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Wall and fluid inlet temperature effect on heat transfer in incompressible laminar oscillating flows

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Abstract

This paper deals with the problem of oscillating flows occurring in devices such as Stirling or thermoacoustic engines and refrigerators. Since the global governing equations cannot be solved, the authors propose to introduce a few simplifications; the most simplifying reduction is that the fluid is assumed to be incompressible. However, specific attention is paid to describing the flow characteristics that's why the Lagrangian formalism which allows the individual study of each fluid particle is adopted. Thereby each particle contribution to global thermal effects can be evaluated and the gas temperature profiles along the exchanger can be computed. Various situations are presented including the case of a non-uniform temperature at the wall and a phase lag between pressure and temperature at the fluid entrance. The efficiency of the wall to fluid thermal exchange is analyzed. The authors show that this exchange depends upon two important parameters: the geometric ratio between the exchanger length and the particle oscillating displacement, and a thermal parameter "β", governing the temperature profiles and related to the Prandtl number, the operating frequency and the phase lag between the instantaneous heat flux and the wall to fluid temperature difference.

Keywords: Thermoacoustic; Pulse tube; Stirling; Research; Heat transfer; Laminar flow; Periodic phenomenon; Wave; Parameter; Temperature; Efficacy; Heat exchange

Effet des températures d'entrée de fluide et des températures de paroi sur l'échange thermique d'un fluide incompressible en écoulement oscillant

Mots clés : Thermoacoustique ; Tube à pulsation ; Stirling ; Recherche ; Transfert de chaleur ; Écoulement laminaire ; Phénomène périodique ; Onde ; Paramètre ; Température ; Efficacité ; Échange de chaleur

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Nomenclature

c	specific heat capacity at constant pressure ($\mathrm{J\,kg^{-1}\,K^{-1}}$)
C_f	friction factor
D	gas displacement (m)
d_h	hydraulic diameter (m)
e	thickness (m)
E	exchange effectiveness
F_1, F_1'	functions Eqs. (22) and (34)
H	convection heat coefficient ($\mathrm{W\,m^{-2}\,K^{-1}}$)
$\dot{h}, \dot{H}$	local and radial average of enthalpy flux ($\mathrm{W\,m^{-2}}$)
j	complex imaginary ($j^2 = -1$)
l	length (m)
Nu, Nu_{R1}, Nu_{I1}	Nusselt number, real part, imaginary part
P	pressure (Pa)
Pr	Prandtl number
Q	thermal energy (J)
Re	Reynolds usual number
S_{pass}	internal cross-section ($\mathrm{m^2}$)
T	time (s)
T	temperature (K)
$t^* = \omega t$	non-dimensional time
u, v	velocities for x and y directions ($\mathrm{m\,s^{-1}}$)
$u^* = (2u/\omega D)$	non-dimensional fluid velocity
W_{om}	Womersley/Stokes number
x	axial coordinate (m)
$x^* = (x/(l_0/2))$	non-dimensional axial coordinate
y, z	co-ordinates (m)
$y^* = (y/(e/2))$	non-dimensional co-ordinates

Greek

λ	parameter Eq. (11)
λ_i	thermal conductivity i ($\mathrm{W\,m^{-1}\,K^{-1}}$)
μ	dynamic viscosity ($\mathrm{kg\,m^{-1}\,s^{-1}}$)
ρ	density ($\mathrm{kg\,m^{-3}}$)
ω	pulsation ($\mathrm{rad\,s^{-1}}$)
ξ	Lagrangian variable
Δ	thermal gradient ($\mathrm{K\,m^{-1}}$)
ΔT_{rad}	radial gradient $T_g - T_{-1}$ (K)
ΔT_{long}	longitudinal gradient $T_1 - T_{-1}$ (K)
Λ_L	fluid relative displacement

Subscripts and superscripts

g	gas
in, in′	entrance in $x^* = -1$ and in $x^* = 1$
l_0	extremity in $x = l_0$
rad, long	radial and longitudinal
t	tube
0	initial, for $x = 0$, of reference
out, out′	Exiting in $x^* = -1$ and in $x^* = 1$
1, −1	to right or left extremity

Operators

p^*, T^*	normalized magnitude with respect to reference value p_0, T_0
$\Im$	imaginary part
$\Re$	real part
$\bar{x}$	x cross-section average
$\lvert x \rvert$	x magnitude
$\langle x \rangle$	x temporal average on a semi-period

1. Introduction

Numerous varieties of thermal machines operate with an internal oscillating fluid flow: Stirling cooler [1], all type of Stirling engines including the kinematic engines and the free piston or free displacer engines: Martini model [2], Ringbom model [3] and hot air flow engine like Manson model. All versions of Pulse Tube Refrigerators (PTR, OPTR, DIOPTR [4,5], etc.) and thermoacoustic systems based on standing or progressive waves (Ceperley ring [6]) are also concerned. There are also a lot of hybrid systems like Gifford-Mc Mahon and Vuilleumier coolers [7] or fluidynes pumps and liquid piston Stirling machines [8,9]. All these machines are known for many years but are not of a common use yet. However, they may take a bigger place in a near future because of their abilities to operate with various energy sources brought by external ways (hence removing most technical problems occurring in internal combustions): biogas, solar energy, geothermic, heat collected downstream industrial processes, and thermoacoustic waste heat upgrading.

In most cases, the mechanical conception of these machines is quite simple compared to gas turbines, internal combustion engines or classic refrigerating and gas liquefaction devices. Note that some of them (purely thermoacoustic devices) do not require any mobile part. In this paper the authors propose to focus their interest on a particular part of these systems: the heat exchangers where at least one fluid operates in oscillating flow mode. Moreover, such devices can improve many industrial processes; for example numerous works [10,11] show the interest of oscillating flows in micro-heat exchangers used as coolers for electronic components.

The oscillating fluid flow causes thermal exchange phenomena which are not well estimated yet. Indeed, due to a lack of knowledge, most researchers and engineers use classic heat exchangers calculus, assuming fluids moving in a steady state flow regime which is not in connection with reality. As mentioned above, very few theoretical or