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A POROUS MEDIA APPROACH FOR NUMERICAL OPTIMISATION OF THERMAL WHEEL

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ABSTRACT

The experimental investigations of rotating heat exchangers are usually too costly and provide limited understanding for the phenomena of heat and fluid flow within them; hence, a less expensive and more comprehensive method is required to investigate what can affect their overall performance. In the current study, a porous media concept is presented as an alternative way to numerically analyse the fluid flow and heat transport through a rotary thermal regenerator. An aluminum core formed of multi-packed square passages is simulated as a porous medium of an orthotropic porosity in order to allow the counter-flowing streams to flow in a way similar to that inside the regenerator core. The geometric properties of the core were transformed into the conventional porous media parameters such as the permeability and inertial

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coefficient based on empirical equations; so, the core has been dealt with as a porous medium of known features. Fluid and solid phases are assumed to be in a local thermal non-equilibrium state with each other. A commercial CFD code "STAR CCM+" was used to solve the current problem numerically, where heat is allowed to be exchanged between the two phases and tracked by creating a heat exchanger interface in the core region. The results are presented by means of overall thermal effectiveness, pressure drop, and coefficient of performance COP. Using porous media approach has been found to be sufficient to simulate the current problem, where the currently computed data were found to deviate by up to 2.7% only from the corresponding analytical and experimental data. The data obtained reveal an obvious impact of the core geometrical parameters on both the heat restored and pressure loss; and hence, the overall efficiency of the regenerator system.

KEYWORDS

Porous media approach, Rotating Regenerator, Overall performance, CFD analysis, Heat exchanger optimisation.

1. INTRODUCTION

Improving performance of power generation plants while reducing the environmental damage caused by using fossil fuels remains of great concern. For achieving a high thermal efficiency in gas turbine systems, there are mainly two options available. The first is via applying a high compression ratio, which is accompanied by an intercooling process but requires higher fuel consumption. The second option is by applying a low compression ratio combined with recirculating heat between the cold stream and exhaust gas to recover a part of its thermal energy instead of releasing it directly to the environment.

Heat recirculation is usually accomplished by means of two alternatives. A recuperator, in which heat exchange between the cold and hot streams takes place across a separating wall between them. The other proposal is the regenerator, which allows heat exchange between the two streams through a common solid surface exposed to the hot and cold streams alternatively, (Organ, 1997). Switching between the two streams can be achieved by several means. One of them, which is known as the “thermal wheel” or “rotary regenerator”, is composed of a permeable core that rotates between two fixed channels carrying hot and cold gases as shown in Fig. 1, (Organ, 1997). The two streams flow continuously in opposite directions through the core, which has an orthotropic porosity to allow both gases to flow simultaneously in separate portions. This configuration was examined experimentally by (Iwai et al., 1987) and later by (Sayama and Morishita, 1993) in the context of applying gas turbines into vehicular propulsion systems.

Regarding air-conditioning applications, rotary regenerators have distinct advantages over ordinary recuperators in recovering heat from exhaust air (Worsϕe-Schmidt, 1991). They have a substantially larger and comparatively less costly heat transfer area. In general, their efficiency is relatively higher, and hence, they have the potential to combine both compactness and high performance (Skiepko, 1989). Also, the amount of heat transported can be regulated by means of adjusting the speed of rotation. A numerical, analytical and experimental study for the impact of rotation rate on the effectiveness of rotary regenerators was presented by (Büyükalaca and Yilmaz, 2002). It was found that thermal effectiveness is enhanced sharply with increasing the rotation level if the heat capacity of the rotating solid core and its rotational speed are both small enough. A mathematical procedure for evaluating the thermal effectiveness of rotary regenerators was proposed by (Yilmaz and Büyükalaca, 2003). The model presented can be applied for different geometrical shapes of channels that form the core of a rotary regenerator. Also, it is valid for a wide range of rotation levels because the influence of rotational speed was taken into account in calculating thermal effectiveness values.

Furthermore, rotary regenerators are used in steam power plants, where their influence on the overall efficiency of the system was investigated experimentally and numerically by (Alagić *et al.*, 2005) and then numerically by (Eljšan *et al.*, 2013). It was found that the regenerator performance relies on its rotation speed, where within a limited range of angular velocity values, its effectiveness is promoted considerably with increasing the rotation rate.

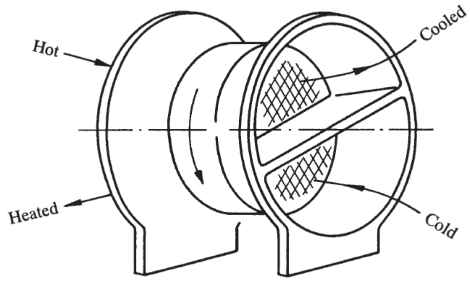

Fig. 1. Operation principle of rotary regenerator, (Organ, 1997)

Phenomena of heat and fluid flow in microstructure systems have gained extensive attention due to their wide applications in micro-mechanical systems such as fuel cells, high-performance cooling devices and heat exchangers, and chemical industries, to name a few, (Kim and Hyun, 2005). Therefore, it is highly desired to use an accurate method in modeling convective flow through microstructure systems, but simple enough to avoid high costs required for traditional experimental or numerical analyses. A modelling way called "porous medium approach" was presented as a candidate capable to address the aforementioned concerns, which is based on a volume averaging method for a microstructural heat transfer device and dealing with it as a fluid-saturated porous medium. This approach was originally evolved out by Koh and Colony (Koh and Colony, 1986) and then has been employed in different thermal aspects such as analysing heat sinks formed of either microchannels [(Kim and Hyun, 2005), (Zhao and Lu, 2002), (Chen, 2007), (Lim, Abdullah and Ahmed, 2010)] or pin-fins [(Jeng and Tzeng, 2005), (Jeng, 2008), (Feng, Kim and Lu, 2013)] in addition to studying the internally finned tubes [(Kim, Yoo and Jang, 2002), (Do, Min and Kim, 2007)].

The aforementioned studies pointed out that this methodology can reasonably describe the convective flow in microstructure systems. This is why the current paper aims to analyse the rotary regenerator numerically by means of this approach, which “to the best of the authors' knowledge” has not been addressed before.

2. MATHEMATICAL MODELLING

2.1. Problem description

A schematic diagram for the studied problem is shown in Fig. 2, where two identical channels carry two separated streams connected thermally in an indirect way by means of a rotating porous core called the regenerator. The parts of both channels located behind the regenerator are considered long enough to establish the condition of fully developed convective flow at the outlet of each one of them, while both the flow and temperature fields are assumed to be uniform at the inlet sections. The regenerator zone, which is formed of multi-packed square aluminum passages, has been dealt with as a permeable medium of orthotropic porosity to allow both streams to flow unidirectionally and separately from each other.

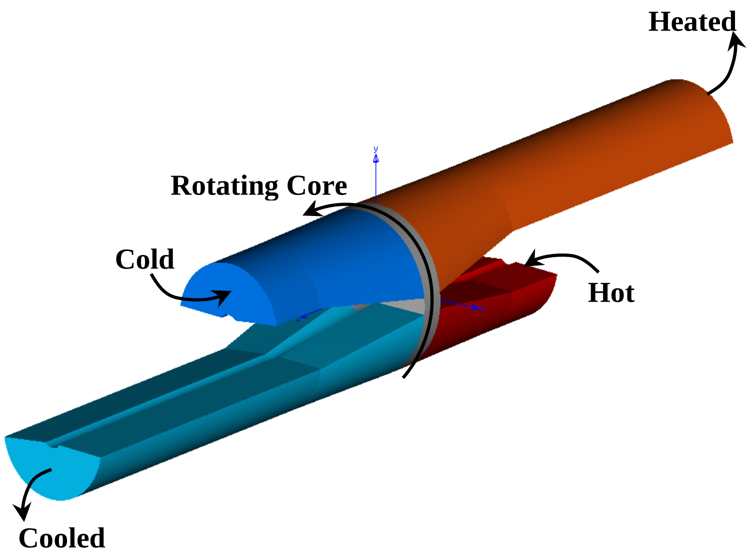

Fig. 2. Geometrical shape of the studied problem

2.2. Assumptions and governing equations

The characteristic diameter of the flow passages used in the rotary regenerators is usually small to provide compactness. This usually leads to laminar flow within the heat exchanger channels (Yilmaz and Büyükalaca, 2003), while the flow in the remaining zones is turbulent. All lateral walls of the channels as well as the regenerator are impermeable and thermally insulated, while heat is only allowed to transport between the two streams via the rotating solid core. The generalised model is used to simulate the momentum equations within the porous matrix. For tracking heat transport, the two-equations model is used, which is indispensable due to the significant difference in thermal conductivity between the solid and fluid phase, (Kim and Hyun, 2005). The working fluid of both streams is air; so, it is possible to neglect the thermal dispersion due to its low effect compared to the relatively high thermal conductivity of aluminum matrix as proposed by (Calmidi and Mahajan, 2000).

Thus, the unsteady dimensional forms of the equations that govern the conservation of mass, momentum, and energy in each of fluid- and solid-phase are respectively as follows:

$$\frac{\partial \rho_f}{\partial t} + \nabla \cdot (\rho_f \boldsymbol{v}) = 0 \tag{1}$$

$$\frac{\rho_f}{\phi}\left[\frac{\partial \boldsymbol{v}}{\partial t} + \frac{1}{\phi}(\boldsymbol{v} \cdot \nabla)\boldsymbol{v}\right] = -\nabla p + \frac{\mu_f}{\phi}\nabla^2 \boldsymbol{v} - \gamma\left(\frac{\mu_f}{K}\boldsymbol{v} + \rho_f \frac{F}{\sqrt{K}}|\boldsymbol{v}|\boldsymbol{v}\right) + (1-\gamma)\nabla \cdot \left(-\rho_f \overline{u_i' u_j'}\right) \tag{2}$$

$$\rho_f c p_f \left(\phi \frac{\partial T_f}{\partial t} + (\boldsymbol{v} \cdot \nabla) T_f\right) = (1-\gamma) k_f \nabla^2 T_f + \gamma\left[k_{fe}\nabla^2 T_f + a_{sf} h_{sf}(T_s - T_f)\right] + (1-\gamma)\nabla \cdot \left(-\rho_f c p_f \overline{u_i' T_f'}\right) \tag{3}$$

$$(1-\phi)\rho_s c_s \frac{\partial T_s}{\partial t} = k_{se}\nabla^2 T_s + a_{sf} h_{sf}(T_f - T_s) \tag{4}$$

Where $\mathbf{v}$, $\boldsymbol{p}$, $\boldsymbol{\mu_f}$, $\boldsymbol{T_f}$ and $\boldsymbol{T_s}$ are the flow velocity, pressure, dynamic viscosity, fluid-phase and solid-phase temperature, respectively, while $\boldsymbol{k_{fe}}$ and $\boldsymbol{k_{se}}$ are the effective thermal conductivities for the fluid and solid-phase, respectively. Also, $\boldsymbol{\rho_f}$ and $\boldsymbol{\rho_s}$ are the fluid and solid densities, respectively, while $\boldsymbol{cp_f}$ and $\boldsymbol{c_s}$ refer to their corresponding specific heats. In Eqs.[(2)-(3)], $\boldsymbol{\gamma}$ is a distinguishing parameter equals to either zero or one depending on whether the flow takes place in a clear or porous region, respectively; while ϕ represents the medium porosity, which equals to one just in the clear flow regions. The last term on the right side of each of Eqs.[(2)-(3)] refer to the turbulent shear stresses and heat fluxes, respectively, where the problem of turbulence has been solved depending on the standard *k*-*ε* model with using a high y+ treatment at the channels walls. The core permeability $\boldsymbol{K}$ and inertial coefficient $\boldsymbol{F}$ have been computed according to the semi-empirical model proposed by (Jeng and Tzeng, 2005), where the values of both viscous and inertial resistance in the transverse directions are three orders higher than them in the axial direction to allow both streams to flow unidirectionally; while $\boldsymbol{h_{sf}}$ and $\boldsymbol{a_{sf}}$ are the solid-to-fluid interfacial heat transfer coefficient and specific surface area, respectively. Following the analytical model of (Muzychka and Yovanovich, 2004), $\boldsymbol{h_{sf}}$ is computed locally depending on the local value of the velocity. Overall, and for seeking conciseness, a complete/closed model of the mathematical procedure proposed is not given herein, which has in turn been detailed elsewhere (Alhusseny and Turan, 2016).

3. NUMERICAL PROCEDURE

3.1. Discretisation method and grid generation

The governing equations mentioned earlier were discretised using the finite volume method. The second-order upwind differencing scheme was employed to represent convection terms of the discretised governing equations, while the problem of pressure–velocity coupling was resolved by using SIMPLE algorithm. Steep gradients expected at the walls and interfaces were captured by refining the mesh at these regions. Dependence of numerical results on the grid size was checked by examining four sets of solution grid having a total number of cells (0.76×106, 1.03×106, 1.28×106, and 1.47×106) at **$\dot{m}$=0.5 kg/sec**, where the fluid temperature profile is plotted at ***x*=200mm** in the vertical mid-plane at ***y*=0** as shown in Fig. 3. It is noticed that the deviation in the obtained data becomes marginal between the third and fourth set of grid and thereafter. Accordingly, the mesh size of (1.28×10^6) can be considered sufficient for the accuracy of the current study.

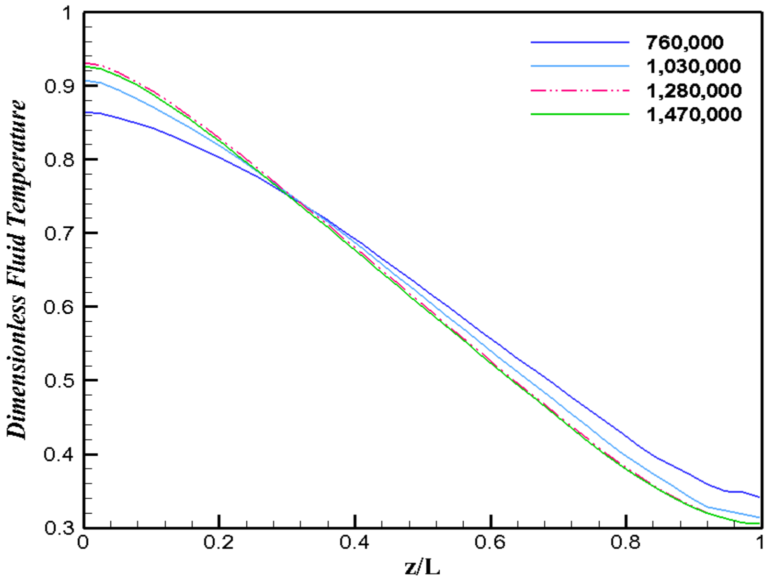

Fig. 3. Grid dependency

3.2. Validation of the computational program

To verify the validity of using the porous media approach in the current study, the results computed numerically using the Star-CCM+ CFD commercial code were compared with some former works. The hot-side pressure drop was validated against the experimental data presented by (Sayama and Morishita, 1993) for different values of mass flow rate and hot inlet temperature as shown in Fig. 4. Moreover, the thermal effectiveness for different values of core cell density (n) (the number of packed passages per in^2) was compared with analytical data introduced by (Kays and London, 2018) as shown in Table 1. It is apparent that the computed results using the current approach are in a fair agreement with the earlier studies, where the error found in values computed for the thermal effectiveness up to 2.7%. Such deviation might be attributed to the approximations followed in both the former NTU analysis and present CFD simulation in addition to the probable inaccuracy in the experimentally measured data.

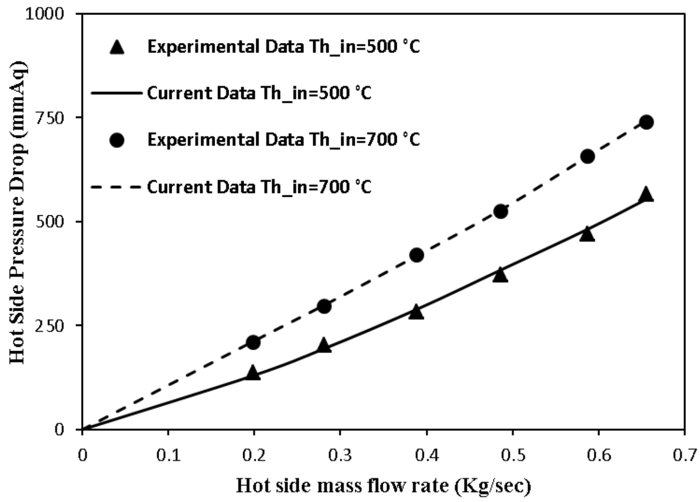

Fig. 4. Comparing the pressure drop with experimental data.

Table 1. Comparison between the numerically and analytically computed effectiveness for square cross-sectional passages core of L=70mm, s=0.09mm, and Ω=20rpm besides the input data applied

n (cell/in^2)	a (mm)	φ	K (×10^8 m^2)	F (×10^3)	a_{sf} (m^2/m^3)	C_s (W/°K)	C^*	C_{sf}^*	h_c (W/m^2°K)	h_h (W/m^2°K)	$\bar{U}A_{s,r}$ (W/°K)	NTU	ε_0	ε_t (Kays and London, 2018)	ε_t Currently numerical data	ε_t error %
400	1.180	0.863	4.232	0.414	2926	1173	0.963	2.362	130.7	143.6	849.7	1.702	0.638	0.625	0.643	2.696
600	0.947	0.834	2.633	0.433	3523	1424	0.967	2.863	159.8	173.2	1163	2.339	0.709	0.698	0.703	0.624
900	0.757	0.799	1.61	0.476	4222	1726	0.973	3.469	198.1	211.9	1718	3.451	0.783	0.776	0.767	1.053
1200	0.643	0.77	1.121	0.498	4786	1976	0.972	3.985	233.0	245.4	2273	4.582	0.830	0.824	0.81	1.694
1500	0.566	0.744	0.839	0.525	5262	2192	0.973	4.439	265.1	275.9	2827	5.726	0.861	0.856	0.843	1.466
1800	0.509	0.722	0.658	0.556	5677	2383	0.970	4.870	294.9	305.0	3382	6.911	0.895	0.880	0.864	1.786

4. RESULTS

The results presented in this section were computed for an identical value of inlet mass flow rate for both streams $\dot{m}$=0.5 kg/sec and constant angular velocity Ω=20 rpm, while the temperature at the inlet of cold and hot stream channel is considered uniform and equal to 250°C and 500°C, respectively. The values of inner and outer diameters of the rotating regenerator are 100mm and 420mm, respectively, while its length is L_{reg}=70mm.

The temperature distribution across the rotating matrix is shown in Fig. 5 for both the fluid and solid phases. The heat recirculation is quite obvious, where the solid matrix plays the role of an intermediate medium that absorbs heat from the hot stream and makes it cooler and then dissipates the gained heat into the cold stream and makes it hotter.

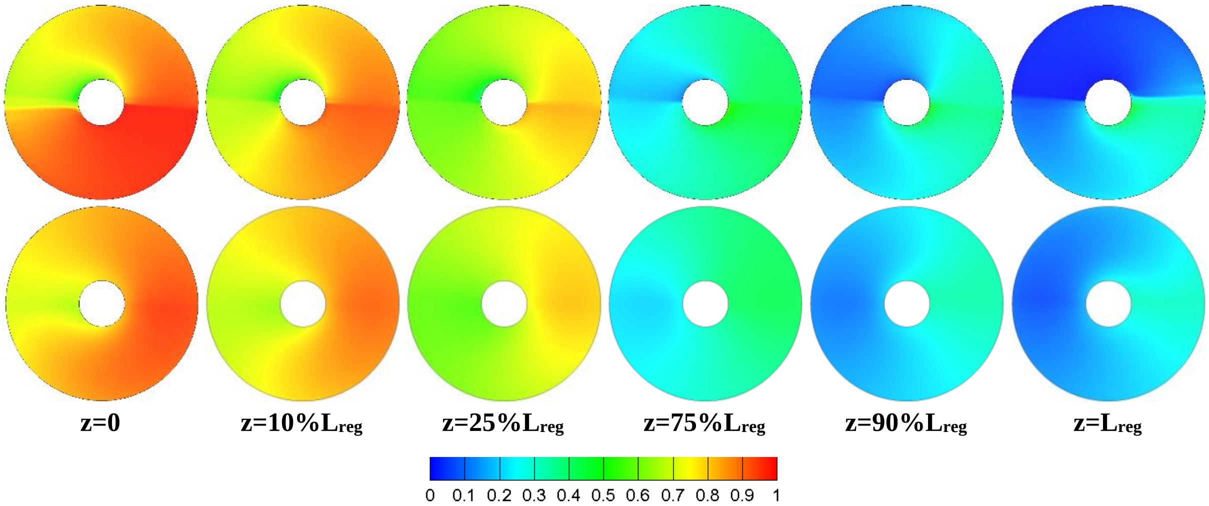

Fig. 5. Dimensionless temperature distribution for fluid (up) and solid phase (down) at various longitudinal locations of the core.

The efficiency of that process is mainly measured in terms of two key factors. The first is the overall thermal effectiveness of rotating heat exchanger and can be computed as:

$$\varepsilon_t = \frac{\varepsilon_c + \varepsilon_h}{2} \quad (5)$$

While the other factor is the total pressure loss occurring across the rotating core, which is used to determine the required pumping power. Overall, examining the practical worth of using specific geometric core characteristics is essential in designing rotary regenerators. This can be achieved through introducing a performance measure similar to that used in heat pumps, which is called the coefficient of performance *COP* and defined as the ratio of the heat recovered from hot stream to the pumping power required.

Among the geometrical characteristics of the rotating core that have a direct impact on the overall performance are the cell density n, which is the number of packed passages per in^2, and the passages wall thickness s. The effect of cell density on thermal effectiveness, pressure drop, and the *COP* of a rotary regenerator formed of passages having s=0.09mm wall thickness is shown in Fig. 6 for n_0=900/in^2 as a reference cell density. Obviously, the effectiveness is enhanced significantly with increasing the cell density due to the increase in the surface area exposed to the fluid flow within the passages. However, this augmentation has a serious drawback, which is the increase in the pressure drop produced across the core. This increase is attributed to the reduction in the matrix porosity, which leads to increasing the local velocity within the passages and reducing the permeability of the core matrix, and hence, increasing the core resistance to the fluid flowing through it according to Eq. (2). Thus, the overall performance, or in other words the *COP* value, deteriorates significantly with increasing the cell density despite the enhancement attained in the heat restored from exhaust gas.

Similarly, the influence of passages wall thickness on thermal effectiveness, pressure lose, and *COP* is demonstrated in Fig. 7 for a cell density n=1200/in^2 and reference wall thickness s_0=0.15mm. It is noticed that increasing the wall thickness of the used passages reduces the thermal effectiveness as a result of decreasing the total surface area exposed to the fluid flow. Meanwhile, it causes a reduction in the core porosity, and hence, increases the pressure drop for the same reasons mentioned earlier. Consequently, utilising flow passages that have thicker walls leads to deteriorating the overall performance of the system. However, affording high pressure losses is sometimes an indispensable choice due to the need for achieving high rates of heat recovery required in certain applications.

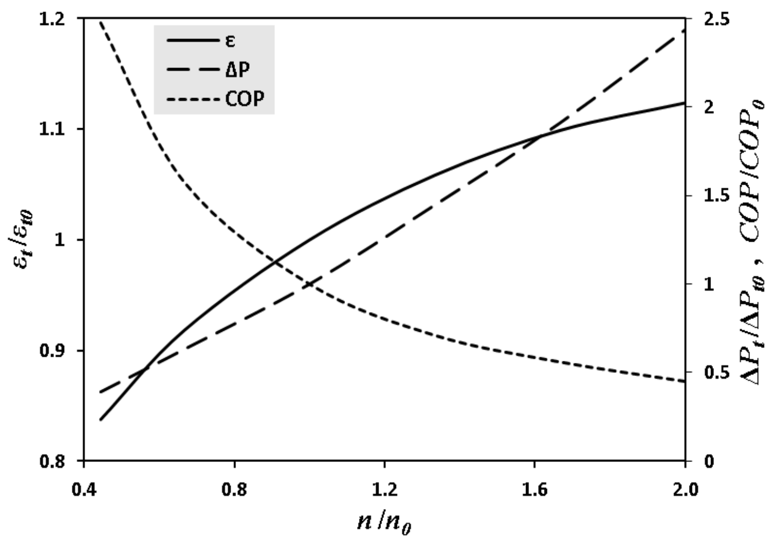

Fig. 6. Cell density effect at ε_0=0.77, ΔP_{t0}=6,088Pa, COP_0=16.3

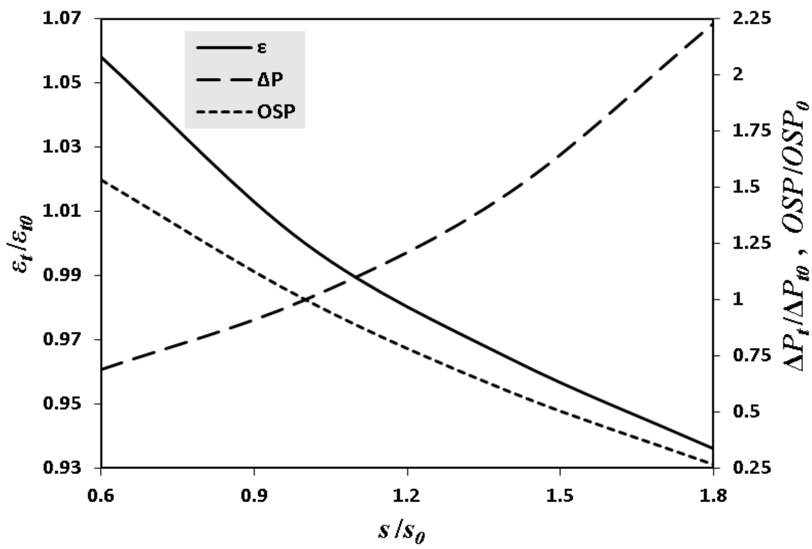

Fig. 7. Wall thickness effect at ε_0=0.768, ΔP_{t0}=12,777Pa, COP_0=9.35

5. CONCLUSIONS

The current study presents a numerical simulation for the heat and fluid flow through a rotating heat exchanger by means of the porous media concept. An aluminum core formed of multi-packed square passages is simulated as a porous medium of an orthotropic porosity, where heat is allowed to transfer from fluid to solid phase and vice versa. The geometric properties of the core were transformed into the conventional porous media parameters based on empirical equations; so, the core has been dealt with as a porous medium of known features. Using porous media approach was found to be sufficient to simulate the rotary regenerator problem. The data obtained reveals an obvious impact of core geometrical parameters on both the heat restored and pressure loss; and hence, overall efficiency of the regenerator system. Thermal effectiveness can be improved considerably by increasing the cell density although pressure loss will be augmented. Thus, care must be taken in selecting the geometrical properties of rotary regenerators to avoid unjustified expenses.

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