FLUID FLOW AND HEAT TRANSFER IN A REGENERATIVE HEAT EXCHANGER MATRIX: ANALYSIS OF THE OPTIMUM FLOW FREQUENCY

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Abstract. *The present work explores the fundamental aspects of the fluid flow and heat transfer between a working fluid and the matrix of a regenerative heat exchanger. Regenerators are a time dependent class of thermal energy storage systems that facilitates the transfer of heat from a hot fluid to a cold fluid through temporary retention of thermal energy in high thermal capacity matrices. They are an attractive technology and constitute the crucial part of many thermal systems that operate with regenerative cycles, such as the Stirling and the magnetic refrigeration cycles. A simplified model was put forward to investigate the behavior of the similarity parameters that describe the heat transfer between a solid cylinder and a periodic cross-flow. An analitycal and a finite volume-based procedures were used to investigate the behavior of the temperature field as well as the transient thermal energy profiles. The flow frequency that yields the optimum heat exchange was determined for each corresponding convective heat transfer coefficient, using a numerical correlation written in terms of the dimensionless Biot and Fourier numbers. The analysis is complemented with a numerical assessment of the complete problem using a commercial CFD package.*

Keywords: *regenerative heat exchanger, numerical analysis, optimization.*

1. INTRODUCTION

Regenerators are a time dependent class of thermal energy storage systems that facilitates the transfer of heat from a hot fluid to a cold fluid through temporary retention of thermal energy in high thermal capacity matrices. They are an attractive technology due to their simple construction, low cost, flexibility, self-cleaning operation and the wide range of temperature over which they can be used (Roy and Das, 2001). Besides, they constitute the crucial part of many thermal systems that operate with regenerative cycles, such as the Stirling, pulse tube and the magnetic refrigeration cycles. The energy storage matrix, usually a solid or a porous medium, is heated by the hot fluid that flows through it during the first half of the cycle and subsequently, during the second half of the cycle, it is cooled by the cold fluid flowing now in the opposite direction. Geometry, thermophysical properties of the matrix and the working fluid as well as the cycle characteristic time scale determine the performance of these devices. Periodic changes in the temperature of the hot and cold fluids prevent these heat exchangers from operating under true stationary conditions. Nevertheless, after a sufficient number of cycles, a fully developed regime can be identified in which the solid and fluid temperatures vary periodically with time (Patankar, 1979).

As far as the mathematical modeling of regenerators is concerned, several papers published in the past decades have focused on the heat transfer characteristics using a First-Law-based approach, and have proposed analytical solutions for the fluid and matrix temperature profiles, thermal efficiency, Nusselt number and number of transfer units, NTU, for various geometric configurations (Kardas, 1966; Chase et al., 1969; Saastamoinen, 1999; Roy and Das, 2001; Klein and Eigenberger, 2001; Chen et al., 2007; Monte and Rosa, 2008).

Studies devoted to the numerical modeling and simulation of regenerators (Muralidhar, 1998; Muralidhar and Suzuki, 2001; Zhang et al., 2002; Zhu and Matsubara, 2004; Heidrich et al., 2005; Gao et al., 2006; Nellis and Klein, 2006) have improved the modeling of the fluid flow and heat transfer phenomena, and contributed to the prediction of optimum operational conditions for a variety of applications. In this context, Muralidhar (1998) presented a modeling approach which consisted of evaluating the fluid flow and coupled heat transfer (solid and fluid) in an elementary cell of the regenerator matrix. The two major advantages of this fundamental approach are: (i) the possibility to prescribe in detail the local velocity field and, at the same time, (ii) use no interface approximation for the calculation of the heat transfer rate between the fluid and solid media.

As an attempt to develop further the approach of Muralidhar (1998), the present work investigates the influence of heat transfer similarity parameters, namely the Biot and Fourier numbers, on the performance of a thermal regenerator using a first-principles approach. The regenerator is considered as a bank of cylinders (e.g., a wire mesh) and it is represented by its basic element, a solid cylinder of radius R around which a fluid with a time-dependent (periodic) temperature distribution flows in a reciprocating fashion, as illustrated in Fig. 1. In the first part of the cycle, i.e., half a period, heat is transferred from the hot fluid coming from the left to the cylinder and, in the second part, heat is transferred from the cylinder to the cold fluid coming from the right. In this problem, the aim is to optimize the parameters so as to obtain, in view of the desired applications, the maximum heat transfer between the fluid and the solid matrix. Therefore, it is necessary to know the behavior of the transient temperature distribution and the correspondent thermal energy that is absorbed and released by the cylinder.

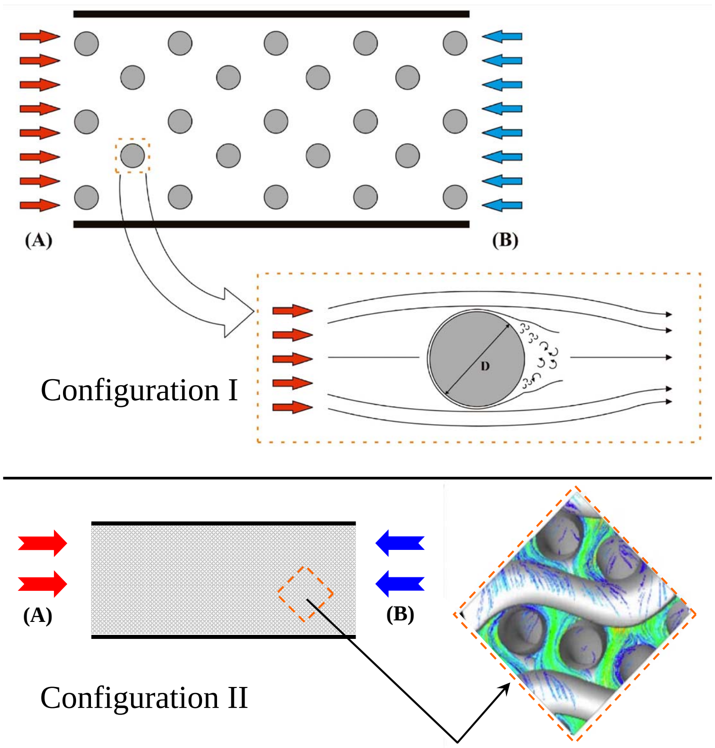


Figure 1. Regenerator configurations: I (a bank of cylinders) e II (a porous medium). In detail: (A) hot fluid stream that heats the matrix during the first half of the cycle and (B) cold fluid stream that is heated by the matrix during the second half of the cycle. Source: Configuration II is adapted from Harvey (2003).

2. MATHEMATICAL FORMULATION

The flow is assumed to be time-dependent, laminar, incompressible, with constant physical properties and two-dimensional. Unsteady heat conduction occurs in the cylinder, which is considered homogeneous, isotropic and with constant physical properties. Heat generation and viscous dissipation effects are neglected. At the inlet boundaries, velocities and temperatures are prescribed in such a way that the former exhibits a sinusoidal time dependency (with amplitudes $+U$ from left to right and $-U$ from right to left) and the latter has a square-wave (with extremes T_H and T_C) behavior. Here, T_H is the hot heat exchanger temperature and T_C is the cold heat exchanger temperature.

The governing partial differential equations for the fluid dynamics and the heat transfer in the fluid and in the solid are given by,

$$\nabla \cdot \mathbf{V} = 0 \tag{1}$$

$$\rho_f \left(\frac{\partial \mathbf{V}}{\partial t} + \mathbf{V} \cdot \nabla \mathbf{V} \right) = -\nabla p + \mu \left(\nabla^2 \mathbf{V} \right) \tag{2}$$

$$\rho_f c_p \left(\frac{\partial T_f}{\partial t} + \mathbf{V} \cdot \nabla T_f \right) = \nabla \cdot \left(k_f \nabla T_f \right) \tag{3}$$

$$\left(\rho_s c \right) \frac{\partial T_s}{\partial t} = \nabla \cdot \left(k_s \nabla T_s \right) \tag{4}$$

where t, $\mathbf{V}$, T, p, ρ, μ, k, are time, flow velocity vector, temperature, pressure, density, dynamic viscosity and thermal conductivity, respectively. The variables c_p and c are the fluid and solid specific heat capacities. The subscripts f and s stand for fluid and solid.

In great part of the present study, the convective heat transfer coefficient between the fluid and the solid, h, is assumed constant. In this case, the partial differential equations for the fluid – Eqs. (1) to (3) – do not need to be solved and the attention is turned on to the thermal problem in the solid. Evolutions of the cylinder temperature $T_s(r,t)$ in space and time are obtained from the solution of the unsteady heat conduction equation, Eq. (4). In conservative form, this equation is given by,

$$\frac{\partial}{\partial t}\left(\rho_s c T_s\right) = \frac{1}{r}\frac{\partial}{\partial r}\left(k_s r \frac{\partial T_s}{\partial r}\right) \tag{5}$$

Equation (5) is subjected to the following boundary and initial conditions,

$$\left.\frac{\partial T_s}{\partial r}\right|_{r=0} = 0 \tag{6}$$

$$-k_s \left.\frac{\partial T_s}{\partial r}\right|_{r=R} = h\left[T_s(R,t) - T_\infty(t)\right] \tag{7}$$

$$T_s(r,0) = T_i \tag{8}$$

In Eq. (7), $T_\infty(t)$ can assume the values of T_H and T_C, depending if the cylinder is in contact with the hot or cold fluid, respectively. For the cylinder heat transfer problem, a set of dimensionless parameters are conveniently defined as follows,

$$\theta(\xi,\tau) = \frac{T_s[r(\xi), t(\tau)] - T_C}{T_H - T_C} \tag{9}$$

$$\xi = \frac{r}{R} \tag{10}$$

$$\tau = \frac{t}{P} \tag{11}$$

$$Fo = \left(\frac{k_s}{\rho_s c}\right)\frac{P}{R^2} = \frac{\alpha_s P}{R^2} \tag{12}$$

$$Bi = \frac{hR}{k_s} \tag{13}$$

where r is the radial coordinate, P is the cycle period, α_s is the solid thermal diffusivity and Fo and Bi are the radius-based Fourier and Biot numbers, respectively.

In dimensionless form, the governing equation, boundary and initial conditions are given by,

$$\frac{\partial \theta}{\partial \tau} = \frac{Fo}{\xi}\frac{\partial}{\partial \xi}\left(\xi \frac{\partial \theta}{\partial \xi}\right) \tag{14}$$

$$\left.\frac{\partial \theta}{\partial \xi}\right|_{\xi=0} = 0 \tag{15}$$

$$\left.\frac{\partial \theta}{\partial \xi}\right|_{\xi=1} = -Bi\left[\theta(1,\tau)\right] \Rightarrow T_\infty(t) = T_C \tag{16}$$

$$\left.\frac{\partial \theta}{\partial \xi}\right|_{\xi=1} = -Bi\left[\theta(1,\tau) - 1\right] \Rightarrow T_\infty(t) = T_H \tag{17}$$

$$\theta_i = \theta(\xi,0) = \frac{T_s(r,0) - T_C}{T_H - T_C} = \frac{T_i - T_C}{T_H - T_C} \tag{18}$$

It is also necessary to determine the amount of thermal energy exchanged between the cylinder and the working fluid. Considering that at $t = 0$ initially the solid is at a temperature $T_i \in [T_C, T_H]$, the thermal energy per unit length Q stored in it and released to the fluid at any instant t is given by,

$$Q(t) = \pi R^2 \rho_s c T_i - \left[\int_0^R 2\pi \rho_s c T_s(r,t) r dr\right]_t \tag{19}$$

Equation (19) provides the temporal evolution of the thermal energy exchanged between the working fluid and the solid during the successive heating and cooling processes. It represents the change in the solid total internal energy with respect to the energy stored at the initial instant (i.e., the first term on the right hand side). This variation is negative when the cylinder is at higher temperatures than its initial temperature T_i. On the other hand, it is positive when the solid temperature T_s is lower than T_i. Equation (19) was obtained from a volumetric integration of Eq. (5). It should also be mentioned that the maximum thermal energy per unit length that can be exchanged between the fluid and the solid, per cycle, is given by

$$Q_{max} = \pi R^2 \rho_s c (T_H - T_C). \tag{20}$$

3. SOLUTION PROCEDURE

Equations (14) to (18) can be solved analytically via the separation of variables method (Arpaci, 1966; Poulikakos, 1994). Assuming that the solid is at an initial temperature T_i and the operation starts with the heating of the storage matrix, the cylinder temperature fields during the heating and cooling intervals, θ_H and θ_C, are as follows,

Heating intervals:

$$\theta_H(\xi,\tau) = 1 + \sum_{n=1}^{\infty} \frac{2\big(\theta_m\big|_{initial} - 1\big) Bi J_0(\lambda_n \xi) \exp[(-\lambda_n^2 Fo)\tau]}{J_0(\lambda_n)(Bi^2 + \lambda_n^2)} \tag{21}$$

Cooling intervals:

$$\theta_C(\xi,\tau) = \sum_{n=1}^{\infty} \frac{2\big(\theta_m\big|_{initial}\big) Bi J_0(\lambda_n \xi) \exp[(-\lambda_n^2 Fo)\tau]}{J_0(\lambda_n)(Bi^2 + \lambda_n^2)} \tag{22}$$

The following auxiliary equation is used to compute the eigenvalues λ_n of the Bessel function J_0,

$$\lambda_n = Bi \frac{J_0(\lambda_n)}{J_1(\lambda_n)} \Rightarrow n = 1, 2, 3, ... \tag{23}$$

The previous equations are valid in the spatial domain $0 \leq \xi \leq 1$. With respect to the time domain, Eq. (21) is valid for $m \leq \tau \leq m + \frac{1}{2}$ and Eq. (22) is valid for $m + \frac{1}{2} \leq \tau \leq m + 1$, where $m = 0, 1, 2, 3, ...$ is an index to identify successive cycles. It is important to clarify that $\theta_m|_{initial}$ refers to the radial temperature profile existing in the solid at the start of the heating and cooling intervals. Thus, the initial condition to the cooling interval is the solid temperature profile at the last instant of the heating interval and vice-versa. Then, $\theta_m\big|_{initial} = \theta_m\big|_{initial}(\xi)$, and it should be noted that when $m = 0$ one has $\theta_m\big|_{initial} = \theta_i$.

Despite the availability of an exact solution, a numerical procedure was adopted due to its flexibility with respect to the introduction of non-homogeneities, which will be the objective of future investigations. The equations for the thermal problem are solved using a totally implicit finite volume method (Maliska, 2004; Versteeg and Malalasekera, 1995). A computational code was produced and grid sensitivity tests were carried out in order to obtain a mesh-independent solution. The mesh uses evenly spaced elements in the radial and temporal domains. It is also important to mention that time step Δt is defined here as the ratio between the cycle time P and the desired number of time steps. The exact solution was used to validate the numerical model, verifying its physical consistence, accuracy and serving as a decision tool in the performed grid refinement tests.

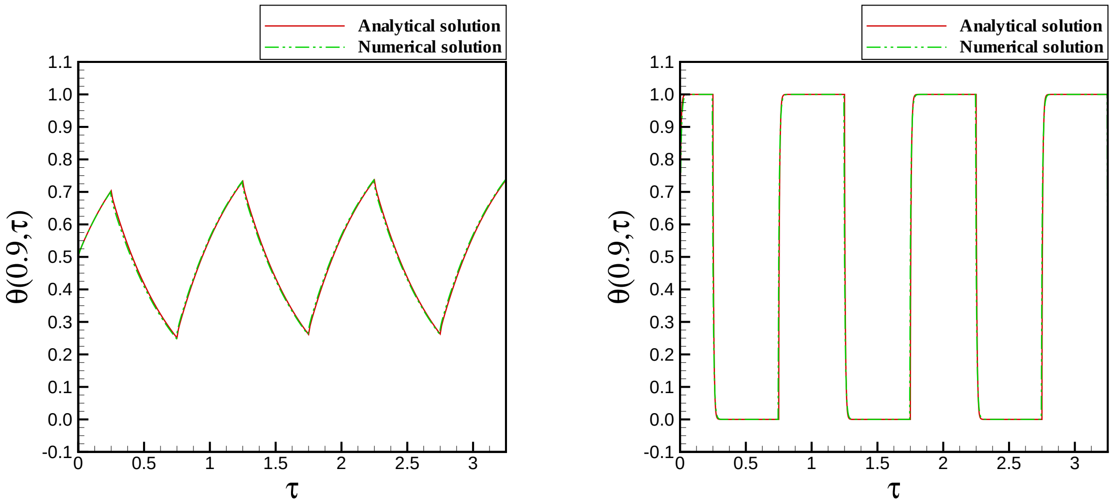
4. RESULTS AND DISCUSSIONS

Table 1 shows the matrix (made of *phosphor-bronze*) thermophysical properties as well as the geometric parameters used in the simulations performed. T_i was taken as the mean value between the extreme temperatures T_H and T_C.

Table 1. Parameters used in the simulations.

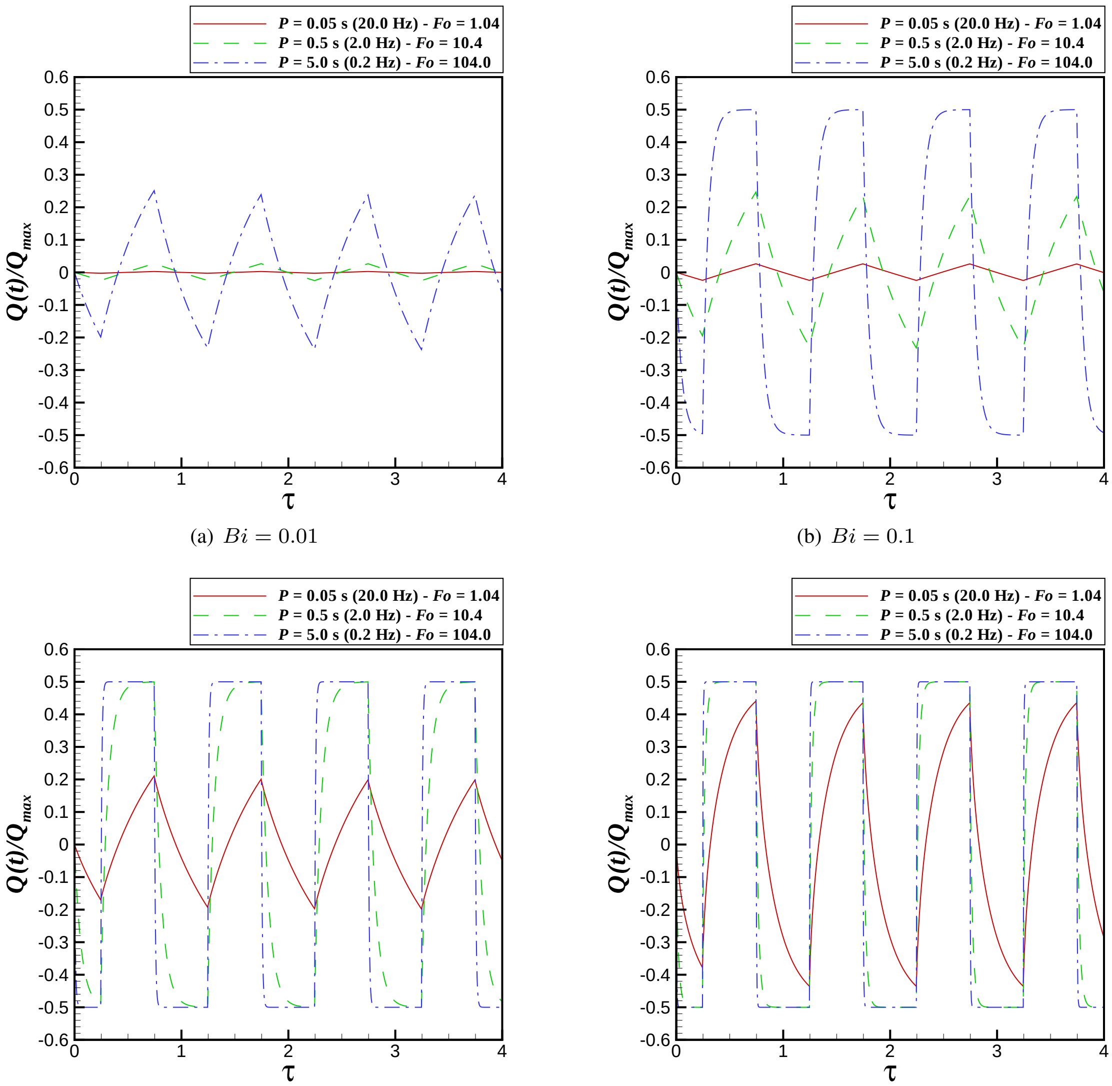
Parameters	R [mm]	T_C [°C]	T_H [°C]	k_s [W/(mK)]	c [J/(kgK)]	ρ_s [kg/m^3]
Values	1	5	60	70	378	8900

The results are evaluated for different values of Bi (0.01, 0.1, 1, 10 and 100) and Fo (1.04, 10.4 and 104), the latter corresponding to the values of the period P of 0.05 s, 0.5 s and 5 s, respectively. Figure 2 confirms the existence of an excellent agreement, in terms of physical behavior and numerical accuracy, between the analytical and numerical solutions using a 100 X 100 mesh for specific conditions of Bi and Fo.



(a) $Bi = 0.1$ e $Fo = 10.4$ (b) $Bi = 1$ e $Fo = 104$

Figure 2. Comparison between the analytical and numerical solutions of the transient temperature, evaluated at $\xi = 0.9$, for distinct combinations of Bi and Fo: (a) $Bi = 0.1$ and $Fo = 10.4$, and (b) $Bi = 1$ and $Fo = 104$.



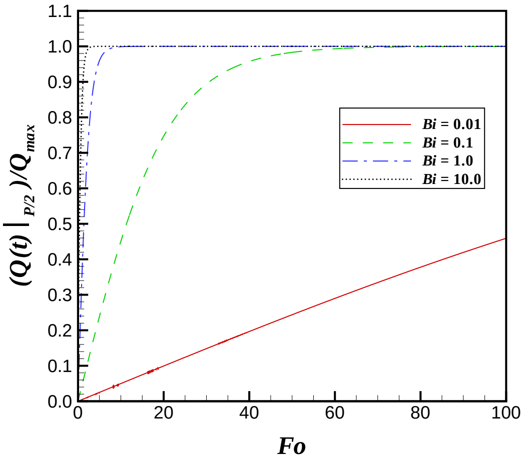
(a) $Bi = 0.01$ (b) $Bi = 0.1$

(c) $Bi = 1$ (d) $Bi = 10$

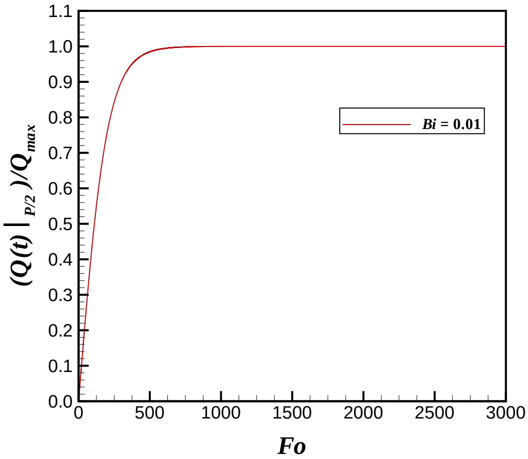
Figure 3. Thermal energy transients for different values of Bi: (a) $Bi = 0.01$, (b) $Bi = 0.1$, (c) $Bi = 1$ and (d) $Bi = 10$.

Figure 3 portrays the dimensionless heat exchange between the fluid and the solid. As can be seen, for $Bi < 1$ the highest Fo curve presents the largest energy exchanged as the heating and cooling intervals are larger. For $Bi > 1$, the existence of plateaus can be perceived in the curves with large Fo, indicating that thermal equilibrium was reached. In all Bi range reported in Fig. 3, the lowest Fourier numbers $(Fo = 1.04)$ are not sufficiently large to enable the attainment of thermal equilibrium between the solid and the fluid.

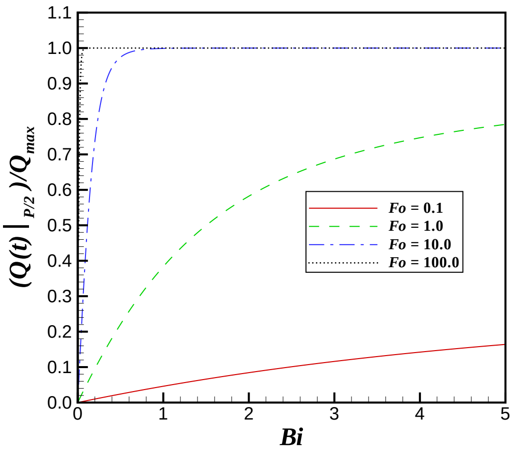
In the following, Fig. 4 illustrates the behavior of the heat transferred in half a cycle, evaluated in the cyclic fully developed regime, for different values of Bi and Fo. For the curves with constant Bi, Figs. 4 (a) and 4 (b), the increase of Fo yields to the maximum thermal energy that can be transferred between the fluid and the solid matrix, even when Bi assume very low values $(Bi = 0.01)$. Nevertheless, observing the curves with constant Fo, Figs. 4 (c) and 4 (d), it can be seen that the increase in Bi yields to the maximum dimensionless thermal energy only in the curves correspondent to high Fo values. The curves with low Fo values (0.1 and 1.0) never reach thermal equilibrium with the hot and cold fluid streams in spite of a drastic increase in the Bi number.



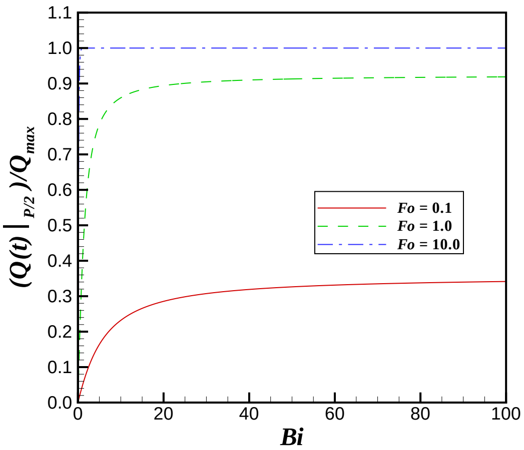
(a) Heat exchanged *versus* Fo



(b) Heat exchanged *versus* Fo (emphasis on $Bi = 0.01$)



(c) Heat exchanged *versus* Bi



(d) Heat exchanged *versus* Bi (emphasis on $Fo = 0.1$ and 1)

Figure 4. Heat transferred in half a cycle in the cyclic fully developed regime as a function of Fo – (a) and (b) – and Bi – (c) and (d).

The behavior illustrated in Figs. 4 (c) and 4 (d) is due to the fact that the radial temperature profile is not uniform because the heating and cooling times are quite short. It can be noticed that for high frequencies (low Fo), even a significant increase in the value of Bi (say, from 100 to 10000) does not affect the temperature distribution in the solid, as shown in Fig. 5. Figure 6 depicts the radial temperature profiles as a function of time during heating, (a) and (c), and cooling intervals, (b) and (d), in the cyclic steady-state regime. They exhibit a sequence of time instants for $Fo = 0.1$, (a) and (b), as well as $Fo = 1$, (c) and (d), for a fixed value of the Biot number $(Bi = 100)$. For small Fo, the temperature distributions go through minima and maxima at some point within the solid domain during the heating and cooling intervals, respectively. Conversely, due to an increase of the period, this trend is barely inexistent for $Fo = 1$. By observing Figs. 6 (a) and 6 (b), it is interesting to notice that at the positions near to the center of the cylinder the temperature is almost constant and insensitive to the heat transfer through its surface. In addition to this, it is worth noting that due to the high value of Biot number, the temperature at the cylinder surface is always very close to the fluid temperature θ_H and θ_C.

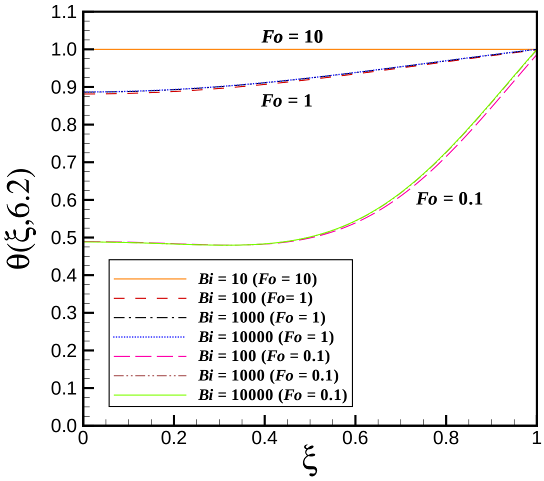
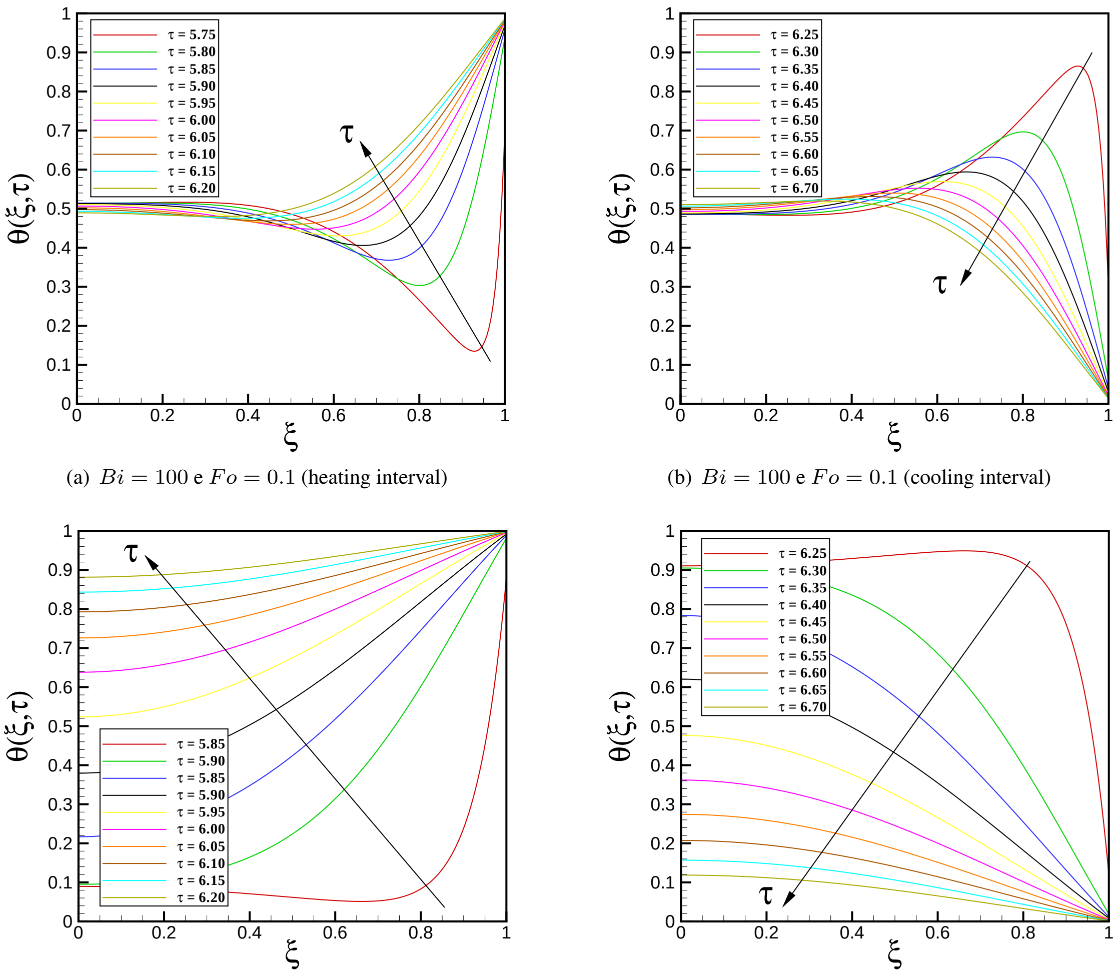


Figure 5. Radial temperature profiles presented for distinct Bi and Fo values, considering a particular instant at the beginning of the heating interval ($\tau = 6.2$).



(a) $Bi = 100$ e $Fo = 0.1$ (heating interval)

(b) $Bi = 100$ e $Fo = 0.1$ (cooling interval)

(c) $Bi = 100$ e $Fo = 1$ (heating interval)

(d) $Bi = 100$ e $Fo = 1$ (cooling interval)

Figure 6. Time evolution of the radial temperature for distinct combinations of Bi and Fo during heating, (a) and (c), and cooling, (b) and (d), intervals.

Based on the information provided in Fig. 4, it was possible to determine the curve for $Fo_{optimum}$ as a function of Bi, where $Fo_{optimum}$ corresponds to the Fourier number at which the heat transferred reaches its maximum value at the end of the cycle. For Fourier numbers larger than $Fo_{optimum}$, part of the cycle will operate with the cylinder at the temperature of the fluid and there will be no heat transfer. Since for a fixed Biot number Fo tends asymptotically to the maximum heat exchange, $Fo_{optimum}$ was computed when the thermal energy exchanged between the fluid and the solid reaches 90% of Q_{max}. The choice for this value was motivated by practical engineering purposes.

In the following, Fig. 7 shows the inversely proportional relation between $Fo_{optimum}$ and Bi. This curve indicates that for each velocity there is an optimum time period for the solid to be exposed to the flow in order to maximize the heat transfer. From a first-law perspective, the optimum heat transfer is such that the cylinder is in contact long enough with the working fluid so the temperature of the former becomes equal to that of the latter.

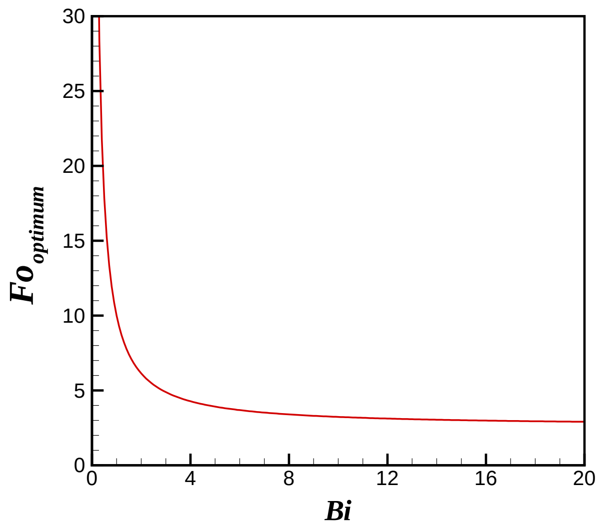


Figure 7. Correlation of the optimum Fo as a function of Bi.

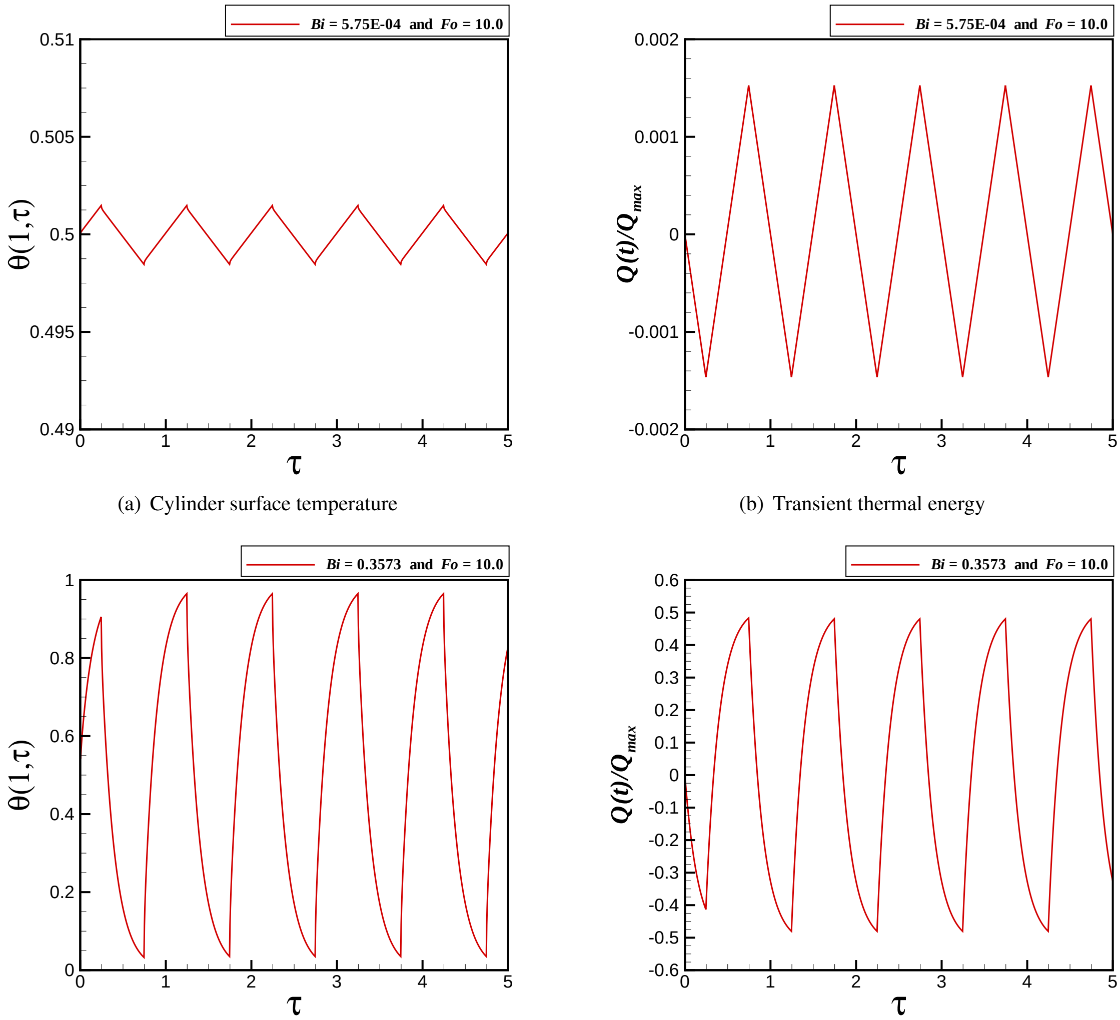
The following equation (Eq. 24) provides a best-fit of the curve shown in Fig. 7 over the interval $0.01 \leq Bi \leq 100$. Thus,

$$Fo_{optimum} = 0.864 + \frac{3.33}{Bi} - \frac{0.0234}{Bi^2} \Rightarrow \delta^2 = 0.9926 \tag{24}$$

where δ^2 is the curve fit coefficient of determination.

The present analysis is complemented with a numerical assessment of the fundamental problem using a commercial CFD package. In order to establish a comparison between the simplified model and the complete one, Eqs. (1) to (4) were solved using FLUENT 6.3 for the same set of dimensionless parameters utilized by Muralidhar (1998), which are $Re_D = 40$, $Pr = 0.7$, $Pe_D = 28$ e $Fo = 10$ where Re_D is the cylinder diameter-based Reynolds number and Pr and Pe are the Prandtl and Peclet numbers, respectively. It is important to mention that one is dealing with a two-dimensional problem limited by the following boundary conditions: (1) a sinusoidal transient velocity profile at the inlet, (2) zero pressure difference at the outlet boundary, (3) periodic conditions at the bottom and the upper boundaries, and (4) thermal interfacial coupling (i.e., continuity of heat fluxes and temperature) at the cylinder surface. The mesh used in the simulations is composed by hexahedral volumes for the fluid domain and tetrahedral volumes in the solid, with a grid refinement in the region adjacent to the cylinder in order to capture boundary layer effects. The fluid is air and the cylinder radius as well as the solid material are the same as the ones used in the one-dimensional problem in order to satisfy the above mentioned dimensionless parameters.

The simulation resulted in a cycle average heat transfer coefficient, $\overline{h}_{cycle}$, of 40.23 W/(m^{2}K) for the cyclic fully developed regime, which provides $Bi = 5.75 \times 10^{-4}$. With this value of Bi and for $Fo = 10$ ($P = 0.5$ s) the solution of the simplified model for the solid resulted in the curves of Figs. 8 (a) and 8 (b), which show a small fluctuation amplitude for the surface temperature $\theta(1, \tau)$, and low values of $Q(t)/Q_{max}$, respectively. This is in line with the results presented by Muralidhar (1998). However, by using Bi corresponding to $Fo_{optimum} = 10$, i.e. $Bi = 0.3573$, one observes a significant increase in the fluctuation amplitude of $\theta(1, \tau)$ as well as in $Q(t)/Q_{max}$ values, as exhibited in Figs. 8 (c) and 8 (d). This result illustrates the relevance of operating in conditions close to the Fourier optimum values for a given Biot value.



(a) Cylinder surface temperature (b) Transient thermal energy

(c) Cylinder surface temperature (d) Transient thermal energy

Figure 8. Surface temperature and transient thermal energy to two situations characterized by the following parameters: (a) - (b) $Bi = 5.75\times10^{-4}$ and $Fo = 10$, and (c) - (d) $Bi = 0.3573$ and $Fo = 10$.

5. CONCLUSIONS

A detailed study of the heat transfer between a solid cylinder and a transversal periodic flow was presented using a simplified model to investigate the behavior of temperature and thermal energy profiles. Despite its simplicity, the proposed model integrates all the phenomenological aspects that characterize the thermal problem. This analysis supports the understanding of aspects that are of concern in the optimization of thermal regenerators. The flow frequency that yields to optimum heat exchange was determined for each corresponding convective heat transfer coefficient, using a numerical correlation written in terms of the dimensionless Biot and Fourier numbers.

6. ACKNOWLEDGEMENTS

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