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Numerical Modeling of Regenerative Rotary Heat Exchanger: A Review

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Abstract

Background: Heat recovery is one of the prominent ways to save a considerable amount of conventional fossil fuel and minimize its adverse effects on the environment. The rotary heat exchanger is one of the most effective and efficient devices for heat recovery or heat exchanging purposes. It is a regenerative type of heat exchanger, which has been studied and used for many heat recovery purposes. However, regenerative thermal wheels have been mostly used as heat recovery systems in buildings. For modeling a rotary regenerator, it is very important to numerically consider all the factors involved, such as effectiveness, rotational speed, geometrical size and shape, and pressure drop (Δp). In recent times, several researchers have actively studied the rotary heat exchangers, both theoretically and experimentally. **Reviews:** In this paper different advances in the numerical modeling of regenerative rotary heat exchangers in relation to fluid flow and heat transfer have been discussed. Researchers have indicated that the effectiveness of the regenerative rotary heat exchanger depends on various factors including, among many others, rotational speed, rotational period and combustion power. It is reported that with the increase of periodic rotation the deviation of theoretical results from the experimental result increases. The available literature indicates that regenerative heat exchangers are having relatively more effectiveness (60–80%), compared to other heat exchangers. It is also observed that the finite difference method and finite volume methods are mostly used for discretizing the heat transfer governing equations, under some assumptions. Research also indicates that for the effectiveness calculation the ε-NTU method is the most popular and convenient.

Keywords: Effectiveness, Governing Equation, Heat Recovery, Numerical Modeling, Regenerative Rotary Heat Exchanger.

Symbols

NTU	number of transfer unit
Cr	total heat capacity rate of a matrix (W/K)
Cr^*	matrix heat capacity rate ratio (Cr/C_{min})
h	heat transfer coefficient (W/m^2 K)
T	temperature (K)
ROT	rotational speed
m_{matrix}	matrix mass (kg)
η	fraction of the phase change energy that enters the air
Cp	specific heat, J/(kg·K)
A	cross-sectional area, m^2
U	mean airflow velocity in tube, m/s
x	axial coordinate, m
W	humidity ratio, kg(water vapor)/kg (dry air)
Wm	empirical coefficient used in sorption isotherm describing maximum moisture capacity of desiccant, kg (water)/kg (desiccant)
h	convective heat transfer coefficient, $W/(m^2{\cdot}K)$
p	perimeter of each tube, m
k	thermal conductivity, W/(m·K)
M_r	mass ratio of the energy wheel

Subscript

a	air
c	cold
h	hot

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H	total enthalpy per mass of dry air, J/kg
w	weight
Cp	specific heat, J/(kg·K)
ε	effectiveness
ϕ_r	correction factor for rotational speed (–)
Δp	pressure drop (kpa)
ρ	density, kg/m^3
D_h	hydraulic diameter of one tube in the energy exchanger
α	thermal diffusivity, m^2/s
$\dot{m}/$	rate of phase change per unit length, kg/(s.m)
h_{ad}	heat of sorption (adsorption and desorption), J/kg
out	outlet
in	inlet
max	maximum
min	minimum
d	desiccant
m	constant
s	solid
f	fluid
w	solid wall
Al	aluminum
t	matrix thickness (m)
v	vapor
g	gas

Introduction

The study of heat transfer and heat exchangers plays a major role in energy conservation, reduction of greenhouse gas emissions, and in sustainable development (Suden and Wang, 2003). In developed countries, buildings are responsible for the maximum portion (almost 40%) of the primary energy consumption, whereas in developing countries it is about 20%-40% (Mardiana and Riffat, 2013). The rotary regenerative heat exchanger may play a major role in reducing the use of primary energy by recovering some portion of the exhaust heat energy of the buildings by introducing a heat exchanger between exhaust and fresh air stream (Valancius et al., 2013; Airaksinen et al., 2011). Regenerative rotary heat exchangers (Thermal wheel) are the rotating type of the compact heat exchanger. The heat exchanger consists of a rotary regenerator with a matrix of a good heat transfer material. The matrix of the heat exchanger is the heat transfer element. The matrix can be made up of various materials such as aluminum, plastic, and PVC. The shape of the heat transfer matrix may be of different types, for instance, hexagonal, triangular, square, and circular. Desiccant materials like silica can be used for dehumidification. In most cases, aluminum is preferred. The wheel is rotated by using a motor, which is run by some external power.

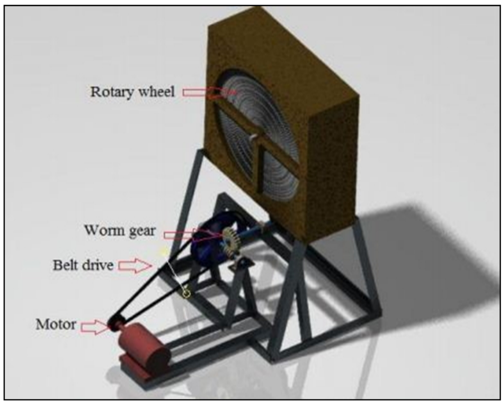

Figure 1. The solid 3-D view of a regenerative rotary heat exchanger.

The rotary heat exchangers have been extensively used and studied for a long time. The increasing interest on the thermal wheel as a heat recovery device is due to its low pressure drop and high effectiveness (Stefano et al., 2014). Several researchers have done different studies on the rotary heat exchanger, both numerically and experimentally. The regenerative rotary heat exchanger has certain advantages over other types of heat exchangers for the recovery of heat from the exhaust air of any engine (Schmidt, 1991). Two major advantages that rotary heat exchangers have are (i) a considerably larger heat transfer area per unit volume and (ii) a comparatively lower cost, and it has a higher effectiveness compared to other types of heat exchangers. Effectiveness of plate-type heat exchangers is 45% to 65%, compared to between 60% and 80% for rotary regenerators (Heinrich and Franzke, 1993). The rotational speed of the matrix can be used to regulate the amount of the heat transferred. Thus, rotary heat exchangers are characterized by both compactness and high performance (Skiepko, 1989). Rotary heat exchangers are frequently used in gas turbine engines and air-conditioning systems. For a long time, they have been used as air pre-heaters in steam power plants (Romie, 1988).

Many studies have considered rotary regenerative heat exchangers, which transfer sensible heat at steady state operating conditions (Klein et al., 1990; Simonson et al., 1999). Kays and London in 1984, presented an "effectiveness-number of transfer unit" (ε-NTU) method for predicting the effectiveness of rotary regenerative heat exchangers and sensible recuperative heat exchangers as a function of dimensionless numbers. They showed that the effectiveness depends on two dimensionless numbers for sensible (recuperative) heat exchangers (NTU and total heat capacity rate of a matrix (Cr)) and four dimensionless numbers for rotary heat exchangers (NTU, Cr, Cr*, h_hA_h/h_cA_c) (Kays and London, 1984), where, Cr* is the matrix heat capacity rate ratio (Cr/Cmin) and h_hA_h/h_cA_c is the ratio of heat transfer coefficient for hot and cold fluids. Researchers have pointed out the energy saving and operational benefits of rotary regenerators for large and small buildings in Florida (Rengarajan et al., 1996; Shirey and Rengarajan, 1996). Bowlen (1993) stated that the energy recovery in the cooling/dehumidifying mode can be up to 2.5 times greater than for an identically sized sensible heat exchanger, and approximately 40% greater in the heating/humidifying mode.

The main purpose of this paper is to study and review different methods and approaches for modeling a regenerative rotary heat exchanger. Before designing a heat exchanger or an air pre-heater it is the utmost priority for the researchers to model the governing equations with proper boundary conditions to get the optimum enthalpy exchange between the heat exchanger and the working fluid. Therefore, it is very important to study the previous works done by the researches with different approaches.

Governing Equations and Numerical Modeling

Klein et al. (1990) developed a mathematical model of a rotary regenerator and established that to obtain the optimum enthalpy exchange between two air streams the regenerator must be operated at such conditions that neither of the two transfer waves reaches the outlet of the exchanger. The enthalpy exchange effectiveness is determined by the number of transfer units. They modified the finite difference solution to study the behavior of the enthalpy exchangers at very cold conditions.

Stiesch et al. (1995) presented the heat and mass transfer governing equations for rotary regenerative heat exchangers. They formulate those equations based on the mass and energy conservation equations combined with some assumptions for the rotary regenerative heat exchanger.

The heat and mass transfer governing equations are as follows,

Mass conservation:

$$\frac{\partial w_a}{\partial Z} + M_r \frac{\partial w_a}{\partial \tau} + \frac{\partial w_m}{\partial \tau} = 0 \tag{1}$$

Energy conservation:

$$\frac{\partial H_a}{\partial Z} + M_r \frac{\partial H_a}{\partial \tau} + \frac{\partial H_m}{\partial \tau} = 0 \tag{2}$$

Mass transfer rate:

$$\frac{\partial w_m}{\partial \tau} = NTU_m (W_a - W_m) \tag{3}$$

Thermal energy transfer rate:

$$\frac{\partial H_m}{\partial \tau} = NTU_t \frac{\partial H_a}{\partial T_a} (T_a - T_m) + NTU_m (W_a - W_m) H_m \tag{4}$$

For designing the flow geometry of the heat exchanger, the numerical modeling and solving of the governing equations is very important. The above cited heat and mass transfer governing equations can be solved by using different numerical methods as discussed below.

Different researchers have used different methodologies to solve the above governing equations. The governing

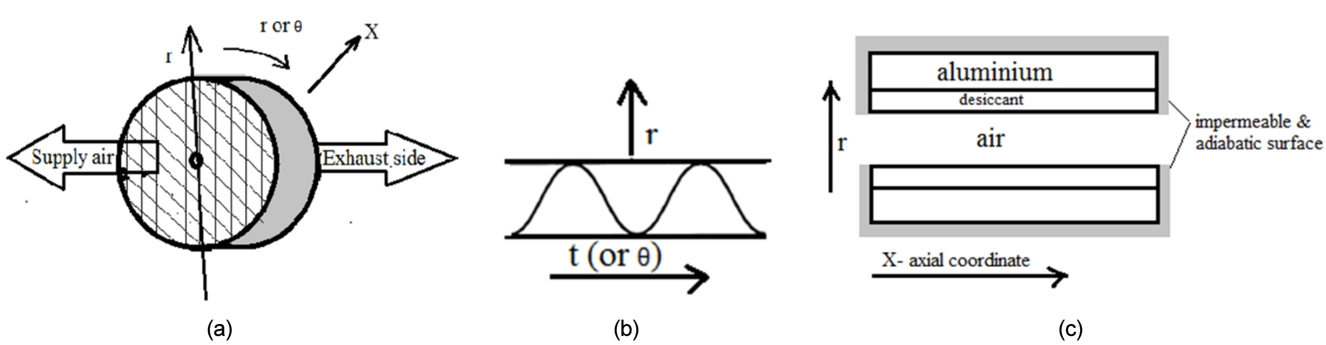

Figure 2. Schematic diagram of the rotary heat exchanger wheel showing (a) entire wheel, (b) the tube geometry, (c) a side view of one of the tubes (Simonsons and Besant, 1999).

Table 1. Assumptions for modeling the governing equations (Simonson and Besant, 1997; Simonson and Besant, 1998)
· In the air, the axial heat conduction and water vapor diffusion are negligible.
· Molecular diffusion and capillary motion of moisture within the desiccant are negligible in the axial direction.
· In the matrix, there are no radial temperature and moisture content gradients.
· The ends and half plane of each matrix tube are adiabatic and impermeable.
· The heat of sorption is assumed constant and equal to the heat of vaporization.
· For the desiccant coating, hysteresis in the sorption isotherm is neglected.
· The tubes of the rotary heat exchanger are similar with constant heat and mass transfer surface area.
· The thermal and moisture properties of the matrix are constant.

equations, by using the coordinate system, were presented as shown in Figure 2. They considered the equations for continuous and coupled heat and moisture transfer in one tube as it rotates around the thermal wheel axis. Warm humid supply air is taken as a typical operating condition for transferring energy and water vapor to the matrix. Energy and moisture are being transferred to exhaust air from the matrix in the second half cycle (Simonsons and Besant, 1999).

The governing equations of the regenerative heat exchangers are nonlinear and are not possible to solve by any direct methods. For solving the equations different researchers have assumed different conditions for making them simpler. The governing equations are derived based on the assumptions given in Table 1 below.

Based on the above assumptions, the governing equations for coupled heat and moisture transfer for one flow tube in rotary energy exchangers are presented by Simonson and Besant (1997). The tube geometry and pattern are shown in Figures 2 (b) and (c). Energy storage, convection, conduction, and energy associated with phase change are the energy equations for the air and matrix, which are given in equation 5 and 6, respectively.

$$\rho_g c_{P_g} A_g \ \frac{\partial T_g}{\partial t} + U \rho_g \, c_{P_g} A_g \ \frac{\partial T_g}{\partial x} - \dot{m}/h_{ad}\eta + hp(T_g - T_m) = 0 \tag{5}$$

$$\rho_g \, c_{P_m} A_m \ \frac{\partial T_m}{\partial t} - \dot{m}/h_{ad}(1-\eta) - \dot{m}/C_{Pw}(T_g - T_m) - \ hp(T_g - T_m) = \frac{\partial}{\partial x}(k_{Al} A_{Al} \ \frac{\partial T_m}{\partial x}) \tag{6}$$

These energy equations, which are written with temperature as the dependent variable, are almost identical to those derived by Stiesch et al. (1995) and Zheng and Worek (1993), who use enthalpy as the dependent variable (Simonsons and Besant, 1997).

The two equations given above are the energy equations where is the fraction of the phase change energy, which is convected directly into the air (Simonsons and Besant, 1998).

$$\eta = \frac{hD_h/\sqrt{\alpha_s}}{\frac{hD_h}{\sqrt{\alpha_a}} + k_d/\sqrt{\alpha_d}} \tag{7}$$

If η=1, all the energy of the phase change is directly convected to air. On the other hand, if η=0, then all the energy of the phase change is conducted into the matrix. The value of η is not the same for all the energy wheels, but it is expected to range between 0 and 0.1. When η lies between 0 and 1, some fraction of the energy will be convected to the air and other part will be conducted to the matrix.

A rotary regenerator was simulated using an already developed mathematical model by assuming constant physical properties of fluid and solid, a constant heat transfer coefficient, and considering the thermal conductivity of the solid matrix to be zero and infinitely parallel and normal to the flow respectively. The heat and amass transfer governing equations (Equations (1), (2), and (3)) are solved computationally by using numerical techniques, which provide the distribution of temperature over time for the fluid flow and the solid matrix. Then the effectiveness of the regenerator is calculated that replaces the analytical solutions given by Kays and London (1984). Using an experimental model of the regenerator, they have calculated the temperature ranges and using the analysis of variance (ANOVA) method, the experimental results are obtained for a range of parameters. It was concluded that rotational speed, hot air velocity, and cool air velocity are the only three parameters affecting regenerator efficiency (Ghodsitour and Sadrameli, 2003).

In a rotary heat exchanger, a one-dimensional numerical model of the heat transfer between the hot combustion air and solid matrix, on one side, and the cold air and solid body on another, can be applied to estimate the heat exchangers performance conveniently. The control volumes are positioned one after another and a constant rotational speed is assumed for all (Zafeiriou and Wurz, 1996).

The numerical model that is used consists of two energy balance equations.

Energy balance equation for the solid body:

$$\underbrace{m_s c_s \frac{\partial T_s}{\partial t}}_{\text{accumulation in solid body}} + \underbrace{k_s A_l \frac{\partial^2 T_s}{\partial x_l^2}}_{\text{conduction in solid}} = \underbrace{hA\left(T_g - T_s\right)}_{\text{heat source}} \tag{8}$$

Energy balance equation for the fluid phase:

$$\underbrace{m_g c_p \frac{\partial T_g}{\partial t}}_{\text{accumulation in gas}} + \underbrace{\dot{m}_g c_p (T_{in} - T_{out})}_{\text{convection}} + \underbrace{k_g A_l \frac{\partial^2 T_g}{\partial x_l^2}}_{\text{Conduction in gas}} = \underbrace{hA\left(T_g - T_s\right)}_{\text{heat source}} \tag{9}$$

The interface between the above two energy balance equations is the convective heat transfer between the fluid and the solid.

The fluid temperature change in the direction perpendicular to the main flow can be neglected as each flow passage is narrow and surrounded by thin material. It is also assumed that there is no exchange in the radial direction with the surroundings, and thus, it can be considered that conduction in the solid body only occurs along the axial coordinate z. Again, because of the low fluid, thermal conductivity heat transfer by conduction in the fluid is neglected. By means of these assumptions, the equations (8) and (9) are simplified and reduced to those of unsteady 1-D flow, as shown in equations (10) and (11). By using a finite difference method, the 1-D unsteady can be solved (Sandira et al., 2005).

Simplified energy balance equations:

$$m_s c_s \frac{\partial T_s}{\partial t} + k_s A_z \frac{\partial^2 T_s}{\partial z^2} = hA\left(T_g - T_s\right) \tag{10}$$

Yilmaz and Ukalaca (2003) presented a calculation method for the design of rotary regenerators having different flow channel geometries. This method is focused on correcting the traditional heat exchanger calculation method "Effectiveness-NTU" by introducing two factors for correction, one for the rotational correction and the other for the purge area. They showed that this method is valid for rotational speeds between 0.05 and 7 rev/min.

Pascal and Dominique (2008) developed a model for a desiccant wheel, which consists of two parts; one is for process and the other is for regeneration. They solved the heat and mass transfer equations of the desiccant wheel by a characteristic method. The model has been validated by comparing the data obtained from the manufacturers and the experimental results.

Zheng et al. (2012) carried out an experimental study on a plate fin air-pre heater to investigate the pressure drop and heat transfer. They used a genetic algorithm method to separate the heat transfer coefficient in the gas and air sides of the heat exchanger. The results obtained showed that the increase in the gas side inlet temperature slightly decreases the gas side pressure drop, while the air side pressure drop remains the same.

Armin and Ebrahim (2014) treated the pre-heater matrix as a porous media and studied the thermal behavior of a rotary air pre-heater in a steam power plant using three dimensional approaches. To incorporate the effect of rotational speed of the matrix, mass, momentum, and energy equations are solved using a moving reference frame (MRF). They discussed how the performance of the air pre-heater is affected by the rotational speed, air and gas flow rate, material type and inlet air temperature.

Numerical Methods for Solving the Fundamental Governing Equations

Many researchers have developed several methods to solve the governing equations (hyperbolic differential equations), which describe the performance of a regenerative heat exchanger. Some of the theoretical approaches are discussed below.

Finite difference method (FDM)

FDM is one of the most popular methods to discretize the heat and mass transfer governing equations of the rotary heat exchanger. It is the oldest method and also the easiest to apply in simple geometries (Borah et al., 2013). FDM is one of the most important numerical methods to solve differential or partial differential equations. The equations can be solved by approximating them with the difference equation, where the derivatives are approximated by the finite differences. In this method, the Taylor series expansion or polynomial fitting is used to approximate the derivatives of the variables with respect to coordinates at each grid point of the computational domain. However, according to Stabat and Marchio (2009), finite difference models require very long computational time. In FDM the computational domain is covered by a grid and at each grid point the Tailor series expansion or polynomial fitting is applied to approximate the variables with respect to the coordinates (Smith, 1978).

Finite Volume Method (FVM)

The FVM is also a well-known and effective numerical method for representing and solving partial differential equations in the form of algebraic equations (LeVeque, 2002; Toro, 1999). This method is the same as the finite difference method, where values are calculated at discrete points on a meshed geometry. This method is termed "finite volume method" because each node point on the mesh is surrounded by a small volume. In this method, the divergence theorem is used to convert volume integrals in a partial differential equation that contain a divergence term in the surface integrals. Then, at the surfaces of each finite volume, these terms are evaluated as fluxes. FVM methods are conservative. The flux entering a given volume is similar to that leaving the adjacent volume. The main advantage of the FVM is that for unstructured meshes, the equations are easily formulated. In this method, the integral form of the conservation equations is applied. The variables are located at the node point, which is placed at the center of the control volume. The values of the variables at the faces of the control volumes are determined by interpolation. The finite volume method is particularly suitable for complex geometries. For heat transfer and convective flow, the FVM method is very popular and useful (Borah et al., 2013). Simonsons and Besant (1997) have developed a one-dimensional transient numerical model, and is formulated using the finite volume method with an implicit time discretization for the coupled heat and moisture transfer in rotary energy exchangers. FVM is also applied in many commercial CFD codes (Patankar, 1980; Versteeg and Malalasekara, 1995).

Control volume FVM (CVFVM)

This is a hybrid method, which is derived from FVM and FEM. In this method, the computational domain is divided into triangular elements for two dimensional cases where the nodes are located at the vertices of the triangle. By joining centroids and mid points of the element edges, the control volumes are formed around the nodes. As outlined for the FVM, the conservation equations in integral form are applied to the control volumes (Masson et al., 1994).

Finite Element Method (FEM)

The FEM is a very important numerical tool for solving engineering and mathematical problems. FEM is also referred to as the finite element analysis (FEA). Heat transfer, fluid flow, mass transport, structural analysis, and electromagnetic potential problems can be solved by using FEM. To solve these problems analytically, the solution to boundary value problems for partial differential equations is required. The FEM formulation of the particular problem results in a system of algebraic equations. At a discrete number of points over the domain FEM provides approximate values of the unknowns. This method is referred to as FEM because, to solve the problem, it subdivides a large problem into smaller, simpler parts called finite elements. Then these simple equations are assembled into a larger system of equations that models the entire problem. To approximate a solution by minimizing an associated error function, it then uses variational methods from the calculus of variations. For complex geometries, FEM is very effective (Sanaye and Hajabdollahi, 2009; Wu et al., 2012). Baclic (1995) used the Galerkin method to get a closed form approximate solution.

Boundary Element Method (BEM)

BEM is also a numerical computational method for solving linear partial differential equations that have been formulated in a *boundary integral* form. It can be applied in many areas of engineering, such as fluid mechanics, acoustics, electromagnetic, and fracture mechanics.

The BEM method transforms the governing equations that are to be solved into boundary integrals. It is well suited for heat conduction problems, but for convective flow and heat transfer problems it becomes too complex (Power and Wrobel, 1994). In BEM, the integral equation may be considered as an exact solution of the partial differential equation (governing equations). This method uses the given boundary conditions to fit boundary values into the integral equation. Then, in the post-processing step, the integral equation can be used again to calculate the numerical solution directly at any desired point within the solution domain.

Empirical approach

Some researchers have developed some correlations for rotary heat exchangers based on the performance curves of manufacturers or experimental data. These correlations state the performance of a particular thermal wheel in various conditions of temperature and humidity ratio at process and regeneration inlets. An empirical formula was presented based on operational data to correlate the effect of rotation with regenerator performance (Chi, 2002).

Analytical method

Based on the Laplace transformation of heat and mass, Mathiprakasam and Lavan (1980) have developed an analytical method to solve the governing equations. The Taylor series expansion or polynomial fitting is used to approximate the derivatives of the variables with respect to the coordinates at each grid point (Stabat and Marchio, 2009). Lamberston (1958) used a standard numerical iterative method to obtain a numerical solution. Leng (2005) used the Laplace transformation method to solve the fundamental equation, where the results show the temperature distribution of metal in steady state.

The method of characteristics

The method of characteristics is a technique for solving different partial differential equations. It is basically applicable to first-order differential equations. Generally, this method is applicable for any hyperbolic partial differential equation. The main function of this method is to reduce a partial differential equation to a family of ordinary differential equations, along which the integration of the solution can be obtained from some initial data given on a suitable hyper-surface. To transform the coupled partial differential equations (non linear) in a set of uncoupled differential equations, the method of characteristics has been developed. The new uncoupled equations depict new independent variables, which are called characteristic potentials (Mathiprakasam and Lavan, 1980).

Commercial computer packages

There are different CFD packages that can be used for solving these equations. Nowadays several industries are using commercially available CFD-codes (computational fluid dynamics) for simulation of fluid flow and heat transfer and for investigation on enhanced heat transfer, electronics cooling, gas turbine heat transfer, etc. Among these codes, the most commonly used are: FLUENT, CFX, STAT-CD, FIDAP, PHOENICS, ADINA, CFD2000, etc. (Mathiprakasam and Lavan, 1980).

Optimization Techniques

Optimization techniques are mathematical methods or tools, which can be used to select the best element from a given set of available alternatives. The optimization techniques can be used in any engineering problem to find the best available values of some objective function. Basically there are two types of optimization techniques, the classical optimization and evolutionary algorithm (EA). The classical optimization techniques are useful in finding the optimum solution for continuous and differentiable functions (unconstrained). These are analytical methods and they use differential calculus to locate the optimum value. On the other hand, an EA is a subset of evolutionary computation, a generic population-based random optimization algorithm. The EA techniques are inspired by biological evolution (for instance, reproduction, mutation, recombination, and selection). In a population, the roles of individuals are played by the candidate solutions to the optimization problem, and their fitness values determine the quality of the solutions. Then, after the repeated application of the above operators, evolution of the population takes place. Different researchers used different optimization techniques for optimizing the regenerative heat exchanger. For obtaining the optimum performance from a rotary heat exchanger, effectiveness and pressure drop are the two crucial parameters that need to be optimized for any industrial application. Sanaye and Hajabdollahi (2009) have modeled the rotary regenerator using the Effectiveness-NTU method to estimate its pressure drop and effectiveness. Frontal area, rotational speed, ratio of hot and cold frontal heat transfer area, matrix thickness, matrix rod diameter, and porosity were the main parameters that they considered for the design. They used a non-dominated sorting genetic algorithm (NSGA-II) for optimizing the design parameters. Their main objective functions are effectiveness and pressure drop.

For air conditioning applications, Sanaye et al. (2008)

obtained optimal operational conditions of an air-to-air rotary regenerator, considering thermal effectiveness as a single objective function. To maximize the heat transfer rate and to minimize the pressure drop, Hilbert et al. (2006) applied a multi-objective optimization technique. Minimization of energy consumption and material cost were also studied on an air cooled heat exchanger (Gholap and Khan, 2007).

Design Methods

Heat exchanger design becomes somewhat more complex when it is combined with both sensible energy transfer and mass transfer. There is no simple design methodology for a regenerative rotary heat exchanger (Simonson and Besan, 1999). Regenerative rotary heat exchangers are widely used in HVAC systems, though it does not have a simplified design or effectiveness correlation, because they can considerably reduce the heating and cooling loads of a building (Rengarajan et al. 1996; Shiery and Rengarajan, 1996). The thermal wheel can also increase the thermal comfort and decrease both the capital and operating costs.

Effectiveness (ε) is the best-known and widely accepted method to characterize and design heat exchangers under a wide range of operating conditions, when only inlet operating conditions are known (Shah et al., 1988; Downing and Bayer, 1993). The ε-NTU method is one of the most favoured design methods for regenerative heat exchangers.

Analytical study of effectiveness for regenerative rotary regenerator

Different researchers have introduced different approaches for finding out the thermal effectiveness of the thermal wheel or desiccant wheel. The first expression for a thermal wheel or desiccant wheel effectiveness was given as follows (Antonellis et al., 2010; Kangolu et al., 2004).

Thermal effectiveness,

$$\varepsilon_T = \frac{T_{po} - T_{pi}}{T_{ri} - T_{pi}} \tag{12}$$

Where,

T_{po} = Temperature of the output process air.
T_{pi} = Temperature of the input process air.
T_{ri} = Regenerative temperature of the input air.

The ε-NTU method is often used for designing heat exchangers and predicting their performance. This method is already well established for sensible heat exchangers (Kays and. London, 1984; Shah and Subbarao, 1988). In many heat transfer publications the classical definition of effectiveness is applied to sensible heat transfers with no moisture transfer (Shah and Subbarao, 1988; Stiesch et al., 1995). Recent experimental and theoretical evidence show that for a range of inlet temperatures and humidity, the effectiveness of regenerative heat exchangers is not constant (Stiesch et al., 1995; Simonsons et al., 1998).

Results for the effectiveness(ε_r) of a regenerator are acquired by using a simple empirical formulation given by Kays and London, in which (ε) denotes the effectiveness of a counter flow heat exchanger and ϕ_r is a correction factor that takes into account the rotational speed (Kays. and London, 1984; Gale, 1967).

$$\varepsilon_r = \varepsilon.\ \phi_r \tag{13}$$

$$\varepsilon = \frac{1-\exp(-NTU_o(1-C^\circ))}{1-C^\circ\exp(-NTU_o(1-C^\circ))} \tag{14}$$

where, is the ratio of minimum to maximum heat capacity rate of air stream.

Kays and London (1984) suggested the equation given below for the correction factor, which depends on the total heat capacity rate (C_r) of the matrix and the rotational speed (ROT).

$$\phi_r = 1 - \frac{1}{9C_r^{-1.93}} \tag{15}$$

$$C_r = \frac{ROT}{60} \cdot m_{matrix} \cdot C_m \tag{16}$$

Influence of period of rotation on effectiveness

The effect of the period of rotation on the effectiveness of a thermo photovoltaic (TPV) rotary heat exchanger was theoretically and experimentally investigated. It was found that with the increase of periodic rotation the deviation of theoretical results from the experimental result increased. With a rotational period of 3-15/s and a

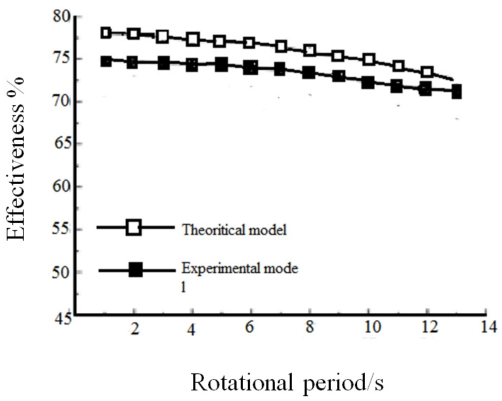

Figure 3. Comparison of theoretical and experimental results.

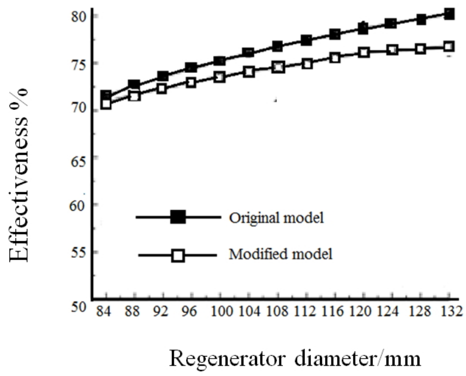

Figure 4. Influence of regenerator diameter on effectiveness.

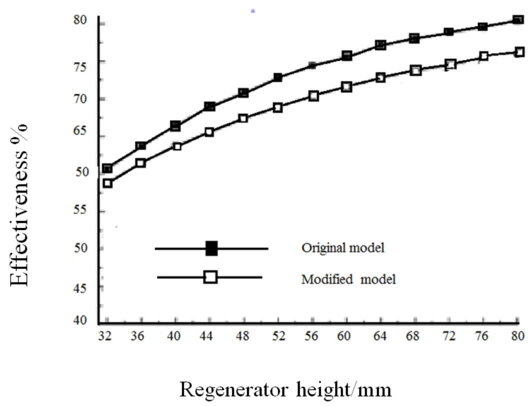

Figure 5. Influence of regenerator diameter on effectiveness.

combustion power of 5 kW, they observed a deviation of 1.3% to 3.9%. In Figure 3 below, the deviation of effectiveness with rotational period/s is shown (Wu et al., 2012).

Influence of fluid turbulence and regenerator geometrical parameters on effectiveness

Nair et al. (1998) in their theoretical study, considered the influence of fluid turbulence on effectiveness. However, the fluid diffusion coefficient must be measured so Wu et al. (2012) established a new theoretical model in which it is showed that fluid turbulence has limited influence in effectiveness. Thus, a new theoretical model ignoring the fluid turbulence was established. Wu et al. (2012) also represented the influence of the regenerator diameter on effectiveness in TPV systems of various combustion powers, when the rotary regenerator rotation period and height were 3 s and 76 mm, respectively. The influence of the regenerator diameter on effectiveness is shown in Figure 4.

They also described the influence of the regenerator height on the effectiveness when the diameter of the wheel is 120 mm and the rotation period is 3 s. In the above figure the solid symbol represents the results of the original model, and the hollow symbols describe the modified outcomes for 5 kW combustion power. It can be clearly seen that with the increase of the height and diameter, the effectiveness increases for both the original and modified models, but the modified values of effectiveness are comparatively less than original ones (Wu et al., 2012). Figure 5 represents the influence of regenerator diameter on effectiveness.

In the classical theory of effectiveness, Kays and London (1984) presented that the effectiveness of a rotary regenerative heat exchanger has a close relationship with three dimensionless parameters: NTU (Number of transfer unit), $\frac{C_r}{C_{\min}}$ (matrix heat capacity rate ratio), and $\frac{C_{\max}}{C_{\min}}$ (heat capacity ratio).

Wu et al. considered the C_{min}=C_c (heat capacity rate of the cold air), and C_{max}=C_h (heat capacity rate of the hot fluid). $\frac{C_{\max}}{C_{\min}}$ is constant under same operating condition. With the increase of height and diameter of the heat exchanger, the heat exchanger surface area, $\frac{C_r}{C_c}$ and NTU also increase, and then the effectiveness of the heat exchanger also increases (Wu et al., 2012).

Conclusion

A detailed review of the literature on regenerative heat exchangers revealed many interesting possibilities. In this paper, advances in the numerical modeling of regenerative rotary heat exchangers related fluid flow and heat transfer have been discussed. It was found that different researchers have used different approaches for solving the fundamental equations. It was also found that for complex geometries, FEM is more effective than the other methods, including FVM, CVFVM, and BEM. The effectiveness of the regenerative heat exchanger depends on various parameters, such as rotational speed, period of rotation, fluid turbulence, and on geometrical parameters like diameter and heat transfer area. For an optimum design, NSGA-II and other optimization techniques have been used. Up to this date, the optimum design shows an effectiveness up to 85%. Therefore, it is of the utmost importance to study and model the regenerative heat exchanger with proper design parameters to get the optimum effectiveness and to save conventional energy.

Conflict of Interest

The authors have no conflicting financial or other interests.

References

Airaksinen M. and P. Matilainen. 2011. A Carbon Footprint of an Office Building. Energies. 4:1197-1210.

Antonellis, S. D., C. M. Joppolo and L. Molinaroil. 2010. Simulation, performance analysis and optimization of dessicant wheel. Energy Build. 42:1386-93.

Armin H. K, H. Ebrahim. 2014. Three-dimensional simulation of rotary air preheater in steam power plant. Applied Thermal Engineering. 73:399-407.

B. Suden and L. Wang. 2008. Relevance of Heat Transfer and Heat Exchangers for Development of Sustainable Energy Systems. WIT Transactions on State of the Art in Science and Engineering. 42:1755-8336.

Baclic, B. S. 1995. The application of the Galerkin method to the solution of the symmetric and balanced regenerato. International Journal of Heat and Mass Transfer. 107:214-220.

Borah, A. K., P. K. Singh and P. Goswami. 2013. Advances in Numerical Modeling of Heat Exchanger Related Fluid Flow and Heat Transfer. American Journal of Engineering Science and Technology Research. 1(9):156 -166.

Bowlen, K. L. 1993. Energy Recovery from Exhaust Air for Year Round Environmental Control. Symposium, ASHRAE Transactions, Volume 80, Part 1, Los Angeles, CA.

Chi, Z. H. 2002. Experimental studies on measuring the cold end metal temperature of 600 MW rotating air heater. China Journal of the Proceedings of the CSEE. 22:128-132.

Downing, C. C. and C. W. Bayer. 1993. Classroom indoor air quality vs ventilation rate. ASHRAE Transactions. 99(2):1099-1103.

Gale, W. K. V. 1967. British iron and steel industry. David and Charles, Newton Abbot. 98-100.

Ghodsipour, N., M. Sadrameli. 2003. Experimental and sensitivity analysis of a rotary air preheater for the flue gas heat recovery. Applied Thermal Engineering. 23(5):571-580.

Gholap, A. K. and J. A. Khan. 2007. Design and multi objective optimization of heat exchangers for refrigerators. Applied Energy 84:1226-1239.

Hilbert, R. 2006. Multi-Objective shape optimization of a heat exchanger using parallel genetic algorithms. International Journal of Heat and Mass Transfer 49:2567-2577.

Kangolu, M., C. M. Oozdinc and M. Yidirim. 2004. Energy and exergy analysis of an experimental open-cycle desiccant cooling system. Applied Thermal Engineering. 24(5):919-31.

Kays, W. M. and A. L. London. 1984. Compact Heat Exchangers. 3rd edition. McGraw-Hill Book Co. New York.

Klein, H., S. A. Klein and J. W. Mitchell. 1990. Analysis of regenerative enthalpy exchangers. International Journal of Heat and Mass Transfer. 33(4):735-744.

Lamberston, T. J. 1958. Performation factors of a periodic-flow heat exchanger, Transactions on ASME. 80:586-592.

Leng, W. 2005. Heat exchange calculation of a regenerative air heater with analytical method. China Journal of the Proceedings of the CSEE. 25:142-148.

LeVeque, Randall. 2002. Finite Volume Methods for Hyperbolic Problems, Cambridge University Press.

Mardiana A. and S. B. Riffat. 2013. Review on physical and performance parameters of heat recovery systems of building applications. Renewable and Sustainable

Energy Review. 28:174-190.

Masson, C., H. J. Saabas and K. B. Baliga. 1994. Co located equal-order control volume Finite Element Method for two-dimensional axisymmetric incompressible flow. International Journal Numerical Methods Fluids. 18:1-26.

Mathiprakasam, B. and Z. Lavan. 1980. Performance predictions for adiabatic desiccant dehumidifiers using linear solutions. Journal of Solar Energy Engineering. 102:73-9.

Nair, S, S. Verma and S. C. Dhingra. 1998. Rotary heat exchanger performance with axial heat dispersion. International Journal of Heat and Mass Transfer. 41(18):2857-2864.

Pascal, S., M. Dominique. 2008. Heat-and-mass transfers modelled for rotary desiccant dehumidifiers. Applied Energy. 85:128-142.

Patankar, S. V. 1980. Numerical Heat Transfer and Flui flow. Mc-Graw Hill Book Company, USA.

Power H. and L. Wrobel. 1995. Boundary Integral Methods in Fluid Mechanics. Computational Mechanics Publications. Southampton. UK.

Rengarajan, K., D. B. Shirey III and R.A. Raustad. 1996. Cost-effective HVAC Technologies to meet ASHRAE Standard 62-1989 in hot and humid climates. ASHRAE Transactions. 102:166-182.

Romie, F. E. 1988. Transient Response of Rotary Regenerators. Transactions of ASME. Journal of Heat Transfer. 110:836-840.

Sanaye, S. 2008. Optimum operation conditions of a rotary regenerator using genetic algorithm, Energy and Buildings. 40(9):1637-1642.

Sanaye, S. and H. Hajabdollahi. 2009. Multi-objective optimization of rotary regenerator using genetic algorithm. International Journal of Thermal Sciences. 48:1967-1977.

Sandira, A., S. Nikola, K. Ahmed, B. Indira. 2005. Numerical analysis of heat transfer and fluid flow in rotary regenerative air pre-heaters. Journal of Mechanical Engineering. 5:411-417.

Shah, R. K., E. K. Subbarao and R. A. Mashelkar. 1988. Heat Transfer Equipment Design. Hemisphere, New York.

Shiery, D. B. III, K. Rengarajan. 1996. Impact of ASHRAE Standard 62-1898 on small Florida offices. ASHRAE Transactions. 102:153-165.

Simonson, C. J. and R. W. Besant. 1997. Heat and moisture transfer in desiccant coated rotary energy exchangers Part 1- numerical model. International Journal of HVAC & R research. 3:325-350.

Simonson, C. J. and R. W. Besant. 1997. Heat and Moisture Transfer in Desiccant Coated Rotary Energy Exchangers: Part I. Numerical Model. HVAC & R Research. Volume 3: Issue 4.

Simonson, C. J. and R. W. Besant. 1998. Heat and moisture transfer in energy wheels during sorption, condensation and frosting conditions. ASME Journal of Heat Transfer. 120:699-708.

Simonson, C. J. and R.W. Besan. 1999. Energy wheel effectiveness: Part I- Development of dimensionless group. International Journal of Heat and Mass Transfer. 42:2161-2170.

Simonson, C. J., D. L. Ciepliski and R. W. Besant. 1999. Determining the performance of energy wheels Part II experimental data and numerical validation. ASHRAE Transactions. volume 105:188-206.

Skiepko, T. 1989. Effect of Parameter Values on Gas and Matrix Temperature Fields in Rotary Heat Exchangers. International Journal of Heat Mass Transfer. 32:1443-1472.

Smith, G. D. 1978. Numerical Solution of Partial Differential Equations. Oxford University Press, UK.

Stabat, P. and D. Marchio. 2009. Heat and mass transfer modeling in rotary desiccant dehumidifiers. Applied Energy. 86:762-771.

Stefano D. A., I. Manuel, M. J. Cesare and L. Calogero. 2014. Design Optimization of Heat Wheels for Energy Recovery in HVAC Systems. Energies. 7:7348-7367.

Stiesch, G., S. A. Klein and J. W. Mitchel. 1995. Performance of Rotary Heat and Mass Exchangers. International Journal of HVAC & R Research. 1(4):308-323.

Suden B. and L. Wang. 2003. Relevance of Heat Transfer and Heat Exchangers for Green House Emmissions. Advances in Heat Transfer Engineering. 101-112.

Toro, E. F. 1999. Riemann Solvers and Numerical Methods for Fluid Dynamics. Springer-Verlag.

Valancius R., A. Jurelionis and V. Dorosevas. 2013. Method for cost-benefit analysis of improved indoor climate conditions and reduced energy consumption in office buildings. Energies. 6:4591-4606.

Versteeg, H. K. and W. Malalasekara. 1995. An Introduction to Computational Fluid Dynamics, The Finite Volume Method. Longman Science & Technology. UK.

WorsÁe-Schmidt, P. 1991. Effect of Fresh Air Purging on the Efficiency of Energy Recovery from Exhaust Air in

Rotary Regenerators. International Journal of Refrigeration. 14(4):233-239.

Wu, X., H. Ye, J. Wang, J. He, J. Yang 2012. Effectiveness analysis and optimum design of the Rotary Regenerator for Thermo Photovoltaic System. Frontiers in Energy. 6(2):193-199.

Yilmaz, T. and O. B. Ukalaca. 2003 . Design of Regenerative Heat Exchangers. Heat Transfer Engineering. 24 (4):32-38.

Zafeiriou, E. and D. Wurz. 1996. Numerical Simulation of Heat Transfer Processes in Rotating Regenerators. VGB Kraftwerkstechnik. 76 (6).

Zheng , M., L. X. Du, D. Liao , W. X. Chu , Q.W. Wang, Y. Luo, Y. Sun. 2012. Investigation on pressure drop and heat transfer performances of plate-fin iron air preheater unit with experimental and Genetic Algorithm methods. Applied Energy. 92:725-732.