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Investigation of synchronous effects of multi-mesh regenerator and double-inlet on performance of a Stirling pulse tube cryocooler

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ABSTRACT

In this paper synchronous effects of multi-mesh regenerator and double-inlet on performance of a Stirling pulse tube cryocooler (SPTC) have been considered. In this respect, a finite volume code was developed to simulate the SPTC. Set of governing equations were written in a general form such that all porous and non-porous sections of the system could be modeled. Results showed that synchronous application of double inlet and multi-mesh regenerator optimizes the phase shift between velocity and pressure at the warm end of the pulse tube, increases the regenerator's outlet pressure amplitude, decreases inertial and viscous losses in the hot end of the regenerator and consequently increases the COP of the system. Furthermore, it was observed that a minimum temperature of 60.3 K and COP of 0.03996 @ 80 K is attainable using optimum multi-mesh regenerator and double inlet; whereas, for a simple SPTC with a uniform mesh regenerator, a minimum temperature of 71.3 K and maximum COP of 0.0227 @ 80 K are concluded.

1. Introduction

Nowadays, pulse tube cryocooler (PTC) is widely used to produce very low temperatures. This is due to its obvious benefits in contrast with other cryocoolers. However, its efficiency is lower than other cryocoolers and, since its invention by Gifford and Longsworth in 1963 [20], many efforts have been dedicated to improve that. As a result of extensive researches in the PTC community, essential improvements achieved by two modifications: adding a reservoir via an orifice valve to the warm end of the pulse tube which led to phase shift between pressure and velocity; adding a second inlet valve, thereby reducing the mass flow through the regenerator led to a second phase shift with resulting improvement in cooling performance [19].

After four decades of experiences with PTC, it is already well-known that the regenerator as a key component plays a critical role on the cooling performance of the system. Therefore, extensive efforts have been focused on improvement in regenerator technology during the last two decades. These efforts are categorized into areas of materials and geometry, modeling, and measurement [9].

Although, there are several sources of energy losses in the regenerator; however, the most important parts of the losses occur due to the viscous and inertial losses [9]. Therefore, many efforts have been dedicated to introduce a good relation for oscillating flow friction coefficient inside the regenerator [14].

Heat transfer between the gas and solid matrices in the regenerator micro-porous media is another important part of the losses. Therefore, exotic alloys and processes were developed so that the regenerator obtained sufficient thermal capacity in the cold region. There are many investigations published in the literature that deal with application of erbium alloys (Er_3Ni, Er_6Ni_2Sn, $ErNi_{0.9}Co_{0.1}$ and ErNi), lead spheres and even multi-mesh regenerators instead of simple micro-porous stainless steel regenerators to improve their performance [12,8,15,17–19].

Although increasing the interfacial heat transfer in the cold end of the regenerator has been the key reason for using multi-mesh regenerator with different materials in PTCs; however, optimum multi-mesh regenerator can reduce the inertial and viscous losses. Furthermore, the multi-mesh regenerator and double inlet can be simultaneously employed for further improvement of COP of SPTC in comparison with simple or multi-mesh regenerator SPTC. In this paper, synchronous effects of multi-mesh regenerator and double inlet on the cooling performance of a SPTC are investigated. In this respect, a 1D CFD code is developed to simulate the SPTCs.

2. Physical model and governing equations

Schematic of the double inlet SPTC is shown in Fig. 1. The physical domain includes all of the components between the piston head of the compressor and the end wall of the buffer. For simplicity three assumptions are applied as listed below:

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Nomenclature

A	cross sectional area of gas flow (m^2)
A_L	interfacial heat transfer area (m^2)
A_1	small area of flow passing section through a joint
A_2	large area of flow passing section through a joint
C_F	Forchheimer's inertial coefficient
C_p	constant pressure specific heat (J kg^{-1} K^{-1})
C_s	specific heat of solid (J kg^{-1} K^{-1})
C_v	constant volume specific heat (J kg^{-1} K^{-1})
E_{or}	energy flow through orifice
J_{or}	correction factor of orifice
K	permeability (m^2)
K_l	local energy loss coefficient
L	conical section length or distance between adjacent nodes of an abrupt joint
Nu	Nusselt number
P	pressure (Pa)
Pr	Prandtl number
$\dot{Q}_c$	cooling power (W)
R	gas constant
Re	Reynolds number
S''_{gen}	entropy generation rate (W m^{-3} K^{-1})
T	temperature (K)
V	volume (m^3)
$\dot{W}$	P–V power of the compressor (W)
X_t	screen transverse pitch
X_u	amplitude of velocity oscillation
d_h	hydraulic diameter
f	friction factor
g	gravity acceleration (m s^{-2})
k	thermal conductivity (W m^{-1} K^{-1})
l	mesh distance
m	mass (kg)
$\dot{m}$	mass flow rate (kg s^{-1})
n	number of control volumes
$\dot{q}_c$	cooling power/CHX solid matrix volume
u	velocity (m s^{-1})
x	longitudinal coordinate
x_{amp}	piston motion amplitude
x_d	length of compressor dead volume
x_{comp}	compressor length
1D	one dimensional
$[A]_b^a$	$(A)_a - (A)_b$
Greek letters	
α	heat transfer coefficient
γ	expansion angle of conical sections
Δx	control volume length
δt	time step
δx	node distance
ε	porosity
φ	opening area ratio of screen
θ	system angle with respect to gravity
λ	matrix conductivity factor
μ	viscosity (Pa s)
μ_w	viscosity at wall temperature (Pa s)
ρ	density (kg m^{-3})
ω	frequency of piston motion (rad s^{-1})
ζ	number of packed screens per length
Subscripts	
cv	center of control volume
f	face of control volume
i	control volume numbered "i"
in	inlet section
or	double inlet orifice
$oscm$	oscillating mean value
out	outlet section
s	solid matrix
$u1$	oscillating velocity at Reg. inlet
$u2$	oscillating velocity at Reg. outlet
Superscripts	
j	current time

- 1D compressible flow of ideal gas;
- Specified heat transfer rate in the cold heat exchanger as cooling power;
- Sink of energy in compressor, transition line, after cooler and hot heat exchanger walls equal to radiative and convective heat transfer with ambient.

Taking an Eulerian point of view, the flow field unknowns are described as $\dot{m}(x,t)$, $P(x,t)$, $\rho(x,t)$ and $T(x,t)$. Finite volumes are defined to construct piecewise uniform approximations to these functions. Control volumes and the corresponding nodal points in their center are shown in Fig. 1. Solid circles ($1 \leqslant i \leqslant n$) store the nodal values of the temperature, pressure, density, mass, and gas properties. The dashed lines ($1 \leqslant i \leqslant n+1$) store values of the mass flow rate, velocity, energy flow, and entropy flow. This staggered arrangement of finite volumes helps to prevent occurring non-physical numerical solutions, Checkerboard, in the Finite Volume Method. With aforementioned assumptions the continuity, momentum, energy of gas, energy of solid and the state equations can be respectively represented as follows:

$$\partial(m)_{cv,i}/\partial t = [\dot{m}]_{f,i+1}^{f,i}. \tag{1}$$

$$\partial(mu)_{f,i}/\partial t = [\dot{m}u]_{cv,i}^{cv,i-1} + A_i[P]_{cv,i}^{cv,i-1} - [\varepsilon\mu/(K\rho) + c_F\varepsilon^2|u|/K^{1/2}]_{f,i}\dot{m}_i\delta x_i + m_{f,i}g\cos\theta. \tag{2}$$

$$\frac{\partial}{\partial t}\left(m\frac{u^2}{2} + mC_VT\right)_{cv,i} = \left[\dot{m}\frac{u^2}{2} + \dot{m}C_pT - kA\frac{\partial T}{\partial x}\right]_{f,i+1}^{f,i} + \alpha_iA_{L,i}(T_{s,i}^j - T_i^j) + P_i\frac{dV_i}{dt}. \tag{3}$$

$$\frac{\partial}{\partial t}(m_sC_sT_s)_{cv,i} = \left[\lambda_sk_s\frac{\partial T_s}{\partial x}\right]_{f,i+1}^{f,i} + \alpha_iA_{L,i}(T_i^j - T_{s,i}^j) + \dot{q}_CV_{s,cv,i}. \tag{4}$$

$$P_i^jV_i = m_i^jRT_i^j. \tag{5}$$

3. Numerical procedure

The implicit control volume method is used to solve Eqs. (1)–(5) which are changed to algebraic equations. The first order method is used for time-direction in first step and the second order for the second step. In x-direction, for diffusive and convective terms, central and second order upwind discretization methods are employed, respectively; except near the boundaries which the first order upwind is used. Herein, only second order method of time-direction discretization is presented.

3.1. Continuity and momentum equations

To explicit continuity equation for the pressure as a function of mass flow rate, the mass of gas is replaced from the equation of