PERGAMON

International Journal of Heat and Mass Transfer 42 (1999) 3523–3527

International Journal of HEAT and MASS TRANSFER

Technical Note

Transient cool-down of a porous medium in pulsating flow

J.W. Paek[a], B.H. Kang[b], J.M. Hyun[a,*]

[a]*Department of Mechanical Engineering, Korea Advanced Institute of Science and Technology, Taejon 305-701, South Korea*
[b]*Thermal/Flow Control Research Center, Korea Institute of Science and Technology, Seoul 130–650, South Korea*

Received 12 June 1998; received in revised form 22 January 1999

1. Introduction

Heat transfer characteristics between porous media and throughflow have been the subject of extensive recent studies. In many modern technological applications of porous media, the fluid flow is often unsteady in nature. Examples can be found in an automobile catalytic reactor attached to the reciprocating-engine exhaust gas and in a thermal regenerator of the Stirling engine [1–3]. However, the majority of existing literature deals with the case of steady throughflows, and reports on heat transfer characteristics involving an unsteady flow passing through porous media are scarce.

A pulsating flow, i.e., an oscillating component added to the mean flow, is a more realistic representation of such innovative engineering applications [3–5]. Sozen and Vafai [4] made numerical investigations into the heat storage by a pulsating compressible flow by using a steady convective heat transfer correlation.

The purpose of the present work is to portray the heat transfer characteristics of a pulsating, in distinction from an oscillating, flow passing through a porous material. A wind-tunnel experiment was carried out. The temperature fields in the porous medium were monitored, and the explicit influence of flow pulsation was delineated.

2. Experimental

A special-purpose wind tunnel was fabricated. The dimension of the test section was 50 mm in diameter (D) and 100 mm in length (L), as sketched in Fig. 1. These dimensions are typical of practical applications [1,2]. The steady main air stream was supplied and regulated by a rotameter to the tapered passage prior to entering the test section. The oscillating component of the inflow was generated by a piston-cylinder mechanism. The cylinder and the tapered passage were connected by a flexible tube, and this arrangement absorbed vibrations of the piston-cylinder assembly. A rigid meshed-screen 20 mm thick was installed as a straightener to produce a uniform pulsating flow.

Two types of packed carbon steel spheres of diameters 3.0 mm and 6.35 mm were employed to constitute the porous material. The density, specific heat, thermal conductivity of carbon steel were 7850 kg m^{-3}, 500 J (kg K)$^{-1}$, and 52 W (K m)$^{-1}$, respectively. A soft insulating layer of 3 mm in thickness was attached to the inside surface wall of the test section to minimize the channeling effect [6]. Negligibly small cross-sectional variations of flow variables were detected by a hotwire anemometer at the inlet and exit of the test section, and these render justification to the one-dimensional flow assumption in the test section.

The specific experimental procedures call for the monitoring of the temperature changes in the porous material. It is, therefore, important to insure that, at the inlet of test section, a spatially-uniform inflow at constant ambient temperature should be maintained. Engineering plastics of very low thermal conductivity were used to construct the walls of the test section with a view toward maximizing insulation effects and minimizing the thermal capacity of the walls. Also, in order to secure a spatially-uniform temperature field in the porous medium at the initial state, a constant-temperature chamber was utilized to house the test section. Before the experiment was commenced, the test sec-

* Corresponding author. Tel.: +82-42-869-3012; fax: +82-42-869-3210.
E-mail address: jmhyun@cais.kaist.ac.kr (J.M. Hyun)

0017-9310/99/$ - see front matter
PII: S0017-9310(99)00039-3

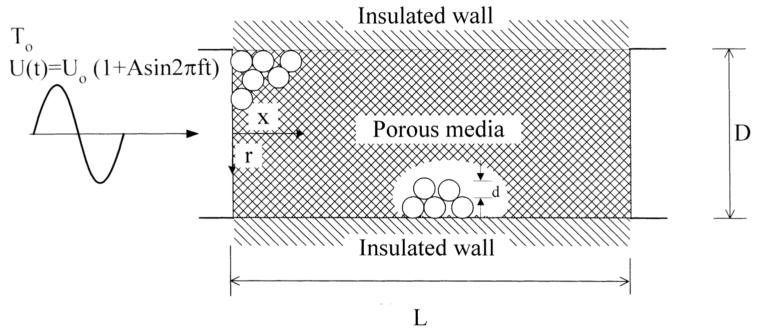


Fig. 1. Flow configuration.

tion, which contained the porous medium, was allowed to reach thermal equilibrium with this chamber at the pre-assigned temperature T_{in}. At the start of the experiment, the test section was instantaneously connected to the coupling nozzle of the regulated air flow of the pre-set temperature T_0. The repeatability errors involved in the hardware setup were carefully analyzed, and they were found to be very small.

Temperature measurements were taken along the centerline of the test section by using T-type thermocouples (0.12 mm in diameter). One thermocouple each was placed at the center of each end of the test section, i.e., $X \equiv x/L = 0.0$ and 1.0. Four thermocouples were inserted inside the porous medium at $X = 0.2$, 0.4, 0.6, and 0.8. Care was exercised to prevent the thermocouple junction from being in direct contact with the packed spheres. For this purpose, each thermocouple was wrapped with a thin steel tube of 1.0 mm in outer diameter. In addition, a helix-shaped guard of 1.5 mm in outer diameter, made of stiff steel, was attached to the tip of this tube.

The output of the thermocouples was processed by an IBM P/C, interfaced with a thermocouple sensing board. The resolution of the temperature measurement was $\pm 0.3°C$. The data acquisition apparatus handled 16 data sets of 50 averaged signals per channel in a second.

3. Results and discussion

To the lowest order of approximation, the pulsating flow at the inlet, measured by a miniature I-type hot-wire anemometer, can be represented as

$$U(t) = U_0(1 + A \sin 2\pi f t) \tag{1}$$

with the base frequency f, the nondimensional amplitude A and the uniform throughflow velocity U_0. The range of the Womersley number, $\beta \equiv d\sqrt{2\pi f/\nu}$, in which d denotes the particle diameter and ν the kinematic viscosity of the fluid, is $\beta \sim O(1)$ in the present study ($f = 0.5$–3.0 Hz). This is typical of the thermal regenerator characteristics of the Stirling engine [1].

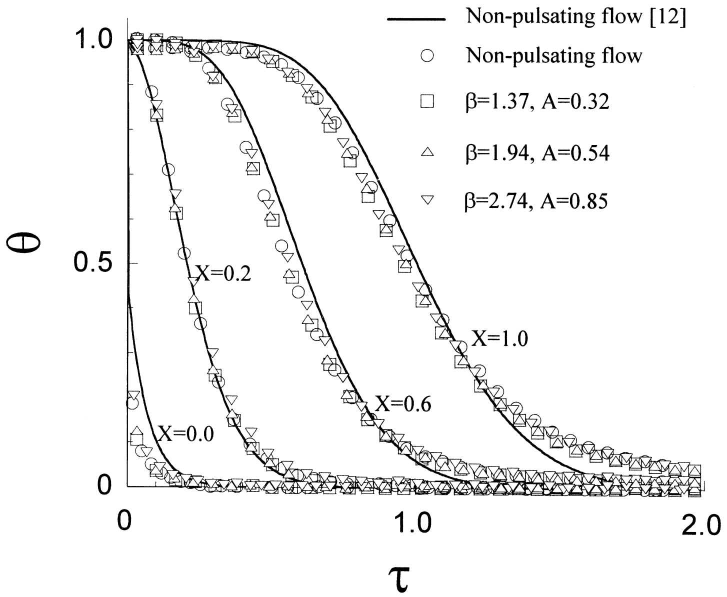
In actual runs, the initial temperature of the entire system was set $T_{in} = 100°C$. At the initial instant $t = 0$, the pulsating air throughflow at temperature $T_0 = 24°C$ was turned on to cool the porous material in the test section. These experimental particulars, including the specifications of the porous medium and throughflow, were selected based on realistic applications and previous works [2,7–9]. The bulk porosity ε of the porous region was measured to be 0.38 and 0.40, respectively, for the packed spheres of 3.0 mm and 6.35 mm in diameter (d).

The Darcy number for packed spheres has been approximated as [4,5]

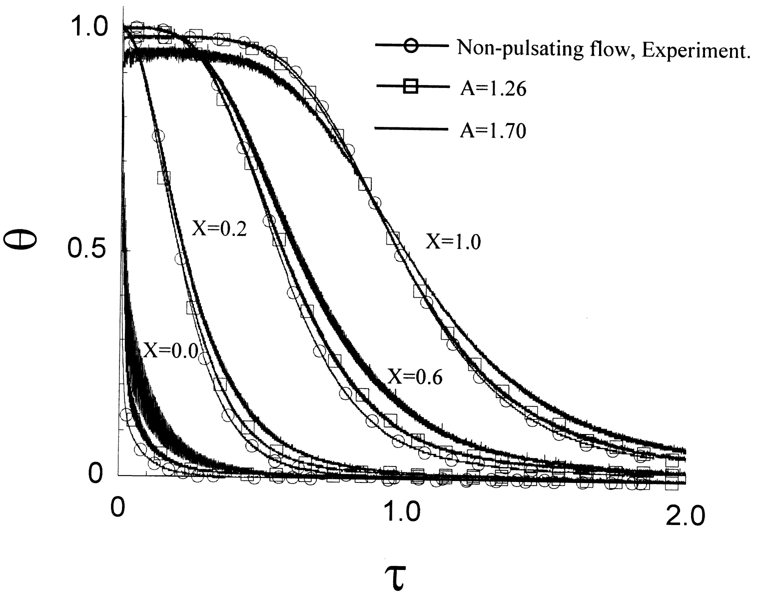
$$Da \cong \frac{\varepsilon^3 d^2}{150(1-\varepsilon)^2 D^2}. \tag{2}$$

The qualitative hydrodynamic features in the porous medium of a pulsating flow can be modeled as a one-dimensional slug flow, and, furthermore, the gross flow properties are little affected by the size of the spheres when the Darcy number is less than 10^{-4} [5]. This gives support to the one-dimensional flow assumption invoked in the present study for the sphere of $d = 3.0$ mm [corresponding to $Da \cong 4.6 \times 10^{-6}$ from Eq. (2)] as well as of $d = 6.35$ mm (corresponding to $Da \cong 2.5 \times 10^{-5}$).

Firstly, a series of runs was made for $A < 1.0$. This implies that no reverse flow occurs. Fig. 2 illustrates the temperature (θ) evolutions in the porous medium vs time (τ). The Reynolds number based on U_0, $Re = U_0 d/\nu$, was set, $Re = 181$. The dimensionless temperature θ and dimensionless time τ are defined as $(T - T_0)/(T_{in} - T_0)$ and $\rho_f c_f U_0 t/(1-\varepsilon)\rho_s c_s L$, in which ρ and c denote density and specific heat, and subscripts f and s refer to fluid and solid particles, respectively. The selection of this particular form of nondimensional time τ was guided by the time scale argument of the preceding studies [9]. The readings from the thermo-



(a)



(b)

Fig. 2. θ–τ curves. $Re = 181$ ($d = 3.0$ mm). Solid line indicates the analytical results of ref. [12]: (a) for $A < 1.0$; (b) for $A > 1.0$. $\beta = 1.73$.

couple probes are indicative of the temperatures of the fluid flowing through the packed bed [7,10,11]. The attenuation of the temperature oscillation by the helix-shaped guard in the probe was measured to be about 15% smaller than that from the probe with no guard. The cool-down characteristics, as exhibited in Fig. 2(a), remain substantially unaltered as β and A vary, in the parameter spaces of present concern. The predic-

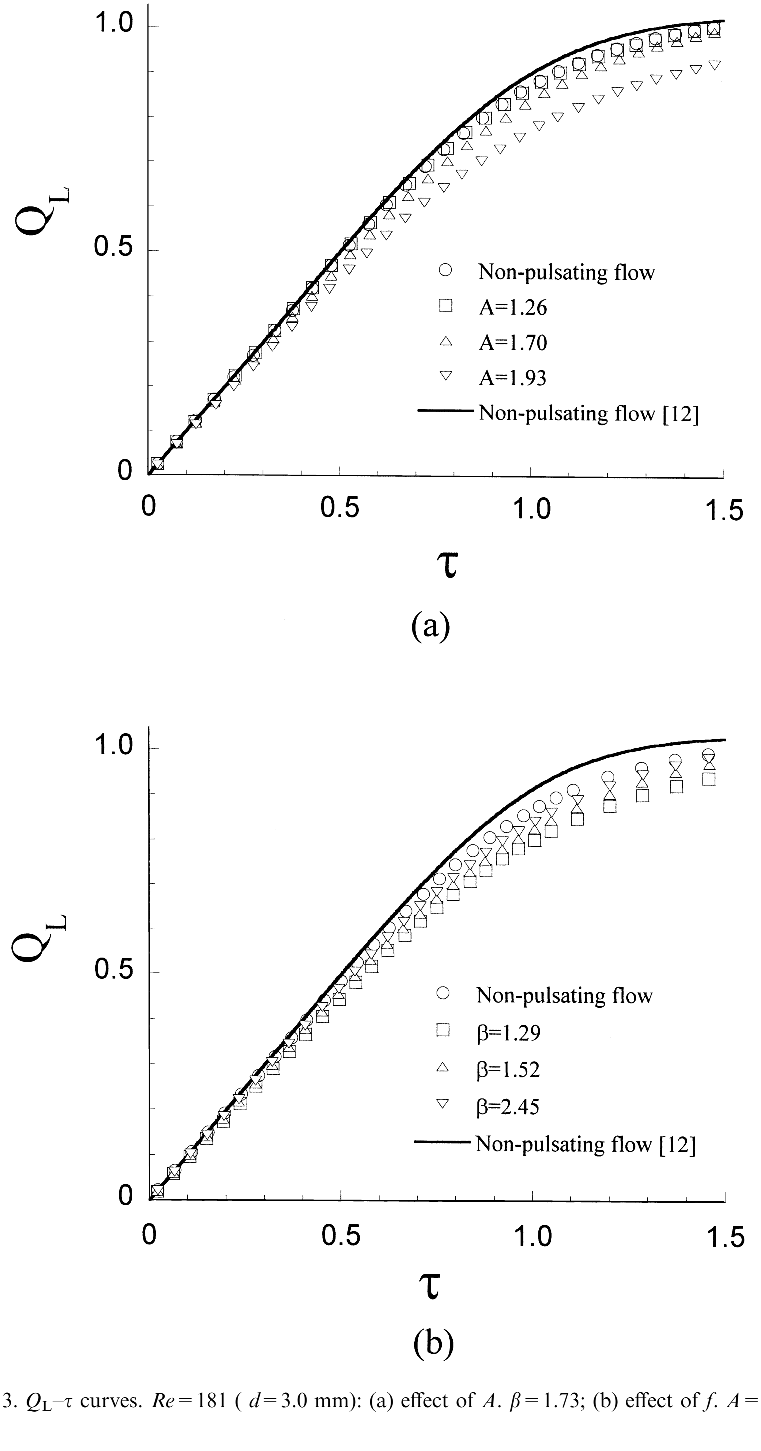


Fig. 3. Q_L–τ curves. $Re = 181$ ($d = 3.0$ mm): (a) effect of A. $\beta = 1.73$; (b) effect of f. $A = 1.8$.

tions of the modified dispersion–concentric model [12] are consistent with the present experimental data for non-pulsating flows.

When there is no reverse flow, in the range of Reynolds number $Re \cong 95$–400 and for the sphere diameter $d = 6.35$ mm, the cool-down characteristics are found to be little affected by changes in β and A. The numerical calculations of Sozen and Vafai [4], in which the experimental correlation based on a non-pulsating flow was incorporated, also showed a similar trend for low frequencies, i.e., $\beta = 0.26$–0.52 ($f \cong 0.05$–0.2 Hz), when no reverse flow was present.

Effects of larger amplitudes A of pulsation, especially when $A > 1.0$ so that a backward flow is induced in the incoming stream, are exhibited in Fig. 2(b). At large values of A, temporal oscillations of θ are evident, in particular in the early stages and in the upstream regions of the porous medium. The amplitude of these temperature oscillations tends to weaken at downstream locations. The overall time-averaged θ–τ curve tends to be less steep as A increases, producing smaller temperature gradients. This is explained by noting that, for the time duration over which a backward flow prevails, the hot air in the downstream positions moves upstream. This, in turn, reduces the difference in temperatures between the spheres and the fluid flow, which lowers the heat transport from the hot porous medium to the cold throughflow. In summary, the global averaged heat transfer rate is decreased by the pulsation when the amplitude A is large ($A > 1.0$) enough to cause a backward flow.

It is of interest to gauge the accumulated heat Q_L transferred from the initial storage in the porous medium to the throughflow from the start-up until time t [13]:

$$Q_L = \frac{\rho_f c_f U_0 \int_0^t \{T(X = 1.0, t) - T_0\}\, dt}{(1 - \varepsilon)\rho_s c_s L(T_{in} - T_0)} \tag{3}$$

where the numerator of (3) indicates the heat obtained by air leaving the porous medium and the denominator shows the initial heat storage in the porous medium. The decrease in heat transport, as the amplitude A becomes large ($A > 1.0$), is also apparent in the Q_L-curves in Fig. 3(a). The effect of β is discernible in Fig. 3(b). In the present experimental setup to generate pulsation, when A and d are fixed, the backward-moving distance of the fluid becomes shorter as β increases, since the time duration of backward flow becomes shorter. Consequently, the temperature alterations in the porous medium are less affected by the backflow as β becomes large. As exhibited in Fig. 3(b), the overall effects of pulsation, as compared with the case of a non-pulsating flow, are decreased as β becomes large.

4. Conclusions

The temperature histories inside the porous medium indicate that the global heat transfer characteristics are little changed by the introduction of flow pulsation if the amplitude A is small so that no reverse flows are induced. For the parameter ranges of present concern, the existing heat transport correlations based on the conventional non-pulsating flows can produce adequate approximations.

When the pulsation amplitude is large so that reverse flows are present, the global heat transfer between the porous medium and the throughflow becomes smaller than for the case of the corresponding non-pulsating flow. For a given amplitude, the heat transport rate from the porous material decreases as the pulsation frequency decreases.

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