TEMPERATURE DISTRIBUTION IN A ROTARY HEAT EXCHANGER

Paulo Cesar Mioralli
State University at Campinas - UNICAMP
Faculty of Mechanical Engineering - Departament of Energy
PO Box: 6122 – ZIP CODE: 13083-970 – Campinas – SP – Brazil
mioralli@fem.unicamp.br

Marcelo Moreira Ganzarolli
State University at Campinas - UNICAMP
Faculty of Mechanical Engineering - Departament of Energy
PO Box: 6122 – ZIP CODE: 13083-970 – Campinas – SP – Brazil
ganza@fem.unicamp.br

Abstract. *The process of heat transfer in a rotary regenerator is numerically investigated in this paper. The aim of the study is to obtain the temperature distribution of the heat exchanging gases and the continuously rotating matrix. The energy partial differential equations for the gas and matrix are solved using the finite volume method and the calculations are performed using a computer program written in Fortran programming language. A correlation for the mean heat transfer coefficient in a duct of the regenerator matrix was obtained for the laminar regime using the commercial software PHOENICS 3.5. The outlet bulk temperature of each flow was obtained and compared to the value calculated by an existing method in literature. The results were analyzed and also compared with field data and a relatively good agreement was observed. Through the numerical simulations, it was possible to obtain the temperature distribution along a duct of the regenerator matrix in different angular positions.*

Keywords*: rotary regenerator, heat exchanger, heat transfer.*

1. Introduction

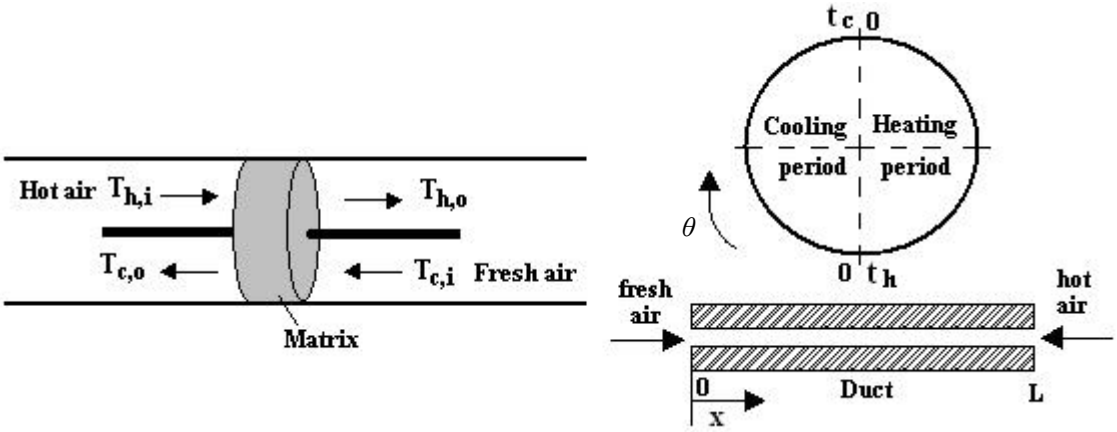
Industrial processes use large quantities of fuels and electricity with aim of promoting heat transfer. Great part of this heat is wasted either to the atmosphere or to water. The rotary regenerator is presented as an option to re-use some of this wasted heat, making possible to save a considerable quantity of fuel and electricity for the industry in general. Many efforts have been concentrated on researches that aim the improvement of this equipment and consequently to achieve a larger economy in the industrial sector.

One of the first articles on this subject was published by Karlsson and Holm (1943), which developed a theoretical investigation about pressure drop across different types of heating surfaces in the regenerator. Coppage and London (1953) contrasted the periodic-flow rotary regenerator to other types of heat exchangers systems and proposed design curves using a set of simple nondimensional parameters (effectiveness-NUT) for this equipment. Shah (1975) studied the effect of longitudinal heat conduction in the rotary heat exchanger and developed an empirical correlation for the effectiveness. Romie (1988) performed a study in which the exit gas temperature responses of the counterflow rotary regenerator were found for a unit step increase of the inlet temperature of either gas. Skiepko (1988) developed a method of measuring and monitoring seal clearances in a rotary heat exchanger. Skiepko (1989) presented two variants of models related to transport phenomema of energy in a rotary heat exchanger: one excluding and the other including longitudinal heat conduction in the matrix. Simonson and Besant (1999) presented effectiveness correlations which allow the designer to predict the sensible, latent and total effectiveness of energy wheels when the operation conditions are known. Büyükalaca and Yilmaz (2002) analyzed the effect of rotational speed on the effectiveness on rotary-type heat exchangers and developed an empirical equation which can be used even for very small rotational speeds. A more recent work is the one of Ghodsipour and Sadrameli (2003), which investigated the effect of dimensionless parameters on the effectiveness of rotary heat exchangers.

Articles like that by Skiepko (1989), which present the temperature distribution in the regenerator, are hardly found in the literature. Besides, pressure leakage effects are usually not included. In the present work, the process of heat exchange in a rotary regenerator is numerically analyzed with the purpose to predict the temperature distribution along a duct of the regenerator matrix in different angular positions. The influence of pressure leakage is also included in the modeling.

2. Numerical Study

Figure 1(a) represents a regenerator. It is constituted of a porous matrix that has the capacity to store a lot of energy. Two air streams are blown in counterflow through the regenerator. The matrix continuously rotates through these adjacent ducts that carry gas streams at different temperatures. As the matrix slowly rotates, the heat is transferred to half matrix by the hot gas and from the other half matrix to the fresh gas.



(a) (b)

Figure 1. (a) Schematic of the rotary regenerator; (b) Nomenclature for boundary conditions.

Nomenclature

A	- free flow cross-sectional area	$[m^2]$
A_m	- matrix cross-sectional area	$[m^2]$
A_T	- total cross-sectional area $(A + A_m)$	$[m^2]$
C	- flow stream heat capacity rate	$[W/K]$
C_{min}	- minimum of C_h and C_c	$[W/K]$
C_r	- total heat capacity rate of a matrix	$[W/K]$
C_r^*	- total matrix heat capacity rate ratio (C_r/C_{min})	
c	- specific heat	$[J/kg.K]$
D_h	- hydraulic diameter $(4r_h)$	$[m]$
h	- convective heat transfer coefficient	$[W/m^2K]$
k	- matrix thermal conductivity	$[W/m.K]$
$\dot{m}$	- gas mass flow rate	$[kg/s]$
m_{matrix}	- mass of the regenerator rotating matrix	$[kg]$
NTU	- number of heat transfer units	
P	- wet perimeter of a regenerator duct	$[m]$
q	- total heat transfer	$[W]$
ROT	- rotational speed for a rotary regenerator	$[rpm]$
r_h	- hydraulic radius (A/P)	$[m]$
t	- time	$[s]$
T	- temperature	$[^oC]$
x	- axial coordinate	$[m]$
θ	- angular coordinate	$[rad]$
σ	- porosity (A/A_T)	

Subscripts

i	- inlet
o	- outlet
c	- cold
h	- hot
g	- gas
m	- matrix

In this study, the mathematical model employed is similar to the models usually found in the literature. The regenerator mathematical model is based on the following simplifying assumptions:

1. Heat transfer between the exchanger and surroundings is negligible. There are no thermal energy sources within the exchanger. No phase change occurs in the regenerator.
2. The velocity and temperature of each fluid at the inlet are uniform over the flow cross section and are constant with time.
3. The fluid velocity on each side is considered constant with position, temperature and time through the matrix.
4. The heat transfer coefficients between the fluids and the matrix wall are constant (with position, temperature and time) throughout the exchanger.
5. The surface area of the matrix as well as the rotor mass is uniformly distributed.
6. The thermal properties of both fluids and the matrix wall material are constant, independent of time and position.
7. The temperature across the wall thickness is uniform at across section and the wall thermal resistance is treated as zero.

Applying the first law of thermodynamics over a small element (*dx*) of the regenerator duct, the following differential energy equations are obtained for the gas and the matrix:

Energy conservation equation for the gas:

$$\rho c_g \frac{\partial T_g}{\partial t} + \rho u c_g \frac{\partial T_g}{\partial x} + \frac{h\left(T_g - T_m\right)}{r_h} = 0 \tag{1}$$

Energy balance for the matrix:

$$\rho_m c_m \left(\frac{1-\sigma}{\sigma}\right)\frac{\partial T_m}{\partial t} - \frac{h\left(T_g - T_m\right)}{r_h} = \left(\frac{1-\sigma}{\sigma}\right)\frac{\partial}{\partial x}\left(k\frac{\partial T_m}{\partial x}\right) \tag{2}$$

Based on nomenclature presented in Fig. 1(b), t_c is the total time regarding the cold period and t_h the total time regarding the hot period of the regenerator. The boundary conditions for differential equations (1) and (2) are written below:

(a) For the gases: during the cooling and heating periods:

$$T_c\left(x = 0, t\right) = T_{c,i} \tag{3a}$$

$$T_h\left(x = L, t\right) = T_{h,i} \tag{3b}$$

(b) For the matrix:

$$\frac{\partial T_m\left(x = L, t\right)}{\partial x} = 0 \tag{3c}$$

$$\frac{\partial T_m\left(x = 0, t\right)}{\partial x} = 0 \tag{3d}$$

(c) The matrix temperature at the beginning of the heating period is equal to the matrix temperature at the end of the cooling period and vice-versa:

$$T_m\left(x, 0\right) = T_m\left(x, t_c\right), \quad for \quad 0 \le x \le L \tag{3e}$$

$$T_m\left(x, 0\right) = T_m\left(x, t_h\right), \quad for \quad 0 \le x \le L \tag{3f}$$

Discretized equations were obtained, using finite volume method, for the coupled differential equations (Eqs. 1 and 2) and the boundary conditions, Eq. (3). At the beginning of the calculations, a constant temperature is assumed. After that, the temperature profiles for the gas and the matrix were obtained, respectively, in distinct angular positions. The

calculations were carried out using a computer program written in Fortran programming language and it continues until the regenerator reaches the steady periodic condition, which means that the temperatures does not change anymore at each angular and axial position. The properties of fluids were evaluated at the mean temperature of each gas stream. The bulk temperature in the exit of each stream was obtained using a numerical integration. These values were compared to the values obtained by the effectiveness-NUT method.

3. Effectiveness-NTU Method for Rotary Regenerator

Results for the effectiveness (ε_r) of regenerator are obtained using a simple empirical formulation given by Kays and London (1984), in which ε denotes the effectiveness of a counterflow heat exchanger and φ_r is a correction factor that takes into account the rotational speed:

$$\varepsilon_r = \varepsilon \cdot \varphi_r \tag{4}$$

Kays and London (1984) suggested Eq. (5) for the correction factor, which depends on the total heat capacity of the matrix and the rotational speed (C_r):

$$\varphi_r = 1 - \frac{1}{9C_r^{*1.93}} \tag{5}$$

$$C_r = \frac{ROT}{60} \cdot m_{matrix} \cdot c_m \tag{6}$$

Flow leakages among the streams occur due pressure differences in the regenerator. Generally, the cold gas is at a higher pressure than that for the hot gas. Figure 2 illustrates the main leakages in the regenerator represented by $\dot{m}_1$ and $\dot{m}_2$. The inlet and outlet bulk temperature of the hot fluid, as it enters and leaves the regenerator matrix, are represented by $T'_{h,i}$ and $T'_{h,o}$.

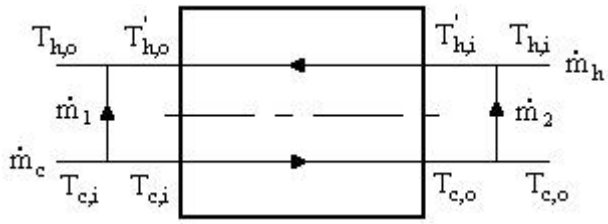


Figure 2. Main leakages in the regenerator.

The bulk temperatures, after mixing the incoming hot fluid stream and the corresponding flow leakage, are found as showed in Eqs. (7) and (8).

$$T'_{h,i} = \frac{T_{c,o} \cdot \dot{m}_2 + T_{h,i} \cdot \dot{m}_h}{(\dot{m}_h + \dot{m}_2)} \tag{7}$$

$$T_{h,o} = \frac{T_{c,i} \cdot \dot{m}_1 + T'_{h,o} \cdot (\dot{m}_h + \dot{m}_2)}{(\dot{m}_h + \dot{m}_2 + \dot{m}_1)} \tag{8}$$

4. Correlation for Convective Heat Transfer Coefficient

Figures 3(a), (b) and (c) illustrates the corrugated surfaces that constitutes the ducts of the regenerator matrix. With the purpose of evaluating the convective heat transfer coefficient (*h*), the flow and heat transfer inside a similar duct was numerically simulated using the commercial software PHOENICS 3.5. Figures 3(d), (e) and (f) illustrates the heat exchange surfaces in PHOENICS environment.

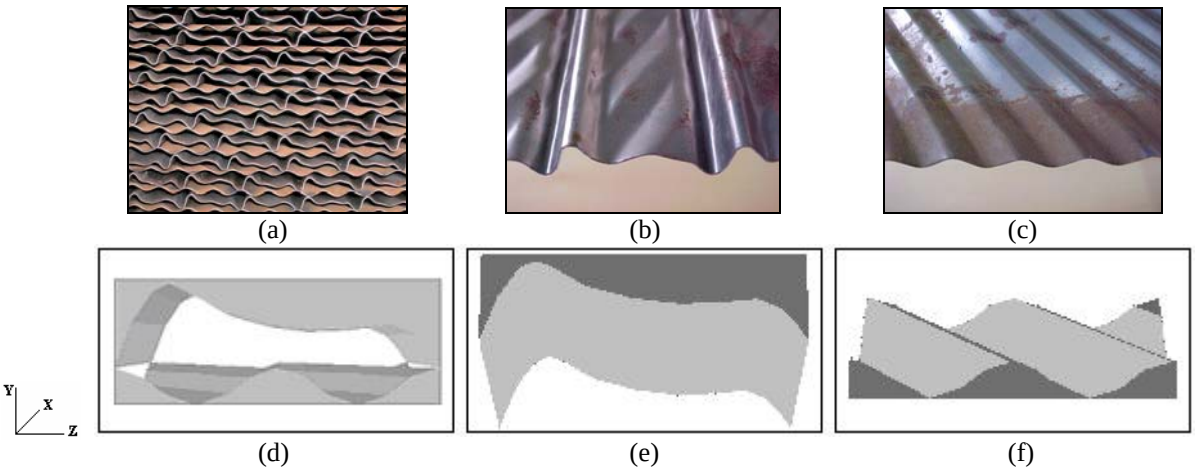


Figure 3. (a) Actual ducts of the regenerator; (b) top surface; (c) bottom surface; (d) Duct of the regenerator in PHOENICS; (e) top surface (PHOENICS); (f) bottom surface (PHOENICS).

The simulation was performed considering laminar flow and constant wall temperature boundary condition. Velocity and temperature are specified at the inlet and the outlet pressure was prescribed. Figure 4 illustrates the boundary conditions used in PHOENICS.

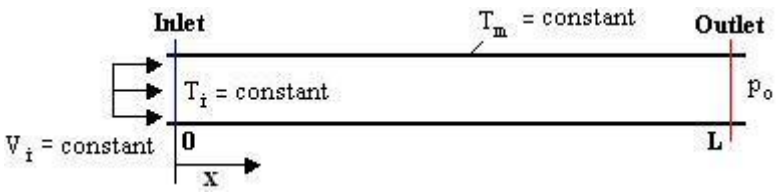


Figure 4. Boundary conditions used in PHOENICS.

The fluid properties were evaluated at the mean temperature of the stream. The mass flow-rate $(\dot{m})$ and the total heat transfer (q) were obtained from the simulations. A cartesian grid was used in PHOENICS and the convective heat transfer coefficient changes significantly with the grid refinement. Thus, Richardson's Extrapolation was performed to get a better value for the heat transfer coefficient. Three different velocities (V) were assumed for simulations and the value of the heat transfer coefficient was found for each case. Figure 5 shows the Richardson's extrapolation for simulated cases. The graph shows Nusselt number (Nu) versus control volume size (Δx) in x axial direction.

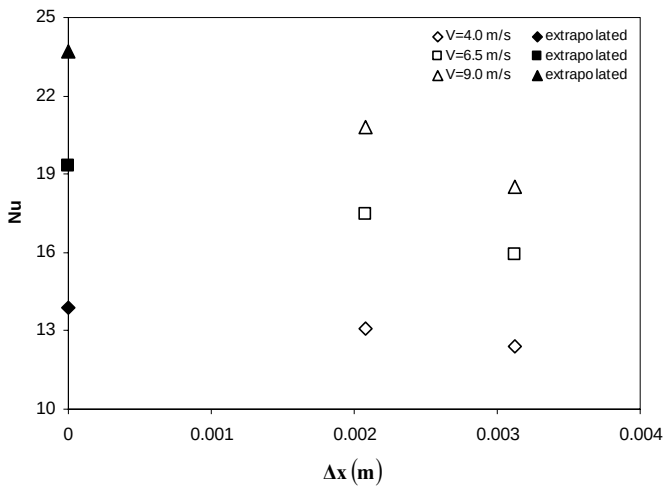


Figure 5. Richardson's extrapolation for simulated cases.

A correlation that allows the calculation of the mean heat transfer coefficient in duct was obtained plotting in a graph the values of Nusselt (*Nu*) versus Reynolds (*Re*) numbers obtained from the numerical simulations. The correlation showed in Eq. (9), valid for laminar flow, was obtained fitting a power law curve to these points.

$$Nu_{D_h} = 0.149 Re_{D_h}^{0.66} \tag{9}$$

5. Results and Discussions

5.1. Comparison with the Effectiveness-NTU Method

Table 1 shows field data, supplied by REPLAN (PETROBRAS), for a rotary regenerator in operation, except for the hydraulic radius and porosity that were calculated. These parameters were obtained from the mass and geometric data of a sample of the matrix illustrated in Figs. 3(a), (b) and (c).

Table 1. Operational conditions of the regenerator.

DATA	VALUE
Rotational speed	3 rpm
Length of the duct	1.35 m
External radius of the regenerator	2.725 m
Internal radius of the regenerator	0.665 m
Inlet mass flow-rate (cold gas)	237046 kg/h
Inlet mass flow-rate (hot gas)	178574 kg/h
Inlet temperature (cold gas)	81.6 ºC
Inlet temperature (hot gas)	483.9 ºC
Leakage 1	28.5% of mass flow-rate of the cold gas
Leakage 2	3.2% of mass flow-rate of the cold gas
Porosity	0.894
Hydraulic radius	0.002

The regenerator was simulated in the computer program with the operational conditions of Tab. 1. In order to validate the computational simulation, the outlet bulk temperature for each stream and the total heat transfer were compared to the values obtained by effectiveness-NTU method (Kays and London, 1984). The results are presented in Tab. 2. The differences between the values obtained by the two methods are very small, being around 0.5%.

Table 2. Comparison of results obtained by computer program and by effectiveness-NTU method.

PARAMETER	Computer Program	Effectiveness-NTU method	Difference (%)
Outlet temperature (cold gas) (ºC)	428.9	430.9	0.47
Outlet temperature (hot gas) (ºC)	142.4	141.8	0.45
Total heat transfer q (W)	1.715×10^7	1.724×10^7	0.58

5.2. Comparison with Field Data

The bulk temperature values were measured by REPLAN in the inlet and outlet of each flow. Table 3 presents a comparison between outlet bulk temperature values obtained by computer program and the field data measured by REPLAN.

Table 3. Comparison between outlet bulk temperatures values.

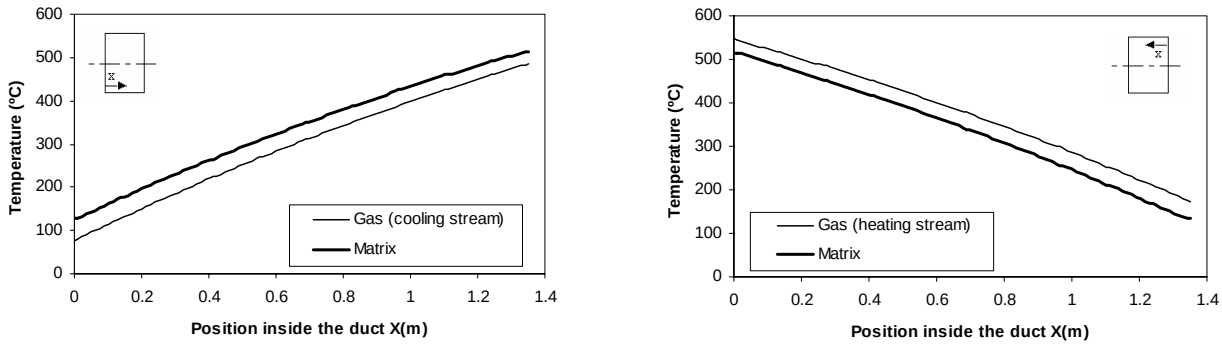
LOCAL	Outlet temperature of flows (ºC)		Difference (%)
	Computer program	Field data	
Cold stream	428.9	405.7	5.42
Hot stream	142.4	194.3	26.68

The computer program overestimates the outlet bulk temperature value of cold stream and underestimates its value in the hot stream. In the last case, the difference is 26.68%. Perhaps this difference can be explained by the fact that the exact positions of the temperature sensors inside the ducts were unknown. Therefore, the measured temperature value

could not represent the bulk temperature. In fact, the temperature sensor might be positioned in such a way that the hot fluid does not have received yet influence of $\dot{m}_1$. Thus, as illustrated in Fig. 2, the measured temperature might be closer to $T'_{h,o}$ than $T_{h,o}$. In this case, $T'_{h,o} = 164.5\,°C$ for the simulated standard case in this work and the difference between $T'_{h,o}$ and the field data is 15.26%.

5.3. Temperature Distribution

Temperature profiles along a duct of the regenerator were obtained in distinct angular positions under operational conditions shown in Tab. 1. Figures 6(a) and (b) illustrates the axial temperature profiles, for gas and matrix, obtained for two angular positions which correspond to the half of the cold and hot periods, respectively. These results were obtained after the regenerator to reach the steady periodic condition.



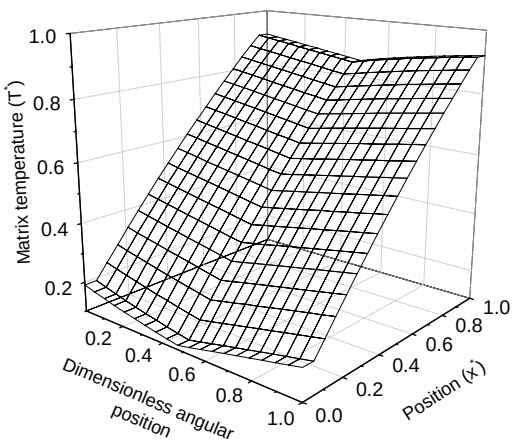
(a) (b)

Figure 6. Profiles temperature along a duct of the regenerator: (a) half of cold period; (b) half of hot period.

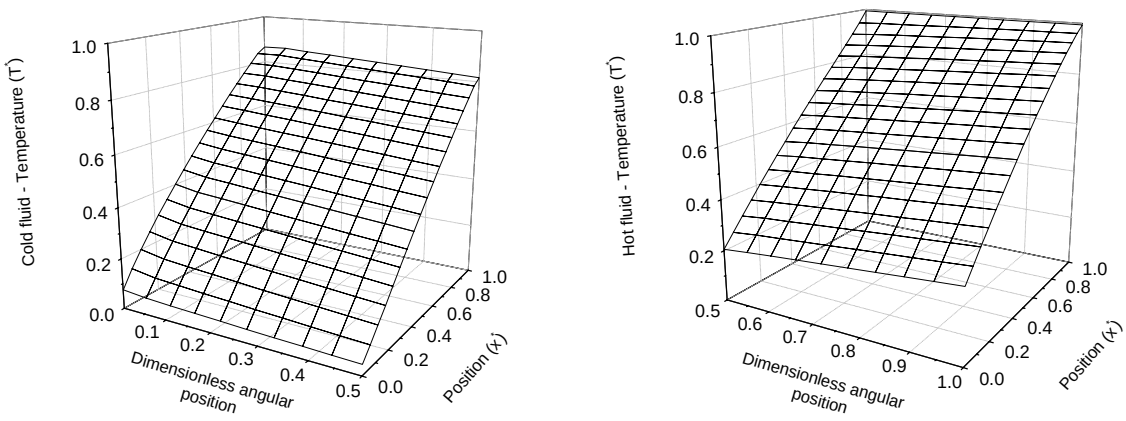
It can be observed in Figs. 6(a) and (b) that the axial temperature variation is approximately linear for the operational conditions covered in the present study. It can be also noticed that temperature variations, for gas as well as for the matrix, are around 350-400 °C from the inlet to outlet region.

Gas and matrix temperature fields in regenerators can be better illustrated by means of 3D charts (Fig. 7). For that, the parameters (temperature T, axial x and angular θ positions) were written in nondimensional form:

$$T^* = \frac{T - T_{c,i}}{T_{h,i} - T_{c,i}}, \qquad x^* = \frac{x}{L}, \qquad \theta^* = \frac{\theta}{2\pi} \tag{10}$$



(a)



(b) (c)

Figure 7. Temperature distribution in 3D charts: (a) matrix; (b) cold gas; (c) hot gas.

The three-dimensional graphs shows that the temperature variation in the angular direction is much smaller than the variation in the axial direction, for gases and matrix. For the studied case, it was verified that the temperature variation in the axial direction is about ten times higher than variation in the angular direction. This aspect indicates that the knowledge of the temperature profile in the axial direction is of great importance in the dimensioning of the rotary regenerator sealing system. The adequate adjustment of such system can minimize seal clearances and, consequently, flow leakages in the regenerator.

6. Conclusions

The heat transfer process in a rotary regenerator was investigated numerically. A mathematical model was solved and a computer program in FORTRAN language was developed for simulation. A correlation to evaluate the mean convective heat transfer coefficient in a regenerator with the presented wavy-wall duct was obtained using the commercial software PHOENICS 3.5. With purpose of validation, the outlet bulk temperature of each flow was obtained and compared to the value calculated by effectiveness- NTU method for rotary regenerator. An excellent agreement was observed. The results were also compared with field data and a relatively good agreement was observed. The temperature distribution was obtained in the equipment and it was verified that the temperature variation in the angular direction is much smaller than the variation in the axial direction for the gases and matrix. Flow leakage effects were also included in the theoretical model.

7. References

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