# Numerical Simulation of Fluid Flow and Heat transfer in a Rotary Regenerator

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## ABSTRACT

In this paper, a numerical simulation of fluid flow and heat transfer in a rotary regenerator is presented. The numerical method is based on the simultaneous solution of the transient momentum and energy equations using a finite volume scheme. To validate the model, the numerical results are compared with those of the analytical results for a simple geometry. The effects of important processing parameters on the rotary regenerator effectiveness were also investigated. The parameters considered were the rotational speed, and the rotor length and diameter. In the design of a regenerator, the mass flow rate of both hot and cold streams and the resulting heat effectiveness are main parameters. The developed model, therefore, is suitable for the design and optimization of rotary regenerators for industrial applications.

## 1. INTRODUCTION

Heat recovery applications are becoming increasingly more attractive as energy prices are raising continuously. Heat exchangers are one of the important equipment that can achieve this purpose. There are many different types of heat exchangers one of which is gas-to-gas used in some applications such as furnaces, air conditioning systems, and power plants. The main issue regarding the design of these types of heat exchangers is the much lower heat transfer coefficients of gases compared to those of liquids. As a result, the required surface area of heat transfers, and in turn, the sizes of these heat exchangers will be much bigger compared to those of gas-to-liquid or liquid-to-liquid heat exchangers. Rotary regenerator is a solution for this problem. It consists of a rotating matrix through which the hot and cold streams flow periodically and alternatively (Fig. 1). First, the hot fluid gives up its heat to the regenerator; then, the cold fluid flows through the same passage picking up the stored heat. The overall efficiency of sensible heat transfer for this kind of regenerator can be as high as 85 percent, while efficiency of plate-type heat exchangers is between 45 and 65 percent [1].

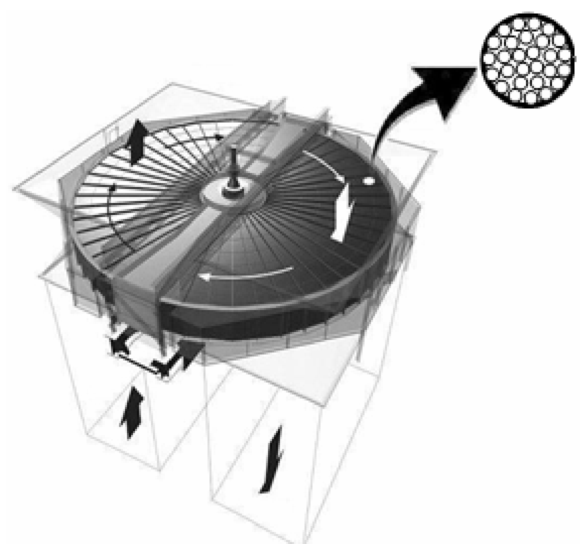

**Figure 1.** A typical rotary regenerator and its extended surface with a large number of channels.

A rotary regenerator has a self cleaning property that removes any fouling problem. This is caused by the periodic reversal flows in the channels. This type of heat exchanger is used widely because of its compactness and high effectiveness.

Studying on the rotary regenerators has a long history. Sheiman and Reznikova [2] introduced a model to calculate the involved heat transfer. They used integral Laplace transformation to solve differential equations. Nahavandi and Weinstin [3] used an elaborate mathematical model, which they called Close method, to solve the differential equations. There have been many attempts to obtain an empirical effectiveness correlation [4-6]. The ε-$NTU_0$ method was used for the rotary regenerator design and analysis. Organ [7] explained a procedure for designing a regenerator; his formulation is the basis for a regenerator operating at an optimum balance between pumping loss and penalty of incomplete temperature recovery. Many numerical solutions are also available in the literature. Kovalevskii [8] solved the conjugate problem on unsteady heat exchange between one-dimensional flows and a two-dimensional matrix wall. He determined the optimum design parameters of such a regenerator and the rotational speed of its rotor for maximum heat efficiency. Finite difference scheme has been used by many researchers [9,10] to solve the involved differential equation. Shah and Skiepko [11] and Drobnic et al. [12] considered the influence of leakage on the thermal performance. The effect of rotational speed on the effectiveness of rotary heat exchanger was studied by Buyukalaca and Yılmaz [13]. They neglected the influence of axial heat conduction. Bahnke and Howard [14] and Romie [15] obtained relationships for estimating the influence of axial heat conduction in the wall. Porowski and Szczechowiak [16] investigated the effect of axial conduction in the matrix on the effectiveness of the rotary regenerator. They showed that a large axial temperature gradient in the wall reduces the regenerator effectiveness and the overall heat-transfer rate. Sphaier and Worek [17] presented a 1D and 2D simulations of a rotary regenerator where they concluded that a 2D formulation is needed for certain design and operating parameters. In this paper, a 2D/axisymmetric model is presented for the simulation of a rotary regenerator and the effects of important processing parameters are investigated.

## 2. NUMERICAL METHOD

The matrix of a rotary regenerator contains a large number of channels as shown schematically in the inset in Fig. 1. Flow velocity, temperature and other variables are nearly the same for all channels. Therefore, to obtain a numerical solution for this problem, we consider a single channel of the matrix where hot and cold streams flow periodically and in the opposite direction of each other. The model is based on the following assumptions:

- Fluid and solid-wall thermophysical properties may vary with temperature.
- The change of the fluid angular momentum as it enters the rotating matrix is negligible.
- The inlet temperature and velocity of hot and cold streams remain unchanged in each period.
- The matrix is divided to two equal sections, so the time duration of hot and cold periods is same.

The fluid flow and heat transfer through the channel is axisymmetric ($r$ and $x$ coordinates) and the channel radius is much smaller than its length. Therefore, the governing equations are fluid flow equations in the channel and energy equations in both fluid and solid wall. These equations are written as:

$$\frac{1}{r}\frac{\partial}{\partial r}(ru_r)+\frac{\partial u_x}{\partial x}=0 \tag{1}$$

$$\frac{\partial u_x}{\partial t}+u_r\frac{\partial u_x}{\partial r}+u_x\frac{\partial u_x}{\partial x}=-\frac{1}{\rho_f}\frac{\partial P}{\partial x}+\upsilon\{\frac{1}{r}\frac{\partial}{\partial r}(r\frac{\partial u_x}{\partial r})+\frac{\partial^2 u_x}{\partial x^2}\} \tag{2}$$

$$\frac{\partial u_r}{\partial t}+u_r\frac{\partial u_r}{\partial r}+u_x\frac{\partial u_r}{\partial x}=-\frac{1}{\rho_f}\frac{\partial P}{\partial r}+\upsilon\{\frac{\partial}{\partial r}(\frac{1}{r}\frac{\partial}{\partial r}(ru_r))+\frac{\partial^2 u_r}{\partial x^2}\} \tag{3}$$

$$\frac{\partial^2 T_f}{\partial x^2}+\frac{1}{r}\frac{\partial}{\partial r}(r\frac{\partial T_f}{\partial r})=\frac{\rho_f c_{pf}}{k_f}(\frac{\partial T_f}{\partial t}+u_x\frac{\partial T_f}{\partial x}+u_r\frac{\partial T_f}{\partial r}) \tag{4}$$

$$\frac{\partial^2 T_w}{\partial x^2}+\frac{1}{r}\frac{\partial}{\partial r}(r\frac{\partial T_w}{\partial r})=\frac{\rho_w c_w}{k_w}\frac{\partial T_w}{\partial t} \tag{5}$$

where $u$ is the velocity, $T$ temperature, $P$ pressure and $\rho$, $\upsilon$, $k$ and $c$ are density, viscosity, thermal conductivity coefficient and specific heat capacity, respectively. The subscript $f$ refers to the fluid and $w$ to the solid-wall of the channel. In the above equations, $u_r, u_x, p, T_f$ and $T_w$ are unknown parameters. The channels are small in radius, therefore; the fluid flow may be well considered to be laminar.

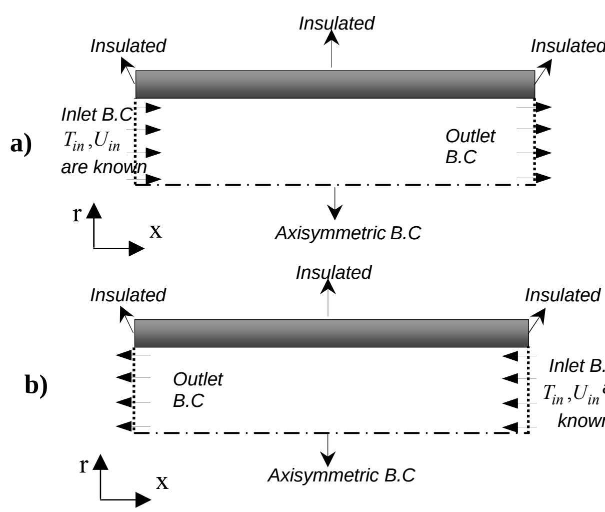


**Figure 2.** A schematic of the computational domain and boundary conditions for **a)** cold period, **b)** hot period.

The equations are discretized and computationally solved using a control volume scheme. The SIMPLE algorithm is employed to solve the pressure-velocity coupling problem. A fully implicit scheme is used to treat the transient calculations, because of its robustness and unconditional stability [18]. The boundary conditions are periodic conditions at the two ends of the channel and adiabatic at the outer surface of the wall (Fig. 2). The input parameters are the entrance velocity and temperature of the two hot and cold streams.

The matrix covers the hot and cold flows equally; therefore, the duration of hot and cold streams is equal to each other (one half of the time duration of one revolution of the matrix). At the beginning of each period, the initial temperature is equal to its quantity at the end time of the previous period. This procedure continues until the temperatures at different points of the channel at each period are very close to the corresponding temperatures at the end of the previous period (convergence criterion for unsteady solution).

## 3. Analytical Method

To validate the numerical model, we consider the fluid flow and heat transfer in a simplified case for which an analytical solution can be obtained. The model results are then compared to those of the analytical solution. Figure 3 displays this simplified scenario.

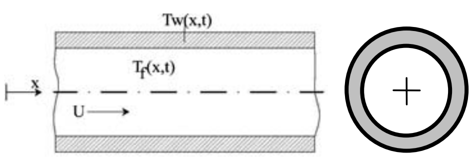


**Figure 3.** A single channel of the matrix of a rotary regenerator (see the inset of Fig. 1).

Since the channel length is much bigger than its radius, the heat transfer problem in the radial direction can be assumed lumped in the analytical problem. A uniform velocity of $U$ is assumed throughout the channel. The conduction heat transfer along the channel in both the fluid and solid-wall is also neglected. The heat transfer equations for the fluid and wall are then simplified as:

$$\frac{\partial T_f}{\partial t} + U_f \frac{\partial T_f}{\partial x} = A(T_w - T_f) \tag{6}$$

$$\frac{\partial T_w}{\partial t} = -B(T_w - T_f) \tag{7}$$

where $A$ and $B$ are defined as:

$$A = \frac{4h}{\rho_f c_f D_h}, \quad B = \frac{h}{\rho_w c_w \delta}$$

where $h$ is the convection heat transfer coefficient, and $\rho, c$ and $\delta$ are density, heat capacity and wall thickness, respectively. To solve the above coupled partial differential equations, two methods are presented; one is based on a simplified case where the initial temperature is assumed constant and the other one is a more elaborate method based on a guess for the initial temperature in the form of power series, first introduced by Nahavandi and Weinstin [3].

### 3.1 Simplified method

To simplify the problem in order to obtain an analytical solution, the initial condition is homogenized by assuming the temperature of the fluid domain to be $T_{hi}$ during the heating period, and $T_{ci}$ during the cooling period. The analytical problem stated above can be solved using the method of Laplace transform. Solving the equations yields:

$$\theta_f(x,t) = \theta_w(x,t) = 0 \qquad t < \frac{x}{U} \tag{8}$$

$$\theta_f(x,t) = \theta_0 e^{\frac{-Ax}{U}} \{e^{-Bt^*} I_0[2(\frac{AB}{U} xt^*)^{1/2}] + B\int_0^{t^*} e^{-B\tau} I_0[2(\frac{AB}{U} x\tau)^{1/2}] d\tau\} \qquad t \geq \frac{x}{U} \tag{9}$$

$$\theta_w(x,t) = B\theta_0 e^{\frac{-Ax}{U}} \int_0^{t^*} e^{-B\tau} I_0[2(\frac{AB}{U} x\tau)^{1/2}] d\tau \qquad t \geq \frac{x}{U} \tag{10}$$

In these equations, $\theta_f, \theta_w$ and $\theta_0$ are defined:

$$\theta_f(x,t) = T_f(x,t) - T' \tag{11}$$

$$\theta_w(x,t) = T_w(x,t) - T' \tag{12}$$

$$\theta_0 = T_{ci} - T_{hi} \tag{13}$$

where $h$ is the convection heat transfer coefficient, and $T'$ is equal to $T_{ci}$ in the cold period and $T_{hi}$ in the hot period.

### 3.2 Close method

Nahavandi and Weinstin [3] presented an analytical solution for the same problem without homogenizing the initial condition. After a rather complex and lengthy mathematics (which they called Close method), they solved the differential equations and provided an estimate for the effectiveness using ε-NTU₀ method. Their result reads:

$$T_f(\xi,\eta) = T_{hi} - e^{-\xi-\eta}\int_0^{\xi}[T_{hi} - f(\varepsilon)]e^{\varepsilon} I_0(2\sqrt{\eta(\xi-\varepsilon)})d\varepsilon \quad (14)$$

$$T_w(\xi,\eta) = T_{hi} - e^{-\eta}[T_{hi} - f(\xi)] + e^{-\xi-\eta}\int_0^{\xi}[T_{hi} - f(\varepsilon)]. \quad e^{\varepsilon}\sqrt{\frac{\eta}{\xi-\varepsilon}} I_1(2\sqrt{\eta(\xi-\varepsilon)})d\varepsilon \quad (15)$$

$$T_f^*(\xi^*,\eta^*) = T_{ci} - e^{-\xi^*-\eta^*}. \int_0^{\xi^*}[T_{ci} - f^*(\varepsilon)]e^{\varepsilon} I_0(2\sqrt{\eta^*(\xi^*-\varepsilon)})d\varepsilon \quad (16)$$

$$T_w^*(\xi,\eta) = T_{ci} - e^{-\eta^*}[T_{ci} - f^*(\xi^*)] + e^{-\xi^*-\eta^*}. \int_0^{\xi^*}[T_{ci} - f^*(\varepsilon)]e^{\varepsilon}\sqrt{\frac{\eta^*}{\xi^*-\varepsilon}} I_1(2\sqrt{\eta^*(\xi^*-\varepsilon)})d\varepsilon \quad (17)$$

$$f(\xi) = \sum_{n=0}^{n=\infty} a_n \xi^n \quad (18)$$

$$f^*(\xi^*) = \sum_{n=0}^{n=\infty} a_n^* \xi^{*n} \quad (19)$$

where parameters with asterisk (*) refer to the hot period and those without it to the cold period. ζ and η are the nondimensionalized length and time defined as:

$$\xi = \frac{x}{\beta} \quad , \quad \eta = \frac{\gamma}{\alpha\beta}\left(\frac{\beta}{\gamma}t - x\right)$$

where $\alpha$, $\beta$ and $\gamma$ are written as:

$$\alpha = \frac{\rho_w c_w \delta}{h} \quad , \quad \beta = \frac{\rho_f c_f U_f D_h}{4h} \quad , \quad \gamma = \frac{\rho_f c_f D_h}{4h}$$

We considered n=3 in the above-mentioned power series; therefore, we needed to solve eight equations to obtain eight unknown parameters of $a_0$, $a_1$, $a_2$, $a_3$, $a_0^*$, $a_1^*$, $a_2^*$, and $a_3^*$. The MATLAB software was used for this purpose.

# 4. Results and Discussion

In this section, first a comparison between the results of the numerical model with those of the analytics for the simple case of a single channel is presented. Having validated the model, then the simulation results for the complete fluid flow and heat transfer for a rotary regenerator will be presented. The effect of important parameters on the effectiveness of a rotary regenerator will also be investigated; they include: rotational speed, fluid inlet velocity, and the rotor length of the regenerator.

## 4.1 Model validation

As mentioned before, the simplified analytical method is presented only for numerical model evaluation. Figure 4 compares the analytical solution with that of the numerical model for this simplified scenario. The time variation of the temperature in the middle of the channel for both solid wall and fluid is shown in the figure during the hot and cold periods. It should be noted that the numerical model provides a temperature profile in each cross section of the channel; therefore, in order to compare the numerical and analytical results, we integrated the temperature profile in *r*-direction to obtain a mean temperature in each cross section.

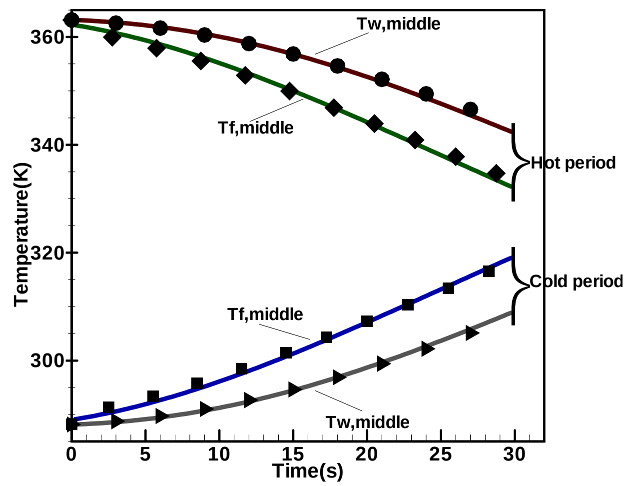


**Figure 4.** Comparison between the results of numerical model (solid lines) and simplified analytical solution (symbols) for a single channel (Fig. 2) during a hot and cold period.

Figure 4 shows a very close agreement between analytical results and simulations for the simplified case of periodic heat transfer in a single channel. Next, we compared the more elaborate analytical results of Close method with that of the numerical model. The result of the analytical solution for one complete period of hot and cold flows is compared with that of the numerical model in Fig. 5. The close comparisons shown in both Fig. 4 and Fig. 5 between the simulations and analytical solutions validate the model and its underlying assumptions.

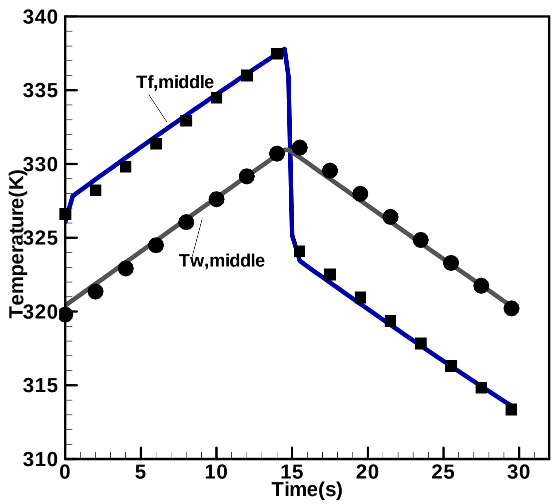


**Figure 5.** Comparison between the results of the numerical model (solid lines) and analytical solution (symbols) of Close method [3] for a single channel (Fig. 2) during a continuous cold and hot period.

## 4.2 Numerical results

A typical rotary regenerator with following geometrical and operational parameters was considered:

- the length 0.45m,
- a single channel diameter 2mm,
- a single channel wall thickness 0.15mm,
- the entrance flow velocity in both hot and cold streams 4m/s (constant during the periods)
- inlet cold air temperature 15$^o$C and inlet hot air temperature 90$^o$C,
- rotational speed 2rpm (i.e., the time duration of each period, hot or cold, is 15s).

Since the hot and cold streams flow periodically and in the opposite direction through the channels, the resulting temperature at any point whether in the fluid or wall will has a periodic shape. The results of simulations for the average temperature of fluid and solid wall at the middle point and at the two ends of the channel (left and right sides) are displayed in Fig. 6. The rotational speed in this case was 2 rpm. As observed, after around 270 s from the start of the flows into the regenerator, we have steady periodic shapes for all temperatures (except for *Tw,middle* which shows an asymptotic behavior). This means that after nearly nine periods (i.e. nine revolutions of the regenerator matrix), the heat transfer arrives at a steady condition.

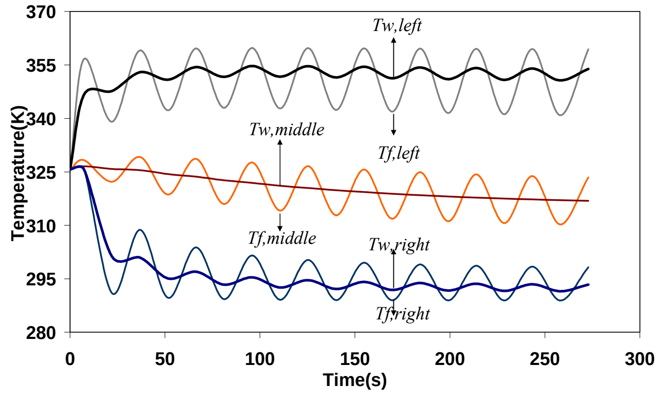


**Figure 6.** The average temperature variation of fluid and solid wall at the middle point and at the two ends of a single channel (left and right sides) of a regenerator matrix. The rotational speed was 2 rpm.

The last revolution (revolution number 9) is the basis of our investigation to obtain velocity, temperature and effectiveness. Velocity vectors and temperature distribution in a channel is shown in Fig. 7 in successive times. The fluid flows are laminar in the channels. During hot and cold periods, the flow has same shape but in the opposite direction. Fluid is entering the channel with a constant velocity and temperature. Due to no-slip condition at the solid wall, the velocity profile gets a parabolic shape in the channel. The fluid flow is fully developed in a distance of about 0.2m from the entrance. It is interesting to note that fluid reaches a steady state condition very rapidly; from $t$=0.25s to $t$=15s the velocity magnitude does not change significantly. Temperature distributions of the fluid are also displayed in Fig. 7 at the bottom of the corresponding velocity profiles. The inlet temperature is assumed constant. At the beginning of a period, the fluid domain has the final temperature obtained from the last period. The hot fluid enters from the left side in the cold period, and the cold fluid enters from the right side in the hot period. After fluid entry, heat exchange occurs between the fluid and solid wall. Hot fluid heats the solid wall during the cold period. Then during the hot period, the heated wall transfers its stored energy to the cold fluid. As seen in Fig. 7, in contrast to the velocity distributions, the temperature values vary in both axial and radial directions. Although the channel radius is too small compared to its length, the temperature variation in radial direction is significant. Therefore, the assumption of constant temperature profile in radial direction in analytical solutions is only an approximation.

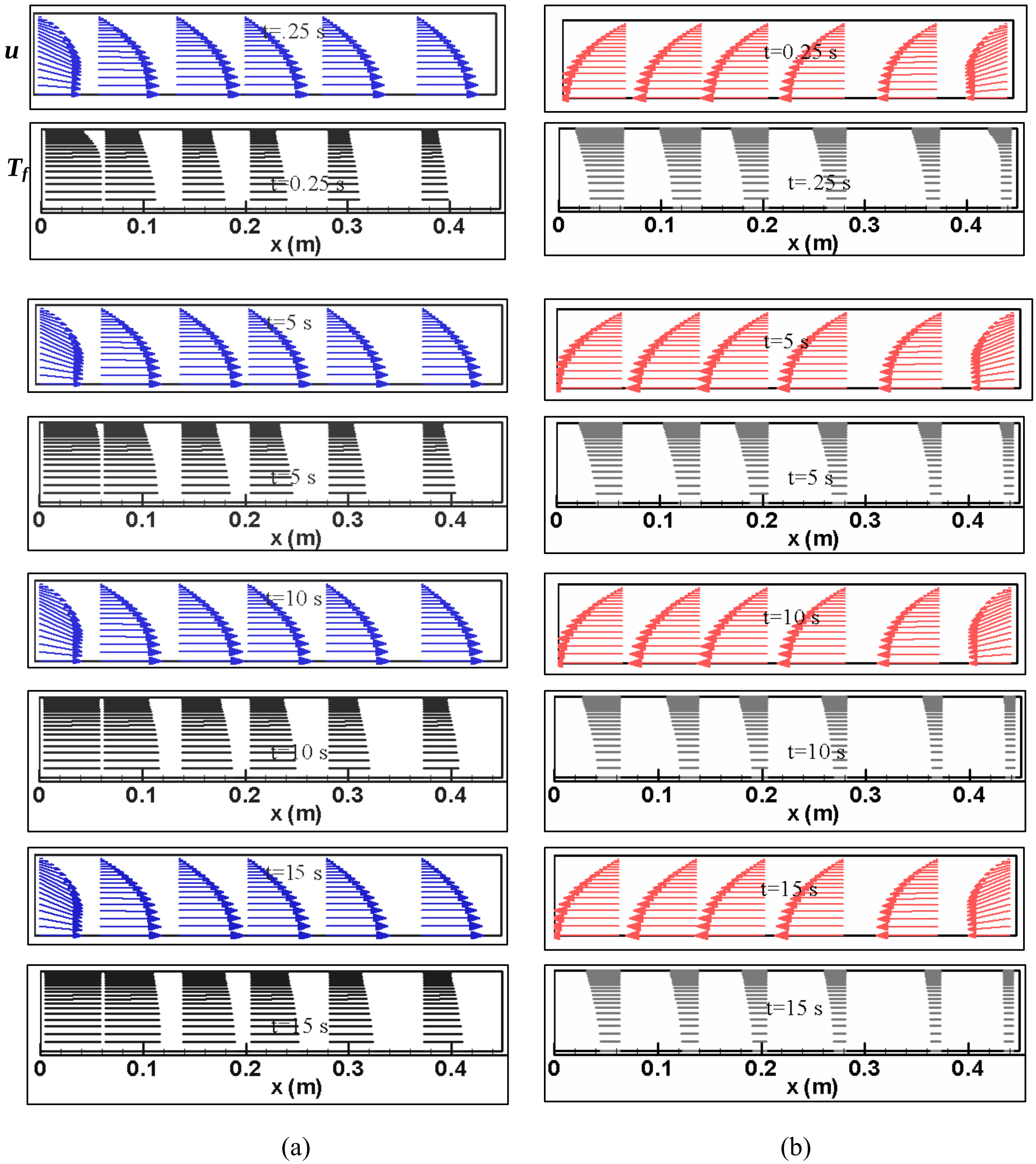


**Figure 7.** Flow velocity vectors in axial direction and temperature distribution (lines without arrow) of the fluid at different locations in a channel at successive times during **a)** cold period and **b)** hot period.

The overall heat-transfer performance of a regenerator is most conveniently expressed as the heat-transfer "effectiveness" $\varepsilon$, which compares the actual heat-transfer rate to the thermodynamically limited maximum possible heat-transfer rate. This value is defined as:

$$\varepsilon = \frac{\overline{T}_{co} - \overline{T}_{ci}}{\overline{T}_{hi} - \overline{T}_{ci}} \tag{20}$$

where $\overline{T}_{co}$ is the outlet bulk temperature in the hot period and is obtained by integrating the temperature with regard to both time and location. With this definition, we can investigate the effect of some parameters on a rotary regenerator. The results can then be compared with those of the empirical correlation given by Kays and London [6]:

$$\varepsilon = \left\{ (1 - \frac{1}{9C_r^{*1.93}})(\frac{1 - e^{[-NTU_{,0}(1-C^*)]}}{1 - C^* e^{[-NTU_{,0}(1-C^*)]}}) \right\} \quad (21)$$

where heat capacity ratio, $C^*$, and the number of transfer units, $Ntu_{,0}$ are defined as follows;

$$C^* = \frac{C_{\min}}{C_{\max}} \quad (22)$$

$$NTU_{,0} = \frac{1}{C_{\min}} \left[ \frac{1}{1/h_c A_c + 1/h_h A_h} \right] \quad (23)$$

***Rotational speed***. First, the effect of rotational speed on the regenerator effectiveness is examined; the result is shown in Fig.8. As seen in the figure, the effectiveness is increasing with the rotational speed of the regenerator. The effectiveness approaches a constant value after a speed of about 1.5 rpm, thus the effect of rotational speed on the effectiveness is bounded. A speed of 2rpm seems to be an optimum value as considered in the previous case study.

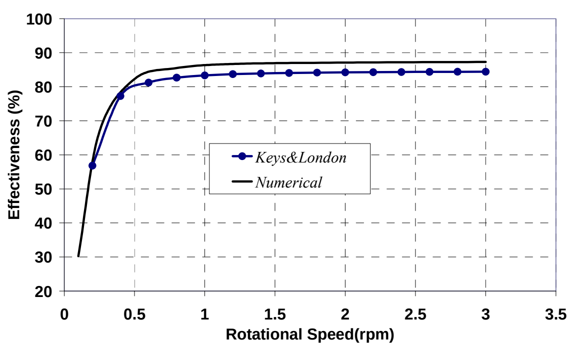


**Figure 8.** Variation of the effectiveness with rotational speed from numerical model and empirical correlations.

***Inlet velocity***. The flow inlet velocity (U) is a factor that has strong effect on the regenerator performance. The results are seen in Fig.9. The effectiveness is decreased as the inlet velocity is increased. This was expected because a lower fluid velocity results in a better heat transfer between fluid and solid wall. It should be mentioned that the velocities examined here were in the range that no turbulence was triggered (for the maximum velocity of 8m/s the Reynolds number was 2000). As a result, a high inlet velocity is not recommended for a rotary regenerator. The inlet velocity is not an independent variable; its value is assigned by the mass flow rate and rotor diameter. In the design of a regenerator, after the mass flow rate is set, the rotor size is selected such that it results in a proper U- velocity.

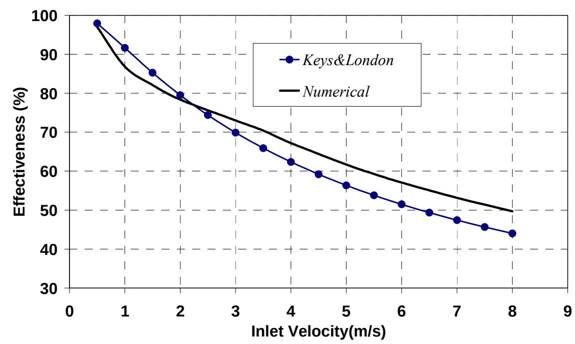


**Figure 9.** Effect of inlet velocity on the effectiveness of a rotary regenerator from numerical model and empirical correlations.

***Rotor length***. The rotor length is another parameter that affects the effectiveness. Its impact is shown in Fig.10. Increasing the rotor length leads to an increase in the effectiveness. Large axial temperature gradient in the solid wall (axial heat conduction) reduces the regenerator effectiveness and the overall heat transfer rate. Theoretically, it is known that an ideal regenerator is the one that has no axial conduction in the wall. Hence a longer rotor decreases the effect of axial conduction and, therefore, increases the effectiveness.

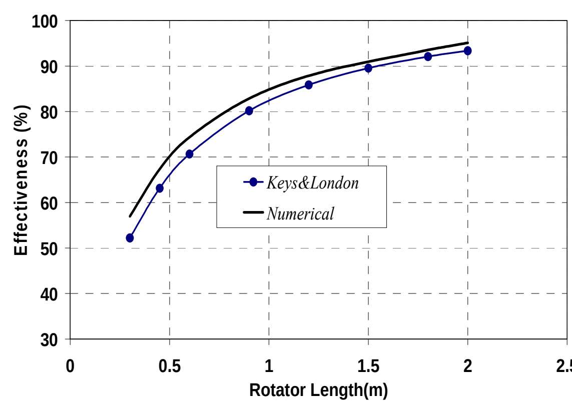


**Figure 10.** Variation of the effectiveness with rotor length from numerical model and empirical correlations.

## 5. CONCLUSIONS

In this study, a mathematical model was developed to simulate the fluid flow and heat transfer in a rotary regenerator. An axisymmetric channel was considered for this purpose. Two analytical methods based on 1D model of heat transfer were presented. They were used to validate the numerical model for a simple geometry. A very close agreement was obtained between numerical results and those of the analytical models. The velocity distribution and the fluid temperature profile were obtained from the model. While the fluid flow in the channels became fully developed in a short distance from the entry, the fluid temperature varied in both axial and radial directions throughout the channel. Effect of some important parameter on the effectiveness of a rotary regenerator was also investigated. These results were also compared with those of the empirical correlations where a good agreement was observed. Increasing the rotational speed resulted in an increase in the effectiveness while increasing the inlet velocity led to a decrease of the regenerator performance. This is the reason that laminar flow conditions are preferred in rotary regenerators. The effect of rotor length was also investigated; increasing the rotor length increased the effectiveness.

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