturation. Then, the dependence of the saturation amplitude upon several important parameters was illustrated. They also investigated the model performance on several examples, including a prime mover, an externally driven thermoacoustic refrigerator, and a combined prime mover/refrigerator system. Both their model and the present model are one-dimensional nonlinear model. However, there are some differences. In the present work, a model based on the finite

* Corresponding author. Tel.: +98 21 8406 3242; fax: +98 21 8867 7274.
E-mail addresses: A.A.Boroujerdi@gmail.com (A.A. Boroujerdi), mzia@kntu.ac.ir (M. Ziabasharhagh).
[1] Tel.: +98 21 8406 3242; fax: +98 21 8867 7274.

http://dx.doi.org/10.1016/j.enconman.2014.05.019
0196-8904/

Nomenclature

A	cross-sectional area of gas flow (m^2)
A_V	heat transfer area per unit volume (m^{-1})
C_F	Forchheimer's inertial coefficient
C_P	isobaric specific heat ($J\,kg^{-1}\,K^{-1}$)
C_s	specific heat of solid ($J\,kg^{-1}\,K^{-1}$)
C_v	isochoric specific heat ($J\,kg^{-1}\,K^{-1}$)
d	diameter (m)
d_h	hydraulic diameter (m)
d_g	plate spacing (m)
d_w	wire diameter (m)
F	general function
f	friction factor
j	imaginary unit,=$(-1)^{1/2}$
K	Darcy permeability (m^2)
k	thermal conductivity ($W\,m^{-1}\,K^{-1}$)
L	length (m)
l	mesh distance (m)
m	mass (kg)
$\dot{m}$	mass flow rate ($kg\,s^{-1}$)
Nu	Nusselt number
n	number of screens per inch ($0.0254^{-1}\,m^{-1}$)
Pr	Prandtl number
$\dot{q}_{ext}$	external heat load per unit volume ($W\,m^{-3}$)
R	gas constant ($J\,kg^{-1}\,K^{-1}$)
Re	Reynolds number
T	temperature (K)
t	time (s)
u	velocity ($m\,s^{-1}$)
V	volume (m^3)
x	longitude coordinate (m)
Z	compressibility factor
$\mathscr{R}\{\}$	real part of

Greek letters

α	heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
β	opening area ratio of screen
μ	dynamic viscosity ($kg\,m^{-1}\,s^{-1}$)
ρ	density ($kg\,m^{-3}$)
ϕ	porosity
ω	angular frequency ($rad\,s^{-1}$)

Subscripts

I	node number
i	face number
S	solid

Superscripts

$\dot{}$ (dot)	time derivative

Abbreviations

PTC	pulse tube cryocooler
SWE	standing wave engine
SS	stainless steel

volume method with upwind scheme is developed to solve 1D equations of a system with non-perfect gas (via considering a compressibility factor) as the working fluid. In many combined systems such as thermoacoustically-driven pulse tube cooler, high amplitude oscillations occur so that high order harmonics must be considered to model these systems. The present model is not limited by a specified order of approximation.

Thermoacoustically-driven pulse tube cooler (TADPTC) has the advantage of structure simplicity with no moving mechanical components and thus promises a potentially long lifetime. High frequency operation of such a system can lead to a much-reduced size, which is very attractive in small-scale cryogenic applications. Godshalk et al. [22] established the first high frequency TADPTC in 1996. However, the cooler just reached a lowest no-load temperature of 147 K with the frequency being 350 Hz.

Before 2005, pulse tube cooler was directly coupled with the thermoacoustic engine and obtainable pressure ratio for the cooler was limited by the capability of the thermoacoustic engine. In 2005, Dai et al. [23] proposed the concept of acoustic amplifier, which is actually a long tube connecting the engine with the pulse tube cooler. They established and studied a 300 Hz TADPTC utilizing the invention of acoustic pressure amplifier tube (APAT) to couple the standing-wave engine (SWE) with the pulse tube cooler (PTC). APAT is simply a thin circular tube with a length smaller than 1/4 of wavelength to connect the cooler to the engine. Their theoretical calculation showed that suitable length and diameter of the tube can lead to a pressure wave amplification effect which means that the pressure wave amplitude coming from the thermoacoustic engine can be much amplified to drive the pulse tube cooler.

With modifications, lowest no-load temperatures of 95 K [24], 79.6 K [25], and 77.8 [26] were sequentially reached. Later, a lowest no-load temperature of 69.5 K has been reached as reported on the work of Zhu et al. [27]. Comparing with the SWE previously used, Zhu et al. made two changes. Firstly, the inner diameters of the hot heat exchanger, the stack and the ambient heat exchanger were increased to increase inlet volume flow of APAT and the amplification ratio. Secondly, a cone-shaped resonator substituted the isodiametric resonator to decrease the viscous resistance. Zhu et al. [28] have investigated the influences of different parameters such as the engine stack lengths, the geometry of the resonator, the length of the APAT as well as the mean pressure on the overall performance of the device, experimentally. Most recently, Wang et al. [29] obtained a cooling power of 1.04 W at 80 K and a no-load temperature of 63 K with 500 W heating power.

The present work reports on the investigations of a 300 Hz frequency thermoacoustically driven inertance tube pulse tube cryocooler by using a numerical nonlinear model. The developed numerical model is based on nonlinear one-dimensional unsteady compressible flow equations solved by the finite volume method. APAT length as a key design parameter and its effect on the pressure ratio, the hot temperature of SWE, the cooling temperature of PTC, mass flow rate, and pressure amplitude are investigated.

2. Physical model

2.1. Geometry of the device

The geometry of the device including the heat exchangers, stack, resonator, and buffers for SWE, and including the heat exchangers, regenerator, pulse tube, straightener, and inertance tubes, for PTC, and also including APAT (Acoustic Pressure Amplifier Tube) as a coupler, is shown in Fig. 1.

2.2. Governing equations

Because the mass flow, momentum flow, and the enthalpy flow in the longitudinal direction is very greater than those of lateral