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MatrixRotation Range of Rotary Regenerators for Good Heat Transfer and Acceptable Fluid CarryoverLeakage

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ABSTRACT

Three typical rotary regenerators with both streams under the laminar flow regime are computationally analyzed from changes in matrix rotation. The mass flow rate, the inlet temperatures of gas streams and the geometric dimensions of the regenerators were fixed in the analysis. The ratio of residence time t_{res} of flow on each side of the equipment to the time t_0 required for a complete matrix rotation is calculated for the regenerators, representing the fluid carryover leakage. The convective heat transfer coefficient is obtained from correlation. The heat transfer rate is determined using the Effectiveness-NTU method specific to rotary regenerators. Ranges of the matrix rotation values and the ratio (t_{res}/t_0) that provide good heat transfer rate and acceptable fluid carryover are chosen for each investigated equipment. Additionally, an analysis of the regenerator effectiveness as function of the time required for a complete matrix rotation is presented. The results reveal that the chosen matrix rotation ranges shorten as the size of the rotary regenerator increases and the limit levels of the chosen matrix rotation decrease as the dimensions and typical operating conditions of the rotary regenerators increase.

Keywords—*rotary regenerator; matrix rotation; carryover leakage; computational analysis.*

NOMENCLATURE

A *free flow cross-sectional area,* m^2
A_m *matrix cross-sectional area,* m^2
A_T *total cross-sectional frontal area* $(A+A_m)$, m^2
A_{tr} *heat exchange area,* m^2
C *heat capacity rate of fluids,* W/K
C_r *matrix heat capacity rate,* W/K
C_r^* *matrix heat capacity rate ratio on the cold or hot side*
c_p *specific heat of gas under constant pressure,* $J/kg\ K$
c_m *specific heat of matrix,* $J/kg\ K$
D_h *hydraulic diameter,* m
e *thickness of the plates that constitute the matrix channels,* m
h *convective heat transfer coefficient,* W/m^2K
k *thermal conductivity,* W/mK
L *length of matrix,* m
$\dot{m}$ *gas mass flow rate,* kg/s
m_m *mass of matrix,* kg
n *rotational speed,* rpm
NTU *number of heat transfer units on the cold or hot side*
Nu *Nusselt number*
P *periphery of the channel,* m
Pr *Prandtl number*
Q *heat transfer rate,* W
Re *Reynolds number*
r_h *hydraulic radius* $(D_h/4)$, m
T *temperature,* K
t_0 *time required for a complete matrix rotation,* s
t_{res} *residence time, s*
u *fluid velocity in the channel,* m/s

Greek Symbols

μ *dynamic viscosity,* Ns/m^2
ε_0 *effectiveness of counterflow heat exchanger*
ε_r *regenerator effectiveness*
φ_r *correction factor*
ρ *fluid density,* kg/m^3
σ *porosity*

Subscripts

i *inlet*
o *outlet*
c *cold*
h *hot*
min *minimum*
max *maximum*

I. Introduction

The rotary regenerator is a device that receives thermal energy from a hot flow, stores it temporarily in a porous matrix, and releases it to a cold flow. The matrix continuously rotates between the two hot and cold streams that flow in opposite directions. A peculiarity in the operation of rotary regenerators is appointed as fluid carryover leakage. It is an unavoidable carryover of a small fraction of the fluid trapped in the passage to the other fluid stream just after switching of the fluids [1]. This characteristic makes the rotary regenerator suitable for gas-to-gas heat transfer operations with focus in waste heat recovery, often in ventilation and air conditioning systems (HVAC) and thermal power plants. These two areas of application involve different sizes and operating conditions of the equipment. HVAC systems generally require smaller rotary regenerators with faster rotational speeds and thermal power plants demand larger equipment at slower rotational speeds. The matrix rotational speed affects the amount of the heat transferred and the contamination from fluid carryover leakage. Higher rotational speeds lead to larger carryover leakage and lower rotational speeds reduce the heat exchange. Both aspects influence negatively the performance of the equipment.

Analysis of rotary regenerators with attention to the matrix rotation have been conducted by researchers. Some authors investigated the influence of matrix rotational speed on the rotary regenerator effectiveness or heat transfer rate. Ghodsipour and Sadrameli [2] simulated a rotary regenerator by solving a mathematical model and analyzed the effect of dimensionless parameters on the effectiveness of equipment. The maximum effectiveness was estimated as a function of matrix rotational speed and air velocities of streams. Sanaye et al. [3] investigated the optimal operational conditions of an air-to-air rotary regenerator using genetic algorithm. The objective function in the optimization technique was the thermal effectiveness. The matrix rotational speed was used as one of the decision variables. Eljšan et al. [4] presented a mathematical model of a rotary air preheater that allows rotational speed to vary and quantify its influence upon heat transfer in the equipment. The results showed that there is the rotational speed at which the maximum heat transfer is achieved. Rao and Patel [5] and Raja et al. [6] used different algorithms to maximize regenerator effectiveness and minimize regenerator pressure drop. The first authors used the artificial bee colony algorithm for optimization of rotary regenerator and the second researchers investigated a tutorial training and self-learning inspired teaching-learning-based optimization (TS-TLBO) algorithm. In these works, seven and six variables, respectively,were considered for optimization, including the matrix rotational speed. O’Connor et al. [7] studied the effect of the matrix rotation of thermal wheel into a wind tower system on the ventilation rate and heat recovery for a range of rotation speeds between 0-500 rpm. Using computational fluid dynamics analysis, the results showed that the optimum operating range of the rotary thermal wheel could be determined between 5 rpm and 20 rpm. Chen et al. [8] investigated how performance parameters of a rotary heat exchanger such as effectiveness, fluid and metal temperature fields, and ammonium bisulfate deposition area vary with matrix rotational speed using the effectiveness–modified number of transfer units (ε-NTU) method and a finite difference method. They verified that the results calculated by ε-NTU method are greater than those obtained by the finite difference method notably at low matrix rotation. In a few works [9-11], empirical correlations that take into account the effects of rotational speed and matrix heat capacity rate on the regenerator effectiveness were developed.

Other studies that take into account the matrix rotation of rotary regenerators focuses on the analysis of different aspects of the equipment. Zheng at al. [12] showed that the matrix rotational speed is a key control parameter to achieve optimal dehumidifier performance in a desiccant wheel. They also presented results of the effect of isotherm shape, maximum desiccant matrix moisture uptake and number of transfer units (NTU) on the dehumidification performance of the equipment. Buyukalaca and Dogruyol [13] investigated experimentally a rotary regenerator constructed with aluminium plates. The experiments were performed at a range of matrix rotational speed in order to verify the influence of some parameters of the heat exchanger. The results showed that the studied parameters were little affected in the experiments. Wu et al. [14] proposed a numerical algorithm to analyze the influence of variations in rotating speed and other characteristics on dynamic responses of the regenerator. Numerical results were compared with experimental measurements and with theoretical predications of energy efficiencies. Bennhold and Wilson [15] presented a numerical analysis of the thermal behavior of rotary regenerator with the matrix rotating in an indexed fashion. The results indicated significant differences in the patterns of thermal gradient development between the indexing regenerator and the continuously rotating regenerator. Grzebielec et al. [16] showed results from modeling a regenerator that normally does not operate under optimal conditions due to climate changes throughout the year. The authors presented dependence regenerator efficiency as a function of the rotational speed. Ruivo et al. [17] used experimental data measured in an air-handling unit and data provided by the manufacturer software to predict the influence of the matrix rotation speed on the performance of a desiccant wheel. The results revealed that the matrix rotation plays a significant role in affecting the effectiveness parameters of rotary wheel. Duan et al. [18] analyzed theoretically and experimentally the influence of the operational parameters on contact heat transfer between granular materials and heating plates inside a rotary regenerator. It was verified that the time-average contact heat transfer coefficient increases with the increase of rotational speed and that this time-average follows a positive power law with matrix rotation. Abroshan and Goodarzi [19] modeled a rotary regenerator by combining mathematical modelling and CFD simulations. The equipment was optimized using a genetic algorithm and, among other parameters, optimal matrix rotational speed was obtained to maximize the heat transfer rate and minimize the pressure drop in the regenerator from the introduction of a fuel saving parameter. Rotary regenerators impregnated with a hygroscopic material are often used in air dehumidification and enthalpy recovery applications. Systems with these applications were studied focusing on the matrix rotational speed of desiccant wheels. Researchers [20-27] verified the existence of an optimal rotation speed that implies in better performance for dehumidification. Niu and Zhang [28] calculated the optimum rotary speeds for sensible heat recovery, latent heat recovery and air dehumidification in a desiccant wheel. The results showed that the optimum rotary speed for air dehumidification is much lower than those for enthalpy recovery and that the optimum rotary speed is a function of matrix wall thickness.Nóbrega and Brum [29] developed a simple mathematical model to describe the heat and mass transfers in rotary regenerators and introduced dimensionless parameters in the analysis. They concluded that the enthalpy recovery reaches a maximum point and exhibits a great dependence on the nondimensional period of revolution. A group of researchers has studied the influence of matrix rotation of rotary regenerators operating under frosting conditions. Jedlikowski et al. [30-31] evaluated the effect of rotor speed on the formation of frost area accumulation in the rotary regenerators. Among the analyzed cases, they verified that the increase in rotor speed results in an increase in regenerator effectiveness with simultaneous decrease in the size of the frost area accumulation. Smith and Svendsen [32] developed a short plastic rotary regenerator for single-room ventilation. Even though significant leakage, the experimental results showed that the lower rotational speeds decrease heat recovery, which is required to prevent frost accumulation.

Although many researchers have investigated the effects of matrix rotation on different performance parameters of rotary regenerators, studies assessing the fluid carryover leakage as a function of matrix rotational speed are less available in the literature. Some authors have proposed or used methods to investigate the rotary regenerator performance. The methods considered the matrix rotation and incorporated the effects of carryover leakage [33-34] and the effects of carryover leakage combined with pressure leakage or fluid bypass [35-42]. The aforementioned works covering carryover leakage do not present results that indicates the matrix rotation and fluid contamination levels that are suitable for the rotary regenerator to operate close to the optimum conditions. Regarding studies that present results with greater emphasis on the relationship between carryover leakage and matrix rotational speed, Kay and London [9] warned that the design of rotary regenerators requires the provision of matrix rotation level with minimum carryover leakage. De Antonellis et al. [43] informed that the optimal matrix rotational speed exists because the contamination from the fluid carryover leakage becomes relevant when the matrix rotates too fast. The authors just reported that the typical rotational speed is around 10-20 rpm, depending on airflow rates and geometric characteristics of rotary regenerator. Mioralli [44] numerically analyzed a rotary thermal regenerator using finite volume method. Alhusseny and Turan [45] numerically analyzed the equipment using a porous media approach. Both works showed graphically that the contamination from fluid carryover leakage tends to increase with increasing matrix rotation. The results also confirmed that there are no significant changes in rotary heat exchanger effectiveness after a certain rotational speed level. Chung et al. [46] optimized the design parameters of a plastic rotary regenerator considering leakage and adsorption. The results showed that the effect of carryover leakage on the heat transfer effectiveness of rotary regenerator is small from changes in matrix rotational speed.Despite appreciable efforts has been dedicated to rotary regenerators analysis, a recommendation of minimum and maximum limits for rotational speed and carryover leakage as suitable for good performance of equipment apparently is absent. This work computationally analyzes typical rotary regenerators focusing on three aspects simultaneously: matrix rotational speed, carryover leakage and heat transfer rate. The mainly purpose is to choose minimum and maximum limits for matrix rotation that imply good heat transfer rate and acceptable contamination from fluid carryover leakage. Furthermore, ranges of a parameter that describes the propagation of carryover leakage also are chosen and an analysis of the regenerator effectiveness as a function of the time required for a complete matrix rotation changes is presented.

II. Problem Setting

A. Rotary Regenerator Features

The schematic of the rotary regenerator is show in Fig. 1. Two gas streams are introducedcounterflow-wise through the parallel ducts of the air preheater. Cold gas is injected inside one duct and hot gas inside the other. The porous matrix, that stores energy, continuously rotates through these parallel ducts. The matrix receives heat from the hot gas on one side and transfers this energy to the cold gas on the other side. The matrix channels were assumed smooth. The fluid velocity was considered constant inside each channel.

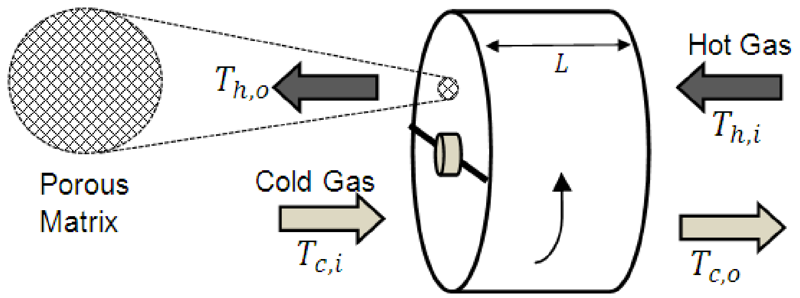

Fig. 1. Schematic of the rotary regenerator.

The actual time that cold gas takes to flow through a hot regenerator matrix is called the cold period and the time that hot gas flows through the cold regenerator matrix is called the hot period. The cold and hot periods are dependent on the matrix rotation. Fig. 2 illustrates a fixed point in the matrix of the regenerator, represented by a channel, which runs through the sides of the cold and hot gas streams during one revolution of the matrix. The parameters t_c e t_h represent the cold and hot periods, respectively.

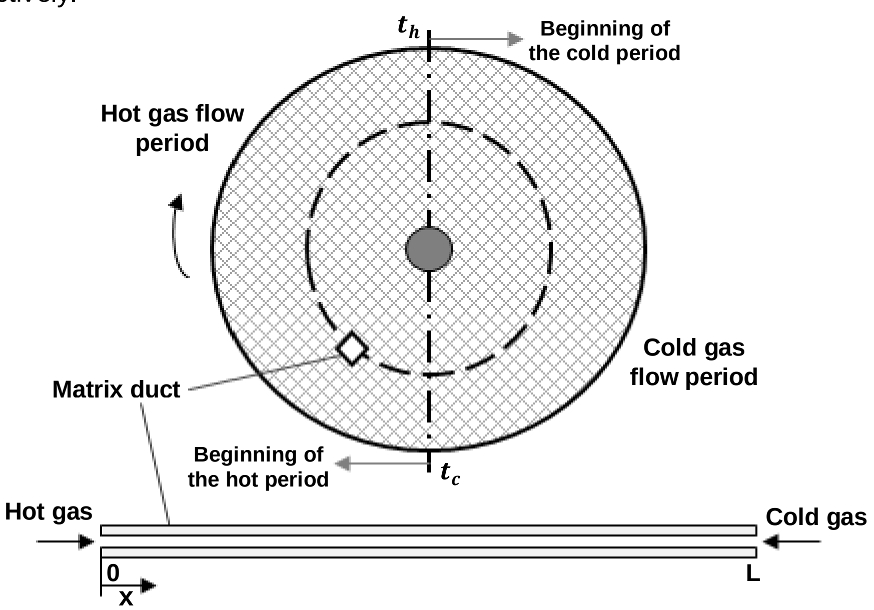

Fig. 2. Cold and hot periods of the rotary regenerator.

Contamination between the fluids streams occurs when an amount of one of the fluids in the matrix passages is lugged from one fluid stream to the other by the rotation of the matrix [37]. For negligible contamination, the time it takes the gas particles to travel through the matrix passages must be sufficiently less than the rotation time of the matrix. Fig. 3 illustrates the contamination from the fluid carryover leakage.

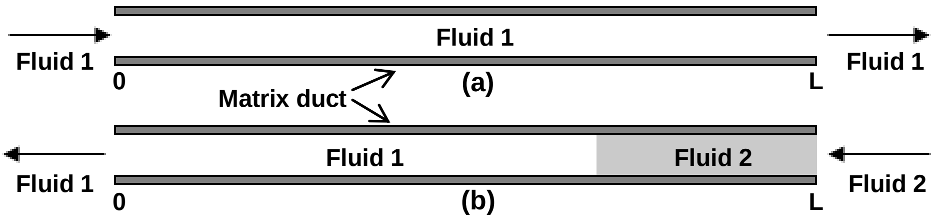

Figure 3. Contamination from the fluid carryover leakage: (a) The end of period 1, (b) The beginning of period 2.

The time of fluid particle passing through the matrix ducts is called the residence time t_{res} defined as the ratio of the length L of the matrix ducts to the bulk stream velocity u in ducts

$$t_{res} = L/u \tag{1}$$

To describe the propagation of carryover leakage, Banks [34] introduced a parameter defined as the ratio of residence time t_{res} of flow on each side to the time t_0 required for a complete matrix rotation.Analysis of typical rotary regenerators operating close to optimal conditions showed that this ratio must be up to 1.5% for negligible contamination between fluids [44], that is

$$\left(t_{res}/t_0\right) \leq 0.015 \tag{2}$$

Geometric parameters of the rotary regenerator can be expressed based on Figs. 1 and 2. The total frontal cross-sectional area A_T is determined by the sum of the free flow cross-sectional area A and the matrix cross-sectional area A_m of the air preheater

$$A_T = A + A_m \tag{3}$$

The matrix porosity σ is defined by the ratio between A and A_T

$$\sigma = \frac{A}{A_T} \tag{4}$$

The hydraulic radius r_h is defined by the ratio between A and the perimeter P of the plates that compose the matrix. The matrix perimeter can be written as function of the matrix cross-sectional area A_m

$$r_h = \frac{D_h}{4} = \frac{A}{P} \tag{5}$$

$$P = \frac{A_m}{(e/2)} \tag{6}$$

where D_h and e are the matrix duct hydraulic diameter and the matrix duct wall thickness, respectively.

The porosity and the hydraulic radius are dependent on each other and influence the thermal exchange in the rotary regenerator. The hydraulic radius can be written as function of the porosity and the matrix duct wall thickness from the definitions above and algebraic manipulations

$$r_h = \frac{\sigma}{1-\sigma}\left(\frac{e}{2}\right) \tag{7}$$

The hydraulic radius is an important parameter and its use is justified in the correlation for Nusselt number. Since the geometric characteristics of the regenerator are known, the heat transfer in the equipment can be calculated using the Effectiveness-NTU method for rotary regenerators.

B. Effectiveness-NTU Method for Rotary Regenerators

The Effectiveness-NTU method for rotary regenerators [9] consists of calculating the effectiveness ε_0 of a conventional counterflow heat exchanger and correcting this effectiveness by a correction factor φ_r that takes into account the rotational speed and the matrix heat capacity rate of the exchanger. Thus, the effectiveness of the regenerator ε_r is given by

$$\varepsilon_r = \varepsilon_0 \, \varphi_r \tag{8}$$

The effectiveness ε_0 of a conventional counterflow heat exchanger is defined by

$$\varepsilon_0 = \begin{cases} \dfrac{1-\exp\left[-NTU\left(1-C^*\right)\right]}{1-C^*\exp\left[-NTU\left(1-C^*\right)\right]}; & C^* < 1 \\ \dfrac{NTU}{1+NTU}; & C^* = 1 \end{cases} \tag{9}$$

where C^* is the ratio between the fluids heat capacity rates and NTU is the number of heat transfer units defined as follows

$$C^* = \frac{C_{mim}}{C_{max}} \tag{10}$$

$$NTU = \frac{1}{C_{min}}\left[\frac{1}{\left(1/hA_{tr}\right)_c + \left(1/hA_{tr}\right)_h}\right] \tag{11}$$

where h is the convective heat transfer coefficient and A_{tr} is the matrix thermal exchange area on the side of the hot or cold stream. The parameters C_{mim} and C_{max} correspond to the minimum and maximum values of the fluids heat capacity rates.

The correction factor φ_r in Eq. (8) is given by Buyukalaca and Yilmaz, [11] that provides good results even for very low rotational speeds

$$\varphi_r = \frac{1}{\left[1+3\left(\varepsilon_0/C_r^*\right)^2 + \left(\varepsilon_0/C_r^*\right)^4\right]^{1/4}} \tag{12}$$

$$C_r^* = \frac{C_r}{C_{min}} \tag{13}$$

$$C_r = \frac{n}{60} m_m \, c_m \tag{14}$$

where C_r is the matrix heat capacity rate, n is the matrix rotational speed, m_m is the matrix mass and c_m is the specific heat of matrix.

Finally, the total heat transfer Q in the rotary regenerator is obtained in the same way as the Effectiveness-NTU method for conventional heat exchangers

$$Q = \varepsilon_r \, Q_{max} \tag{15}$$

$$Q_{max} = C_{min}\left(T_{h,i} - T_{c,i}\right) \tag{16}$$

where Q_{max} is the maximum possible heat transfer and the term between parenthesis corresponds to the difference between the inlet temperature of the hot stream and the inlet temperature of the cold stream.

C. Hydrodynamic and Thermal Analysis

The hydrodynamic and thermal analysis are performed for each gas stream. The mass flow rate and the inlet temperatures of each gas stream are established in the rotary regenerator. The convective heat transfer coefficient is obtained from correlation for Nusseltnumber Nu. Correlation for smooth ducts with circular cross-sectional area was used based on the hydraulic diameter of matrix ducts considering laminar flow regime. The correlation take into account hydrodynamically fully developed flow with thermal entrance length and constant wall temperature boundary condition.

$$Nu = 3.66 + \frac{0.0668\left(\frac{D_h}{L}\right)Re_{D_h}Pr}{1+0.04\left[\left(\frac{D_h}{L}\right)Re_{D_h}Pr\right]^{\frac{2}{3}}} \tag{17}$$

where Re_{D_h} is the Reynolds number and Pr is the Prandtl number.

The convective heat transfer coefficient h is expressed in terms of Nusselt number

$$h = \frac{Nu\,k}{D_h} \tag{18}$$

where k is the fluid thermal conductivity.

D. Fluid and Matrix Properties

The fluid properties were obtained at the average temperature of each gas stream. The fluid density for gases with moderate values of pressure and temperature is well represented by the equation of state of an ideal gas

$$\rho = \frac{p}{RT} \tag{19}$$

where p is the pressure of fluid, T is the average temperature of gas stream and R is the ideal gas constant. The values of air atmospheric pressure $p = 10^5\,Pa$ and ideal gas constant for air $R = 287\ Nm/kgK$ were assumed.

The dynamic viscosity μ and the thermal conductivity k of fluids can be approximated by the Sutherland equations [47] as follows

$$\frac{\mu}{\mu_0} \approx \left(\frac{T}{T_0}\right)^{3/2}\frac{T_0+S}{T+S} \tag{20}$$

$$\frac{k}{k_0} \approx \left(\frac{T}{T_0}\right)^{3/2}\frac{T_0+S}{T+S} \tag{21}$$

where S is the Sutherland constant temperature, which is characteristic of each gas. Considering air $S = 111\,K$ for dynamic viscosity and $S = 194\,K$ for thermal conductivity. The parameters T_0, μ_0 and k_0 are reference constants whose values are $T_0 = 273\,K$, $\mu_0 = 1.716\cdot 10^{-5}\,Pa\cdot s$ and $k_0 = 0.0241\ W/mK$ for air.

The specific heat of gas under constant pressure c_p is obtained by a polynomial equation [48] for several gases in the temperature range between 300 and 1,000 K

$$\frac{c_p}{R} = \alpha_0 + \beta_0 T + \gamma_0 T^2 + \delta_0 T^3 + \lambda_0 T^4 \tag{22}$$

where $\alpha_0 = 3.653$, $\beta_0 = -1.337\cdot 10^{-3}$, $\gamma_0 = 3.294\cdot 10^{-6}$, $\delta_0 = -1.913\cdot 10^{-9}$ and $\lambda_0 = 0.2763\cdot 10^{-12}$ are the air constants.

The Prandtl number Pr is obtained from the ratio between some fluid properties, as follow

$$Pr = \frac{\mu\,c_p}{k} \tag{23}$$

The matrix properties of the regenerative air preheater were assumed constant. The AISI 1010 low alloy carbon steel and the 2024-T6 aluminum alloy materials were considered for the matrix. Table 1 shows the matrix properties used in this study, where ρ_m is the density of matrix.

Table 1. Matrix properties of the regenerative air preheater.

Material	$c_m \ (J/kg\ K)$	$\rho_m \ (kg/m^3)$
2024-T6 aluminum	875	2,770
AISI 1010 alloy carbon steel	434	7,832

E. Computer Program

A computer program written in *C* programming language was developed for the simulation of rotary regenerator. The *Dev-C++* software was used for compilation and recording results. Three typical sizes of equipment were simulated: small, medium-sized and large. The material AISI 1010 low alloy carbon steel was used for the medium-sized and the large heat exchangers in the simulations. The 2024-T6 aluminum alloy was used for the small regenerator. The total heat transfer in the rotary regenerator and the outlet temperatures of gas streams were calculated for different matrix rotation from the prescribed mass flow rate and inlet temperatures of each gas stream. The other geometric parameters of the equipment were fixed.

An iterative process was used to obtain the fluid flow and the heat transfer. An outlet temperature values of each stream was assumed at the beginning of this process. Then, the fluid properties were evaluated at the average temperature of each gas stream. Based on these properties, the fluid flow and the heat transfer were obtained from correlations and the Effectiveness-NTU method for rotary regenerators. The iterative process continued until convergence of the outlet temperatures for both streams. The whole process was repeated for each assumed rotation value of matrix. The subrelaxation factor 0.5 was used to the convergence of the outlet temperature values. The tolerance for convergence iterative procedure was adjusted as 10^{-3} for the outlet temperatures. The calculations were performed considering the steady-periodic condition of the regenerator, indicating that the temperatures no longer changed in any angular or axial position of the matrix. The schematic diagram of the calculation process is shown in Fig. 4.

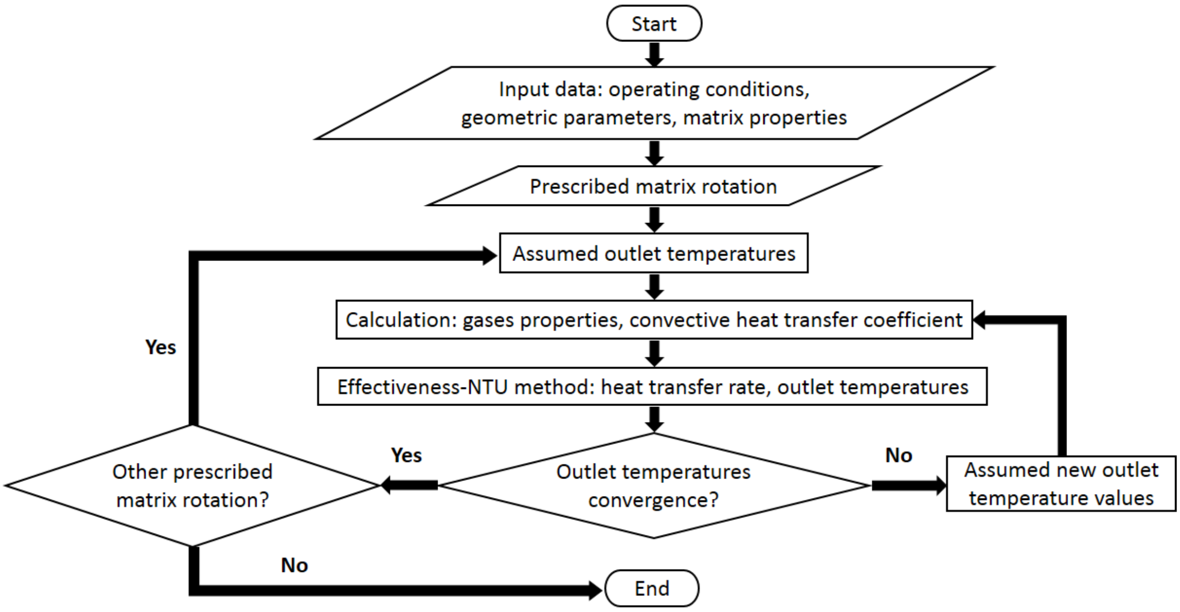

Fig. 4. Schematic diagram of the calculation process.

In order to check the reliability of the developed computer program, the outlet temperatures of gas streams were calculated at a medium-sized rotary regenerator with corrugated ducts. Correlations for corrugated ducts [49] were used for the hydrodynamic and thermal analysis in this case. The results were compared with field data from a rotary regenerator operating in a petroleum refinery. Table 2 shows the comparison between the results of the present study and the field data. It is observed that the results are in reasonable agreement with a greater difference for the hot outlet temperature values. Since the pressure leakage and fluid bypass of rotary regenerator are not considered in this study, this difference could be minimized by including these parameters.

Table 2. Comparison of the present data with field data.

Outlet temperature (°C)	Present work	Field data	Difference
$T_{c,o}$	433.31	405.65	0.068
$T_{h,o}$	164.95	194.27	0.151

III. RESULTS AND DISCUSSION

The input data of the developed computer program are listed in Table 3. The operating conditions of the rotary regenerators are based on information from literature and industry. The simulations were carried out from different matrix rotation values and considering the gas streams under the laminar flow regime.

Table 3. Input data for computer program of typical rotary regenerators.

Rotary regenerator	L (m)	e (m)	D (m)	σ	Inlet Temp. (°C)		Flow Rate (kg/s)	
					$T_{h,i}$	$T_{c,i}$	$\dot{m}_h$	$\dot{m}_c$
Small	0.2	0.00035	0.7	0.83	50	20	0.68	0.76
Medium-sized	1.5	0.00050	6.0	0.90	450	80	39.00	49.00
Large	3.5	0.00060	15.0	0.90	600	150	260.00	292.00

Fig. 5 shows the total heat transfer as a function of the matrix rotation for the small rotary regenerator. The total heat transfer changes significantly up to matrix rotation values close to $n = 6.0\,rpm$. As a comparison, the heat transfer rate is $Q \cong 1.5\,kW$ for the matrix rotation $n = 0.1\,rpm$ while $Q \cong 9.6\,kW$ for $n = 6.0\,rpm$. The increase in the heat transfer rate in this case is close to 540%. The heat transfer rate does not change significantly to matrix rotation values greater than $n = 6.0\,rpm$ .Thus, based on Fig. 5, the matrix rotation values $n \geq 6.0\,rpm$ provide high heat transfer rate in the small rotary regenerator. However, this result does not take into account the contamination from the fluid carryover leakage. Fig. 6 shows the ratio (t_{res}/t_0) of both gas streams as a function of the matrix rotation for the small rotary regenerator.Based on the ratio $(t_{res}/t_0) \leq 0.015$, (2), Fig. 6 demonstrates that the matrix rotation values must be less than $n = 17.5\,rpm$ for negligible contamination between fluids in the small rotary regenerator. Finally, the simultaneous analysis of the results observed in Figs. 5 and 6 indicates that the range $6.0 \leq n \leq 17.5\,rpm$ of matrix rotation values can be chosen as suitable for good heat transfer rate and acceptable fluid carryover leakage in the small rotary regenerator.

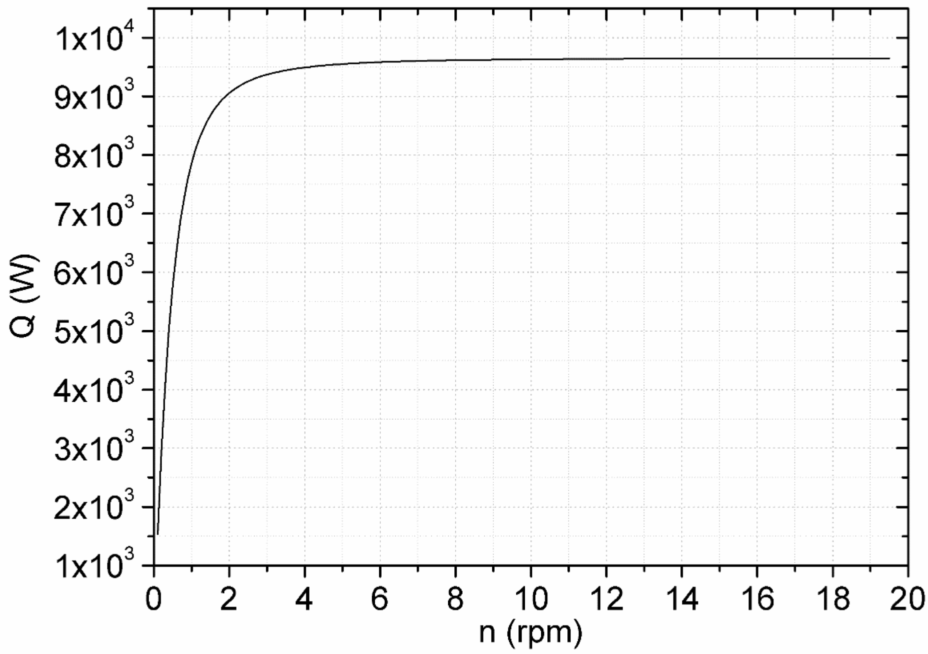

Fig. 5. Heat transfer versus matrix rotation for small rotary regenerator.

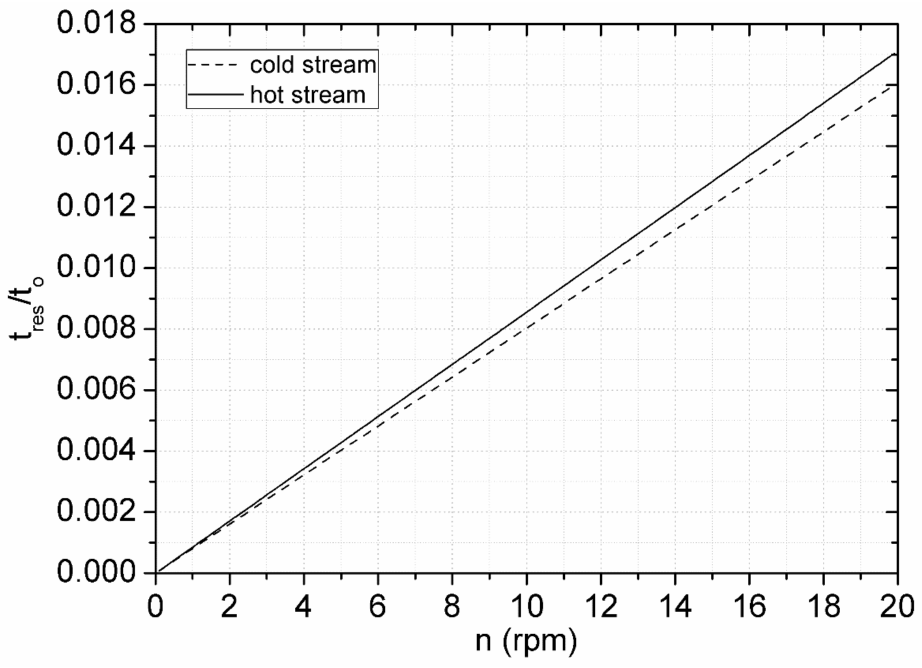

Fig. 6. Contamination from fluid carryover leakage versus matrix rotation for small rotary regenerator.

Fig. 7 shows the total heat transfer as a function of the matrix rotation for the medium-sized rotary regenerator. The total heat transfer changes significantly upto matrix rotation values close to $n = 2.0\,rpm$. It is observed that the heat transfer rate is $Q \cong 5.8\,MW$ for the matrix rotation $n = 0.1\,rpm$ while $Q \cong 9.04\,MW$ for $n = 2.0\,rpm$. In this case, the increase in the heat transfer rate is close to 56%. The heat transfer rate does not change significantly to matrix rotation values greater than $n = 2.0\,rpm$ .Hence, based on Fig. 7, the matrix rotation values $n \geq 2.0\,rpm$ provide high heat transfer rate in the medium-sized rotary regenerator. Taking into account the contamination from the fluid carryover leakage in the medium-sized rotary regenerator, Fig. 8 shows the ratio $\left(t_{res}/t_0\right)$ of both gas streams as a function of the matrix rotation. Note that the matrix rotation values must be less than $n = 3.1\,rpm$ for negligible contamination between fluids in Fig. 8, according to (2). Thus, the simultaneous analysis of the results observed in Figs. 7 and 8 shows that the range $2.0 \leq n \leq 3.1\,rpm$ of matrix rotation values can be chosen as suitable for good heat transfer rate and acceptable fluid carryover leakage in the medium-sized rotary regenerator.

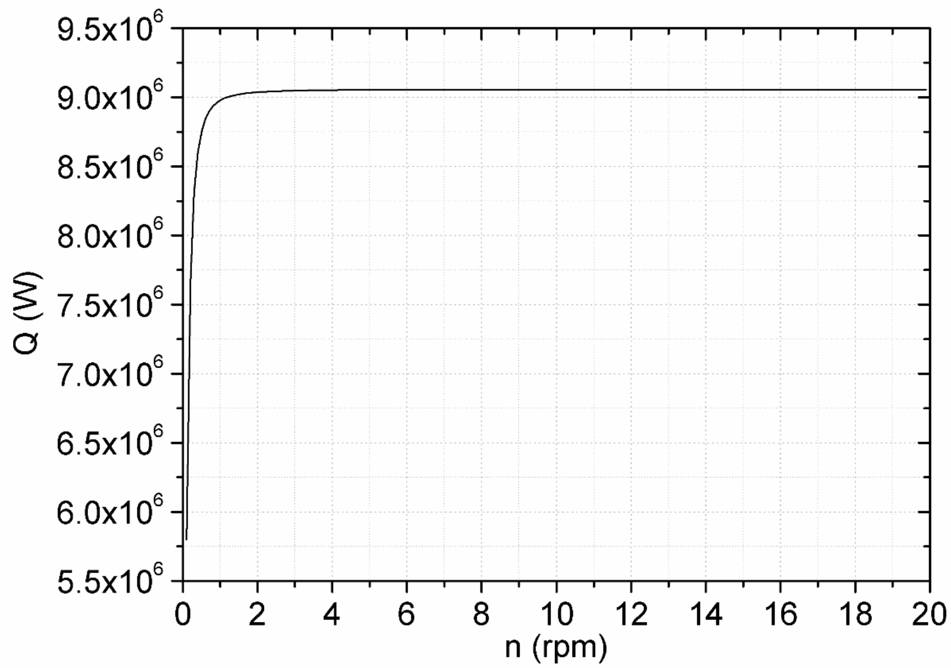

Fig. 7. Heat transfer versus matrix rotation for medium-sized rotary regenerator.

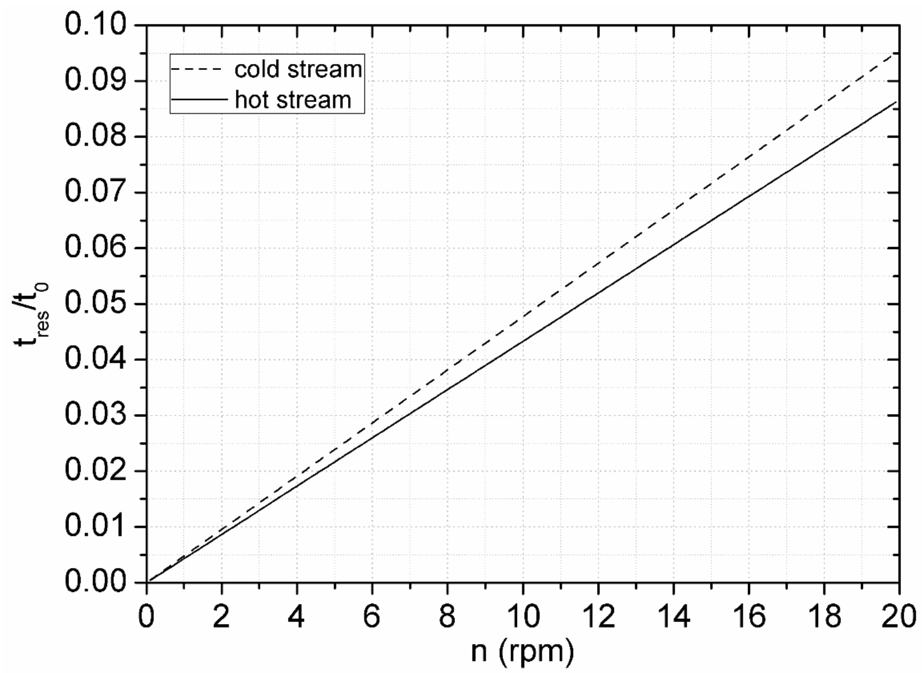

Fig. 8. Contamination from fluid carryover leakage versus matrix rotation for medium-sized rotary regenerator.

Similarly to the analyzed cases for the small and medium-sized rotary regenerators, Fig. 9 shows the total heat transfer as a function of the matrix rotation for the large rotary regenerator. The total heat transfer changes significantly up to matrix rotation values close to $n = 1.4\,rpm$. In this case, the heat transfer rate is $Q \cong 0.07\,GW$ for the matrix rotation $n = 0.1\,rpm$ while $Q \cong 0.084\,GW$ for $n = 1.4\,rpm$, corresponding to an increase in the heat transfer rate close to 20%. The heat transfer rate does not change significantly to matrix rotation values greater than $n = 1.4\,rpm$.Thus, based on Fig. 9, the matrix rotation values $n \geq 1.4\,rpm$ provide high heat transfer rate in the large rotary regenerator. Taking into account the contamination from the fluid carryover leakage in the large rotary regenerator, Fig. 10 shows the ratio (t_{res}/t_0) of both gas streams as a function of the matrix rotation. Considering (2), itis observed that the matrix rotation values must be less than $n = 1.7\,rpm$for negligible contamination between fluids. The simultaneous analysis of the results observed in the Figs. 9 and 10 shows that the range $1.4 \le n \le 1.7\,rpm$ of matrix rotation values can be chosen as suitable for good heat transfer rate and acceptable fluid carryover leakage in the large rotary regenerator.

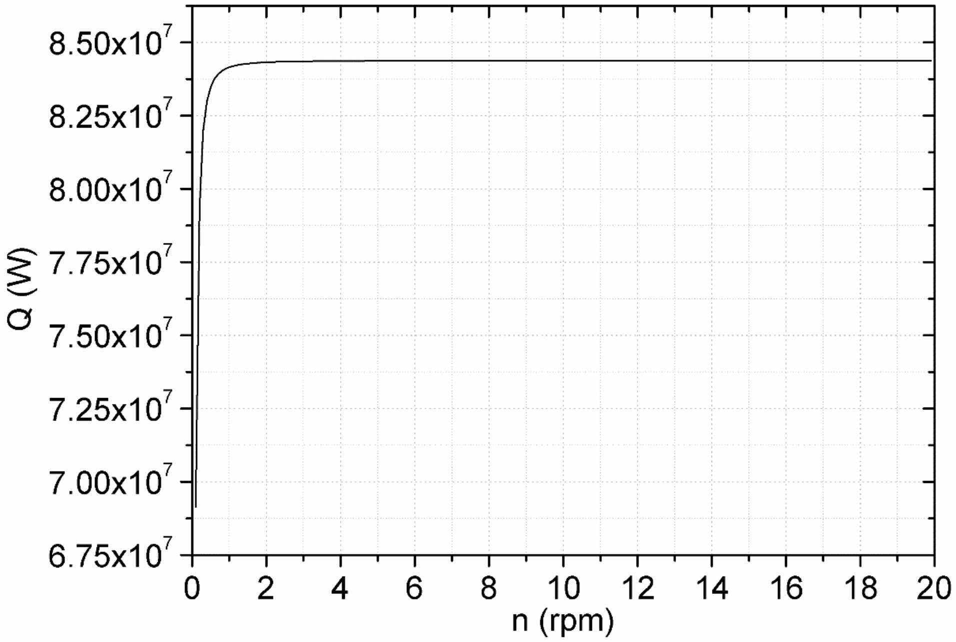

Fig. 9. Heat transfer versus matrix rotation for large rotary regenerator.

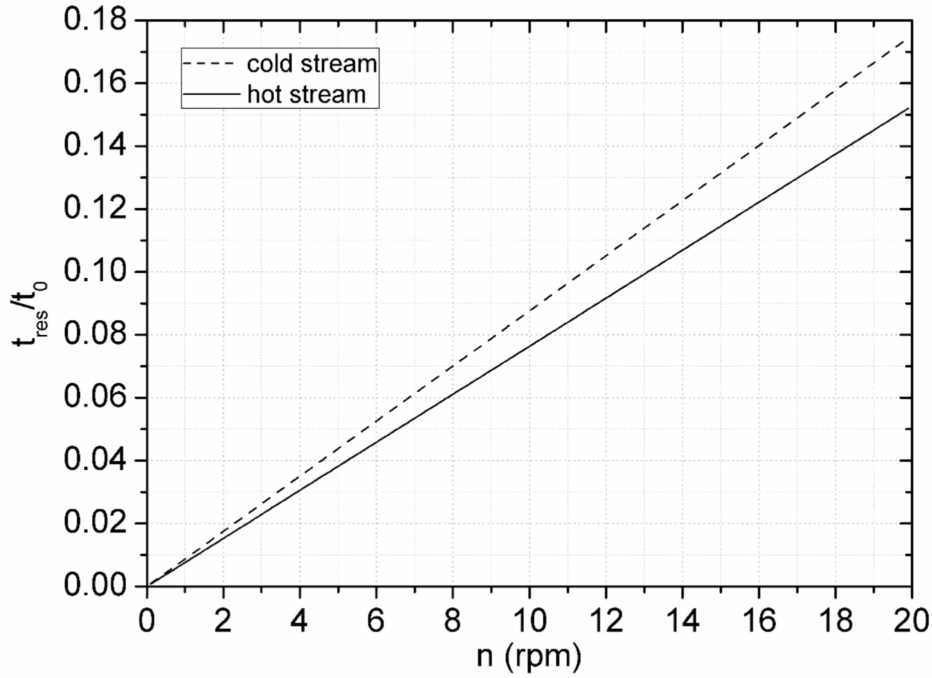

Fig. 10. Contamination from fluid carryover leakage versus matrix rotation for large rotary regenerator.

The analysis of the Figs. 5 to 10 reveals that, when good heat transfer rate and acceptable contamination between fluids are required, the chosen matrix rotation ranges shorten as the size of the rotary regenerator increases. Additionally, these matrix rotation ranges move to the left on the abscissa axis (matrix rotation) as the dimensions and typical operating conditions of the rotary regenerators increase. Another feature is that the chosen matrix rotation range is narrow for the medium-sized and large rotary regenerators whereas it is wide for de small rotary regenerator. The contamination from the fluid carryover leakage in the small equipment is less susceptible as the matrix rotation changes.

The effectiveness of the typical rotary regenerators can be analyzed as a function of the ratio (t_{res}/t_0). The effectiveness of the three rotary regenerators is showed in Fig. 11 as a function of the ratio (t_{res}/t_0) of each gas stream. It is observed that there is a minimum level for this ratio after that the regenerator effectiveness does not change significantly. Since that the $(t_{res}/t_0)\le 0.015$, (2), for negligible contamination between fluids, a range of (t_{res}/t_0) can be chosen as suitable for good effectiveness and acceptable fluid carryover leakage in the rotary regenerators. Based on Fig. 11, the ranges $0.007\le(t_{res}/t_0)\le 0.015$, $0.008\le(t_{res}/t_0)\le 0.015$ and $0.009\le(t_{res}/t_0)\le 0.015$ for the small, medium-sized and large rotary regenerator, respectively, can be chosen to ensure high effectiveness and acceptable fluid carryover leakage. These ranges are valid for both gas streams of each equipment. Fig. 11 also shows that the maximum effectiveness values of each regenerator are close to $\varepsilon_r = 0.47$, $\varepsilon_r = 0.59$ and $\varepsilon_r = 0.67$ for the small, medium-sized and large rotary regenerator. The maximum effectiveness values for each rotary regenerator are associated with the pre-established operating conditions shown in Table 3.

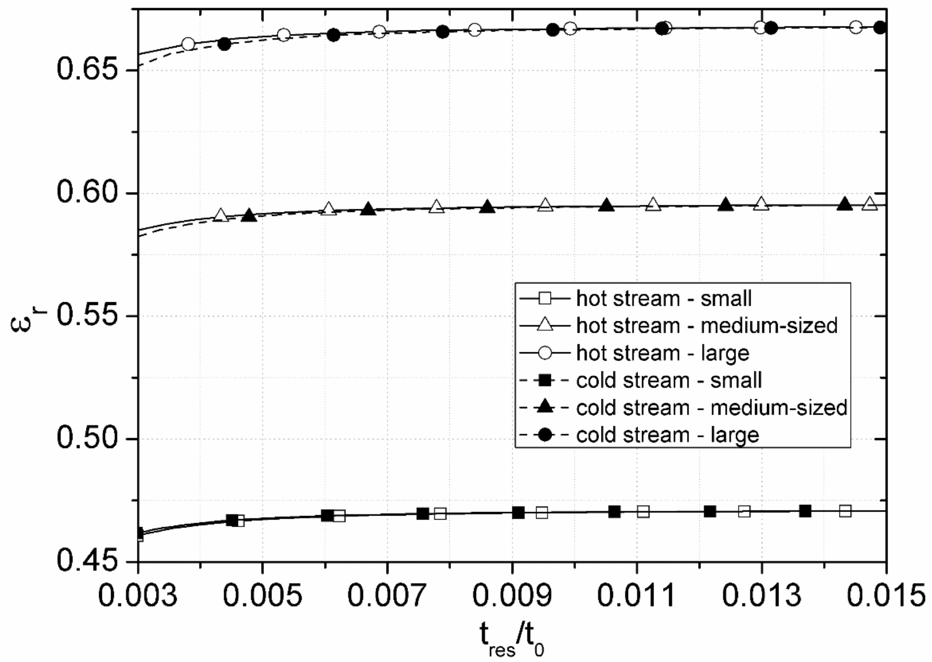

Fig. 11. Regenerator effectiveness versus contamination from fluid carryover leakage.

Fig. 12 shows the effectiveness of the rotary regenerators as a function of the time required for a complete matrix rotation. The effectiveness decreases with increasing time for one matrix revolution in all cases. The decrease in effectiveness has a linear behavior for high t_0 values, which correspond to small matrix rotational speed. The linear behavior can also be seen for small matrix rotation in Figs. 5, 7 and 9. Fig. 12 shows that the decrease in effectiveness is more pronounced for the small regenerator, notably in the interval $10 < t_0 \le 200\ s$. The effectiveness remains high for a wider range of t_0 in the large regenerator. This range decrease as the size of the regenerator decreases. The effectiveness are high at $t_0 \le 10\ s$, $t_0 \le 30\ s$ and $t_0 \le 45\ s$ for the small, medium-sized and large rotary regenerator, respectively. However, there is a minimum t_0 value, in each simulated case, after that the fluid carryover leakage is acceptable. These values are close to $t_0 \cong 3.4\ s$, $t_0 \cong 19.4\ s$ and $t_0 \cong 35.3\ s$ for the small, medium-sized and large regenerator and theyare associated to the maximum matrix rotation values chosen for acceptable contamination between fluids, as shown in Figs. 6, 8 and 10. Finally, a range of t_0 values can be selected as suitable for good heat transfer rate and acceptable fluid carryover in the simulated typical rotary regenerators. These ranges are $3.4\le t_0 \le 10.0\ s$, $19.4\le t_0 \le 30.0\ s$ and $35.3\le t_0 \le 45.0\ s$ for the small, medium-sized and large rotary regenerator, respectively.

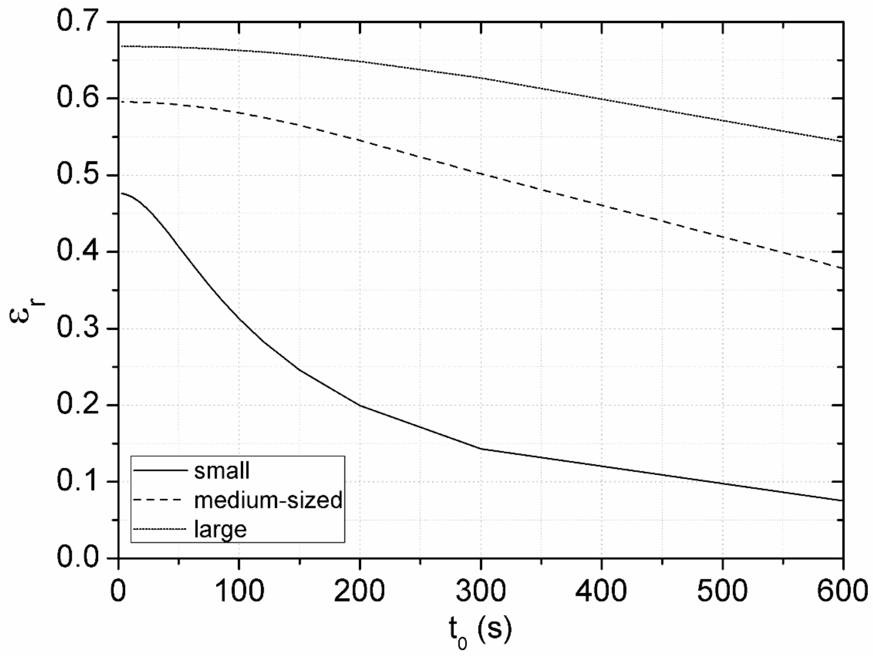

Fig. 12. Regenerator effectiveness versus the time required for a complete matrix rotation.

IV. Conclusion

Typical rotary regenerators were computationally analyzed from the pre-established mass flow rate and inlet temperatures for each gas stream of the equipment and different matrix rotation. The conclusions can be summarized as follows:

- A matrix rotation range that provide good heat transfer rate and acceptable fluid carryoverwas chosen for each simulated typical rotary regenerator.
- The chosen matrix rotation ranges shorten as the size of the rotary regenerator increases. The chosen matrix rotation range is narrow for the large rotary regenerator whereas it is wide for de small rotary regenerator.
- The limit values of the chosen matrix rotation ranges decrease as the dimensions and typical operating conditions of the rotary regenerators increase.
- The propagation of fluid carryover was investigated in the rotary regenerators. A range of the ratio (t_{res}/t_0) for high effectiveness and acceptable fluid carryoverwas also defined for each regenerator.
- Additionally, the range of the time required for a complete matrix rotation that provides high effectiveness and acceptable fluid carryover,was showed for each typical rotary regenerator.
- The decrease in effectiveness is more pronounced in the small regenerator as the time required for a complete matrix rotation increases.
- The results can be used in the prescription of operating conditions of rotary regenerators in search of better performance.

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