THERMAL OPTIMIZATION OF A ROTARY REGENERATOR WITH FIXED PRESSURE DROP

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Abstract. *The thermal exchange in a rotary regenerator is optimized by means of porosity variation given assigned values for the pressure drop in the matrix ducts. The heat transfer was numerically calculated. The matrix ducts were admitted smooth and equilateral triangular. The fluid flow and the mean heat transfer coefficient were obtained from correlations taking into account the entrance region of the matrix ducts. The leakage effects between the streams and the partial flow blockage, caused by part of the equipment structure, were also considered in this work. The matrix porosity was varied and the remaining geometric parameters were kept constant. The calculations were performed using a computer program and the total heat transfer was calculated as function of the matrix porosity. It was verified that, for every assigned pressure drop, there is a value of the porosity which maximizes the heat transfer rate. A scale analysis was performed to estimate the optimal porosity and the maximum heat transfer rate. Based on this analysis, modified parameters were introduced and it was verified that the scale analysis properly predicts the order of magnitude of the optimal porosity and the corresponding maximum heat transfer rate.*

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

1. INTRODUCTION

The rotary regenerator is an important component of energy intensive sectors, which is used in many heat recovery systems. The importance of this equipment has been recognized in applications like thermal comfort and gas turbine power plant among other industrial processes. For decades, many studies have been concentrated in search of improvements in regenerator. Among the recent works, Jassim et al. (2004) demonstrated the importance of exergetic analysis for minimizing irreversibilities in a rotary regenerator. Shang et al. (2005) investigated experimentally the ice formation in a regenerative exchanger used in ventilation systems with very low temperatures. Shang and Besant (2006) verified that the effective *NTU* decreases when manufacturing tolerances in the flow channels are considered for regenerators. Drobnic et al. (2006) proposed an online method to control the radial leakages in the equipment. Nobrega and Brum (2007) investigated the distribution of heat transfer coefficients along the flow direction.

There are several points raised about the rotary regenerator. Meanwhile, works that focusing on the influence of the matrix porosity in the regenerator performance were not found. This work analyses the influence of the porosity in the regenerator performance. The goal is the thermal optimization the heat transfer with established values for the pressure drop in matrix ducts. For every assigned pressure drop, there is a value of the porosity which maximizes the heat transfer rate.

2. DESCRIPTION OF REGENERATOR

Figure 1 represents a rotary regenerator. Two gas streams are blown in counterflow through the side-by-side ducts of regenerator; one for the fresh gas; the other for the hot stream. The porous matrix, that stores internal energy, continuously rotates through these adjacent ducts and receives heat from the hot fluid in one side and transfers this energy to the cold fluid on the other side. For this study, the matrix channels were admitted as being triangular equilateral.

Considering A the free flow cross-sectional area and A_m the matrix cross-sectional area of the regenerator, based in the Fig.1, the total cross-sectional A_T area is calculated as $A_T = \pi\left(R_e^2 - R_i^2\right) = A + A_m$. The relation between A and A_T is known as porosity $\left(\sigma = A/A_T\right)$. The relation between A and the perimeter P of the plates that compose the matrix is known as hydraulic radius $\left(r_h = A/P\right)$. Perimeter P can be expressed as function of the matrix cross-sectional area,

$$P = \frac{A_m}{(e/2)} \tag{1}$$

where e is the thickness of the plates that constitutes the matrix channels. Using Eq. (1) and the definitions above, the hydraulic radius can be express as function of the porosity.

$$r_h = \frac{\sigma}{1-\sigma} \cdot \left(\frac{e}{2} \right) \tag{2}$$

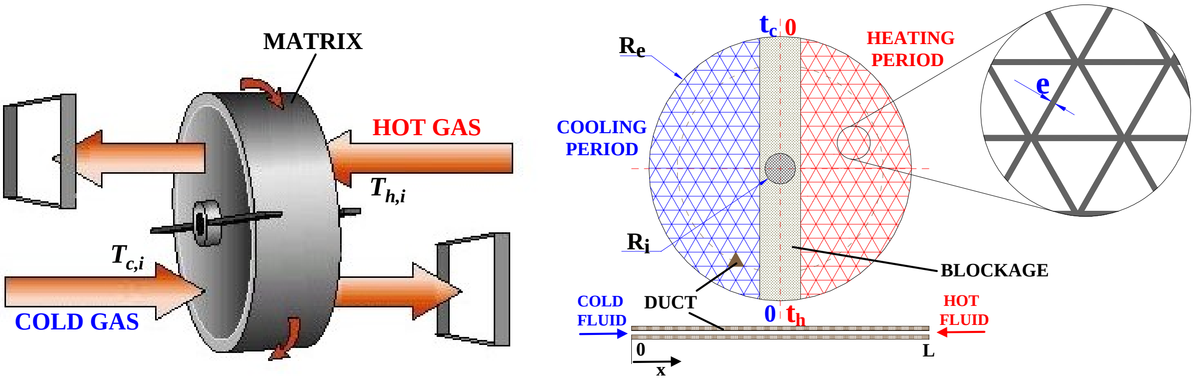


Figure 1. Schematic of the rotary regenerator.

3. NUMERICAL STUDY

A numerical study was conducted to obtain the temperature distribution along a duct of the regenerator matrix in different angular positions. The regenerator mathematical model is based on the following simplifying assumptions:

- 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.
- The velocity and temperature of each fluid at the inlet are uniform over the flow cross section and are constant with time.
- The fluid velocity on each side is considered constant with position, temperature and time through the matrix.
- The heat transfer coefficients between the fluids and the matrix wall were obtained from correlations.
- The surface area of the matrix as well as the rotor mass is uniformly distributed.
- The thermal properties of both fluids and the matrix wall material are constant, independent of time and position.
- The temperature is uniform across the wall thickness.

The following differential energy equations are obtained for the gas and the matrix, respectively:

$$\rho c_g \frac{\partial T_g}{\partial t} + \rho u c_g \frac{\partial T_g}{\partial x} + \frac{h(T_g - T_m)}{r_h} = 0 \tag{3}$$

$$\rho_m c_m \left(\frac{1 - \sigma}{\sigma} \right) \frac{\partial T_m}{\partial t} - \frac{h(T_g - T_m)}{r_h} - \left(\frac{1 - \sigma}{\sigma} \right) \frac{\partial}{\partial x} \left(k_m \frac{\partial T_m}{\partial x} \right) = 0 \tag{4}$$

where u is velocity, c is specific heat and k_m is the matrix thermal conductivity. The boundary conditions for numerical solution were established according to Fig. 1, in which a matrix duct will be accompanied during a revolution. Based on nomenclature presented in Fig. 1, 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 (3) and (4) are written below:

- For the gases: during the cooling and heating periods:

$$T_c(x = 0, t) = T_{c,i} \tag{5}$$

$$T_h(x = L, t) = T_{h,i} \tag{6}$$

- For the matrix:

$$\frac{\partial T_m(x = L, t)}{\partial x} = 0 \tag{7}$$

$$\frac{\partial T_m(x = 0, t)}{\partial x} = 0 \tag{8}$$

- 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:

$$T_m(x, 0) = T_m(x, t_c), \quad for \quad 0 \le x \le L \tag{9}$$

$$T_m(x, 0) = T_m(x, t_h), \quad for \quad 0 \le x \le L \tag{10}$$

4. RESOLUTION METHOD

The calculations were performed using a computer program written in Fortran programming language. The regenerator was simulated with fixed geometric parameters, except the porosity, which was varied. The pressure drop due contraction and expansion in the entrance and exit of the matrix ducts were not been considered. In the simulation was considering the partial flow blockage, caused by part of the equipment structure, which is show in Fig 1. The flow leakages among the streams that occur due pressure differences in the regenerator were also taken into account in the simulation.

The total heat transfer is calculated from fixed values for pressure drop in the matrix ducts. The pressure drop was assumed to be the same for both gas streams. An iterative process was necessary to obtain the fluid flow and heat transfer. In the beginning of this process, a value for the bulk temperature in the exit of each stream was assumed. The fluid properties were evaluated at the mean temperature of each gas stream. According to the prescribed pressure drop, the fluid flow and the heat transfer were obtained from correlations. The coupled differential energy equations (fluid and matrix) were solved using the finite volume method and the temperature profiles for the gas and the matrix were obtained, respectively, in distinct angular positions. The bulk temperature in the exit of each stream was obtained using a numerical integration. The calculations continue until the regenerator reaches the steady periodic condition, which means that the temperatures do not change anymore at each angular and axial position. Then the fluid properties were recalculated and an iterative process continues until convergence of the bulk temperatures in the exit of each stream. The whole process is repeated for each value of the matrix porosity.

4.1. Hydrodynamic Analysis

In this study the pressure drop due contraction and expansion in the entrance and exit of the matrix ducts were not been considered. The distributed pressure drop in the flow can be obtained by equation of Darcy-Weissbach taking into account the hydrodynamic entry length.

$$\Delta P = f \cdot \rho \cdot \frac{L}{r_h} \cdot \frac{V^2}{2} + \rho \cdot K \cdot \frac{V^2}{2} \tag{11}$$

where V is the fluid velocity and K is the pressure drop dimensionless coefficient for the hydrodynamic entry length. Lundgren et al. (1964) developed an analytical method to obtain the pressure drop in ducts with arbitrary cross-section area taking into account the entry region for laminar flow. Result for equilateral triangular duct was $K = 1.818$.

The Fanning friction factor f was obtained from correlations for smooth equilateral triangular duct and fully developed flow. For laminar flow the following correlation was used, Marco and Han (1955).

$$f = \frac{40}{3 \cdot Re_{D_h}} \tag{12}$$

where Re_{D_h} is the Reynolds number based on the hydraulic diameter $(D_h = 4r_h)$ of the channel.

For turbulent flow, the friction factor was obtained using Eq. (13) for equilateral triangular duct proposed by Malák et al. (1975) from experimental measures to Altemani and Sparrow (1980). This correlation is valid in the range $4000 < Re_{D_h} < 8 \cdot 10^4$.

$$f = 0.0425 Re_{D_h}^{-0.2} \tag{13}$$

For turbulent flow in circular duct, Zhi-qing (1982) obtained and scheduled K as a function of geometric parameters of regenerator and Re_{D_h}. In this work, a polynomial equation for K was obtained from Zhi-qing's data. Ahmed and Brundrett (1971) introduced a concept of equivalent diameter $D_e = 1.5D_h$ to triangular duct. This concept provides good results for triangular ducts when used in correlations for circular duct. In this study, D_h was replaced by D_e in the obtained polynomial equation. Results obtained from polynomial equation were compared with results obtained from computational simulation of equilateral triangular duct using a commercial package. It was observed that the results were closer when D_e was used in polynomial equation instead of D_h.

The mass flow rates were calculated from the friction factor and the pressure drop prescribed in the matrix ducts. The Reynolds number was calculated to verify the flow regime. The critical Reynolds number that determines the flow transition is $Re = 2800$ for equilateral triangular duct, Nikuradse (1930).

4.2. Thermal Analysis

For laminar flow, the convective heat transfer coefficient h was obtained using correlation considering constant wall temperature boundary condition and simultaneous development for equilateral triangle. Wibulswas (1966) presented in a table the variation of mean Nusselt number Nu_m as a function of geometric parameters of regenerator and Re_{D_h} for Prandtl number $Pr = 0.72$. A polynomial equation for Nu_m was obtained from data of Wibulswas (1966).

Altemani and Sparrow (1980) proposed a correlation to calculate the mean Nusselt Nu_d number for fully developed turbulent flow with constant heat flux boundary condition. This correlation is valid for $Pr = 0.7$ and $4000 < Re < 6 \cdot 10^4$.

$$Nu_d = 0.019 Re_{D_h}^{0.781} \tag{14}$$

In the literature were not found correlations for Nusselt that take into account the entry region for turbulent flow in triangular ducts. Turbulent flow in smooth ducts with simultaneous development is affected by geometric configuration of the entry region. Mills (1962) proposed, for Nu_m, the following correlation for circular cross section duct with straight section in entry region.

$$\frac{Nu_m}{Nu_d} = 1 + \frac{2.4254}{\left(L/D_h\right)^{0.676}} \tag{15}$$

where Nu_d the Nusselt number for fully developed flow This correlation is valid for $Pr = 0.7$ and $Re < 3 \cdot 10^4$ with constant wall temperature or constant heat flux boundary condition. In this work was used Eq. (15) for circular duct with D_h replaced by D_e and Nu_d was calculated by Eq. (14) for triangular duct.

5. SCALE ANALYSIS

A scale analysis was performed to estimate the optimal porosity and the maximum heat transfer rate. According to Kays and London (1964), the regenerator effectiveness can be obtained using a correction that takes into account the matrix rotational speed and heat capacity. Considering that the order of magnitude of such correction is ≈ 1, the heat transfer q for a counterflow heat exchanger with heat capacity rates of the same order $\left(C_h \approx C_c\right)$ can be evaluated from the $\varepsilon - NTU$ method as function of the maximum temperature difference $\Delta T_{max} = T_{h,i} - T_{c,i}$.

$$q \approx \frac{1}{\left(\dfrac{1}{UA_{tr}} + \dfrac{1}{\dot{m} \cdot c_p}\right)} \Delta T_{max} \tag{16}$$

If ΔT_{max} is a fixed value, the existence of a maximum heat transfer q can be verified examining the sum between parenthesis in the right side of the Eq. (16). For established pressure drop, two asymptotic limits for the porosity values can be identified: for small porosity, the heat exchange area increases and the mass flow rate decreases; on the other hand, if the porosity increases, the mass flow rate also increases and the heat exchange area decreases. Since the two terms between parentheses in Eq. (16) vary in opposite ways as the porosity changes, the magnitude of the maximum heat transfer will be obtained when,

$$UA_{tr} \cong \dot{m} \cdot c_p \tag{17}$$

To find out this optimal porosity, the order of magnitude for both the terms of the Eq. (17) must be found. A similar analysis is presented by Ganzarolli and Altemani (2003) in the study of forced convection in a plate with fins.
The force balance in a channel of matrix allows writing the speed of the fluid as function of the pressure drop Δp and the hydraulic radius. Using still the Eq. (2), in which the hydraulic radius is express as function of the porosity, then:

$$V = \left[\frac{\Delta p \cdot e}{f \cdot L \cdot \rho} \cdot \frac{\sigma}{(1-\sigma)} \right]^{1/2} \tag{18}$$

The term in the right side of Eq. (17) is given by:

$$\dot{m} \cdot c_p = A_L \cdot c_p \cdot \sigma \cdot \left[\frac{\Delta p \cdot e \cdot \rho}{f \cdot L} \cdot \frac{\sigma}{(1-\sigma)} \right]^{1/2} \tag{19}$$

where $A_L = A_T / 2$. The magnitude of UA_{tr} is given by:

$$UA_{tr} = \frac{h}{2} \cdot P \cdot L = \frac{h \cdot A_L \cdot L \cdot (1-\sigma)}{e} \tag{20}$$

where h is the convective heat transfer coefficient in the matrix channel. In this analysis, the convective heat transfer coefficients for the two streams are considered of the same order of magnitude.

For fully developed laminar flow, results from an extrapolation of Wibulswas (1966) and graphics results of Schmidt e Newell (1967) show $Nu_m = 2.47$ for equilateral triangular duct with constant wall temperature boundary condition and $Pr = 0.7$. Using this result, considering both streams in laminar flow and using still Eq. (2), the heat transfer coefficient h is given by:

$$h = \frac{Nu_m \cdot k}{D_h} = \frac{2.47 \cdot k}{4 \cdot r_h} \Rightarrow h = \frac{2.47}{2} \cdot \frac{k}{e} \cdot \frac{(1-\sigma)}{\sigma} \tag{21}$$

where k is the fluid thermal conductivity. Substituting h in Eq. (20) and manipulating, then:

$$UA_{tr} = \frac{2.47}{2} \cdot \frac{k}{e^2} \cdot A_L \cdot L \cdot \frac{(1-\sigma)^2}{\sigma} \tag{22}$$

To express the optimal porosity as function of the pressure drop, the friction factor correlation for fully developed laminar flow, Eq. (12), was used. Combining the Eq. (12) with the Eq. (18) and eliminating speed V, after algebraic manipulations, the friction factor is obtained as function of the pressure drop Δp,

$$f = \left(\frac{20}{3} \right)^2 \cdot \frac{\mu^2}{e^3} \cdot \frac{L}{\rho \cdot \Delta p} \cdot \left(\frac{1-\sigma}{\sigma} \right)^3 \tag{23}$$

A similar process can be applied considering both streams in fully developed turbulent flow. The heat transfer coefficient h can be evaluated using the Colburn analogy.

$$St = \frac{f}{2} \cdot Pr^{-2/3} = \frac{h}{\rho \cdot c_p \cdot V} \tag{24}$$

where St is the Stanton number. Substituting speed V from Eq. (18) in the Eq. (24), then:

$$h = \frac{c_p}{2} Pr^{-2/3} \left[\frac{\Delta p \cdot e \cdot \rho \cdot f}{L} \cdot \frac{\sigma}{(1-\sigma)} \right]^{1/2} \tag{25}$$

Substituting h from Eq. (25) in the Eq. (20) and manipulating, it is obtained,

$$UA_{tr} = \frac{A_L}{2} c_p Pr^{-2/3} (1-\sigma) \left[\frac{\Delta p \cdot \rho \cdot f \cdot L \cdot \sigma}{e(1-\sigma)} \right]^{1/2} \tag{26}$$

Combining the Eq. (13) with the Eq. (18) and eliminating speed V, after algebraic manipulations, the friction factor is obtained as function of the pressure drop Δp for turbulent flow.

$$f = 0.02565 \cdot v^{2/9} \cdot \left(\frac{\sigma}{1-\sigma} \cdot e \right)^{-1/3} \cdot \left(\frac{\Delta p}{L \cdot \rho} \right)^{-1/9} \tag{27}$$

Substituting f from the Eq. (23) in the Eq. (19) and equating Eqs. (19) and (22), it is found the optimal porosity as function of pressure drop when both stream are laminar. Substituting f from the Eq. (27) in the Eq. (19) and equaling Eqs. (19) and (26), it is found the optimal porosity when both stream are turbulent. The expression to estimate the optimal porosity for both cases is:

$$\left(\frac{\sigma}{1-\sigma} \right)_{opt} \approx C_1 \left(\frac{L}{e} \right) \left(\frac{\Delta P \cdot L^2}{\mu \cdot \alpha} \right)^m \tag{28}$$

where C_1 and m are parameters that depends whether the flow is laminar or turbulent. For laminar flow, $C_1=1,7$ and $m = -1/4$. For turbulent flow, $C_1 = 0.038 Pr^{-5/12}$ and $m = -1/12$.

Substituting the term $[\sigma/(1-\sigma)]$ from Eq. (28) in the Eqs. (19) or (22), the scale for the maximum heat transfer, when both streams are laminar, can be determined by the Eq. (16). Substituting the term $[\sigma/(1-\sigma)]$ from Eq. (28) in the Eqs. (19) or (26), the scale for the maximum heat transfer, when both streams are turbulent, can be determined by the Eq. (16). The maximum heat transfer, for both cases, can be estimate by:

$$q_{max} \approx C_2 \cdot k \left(\frac{A_L}{L} \right) \Delta T_{max} \left(\frac{\Delta p \cdot L^2}{\mu \cdot \alpha} \right)^{1/2} \sigma_{opt} \tag{29}$$

where $C_2 = 0.2152$ for laminar flow and $C_2 = \left(1/2\sqrt{2}\right) \cdot Pr^{1/6}$ for turbulent flow. The dimensionless parameters presented in the scale analysis were calculated with properties evaluated at the average of the hot and cold inlet temperatures. The dimensionless parameter $\left(\Delta p \cdot L^2 / \mu \cdot \alpha\right)$ is known as Bejan number Be.

The scales obtained in the Eqs. (28) and (29) are used to define the parameters σ^* and q^* as follows.

$$\sigma^* = \frac{\sigma/(1-\sigma)}{(L/e) Be^m} \tag{30}$$

$$q^* = \frac{q}{k(A_L/L) \Delta T_{max} Be^{1/2}} \tag{31}$$

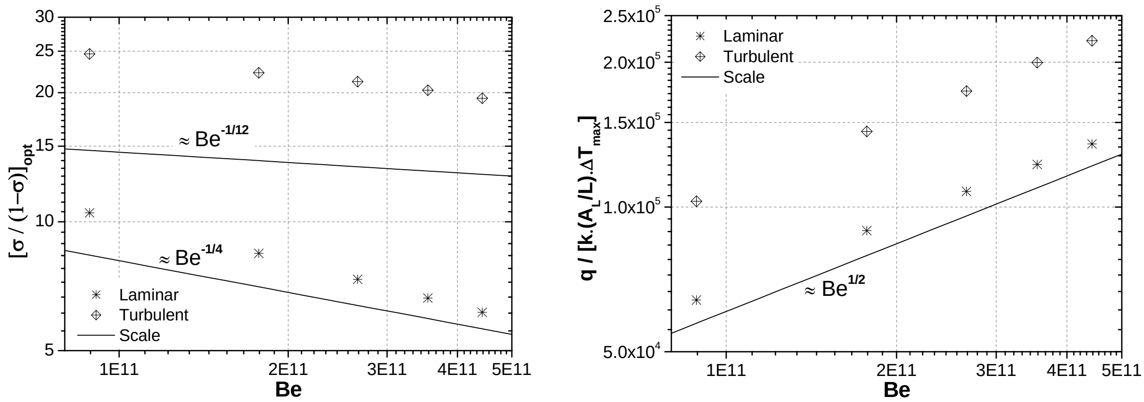
6. RESULTS AND DISCUSSIONS

The simulations were based in a typical rotary regenerator used in an oil refinery with geometric data and operational conditions show in Table 1. The pressure drops are assumed the same in both streams of regenerator with values of *100, 200, 300, 400* and *500Pa*.

Table 1: Geometric data and operational conditions of a rotary regenerator.

$Rotational\ Speed(rpm)$	$L(m)$	$e(m)$	$R_e(m)$	$R_i(m)$	$T_{c,i}(°C)$	$T_{h,i}(°C)$
3	*1.4*	*0.0005*	*2.7*	*0.7*	*80*	*485*

The power laws anticipated in Eqs. (28) and (29) for the variation of σ_{opt} and q_{max} as a function of *Be* can be compared to the optimal values obtained numerically. In Figure 2(a), the power laws $Be^{-1/4}$ and $Be^{-1/12}$, from Eq. (28), for laminar and turbulent flow, are represented by continuous lines. The discrete points represent the optimal values for $[\sigma/(1-\sigma)]_{\text{opt}}$ obtained numerically as function *Be* for both flow regimes. Observing the discrete points for turbulent flow, it can be noticed that the inclination is relatively close to that predicted by the scale analysis. For laminar flow, Fig. 2(a) shows that the power law $Be^{-1/4}$ slight underestimates the influence the Bejan number on σ_{opt}. In this case, the power law obtained from the numerical results is $Be^{-1/3}$.

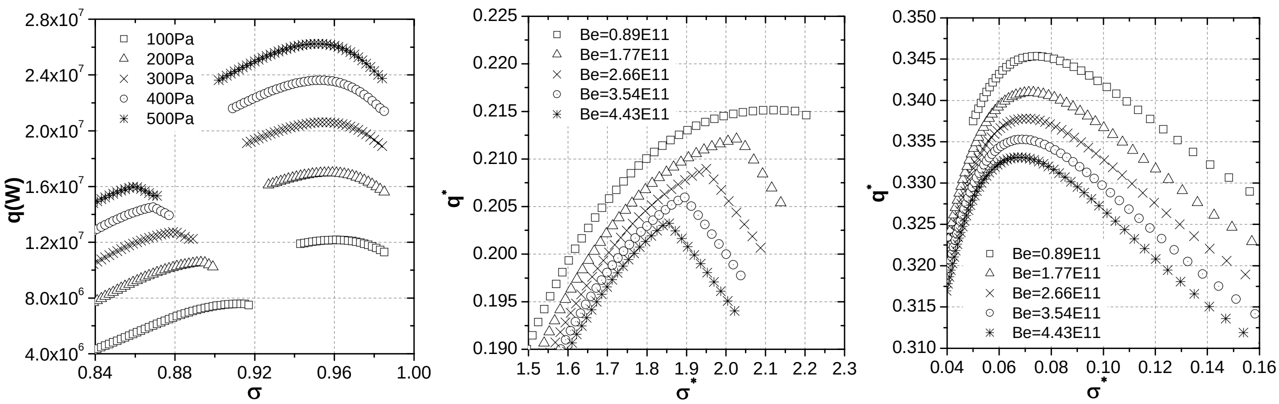


(a) (b)

Figure 2. Comparison between scale analysis and numerical method: (a) optimal porosity, (b) maximum heat transfer.

In Figure 2(b), the power law $Be^{1/2}$, from Eq. (29), is the same for both flow regimes and it is represented by a continuous line. The discrete points represent the optimal values for $q/[k(A_L/L)\Delta T_{max}]$ obtained numerically as a function of *Be*. According to this figure, the optimal values calculated numerically, for both flow regimes, obey very closely the power law $Be^{1/2}$ predict from the scale analysis.

Figure 3 shows the heat transfer as a function of the porosity for each simulated case. Each curve presents a maximum value for the total heat transfer, which corresponds to an optimal porosity. Two regions are observed in Fig. 3(a): the first region where both streams are laminar and the second region where both streams are turbulent. There is a maximum value of the total heat transfer and the corresponding porosity in each region. Notice that the maximum heat transfer always occur for turbulent flow and porosity values closer to unity. Introducing the parameters σ^* and q^*, from Eqs. (30 and (31), Figures 3(b) and 3(c) shows the maximum values of the total heat transfer collapsing in a smaller region with reference to the vertical coordinate.



(a) (b) (c)

Figure 3. Total heat transfer: (a) σ versus q ; (b) σ^* versus q^* - laminar; (c) σ^* versus q^* - turbulent.

Comparing the values for q^*_{max} anticipated by the scale analysis from Eqs. (29) and (31), $C_2 = 0.2159$ for laminar flow and $C_2\ \sigma_{opt} \approx 0.33$ for turbulent flow, to the values displayed in Figs. 3(b) and 3(c), it is observed that they are of the same order of magnitude.

The optimal values anticipated by Eq. (28) and (30) for σ^*_{opt} are $C_1 = 1.7$ for laminar flow and $C_1 = 0.045$ for turbulent flow. These values are also of the same order of magnitude of that showed in Figs. 3(b) and 3(c).

7. CONCLUSIONS

The numerical simulation of the heat transfer in a rotary regenerator confirmed that there is a value of the porosity which maximizes the heat transfer rate for every assigned pressure drop.

A scale analysis was performed to predict the influence of the pressure drop, represented by the Bejan number, on the optimal porosity and maximum heat transfer. For turbulent flow, the scale analysis approaches the power law in accordance with the numerical results. For laminar flow, it underestimates the influence the Bejan number. For the maximum heat transfer obtained numerically, the tested cases obey very closely the power law predict from the scale analysis for both flow regimes.

By means of the introduced parameters $\sigma*$ and $q*$, it was verified that the scale analysis was able to properly predicts the order of magnitude of the optimal porosity and the corresponding maximum heat transfer rate.

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