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Theoretical modeling of a gas clearance phase regulation mechanism for a pneumatically-driven split-Stirling-cycle cryocooler

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ABSTRACT

The concept of a new type of pneumatically-driven split-Stirling-cycle cryocooler with clearance-phase-adjustor is proposed. In this implementation, the gap between the phase-adjusting part and the cylinder of the spring chamber is used, instead of dry friction acting on the pneumatically-driven rod to control motion damping of the displacer and to adjust the phase difference between the compression piston and displacer. It has the advantages of easy damping adjustment, low cost, and simplified manufacturing and assembly. A theoretical model has been established to simulate its dynamic performance. The linear compressor is modeled under adiabatic conditions, and the displacement of the compression piston is experimentally rectified. The working characteristics of the compressor motor and the principal losses of cooling, including regenerator inefficiency loss, solid conduction loss, shuttle loss, pump loss and radiation loss, are taken into account. The displacer motion was modeled as a single-degree-of-freedom (SDOF) forced system. A set of governing equations can be solved numerically to simulate the cooler's performance. The simulation is useful for understanding the physical processes occurring in the cooler and for predicting the cooler's performance.

1. Introduction

The pneumatically-driven split-Stirling-cycle cryocooler, which was invented by Horn et al. in the 1970s [1–3], has been developing rapidly in recent years because of its low cold-tip vibration, low mass and high efficiency [4–9]. With the introduction of many special components and technologies such as linear compressors, non-contact seals, flexure springs, and contamination control techniques into the manufacturing and assembling processes, pneumatically-driven split-Stirling-cycle cryocoolers are quite reliable and have been used in many applications such as infrared detector cooling [10].

One of the key problems in the pneumatically-driven split-Stirling-cycle cryocooler is how to maintain and adjust the motion phase relationship between the active and passive vibrators. The compression piston driven by a linear motor is the active vibrator, and the free displacer driven by synthesizing gas force provided by gas in different chambers and damping forces in various sealing clearances is the passive vibrator. We put forward a new pneumatically-driven split-Stirling-cycle cryocooler with clearance-phase-adjustor to solve this problem.

A theoretical model has been made to simulate the dynamics, fluid dynamics and thermodynamics of a pneumatically-driven split-Stirling-cycle cryocooler with the proposed clearance-phase-adjustor. The linear compressor is modeled under adiabatic conditions, and the displacement of the compression piston is experimentally rectified. The working characteristics of compressor motor and the principal losses of cooling are taken into account. A single-degree-of-freedom (SDOF) forced system is utilized to model the displacer motion. This simulation model is useful for understanding the essentials of the internal physical processes and is reliable in predicting the performances of this new type cryocooler [11].

2. Pneumatically-driven split-Stirling-cycle cryocooler with clearance-phase-adjustor

The motion phase relationship between the compression piston and displacer in the pneumatically-driven split-Stirling-cycle cryocooler has traditionally been maintained or adjusted by frictional damping forces acting on the pneumatically-driven rod by the sealing gap between hot chamber and spring chamber. Fig. 1 shows the structure of the conventional pneumatically-driven split-Stirling-cycle cryocooler.

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Nomenclature

A	heat transfer area, ambient
A_H	heat transfer area for hot chamber
A_n, B_n	constants in Fourier series of current
A_0	clearance section area
a	ambient
a_n, b_n	constants in Fourier series of active coil height
B_g	flux density in gap
cc	mutual inductive coupling coefficient
c_p	gas specific heat
c_m	matrix specific heat
D_h	equivalent diameter
d_h	hydraulic diameter
E	impressed voltage
E_m	induced voltage
E_0	amplitude of impressed voltage
F	force
f	friction factor or frequency
f_1	friction factor
f_2	friction factor
h	height
h_v	effective coil height
h_g	gap height
h_t	total coil height
I	effective electric current
i	electric current or imaginary number unit, $i = \sqrt{-1}$
L	effective coil inductance or length/height of phase-adjusting part
MAX	constants defined by Eq. (12)
m	mass or gas mass
$\dot{m}$	mass flow rate
n	general integer
P_{in}	power input to motor
P_{out}	power output of motor
p	pressure
Pr	Prantle's number
Q	heat or cooling power
R	radius; gas constant
Re	Reynolds number; real part of a complex number
R_L	coil quality factor
R_p	power source resistance
R_T	circuit total resistance
R_W	wire resistance
r	radius
S	stroke
T	temperature or period
U	displacer velocity
u	gas flow velocity
V	volume
$\dot{W}$	work rate
X	displacement amplitude
x	displacement; coordinate of displacer motion direction
y	coordinate of radium direction

Greek letters

α	heat transfer coefficient
ε	efficiency or emissivity; convergent constant
θ	differential phase angle
Φ	motion phase difference
δ	clearance gap; thickness of gas thermal and viscous boundary layer
λ	thermal conductivity or friction factor
μ	dynamic viscidity or modified empirical coefficient
v	motion viscosity
ρ	density
σ	convergent constant
τ	time
ϕ	motor phase shift
γ	ratio of specific heat
ω	angular frequency
ψ	modified resistance coefficient

Subscripts

A	ambient
b	buffer space of compression chamber
C	compression chamber or compressor
Conduction	conduction loss
cp	compressor
d, D	displacer
e, E	cold chamber
Effective	effective value
emit	radiation loss
f	boundary of control volume
gross	gross cooling power
h	hydraulic
H	hot chamber or hot end
i	the ith control volume
input	compressor input power
j	the jth control volume
m	matrix
max	maximum
net	net cooling power
out	outlet
p	compressor piston
pump	pump loss
R	regenerator
RH	high-temperature end of regenerator
RL	low-temperature end of regenerator
reg	regenerator or regenerative loss
rod	relating to drive rod
shuttle	shuttle loss
sol	solenoid
t	tube or hose
W	wall
1	upper spring sub-chamber
2	lower spring sub-chamber

Superscripts

$'$	displacer motor parameter or new value
$-$	mean value
$\cdot$	variation with time

The conventional arrangement has the following disadvantages:

(1) The design and manufacturing process of the seal between the hot chamber and spring chamber is very complicated and difficult, because it acts as not only as a gas seal, but also to maintain or adjust damping of the free displacer.
(2) The cooler operating performances are sensitive to the diameter of the pneumatically-driven rod.
(3) Adjusting the diameter of the pneumatically-driven rod, which is basically equivalent to manufacturing a new pneumatically-driven gas expander, increases cost and manufacturing and assembly time.