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# Brinkman-Forchheimer modeling for porous media thermoacoustic system

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## ABSTRACT

In this paper, analytical studies have been conducted on the flow and thermal fields of unsteady compressible viscous oscillating flow through channels filled with porous media representing stacks in thermoacoustic systems. The flow in the porous material is described by the Brinkman–Forchheimer-extended Darcy model. Analytical expressions for oscillating velocity, temperature, and energy flux density are obtained after linearizing and solving the governing differential equations with long wave, short stack, and small amplitude oscillation approximations. Experimental work is also conducted to verify the temperature difference obtained across the porous stack ends. To produce the experimental results, a thermoacoustic heat pump is designed and constructed where reticulated vitreous carbon (RVC) is used as the stack material. A very good agreement is obtained between the modeling and the experimental results. The expression of temperature difference across the stack ends obtained in the present study is also compared with the existing thermoacoustic literature. The proposed expression surpasses the existing expression available in the literature. The system of equations developed in the present study is a helpful tool for thermal engineers and physicist to design porous stacks for thermoacoustic devices.

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# 1. Introduction

Thermoacoustic devices are energy conversion devices that convert heat to acoustic energy and vice versa. A typical thermoacoustic device consists of a resonator, two heat exchangers, and a stack. The stack supports a longitudinal temperature gradient and is the element in which the desired interchange between thermal and acoustic energies takes place. It is sandwiched between heat exchangers which either supply or remove heat from the stack. Installation of heat exchangers near the stack ends provides an opportunity to supply heat flux inside the resonator (through the cold heat exchanger) and extract the heat flux from the resonator (through the hot heat exchanger) in thermoacoustic heat pumps and refrigerators. Thus a thermoacoustic heat pump can be used to heat the other systems which are connected to the hot heat exchanger, and a thermoacoustic refrigerator can be used to cool the systems which are connected to the cold heat exchanger.

Even though thermoacoustic devices have some inherent benefits; e.g., simplicity, reliability, very long life, and environmental friendliness; they are not very efficient (efficiencies are 20% to 30% of Carnot efficiency) and have low power density when compared with typical compression refrigerators. Several authors have considered different approaches to increase the efficiency (Swift

\* Corresponding author. *E-mail address:* popm.ioan@yahoo.co.uk (I. Pop). [1]), or power density (Poese et al. [2], Bösel et al. [3], and Tijani [4]) of thermoacoustic devices. Poese et al. [2] have designed a thermoacoustic refrigerator to operate at larger pressure amplitudes, since power density in a thermoacoustic device is proportional to the square of the acoustic Mach number (Swift [5]). Another approach to increase the performance of a thermoacoustic device is to decrease the Prandtl number. Viscous friction has a negative effect on the performance of thermoacoustic devices (Swift [5], Tijani [4]). Lower values of Prandtl number can be obtained by using gas mixtures of heavy and light monatomic gases. Tijani [4] showed experimentally that decreasing the Prandtl number (Pr) to 0.2 leads to a coefficient of performance relative to Carnot which is 70% higher than with pure helium (Pr = 0.7). Mahmud and Fraser [6] have examined a new concept of a thermoacoustic system under the influence of a magnetic force. The influence of a magnetic force on the performance of any particular thermoacoustic system has been analyzed using entropy generation analysis.

A different approach to increase the power density of a thermoacoustic device is to use stacks of different arrangements (Bösel et al. [3]) and geometries (Adeff et al. [7]). The stack is the structural heart of a thermoacoustic device. To operate, the stack must conduct thermal energy in a direction perpendicular to the sound wave direction. However, the stack inherently also experiences a finite temperature difference along its length resulting in heat transfer parallel to the sound waves from the hot to the cold heat exchangers. This is an inherent loss mechanism in a stack. In order to reduce this diffusion loss and thereby to enhance the perfor-

<sup>0017-9310/\$ -</sup> see front matter  $\circledcirc$  2011 Elsevier Ltd. All rights reserved. doi:10.1016/j.ijheatmasstransfer.2011.05.004

#### Nomenclature

$a_0$ a complex parameter, $(1 + 1)\sqrt{1} + \phi_1/(2L/d)/\phi_1$ creex symbols $a_0$ a complex parameter, $(1 + 1)\sqrt{1} + \phi_1/(2L/d)/\phi_1$ $a_0$ thermal diffusivity, $m^2 s^{-1}$ $a_0$ pore diameter, $m$ $\beta_1$ thermal diffusivity, $m^2 s^{-1}$ $a_0$ pore diameter, $m$ $\beta_1$ thermal diffusivity, $m^2 s^{-1}$ $a_0$ pore diameter, $m$ $\beta_1$ thermal diffusivity, $m^2 s^{-1}$ $a_1$ pore diameter, $m$ $\beta_2$ thermal diffusivity, $m^2 s^{-1}$ $a_2$ energy flux density vector, $W m^{-2}$ $a_2$ thermal diffusivity, $m^2 s^{-1}$ $a_2$ scond order energy flux density, $W m^{-2}$ $b_2$ tiscoal ponetarian depth, $= \sqrt{2x/\omega}$ $F_2$ global energy flux density, $W m^{-2}$ $b_2$ tiscoal ponetarian depth, $= \sqrt{2x/\omega}$ $F_2$ global energy flux density, $W m^{-2}$ $b_2$ thermal pretation depth, $= \sqrt{2x/\omega}$ $f_2$ scoal diffusivity, $m^{-2} s^{-1}$ $\phi_1$ thermal pretation depth, $= \sqrt{2x/\omega}$ $f_2$ scoal diffusivity, $m^{-2} s^{-1}$ $\nabla T_m$ mean temperature gradient, $= \partial_m/2 s^{-1}$ $f_2$ Swift's function of thermoacoustics, $= tanh (a_0y_0)/(b_0y_0)$ $\nabla T_m$ dynamic viscosity of the fulid, $N m^{-2} s^{-1}$ $f_4$ Swift's function of thermoacoustics, $= tanh (b_0y_0)/(b_0y_0)$ $\sigma$ viscous stress tensor, $N m^{-2}$ $f_4$ Swift's function of thermoacoustics, $= tanh (b_0y_0)/(b_0y_0)$ $\sigma$ iccrual frequency, rad $s^{-1}$ $f_4$ thermal ponetarion displane diffusivity, $m^{-2} s^{-1}$ $\sigma$ heat capacity ratio, $= \phi $	~	$(1+i)\sqrt{1+(1/2iD_{e})}/s$	Creation	who lo
$ \begin{array}{llllllllllllllllllllllllllllllllllll$	a <sub>0</sub>	a complex parameter, = $(1 + i)\sqrt{1 + \phi/(2iDa)/\delta_v}$	Greek sy	mbols
$ \begin{array}{cccc} P & \mbox{spectrac heat of the fluid at constant pressure, j kg * k * j } & \mbox{trunt} p \\ p & \mbox{trunt} p \\ p & \mbox{spectrac} p \\ p$	D <sub>0</sub>	a complex parameter, = $(1+i)\sqrt{o}/o_k$	α	thermal diffusivity, $m^2 s^2$
$ \begin{array}{llllllllllllllllllllllllllllllllllll$	$C_p$	specific heat of the fluid at constant pressure, J kg · K	β	thermal expansion coefficient, K
$ \begin{array}{cccccccccccccccccccccccccccccccccccc$	Da	Darcy number, = $K/\delta_v^2$	γ	isobaric to isochoric specific heat ratio, = $C_p/C_v$
$ \begin{array}{cccccccccccccccccccccccccccccccccccc$	d	pore diameter, m	$\Gamma_0$	temperature gradient ratio in the presence of a porous
$ \begin{array}{llllllllllllllllllllllllllllllllllll$	Ē	energy flux density vector, W m <sup>-2</sup>		medium, = $(T_{ad}/T_{sw}) [1 + \phi/(2iDa)]$
$ \begin{array}{llllllllllllllllllllllllllllllllllll$	$E_2$	second order energy flux density, W m <sup>-2</sup>	$\delta_v$	viscous penetration depth, = $\sqrt{2v/\omega}$
$ \begin{array}{llllllllllllllllllllllllllllllllllll$	E <sub>2</sub>	global energy flux, W	$\delta_k$	thermal penetration depth, = $\sqrt{2\alpha_f/\omega}$
$ \begin{array}{llllllllllllllllllllllllllllllllllll$	F	Forchheimer's coefficient	$\nabla T_m$	mean temperature gradient, = $\partial T_m / \partial x$
$ \begin{array}{llllllllllllllllllllllllllllllllllll$	f	frequency of oscillation, Hz	$\nabla p_1$	pressure gradient, = $\partial p_1 / \partial x$
$ \begin{array}{lll} \hline f_v & \mbox{complex conjugate of } f_v & \mu & \mbox{dynamic viscosity of the fluid, N m^{-2} s} \\ \hline f_k & \mbox{complex conjugate of } f_k & \mbox{dynamic viscosity, m^2 s^{-1}} \\ \hline f_k & \mbox{complex number, = } \sqrt{-1} & \mbox{dynamic viscosity, and s^{-1}} \\ \hline f_k & \mbox{ermeability of the porous medium, m^2} & \mbox{dynamic viscosity of the fluid, kg m^{-3}} \\ \hline f_k & \mbox{ermeability of the porous medium, m^2} & \mbox{dynamic viscosity of the fluid, kg m^{-3}} \\ \hline f_k & \mbox{thermal conductivity, W m^{-1} K^{-1}} & \mbox{the max conductivity, W m^{-1} K^{-1}} & \mbox{the regular of the stack, m} & \mbox{a dimensionless parameter, = } \phi(2iDa)^{-1} + \frac{2iu_0\phi}{i\omega\sqrt{K}} \\ \hline f_k & \mbox{length of the stack, m} & \mbox{a vavelength, m} & \mbox{dynamic viscosity of the porous medium, = void volume/total volume/total volume/total volume/total volume/total volume/total volume/total volume \\ \hline f_k & \mbox{the max stail component of the fluid, =} \delta_v^2/\delta_k^2 & \mbox{dynamic viscosity of the porous medium, = void volume/total value time, s & \mbox{transverse velocity, m s^{-1}} & \mbox{transverse value} & t$	$f_v$	Swift's function of thermoacoustics, = $tanh (a_0y_0)/(a_0y_0)$	$\nabla T_{\rm cr}$	critical temperature gradient, °C m <sup>-1</sup>
	$\tilde{f}_v$	complex conjugate of $f_v$	μ	dynamic viscosity of the fluid, N m <sup>-2</sup> s
$ \begin{array}{cccc} \hline f_k & \mbox{complex conjugate of } f_k & \mbox{complex number, } = \sqrt{-1} & complex number$	$f_k$	Swift's function of thermoacoustics, = $tanh (b_0y_0)/(b_0y_0)$	v	kinematic viscosity, m <sup>2</sup> s <sup>-1</sup>
icomplex number, = $\sqrt{-1}$ $\sigma$ heat capacity ratio, = $\phi + (1 - \phi)(\rho_{sm} C_{sm})/(\rho C_p)$ henthalpy, J kg <sup>-1</sup> $\sigma$ circular frequency, rad s <sup>-1</sup> Kpermeability of the porous medium, m <sup>2</sup> $\rho$ density of the fluid, kg m <sup>-3</sup> kthermal conductivity, W m <sup>-1</sup> K <sup>-1</sup> $\tau$ time period, $= 2\pi/\omega$ k0inverse of the wave number, $= \omega/c$ $\Psi$ a dimensionless parameter, $= \phi(2iDa)^{-1} + \frac{2Fu_{0}\phi}{i\omega\sqrt{K}}$ Lslength of the stack, m $\lambda$ wavelength, mppressure, N m <sup>-2</sup> $\Pi$ width of the plate, mPrPrandtl number of the fluid, $= \delta_v^2/\delta_k^2$ $\phi$ porosity of the porous medium, = void volume/total volume/total volume/total volume, swift number, $= y_0/\delta_v$ SwSwift number, $= y_0/\delta_v$ uaxial component of the fluctuating velocity inside the channel, m s <sup>-1</sup> 1Udimensionless axial velocity (see Eq. (35)) $av$ average valueua reference velocity, m s <sup>-1</sup> mmean valueV, vvelocity vector, m s <sup>-1</sup> mmean valueV, vvelocity vector, m s <sup>-1</sup> smproperties of the solid matrix material in the porousW2total work flux, Wmvxaxial distance, mwvalue at wallytransverse distance in the porousSymbolsmedium, $= y/y_0$ $\Re$ []real part of an expressiony0half width of a channel, m $\Im$ []imaginary part of an expression	$\tilde{f}_k$	complex conjugate of $f_k$	σ	viscous stress tensor, N m <sup>-2</sup>
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$ \begin{array}{cccccccccccccccccccccccccccccccccccc$	$k_0$	inverse of the wave number, = $\omega/c$	$\Psi$	a dimensionless parameter, = $\phi(2iDa)^{-1} + \frac{2Fu_0\phi}{4\pi}$
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	$y_0$	half width of a channel, m	รบไ	imaginary part of an expression
		·	-11	

mance of a thermoacoustic device; Bösel et al. [3] have proposed an alternate stack arrangement. Bösel et al.'s approach [3] is to replace the typical parallel plate stacks that are much longer than the acoustic displacement amplitude, by parallel plate segments which are only a fraction of the displacement amplitude long and randomly oriented to each other. This simplified stack arrangement is expected to benefit from improved heat transfer characteristics and an anisotropic thermal conductivity (Bösel et al. [3]).

Adeff et al. [7] have used porous RVC as a stack material in thermoacoustic devices. RVC is a rigid glassy carbon material, with a porous sponge like structure, a large surface area, and a low thermal conductivity. Based on the experimental measurements, Adeff et al. [7] shows that RVC has a definite potential as a low cost, easy to fabricate material suitable for use in all types of thermoacoustic devices.

Recently, Mahmud and Fraser [8] and Tasnim and Fraser [9] have presented investigations that consider the stack plate to be composed of multiple parallel plates and the space between the stack plates is occupied by porous materials. For a steady state case, Hunt and Tien have showed experimentally [10] that the porous medium enhances heat transfer from a surface as compared to predicted results for slug or for laminar flow in a channel. Mahmud and Fraser [8] consider a simplification of Vafai and Tien's porous medium model [11] for the modeling of momentum equation. Mahmud and Fraser's work [8] show that if a porous medium is

used in between the thin parallel plates, much more efficient thermoacoustic engine designs are possible in the  $Da (= K/\delta_v^2$ , where Kis the permeability of the porous medium and  $\delta_v$  is the viscous penetration depth [5]) range 0.1–1.0 and for  $S_w$  (non-dimensional stack plate spacing)  $\geq 2$ . Tasnim and Fraser [9] have presented an analytical investigation considering Darcy flow model for the modeling of momentum equation. The thickness of the channel wall is considered to be nonzero in Tasnim and Fraser [9] and the entire problem is treated as a conjugate heat transfer problem. Tasnim and Fraser [9] have showed that refrigeration performance comparable to that of a traditional plastic parallel plate stack is achievable using RVC foam ( $\phi = 0.95$ ,  $Lc_k = 2.11$ ,  $Lc_k = y_0/\delta_k$ , where  $y_0$  and  $\delta_k$  are the half width of the stack plate and the thermal penetration depth [5], respectively) as a porous medium.

A thermoacoustic theory for a bulk porous medium is developed by Roh et al. [12] based on parallel, capillary-tube-based theories. The authors have introduced the tortuosity, the viscous dynamic shape factor, and the thermal dynamic shape factor to extend thermoacoustic theory to a bulk porous medium. Comparisons of thermoacoustic properties (of RVC and aluminum foam developed by Roh et al. [12]) show good agreement with experiment.

A simplified model based on steady-flow correlations and Lagrangian approach is developed by Jung and Matveev [13] for thermoacoustic analysis of short transverse-pin stacks with moderate temperature differences. This method provides fast Download English Version:

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