



## Oscillatory flow and temperature fields in an open tube with temperature difference across the ends

T.R. Ashwin<sup>a</sup>, G.S.V.L. Narasimham<sup>a,\*</sup>, Subhash Jacob<sup>b</sup>

<sup>a</sup> Department of Mechanical Engineering, Indian Institute of Science, Bangalore 560 012, India

<sup>b</sup> Centre for Cryogenic Technology, Indian Institute of Science, Bangalore 560 012, India

### ARTICLE INFO

#### Article history:

Received 5 July 2010

Received in revised form 12 March 2011

Accepted 21 March 2011

Available online 15 April 2011

#### Keywords:

Pulsating flow

Pulse tube

Heat transfer

Laminar

Circular tube

Conjugate convection

### ABSTRACT

The oscillating flow and temperature field in an open tube subjected to cryogenic temperature at the cold end and ambient temperature at the hot end is studied numerically. The flow is driven by a time-wise sinusoidally varying pressure at the cold end. The conjugate problem takes into account the interaction of oscillatory flow with the heat conduction in the tube wall. The full set of compressible flow equations with axisymmetry assumption are solved with a pressure correction algorithm. Parametric studies are conducted with frequencies of 5–15 Hz, with one end maintained at 100 K and other end at 300 K. The flow and temperature distributions and the cooldown characteristics are obtained. The frequency and pressure amplitude have negligible effect on the time averaged Nusselt number. Pressure amplitude is an important factor determining the enthalpy flow through the solid wall. The frequency of operation has considerable effect on penetration of temperature into the tube. The density variation has strong influence on property profiles during cooldown. The present study is expected to be of interest in applications such as pulse tube refrigerators and other cryocoolers, where oscillatory flows occur in open tubes.

© 2011 Elsevier Ltd. All rights reserved.

### 1. Introduction

Oscillating flow through passages like pipes and conduits is of interest in biological applications in relation to blood flow, industrial heat exchangers, manifolds, combustion and regenerators of Stirling and pulse tube cryocoolers. In the studies aimed at determining the heat transfer and friction between the wall and oscillating flow, the pulsatile flow field is assumed to consist of a steady Poiseuille flow part and a purely oscillatory part. Such a flow could enhance heat transfer by breaking down the boundary layer. In other words, the pulsations can have considerable effect on the rate of heat transfer and thermal resistance due to the alteration of thickness of the thermal boundary layer [1]. The flow and temperature distributions that occur in oscillatory flow in a tube with the end portions of the tube maintained at different temperatures is of interest in pulse tube refrigerators. The present study is motivated by such an application. Due to the nature of the problem and the large temperature differences that exist, the problem is analyzed using the full set of compressible flow governing equations with the assumption of axis symmetry.

Faghri et al. [2] reported a study on the heat transfer from a cylindrical pipe where periodic flow is superimposed on a fully developed steady laminar flow. It was shown that the interaction between velocity and temperature pulsations generates an extra diffusivity akin to eddy diffusivity, contributing to larger heat transfer rates.

There are other numerical investigations such as those of Cho and Hyun [3] which focused attention on heat transfer characteristics of a pulsating flow in a pipe. Using unsteady boundary-layer equations, Cho and Hyun [3] found that the time-mean axial velocity profiles and the Nusselt number were largely unaffected by the changes in frequency parameter and amplitude of oscillation. The skin friction was found to be greater than that of steady flow. The influence of oscillation on skin friction is appreciable both in terms of magnitude and phase relation.

Al-Haddad and Al-Binaly [4] presented a correlation for the prediction of the heat transfer coefficient in a heating process for steady and pulsating flow of air through a circular pipe. The pulsating frequency is between 5 and 60 Hz and a combined dimensionless number composed of the Reynolds number and a dimensionless flow frequency was used to correlate thermal behavior with flow parameters. When this number was below  $2.1 \times 10^5$ , there was no significant improvement in heat transfer. Similar work was reported by Chattopadhyay et al. [5] for simultaneously developing pulsating incompressible laminar flow in a pipe with constant wall temperature. The analysis proved that pulsation has a negligible ef-

\* Corresponding author. Tel.: +91 80 22932971; fax: +91 80 23600648.

E-mail address: [mecgsvln@mecheng.iisc.ernet.in](mailto:mecgsvln@mecheng.iisc.ernet.in) (G.S.V.L. Narasimham).

## Nomenclature

|           |                                                                      |
|-----------|----------------------------------------------------------------------|
| $c_p$     | constant pressure specific heat ( $\text{J kg}^{-1} \text{K}^{-1}$ ) |
| $D, d$    | diameter (m)                                                         |
| $\dot{E}$ | enthalpy flow (W)                                                    |
| $Ec$      | Eckert number, $v_c^2/c_p, c T_c$ dimensionless                      |
| $f$       | frequency (Hz)                                                       |
| $g$       | acceleration due to gravity ( $\text{m s}^{-2}$ )                    |
| $H$       | length of tube (m)                                                   |
| $h$       | heat transfer coefficient ( $\text{W m}^{-2} \text{K}^{-1}$ )        |
| $k$       | thermal conductivity ( $\text{W m}^{-1} \text{K}^{-1}$ )             |
| $L$       | length (m)                                                           |
| $\dot{M}$ | mass flow rate ( $\text{kg s}$ )                                     |
| $Nu$      | Nusselt number, $hR_i/k$ (dimensionless)                             |
| $p$       | pressure (Pa)                                                        |
| $Pr$      | Prandtl number, $\mu c_p/k$ (dimensionless)                          |
| $R$       | radius (m)                                                           |
| $Re$      | Reynolds number, $\omega R_i^2 \rho / \mu$ (dimensionless)           |
| $T$       | temperature (K)                                                      |
| $t$       | time (s)                                                             |
| $v$       | velocity ( $\text{m s}^{-1}$ )                                       |

### Greek symbols

|              |                                           |
|--------------|-------------------------------------------|
| $\Delta T_c$ | characteristic temperature difference (K) |
| $\gamma$     | ratio of specific heats (dimensionless)   |
| $\epsilon$   | overheat ratio (dimensionless)            |

|          |                                                        |
|----------|--------------------------------------------------------|
| $\mu$    | dynamic viscosity ( $\text{kg m}^{-1} \text{s}^{-1}$ ) |
| $\nu$    | kinematic viscosity ( $\text{m}^2 \text{s}^{-1}$ )     |
| $\rho$   | density ( $\text{kg m}^{-3}$ )                         |
| $\omega$ | angular frequency ( $\text{rad s}^{-1}$ )              |

### Subscripts

|    |                        |
|----|------------------------|
| a  | amplitude              |
| av | average                |
| c  | characteristic         |
| e  | evaporator or cold     |
| f  | fluid region           |
| h  | hot or ambient         |
| i  | inner                  |
| o  | outer, charge pressure |
| p  | time period            |
| r  | radial direction       |
| s  | solid region           |
| w  | wall                   |
| z  | axial direction        |

### Superscript

|   |                        |
|---|------------------------|
| * | dimensionless quantity |
|---|------------------------|

fect on time-averaged heat transfer. Moschandreou and Zamir [6] reported that the rate of heat transfer is not significantly affected when the frequency is very low or very high, while Guo and Sung [7] reported that heat transfer augmentation occurs at large amplitudes of oscillation. For turbulent pulsating flow, Wang and Zhang [8] found that there exists an optimum Womersley number at which heat transfer is enhanced. Heat transfer characteristics of pulsating turbulent air flow in a pipe heated at uniform heat flux were experimentally investigated [9]. The results show that the Nusselt number is strongly affected by pulsation frequency, the relative Nusselt number increasing or decreasing depending on the frequency range. Zhao and Cheng [10] presented a heat transfer correlation in terms of the kinetic Reynolds number, dimensionless amplitude of oscillation, length-to-diameter ratio and the Prandtl number. Khabakhpasheva et al. [11] experimentally observed considerable phase shift between the velocity and pressure gradient in pulsating flows of viscoelastic fluids.

Hemida et al. [12] studied the pulsation effect on heat transfer in laminar incompressible flow in a duct for both thermally fully developed and developing conditions. The results show that the pulsation has negligible role in fully developed region but it has greater sensitivity in thermally developing region. For isothermal and isoflux boundary conditions, the effect of pulsation on the time average heat transfer coefficient tends to be negative, but remains relatively small. But non-linear boundary conditions (e.g. radiation and natural convection) combined with pulsation resulted in a noticeable enhancement of the time average Nusselt number.

Other studies on pulsating turbulent flows include those of Li and Xu [13], Bouvier et al. [14] and He and Jackson [15].

From the literature, it can be seen that pulsating gas flow in open tubes subjected to end-to-end temperature difference and its interaction with the tube wall has not received attention. Such flows find practical application in various cryocoolers and other thermal systems. In an earlier paper, the present authors reported a numerical simulation for the complete pulse tube refrigerator operating high frequencies [16]. The present paper is motivated by the need for a detailed study of the pulse tube for systems in the frequency range 5–15 Hz.

## 2. Formulation

### 2.1. Geometry

Fig. 1 shows the physical model and coordinate system. Cylindrical polar coordinate system is chosen with the assumption of axisymmetric flow and temperature distributions. The model consists of a cylindrical pipe with finite wall thickness and both the ends open. The flow and temperature distributions are assumed to be axisymmetric. The  $r$ -axis is perpendicular to the vertical axis and points outwards in a radial direction. The positive direction of the  $z$ -axis coincides with the vertical axis and points upwards. The gravity vector is parallel to the  $z$ -axis and acts vertically downwards. The portions of the tube beyond the open ends belong to the cold and warm heat exchangers of a pulse tube refrigerator. Thus the working medium when entering one of the ends does so at a cryogenic temperature  $T_e$ . Similarly the working medium entering the other end of the tube is at a higher temperature  $T_h$ . The oscillating flow in the tube is driven by a sinusoidally varying pressure at the cold end of the tube. Since the so called DC component is absent, the fluid flow during a cycle takes place partly into and partly out of the tube at either end. The height of the tube and the inner and outer radii are  $H$ ,  $R_i$  and  $R_o$ , respectively. Clearly the wall thickness  $\delta_w$  is  $R_o - R_i$ . The oscillating heat transfer between the wall and the gas is taken into account through the coupling between the fluid and the solid at the interface. The annular surfaces of the solid at the tube ends are assumed to be insulated.

### 2.2. Non-dimensionalisation

The characteristic length  $L_c$  is taken as the inner radius  $R_i$  of the tube and the characteristic temperature  $T_c$  as the hot heat exchanger temperature  $T_h$ , which is the ambient temperature. The geometrical parameters of the problem are the dimensionless inner radius  $R_i^*$  of the tube, dimensionless height  $H^*$  and the dimensionless wall thickness  $\delta_w^*$ . The characteristic density  $\rho_c$  corresponds to the state of helium at the charge pressure  $p_o$  and ambient temperature  $T_h = T_c$ . The characteristic thermal conductivity, specific heat

Download English Version:

<https://daneshyari.com/en/article/659970>

Download Persian Version:

<https://daneshyari.com/article/659970>

[Daneshyari.com](https://daneshyari.com)