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Numerical investigation of thermoacoustic refrigerator at weak and large amplitudes considering cooling effect

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ABSTRACT

In this paper, OpenFOAM package is used for the first time to simulate the thermoacoustic refrigerator. For simulating oscillating inlet pressure, we implemented cosine boundary condition into the OpenFOAM. The governing equations are the unsteady compressible Navier–Stokes equations and the equation of state. The computational domain consists of one plate of the stack, heat exchangers, and resonator. The main result of this paper includes the analysis of the position of the cold heat exchanger versus the displacement of the pressure node at large amplitude for successful operation of the refrigerator. In addition, the effect of the input power on the successful operation of the apparatus has been investigated. It is observed that for higher temperature difference between heat exchangers, the time of steady state solution is longer. We show that to analyze and optimize the thermoacoustic devices, both heat exchangers should be considered, coefficient of performance (COP) should be checked, and the successful operation of the refrigerator should be evaluated.

1. Introduction

A thermoacoustic refrigerator is a device that transfers heat from a low-temperature reservoir to a high-temperature reservoir by utilizing acoustic power. The standing wave thermoacoustic refrigerators consist mainly of four parts: acoustic driver, resonator, heat exchangers, and stack. The acoustic driver is attached to the resonator filled with a gas. In the resonator, the stack consisting of many parallel plates and two heat exchangers are installed as illustrated in Fig. 1. The acoustic driver sustains an acoustic wave in the gas at the fundamental resonance frequency of the resonator. The standing wave displaces the gas in the channels of the stack. The thermal interaction between the oscillating gas and the surface of the stack transfers heat from the cold side to the hot edge. The heat exchangers exchange heat between the apparatus and reservoirs.

Thermoacoustic devices use no moving parts, no exotic and poison materials; therefore, they seem to have the immediate potential to have comparably high reliability and low cost [1]. Cao et al. [2] simulated an oscillating gas near a 1D isothermal stack and computed energy flux density in thermoacoustic devices. Unusual vertical energy flux is found near the ends of the stack plate within an area whose length scale is proportional to the gas displacement amplitude. Worlikar and Knio [3] used an overall method consisting of a quasi-1D computation scheme for resonator and a multi-dimensional vorticity/stream-function potential formulation for the detailed simulation of flow around the stack. They demonstrated that the 1D code is capable of representing wave amplification through heat addition for weakly-nonlinear acoustic. Worlikar et al. [4] further extended their previous work by solving the energy equation in the fluid and the stack plates. They implemented fast Poisson solver for the velocity potential based on the domain decomposition/boundary Green's function technique. They predicted the steady state temperature gradient across a two-dimensional couple and analyzed its dependence on the amplitude of the resonant wave. Ishikawa and Mee [5] used PHOENICS commercial code and solved 2D full Navier–Stokes equations and simulated flow near an isothermal zero thickness stack. They examined solver results in the form of energy vectors, particle paths, and overall entropy generation rates. It is observed that, the time-averaged heat transfer to and from the plates is concentrated at the edges of the plates. In constant Mach number, the width of the region where there is substantial heat transfer decreases as the plate spacing is reduced. Tasnim and Fraser [6] simulated a conjugate heat transfer in the thermoacoustic refrigerator. They solved unsteady compressible Navier–Stokes and energy equations with commercial code STAR-CD, and illustrated flow and thermal fields during a cycle. Ke et al. [7] used self-written program of the compressible SIMPLE algorithm and carried out

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Nomenclature

x	longitudinal direction, m
y	transversal direction, m
$\mathbf{u}$	velocity field components, m s^{-1}
u	velocity in x direction, m s^{-1}
v	velocity in y direction, m s^{-1}
V	velocity magnitude, m s^{-1}
p	pressure, Pa
n	normal
DR	driven ratio, p_A/p_m
$\dot{e}_y$	energy flux density, W m^{-2}
f	frequency, Hz
k	thermal conductivity, W m^{-1} K^{-1}
k_w	wave number, m^{-1}
p_A	pressure amplitude, Pa
T	temperature, K
T_m	mean temperature, K
T_c	cold heat exchanger temperature, K
T_h	ambient heat exchanger temperature, K
h	enthalpy, J kg^{-1}
h_0	total enthalpy, J kg^{-1}
a	speed of sound, m s^{-1}
c_p	specific heat, J kg^{-1} K^{-1}
R	gas constant, J kg^{-1} K^{-1}
pr	Prandtl number
L	stack length, m
L_1	ambient heat exchanger length, m
L_2	cold heat exchanger length, m
L_3	distance between driver and cold heat exchanger, m
L_4	plates thickness, m
L_5	half distance between plates, m
L_6	gap between stack and ambient heat exchanger, m
L_7	gap between stack and cold heat exchanger, m
L_8	computational domain length, m
q_c	cooling load, W
q_h	rejected heat, W
Δt	time step, s
COP	coefficient of performance

Greek symbols

μ	dynamic viscosity, N s m^{-2}
ω	angular frequency, rad s^{-1}
α	diffusivity, $k/\rho c_p$
ρ	density, kg m^{-3}
τ	cycle time, s
λ	wave length, m
γ	ratio of specific heats
ζ	gas parcel displacement, m

numerical simulation of the thermoacoustic refrigerator driven at large amplitude to consider nonlinear effects. Then the parameters affecting the refrigerating performance, including position of the stack, length of the stack and the heat exchanger, thickness of the parallel plates, and the spacing, were investigated in detail. Zink et al. [8] considered a full 2-D thermoacoustic engine that also includes a refrigerator stack. Acoustic power that was generated in the engine part is used for cooling by the refrigerator. They used k-ε turbulence model for their CFD simulation in the Fluent commercial code. They showed that locating cooling stack closer to the pressure node would yield to a better performance.

In the current work, we use an open source CFD software, namely, OpenFOAM. OpenFOAM benefits from an efficient and flexible implementation of complex physical models in the framework of finite volume discretization. It supports unstructured, polyhedral meshes and massively parallel computing [9]. The full 2D simulation of the thermoacoustic refrigerator is attempted. We aim to investigate successful operation of the device at weak and large amplitude. The thermoacoustic phenomenon is illustrated in details with variations of temperature and velocity profile. The energy flux density over stack and heat exchangers at weak and large amplitude, for the first time, is computed with CFD simulation and validated using the analytical solution.

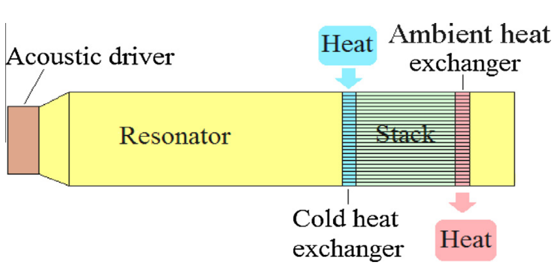

Fig. 1. Thermoacoustic refrigerator.

In this paper, two expressions are frequently employed; "weak amplitude" and "large amplitude". For the conditions that inlet dynamic pressure increases and Mach number becomes greater than 0.1, the results show that nonlinear effects (e.g., pressure node displacement) influence on the operation of the apparatus and are not negligible [1]. Such conditions are defined as "large amplitude". In contrast, "weak amplitude" is refereed to conditions that inlet dynamic pressure cannot delivers enough heat to the ambient heat exchanger or cannot absorbs heat from the cold heat exchanger to makes COP > 0.

2. Governing equations and numerical method

The continuity, momentum and energy equations for compressible flow in a two-dimensional Cartesian coordinate system are as follows:

$$\frac{\partial \rho}{\partial t} + div(\rho \mathbf{u}) = 0, \quad (1)$$

$$\frac{\partial (\rho u)}{\partial t} + div(\rho u \mathbf{u}) = -\frac{\partial p}{\partial x} + div(\mu \; grad \; u) + S_{Mx}, \quad (2)$$

$$\frac{\partial (\rho v)}{\partial t} + div(\rho v \mathbf{u}) = -\frac{\partial p}{\partial y} + div(\mu \; grad \; v) + S_{My}, \quad (3)$$

$$\frac{\partial (\rho h_0)}{\partial t} + div(\rho h_0 \mathbf{u}) = \frac{\partial p}{\partial t} + div(k \; grad \; T) + S_h, \quad (4)$$

where $h_0 = h + \frac{1}{2}(u^2 + v^2)$, $h = c_p(T - T_m)$. The viscosity μ and thermal conductivity of fluid k are temperature dependent. Working gas is air, assumed to be an ideal gas, therefore, the state equation is:

$$p = \rho R T. \quad (5)$$

Energy equation in the solid domain of the parallel-plate stack and heat exchangers is given by:

$$\frac{\partial (\rho c_p T)}{\partial t} = div(k \; grad \; T). \quad (6)$$