International Journal of Heat and Mass Transfer 103 (2016) 382–389

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Refrigeration mechanism of the gas parcels in pulse tube cryocoolers under different phase angles

Xiaoqin Zhi [a], Limin Qiu [a,*], John M. Pfotenhauer [b], Zhihua Gan [a], Yan Yan [c]

[a] Institute of Refrigeration and Cryogenics, Zhejiang University, Hangzhou 310027, China
[b] College of Mechanic Engineering, University of Wisconsin, Madison 53706, USA
[c] School of Mechanical and Engineering, Southeast University, Nanjing 21000, China

A R T I C L E I N F O

Article history:
Received 8 April 2016
Received in revised form 21 June 2016
Accepted 12 July 2016

Keywords:
Pulse tube cryocooler
Heat transfer
Refrigeration mechanism
Thermodynamic cycles
Phase angle

A B S T R A C T

The thermodynamic behavior of gas parcels in the key parts of the pulse tube cryocooler (PTC) is studied. The refrigeration mechanism of PTCs under different cold end phase angles is revealed by analyzing and comparing the heat transfer characteristics of the gas parcels at both sides of the cold end heat exchanger. Results show that the cold end phase angle, an important parameter of the PTC, determines the cooling performance of the PTC by affecting the heat transfer characteristics of the gas parcels in a cycle. In PTCs with a cold end phase angle normally between −30° and 60°, the gases pump heat from the cold end heat exchanger to the aftercooler all the way through the regenerator. In contrast, for the basic PTC with a cold end phase angle close to 90°, the heat transfer direction is reversed, the gases pump heat from the cold end heat exchanger to the hot end heat exchanger all the way through the pulse tube. This study unifies the various perspectives on the PTC's refrigeration mechanism by revealing the inherent effect of the cold end phase angle, indicating that enhancing the heat pumping function of the gas in the regenerator is an essential way to improve the cooling performance.

1. Introduction

The pulse tube cryocooler (PTC) with no moving parts at the cold chamber has attracted increasing attention due to its promising applications in the military, space and superconductor fields, each having rigorous requirements of compactness and high reliability for the cooling system. The recently published literature shows that the PTC is receiving more interest as a cryocooler than the traditional J-T, Stirling and G-M cryocoolers. It is likely to replace other cryocoolers in more applications because of its simplified structure and reliable performance [1]. However, the simplified structure of the PTC leads to complex working mechanisms due to the phase shifting effect of the gas inside.

As a member of the regenerative cryocooler family, the PTC has a "gas piston" inside the pulse tube that causes gas expansion and decreasing temperatures at the cold end like the piston expander in the Stirling and G-M cryocoolers. Therefore, the early studies of the PTC mainly focused on the pulse tube component, for example, the surface heat pumping theory [2], the phase shifting theory [3], the adiabatic expansion principle, and the thermodynamic non-symmetry analysis [4,5]. These studies consider that it is the gas in the pulse tube that produces cooling by expanding and absorbing heat at the cold end heat exchanger. However, with the development of the thermoacoustic theory, people realized that the regenerator is not only a heat exchanger, but also a work-heat converter, working either as an engine or a refrigerator under different thermoacoustic fields. Since then, theoretical investigations have been carried out on the regenerator based on classic thermodynamic principles [6,7] and thermoacoustic theory [8,9], these demonstrating that the gas in the regenerator can produce cooling by pumping heat away from the cold end heat exchanger. The contradictory nature of the above studies, as to whether the pulse tube or the regenerator produce cooling at the cold end is mainly caused by the over-simplification associated with the one dimensional, ideal regenerator with no dead volume, and the ideal isothermal or adiabatic analyses. These inappropriate methods of analysis may lead to the limitations and uncertain authenticity of the results obtained. Also some results analyses are shown in Euler way of focusing on the time-averaged parameters at fixed cross section, which is not so visual to reveal the periodical behavior of the oscillating gases.

In a PTC, the various gas parcels each experience differing thermodynamic processes through periodic oscillation, and the inhomogeneous characteristics of energy, mass transfer and temperature distribution, especially inside the pulse tube, cannot

* Corresponding author. Fax: +86 571 87952793.
E-mail address: limin.qiu@zju.edu.cn (L. Qiu).

http://dx.doi.org/10.1016/j.ijheatmasstransfer.2016.07.032
0017-9310/

be ignored [10]. It is helpful to reveal the refrigeration mechanism of the PTC by studying the gas parcels' thermodynamic behavior through the use of a multi-dimensional, quantitative approach such as is possible at a micro scale in a Lagrangian way. Such research has been realized in the regenerator and the pulse tube by CFD simulation methods [11,12]. The cooling power is actually the result of a cooperative interaction between the different parts in a PTC. However, most of the previous investigations studying the refrigeration mechanism are carried out on a single component (either the regenerator or the pulse tube) under different working conditions, and these are unable to explain how the net cooling effect generated in the cold end heat exchanger is a result of the combined work of the gas in the regenerator and pulse tube. Additionally, although it is an essential parameter of the oscillating flow, no research has yet investigated how the cold end phase angle affects the thermodynamic cycles of the gas in the regenerator at a micro scale, connecting that behavior to the macro-scale cooling performance.

In this study, the refrigeration mechanism of the PTC is investigated by analyzing the thermodynamic behavior of the gas parcels in its key locations. The heat transfer characteristics of the gas parcels in the pulse tube, regenerator and cold end heat exchanger are compared in order to explain the net cooling effect that is produced. Furthermore, the effects of different cold end phase angles on the gas parcels' periodic heat transfer characteristics are studied at a micro-scale to reveal its influence on the cooling performance.

2. Numerical model of the single stage PTCs

2.1. Mathematic model and boundary conditions

The structure of the two-dimentional, axis-symmetric single stage PTC is shown in Fig. 1. The details are given in Table 1. The PTC model contains two kinds of flow areas, the porous media area as the aftercooler *a*, regenerator *b*, cold end heat exchanger (CHX) *c* and hot end heat exchanger (HHX) *e*; and the non-porous media area as the pulse tube d. For all the flow areas, the mass, momentum and energy equations can be described as follows [13]:

$$\frac{\partial}{\partial t}(\phi\rho_{\rm f}) + \nabla\left(\phi\rho_{\rm f}\vec{u}\right) = 0 \tag{1}$$

$$\frac{\partial}{\partial t}\left(\phi\rho_{\rm f}\vec{u}\right) + \nabla\left(\phi\rho_{\rm f}\vec{u}\,\vec{u}\right) = -\phi\nabla p + \nabla\cdot\left(\phi\tau\vec{u}\right) + S_{x(y)} \tag{2}$$

$$\frac{\partial}{\partial t}(\phi\rho_{\rm f}E_{\rm f} + (1-\phi)\rho_{\rm s}E_{\rm s}) + \nabla\cdot\left(\vec{u}(\rho_{\rm f}E_{\rm f} + p)\right) = \nabla\cdot\left[(\phi k_{\rm f} + (1-\phi)k_{\rm s})\nabla T + (\phi\tau\cdot\vec{u})\right] \tag{3}$$

$$E_{\rm f} = H_{\rm f} + \frac{u^2}{2} - \frac{p}{\rho_{\rm f}}, \quad E_s = C\upsilon_s T \tag{4}$$

$$S_{x(y)} = -\frac{1}{2D_{\rm h}} f_{\rm r}\rho_{\rm f}|\vec{u}|\vec{u} \tag{5}$$

The explanations of each item in the above equations are given in reference [12,13]. For the pulse tube *d*, the porosity $\varphi = 1$, and the momentum source term $S_{x(y)} = 0$; for the areas *a, b, c, e*, their porosity is given in Table 1, the momentum source term is calculated by Eq. (5), in which f_r is the flow resistance coefficient of the porous media. In the regenerator and the hot end heat exchanger, which are packed with stacked screens, the term f_r is treated as isotropic and it can be calculated by the following empirical formula [14]:

$$f_{\rm r} = 129/Re + 2.91Re^{-0.103} \tag{6}$$

For the areas *a, c* which are slit type heat exchangers, the term f_r can be calculated as [15]:

$$f_r = 0.11(\phi/d_h + 68/Re)^{0.25} \tag{7}$$

In order to study the influence of the cold end phase angle θ_C, its value is determined for both the phase shifting type and the basic type of PTCs. The cold end phase angle is defined as the phase angle between the mass flow oscillation and the pressure oscillation at the cold end. It is positive when the mass flow oscillation leads the pressure oscillation. For the phase shifting PTCs, the mass flow rate $\dot{m}_{in}$and the pressure wave p_{out} are respectively used as the inlet and outlet boundary conditions:

$$\dot{m}_{in} = m_a \cdot \sin(2\pi ft + \theta_{\rm h}\cdot\pi/180) \tag{8}$$

$$p_{out} = p_a \cdot \sin(2\pi ft) \tag{9}$$

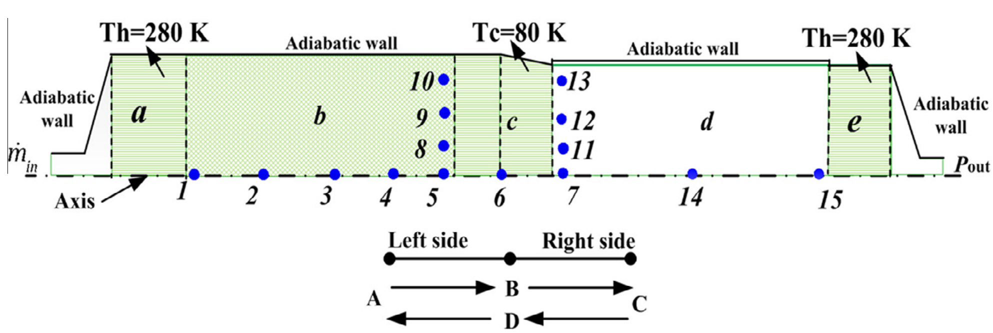

Fig. 1. Sketch of the pulse tube cryocooler and the typical gas parcels at different positions.

Table 1
Details for different parts of the pulse tube cryocooler.

Items	Porosity	Material	Boundary conditions
Aftercooler *a*	0.2	Slit type, copper	Isothermal, 280 K
Regenerator *b*	0.72	SS304 mesh screen	Adiabatic
Cold end heat exchanger (CHX) *c*	0.24	Slit type, copper	80 K (160 K for basic PTC)
Pulse tube *d*	1	Stainless steel	Adiabatic
Hot end heat exchanger (HHX) *e*	0.78	Copper mesh screen	Isothermal, 280 K