Three-dimensional simulation of rotary air preheater in steam power plant

Armin Heidari-Kaydan, Ebrahim Hajidavalloo

Mechanical Engineering Department, Shahid Chamran University, Ahvaz 61357, Iran

HIGHLIGHTS

- Three-dimensional thermal simulation of full-scale rotary air preheater is presented.
- Variation of isothermal lines in the air preheater are shown.
- Effect of separator plate on the temperature distribution is restricted to center of the matrix.
- Rotational speed of matrix has important role in the performance of preheater until certain limit.

ABSTRACT

In this study, thermal behavior of a full-scale rotary air preheater is investigated using three-dimensional approach and treating preheater matrix as a porous media. Mass, momentum and energy equations are solved using moving reference frame (MRF) to incorporate the effect of rotational speed of the matrix. Temperature distributions of the matrix at different conditions have been presented and the effect of essential parameters such as rotational speed of the matrix, fluid mass flow, matrix material and temperature of inlet air on the performance of preheater have been discussed. Numerical results which are confirmed by experimental data show the significant effect of rotational speed, separator plate, fluid flow rate on the performance and temperature distribution of preheater. Increasing the rotational speed of the air heater increases the efficiency up to certain limit, after which it does not significantly change. It was also found that the effect of material change on the efficiency is very limited.

© 2014 Elsevier Ltd. All rights reserved.

1. Introduction

Rotary air preheater is one of the important energy recovery systems in the steam power plant which was first introduced in 1920 by Ljungstrom [1]. It transfers heat from the hot fluid to the cold one by using a rotating matrix of compact plates as shown in Fig. 1. Considering the important effect of the air preheater on the cycle efficiency, there are many studies addressing preheater efficiency. Warren [2] published his studies on Ljungstrom as a particular type of air to air exchanger and base on the experimental results confirmed a minimum reduction of 10% in power plants fuel consumption. Skiepko [3,4] investigated the effects of heat conduction in the matrix, Peclet number and the length of the matrix on the preheater performance [5,6]. Investigating on the effect of separator plate on the preheater performance, Worsoe-Schmidt [7] stated that although the separator decreases the efficiency of the exchanger, but it cannot be removed due to its role in the reduction of the fluid leakage. Based on the several experimental and numerical analyses, Ghodispour and Sadrameli [8] studied the effect of mass flow rate and rotational speed of the matrix on the preheater performance and showed that the flow rate effect was more significant than the rotational speed.


Despite many studies in this area, there are more rooms for better understanding of the periodic nature of heat transfer process...
in the rotary preheater. For example, three-dimensional temperature distribution of a preheater has not accurately been presented yet. Furthermore, the effects of influential parameters on the temperature distribution of the matrix have not been addressed sufficiently. In the present study, using three-dimensional approach and considering rotary matrix as a porous media, the governing equations of a full scale rotary preheater located in Ramin power plant located in Iran is simulated to clarify the exact temperature distribution inside the preheater. Moreover, the effects of some variables such as rotational speed of the matrix, fluid mass flow rate, plates' material, and inlet fluid preheating on the temperature distribution and the exchanger performance are investigated.

2. Mathematical modeling and governing equations

Considering the narrow passages of fluids compared to the overall dimensions of the preheater [Fig. 2], a porous media approach can be used to simulate fluids flows in the air heater matrix [13–15]. Using this approach can reduce the computational time while maintaining the results with acceptable accuracy. By experimental measurements of the volume, weight, and dimensions of the compact plates within the rotary preheater matrix, it was found that the porosity in the hot and cold layers is 0.84 and 0.76, respectively.

The Reynolds number of flow in the porous media based on the Eq. (1) is 10.2, which is less than the critical Reynolds number of 100 [16] which indicate that the flow is laminar. In this equation, $D_H$ is the hydraulic diameter and $V$ is the velocity.

$$Re = \frac{V*D_H*\rho}{\mu}$$

To simulate the flow and heat transfer within the exchanger, Navier–Stokes equations in the porous medium can be used. Continuity and momentum equations are as follows [16]:

$$\gamma \frac{\partial \rho f}{\partial t} + \nabla (\rho f \mathbf{V}) = 0$$

$$\rho f \left[ \gamma^{-1}\frac{\partial \mathbf{V}}{\partial t} + \gamma^{-2} \left( \mathbf{V} \cdot \nabla \right) \mathbf{V} \right] = -\nabla P - \frac{\mu}{K} \nabla$$

Energy equations for the solid and liquid phases are given in Eqs. (4) and (5), respectively [16].

$$(1 - \gamma)(\rho c_P) \frac{\partial T_s}{\partial t} = (1 - \gamma)\nabla \cdot (k_s \nabla T_s) + (1 - \gamma)q_s^v$$

$$\gamma(\rho c_P)_{fl} \frac{\partial T_f}{\partial t} + (\rho c_P)_{fl} \mathbf{V} \cdot \nabla T_f = \gamma \nabla \cdot (k_f \nabla T_f) + \gamma q_f^v$$

In general, the energy equations for the liquid and solid phases must be solved separately (local thermal non equilibrium condition), but if the Sparrow number defined in Eq. (6) in a medium is above 100, local thermal equilibrium condition can be applied [17]. In the present medium the Sparrow number is 6061, so local thermal condition equation can be assumed.

$$Sp = \frac{2h_l^2}{K_m D_h}$$

$$\left(\rho c_P\right)_{m} \frac{\partial T_m}{\partial t} + \left(\rho c_P\right) \mathbf{V} \cdot \nabla T = \nabla \cdot \left(k_m \nabla T_m\right) + \gamma q_m^v$$

The efficiency of a rotary preheater is obtained by calculating the ratio of the exchanged energy to the maximum transferrable energy as follow [12,18]:

$$\varepsilon = \frac{\text{heat transferred}}{\text{maximum possible heat transferred}} = \frac{m_{\text{air, in}} c_{p,\text{air}}^* (T_{\text{air, out}} - T_{\text{air, in}})}{m_{\text{flue, in}} c_{p,\text{flue}}^* (T_{\text{flue, in}} - T_{\text{air, in}})}$$

To model the preheater, computational grids were used as shown in Fig. 3 and different grid sizes were employed for simulation as seen in Table 1. It was found grid number including 1,052,961 wedge cells is appropriate for the simulation and the results are independent of grid number. Boundary conditions were considered similar to that of real conditions. Inlet and outlet pressure conditions were used for the momentum equation boundary conditions and inlet temperatures of air and gas were used for energy equation boundary conditions. Also it was assumed that the surrounding wall of the air heater was insulated.

Moving Reference Frame (MRF) method was used to incorporate the effect of rotational speed of the matrix. Eqs. (9)–(11) were used for continuity, momentum, and energy equations, respectively. To solve the governing equations FLUENT 6.3 software was used employing SIMPLE algorithm to solve the Navier–Stokes equations.

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \rho \mathbf{V} = 0$$

$$\gamma \frac{\partial \rho f}{\partial t} + \nabla (\rho f \mathbf{V}) = 0$$

$$\gamma(\rho c_P)_{fi} \frac{\partial T_{fi}}{\partial t} + (\rho c_P)_{fi} \mathbf{V} \cdot \nabla T_{fi} = \gamma \nabla \cdot (k_f \nabla T_{fi}) + \gamma q_{fi}^v$$
These distribution at the vertical plane of the matrix is shown in Figs. 6 the matrix from cold end which is shown in Fig. 5. Temperature air temperature. The similar trends can be realized when view to rotational region, the matrix temperature approaches to the inlet matrix and consequently is heated less. As seen, at the end of results the other part of the outlet air passing over relatively cold transfer its energy to the air, its temperature decreases and as a appropriately and become warm but as the matrix rotates and over the hot matrix which is coming from the hot section is heated air outlet channel. In fact, that part of the outlet air which is passing outlet air) shows a gradual change in the rotational direction in the channel. However, the matrix temperature (and consequently distribution is almost uniform at the entrance surface of gas the hot end of the preheater is at the top. As seen, the temperature maximum error is 7.5%, which is acceptable.

The dimensional and operational data of the rotary preheater are summarized in Table 2.

3. Results and discussion

Table 3 shows the simulation results for outlet temperatures and mass flow rate and compares them with actual data obtained by measurement. As seen, the results are in good agreement and the maximum error is 7.5%, which is acceptable.

Fig. 4, shows the temperature distribution in the matrix where the hot end of the preheater is at the top. As seen, the temperature distribution is almost uniform at the entrance surface of gas channel. However, the matrix temperature (and consequently outlet air) shows a gradual change in the rotational direction in the air outlet channel. In fact, that part of the outlet air which is passing over the hot matrix which is coming from the hot section is heated appropriately and become warm but as the matrix rotates and transfer its energy to the air, its temperature decreases and as a results the other part of the outlet air passing over relatively cold matrix and consequently is heated less. As seen, at the end of rotational region, the matrix temperature approaches to the inlet air temperature. The similar trends can be realized when view to the matrix from cold end which is shown in Fig. 5. Temperature distribution at the vertical plane of the matrix is shown in Figs. 6–8. These figures show that the hottest and coldest temperatures are happen at the maximum radius of the matrix.

One of the important components of a rotary preheater is a separator plate which is located between the inlet and outlet section and separate the hot and cold fluids. The separator area is used for placing the radial seals to prevent the leakage between hot and cold fluids. The simulation results show that the separator plate has an important effect on the temperature distribution within the matrix. In the presence of separator plate, the temperature distributions seen from the hot and cold sections are illustrated in Fig. 9(a) and (b), respectively. As shown, the shape of temperature isotherms are circular for the surfaces of matrix located under the separator area, while the temperature distribution changes to radial shape for the other part of the matrix. If the separator plate is removed from the preheater, the temperature distribution would be quite radial as shown in Fig. 10. This difference shows how much the separator plate affects the temperature profile in the matrix. The reason behind this difference is the dominance of conduction heat transfer compared to the convection in the area under the separator plate surface. Reducing the separator plate area increases heat transfer compared to the convection in the area under the separator plate surface. Reducing the separator plate area increases the preheater efficiency because it increases the effective heat transfer surfaces between plates and fluids. However, due to the leakage problem, area reduction cannot be less than specific limit which is required for effective radial seal control.

4. The effect of rotational speed on the air preheater performance

One of the important parameter on the rotary preheater efficiency is the rotational speed of the matrix. Fig. 11 shows the effect of rotational speed of the preheater on the temperature distribution if viewed from the hot end. As shown, by increasing the rotational

\[
\frac{\partial}{\partial t} \left( \rho \mathbf{v} \right) + \nabla \cdot \left( \rho \mathbf{v} \mathbf{v} \right) + \nabla \left( \mathbf{v} \times \mathbf{T} - \mathbf{T} \right) = -\nabla p + \nabla \cdot \mathbf{F} \tag{10}
\]

\[
\frac{\partial}{\partial t} \left( \rho E_\mathrm{f} \right) + \nabla \cdot \left( \rho \mathbf{v} \mathbf{E}_\mathrm{f} \rho \mathbf{v} \right) = \nabla \cdot (k \nabla T + \nabla \cdot \mathbf{v} \mathbf{v}) + S_h \tag{11}
\]

The study of computational grid.

<table>
<thead>
<tr>
<th>Number of study</th>
<th>Number of cells</th>
<th>Temperature of air outlet, K</th>
<th>Temperature of gas outlet, K</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>789,016</td>
<td>550.28</td>
<td>426.02</td>
</tr>
<tr>
<td>2</td>
<td>904,020</td>
<td>553.52</td>
<td>424.39</td>
</tr>
<tr>
<td>3</td>
<td>1,052,960</td>
<td>552.63</td>
<td>423.45</td>
</tr>
<tr>
<td>4</td>
<td>1,397,108</td>
<td>553.79</td>
<td>423.30</td>
</tr>
</tbody>
</table>

Table 2

The rotary preheater technical characteristics.

<table>
<thead>
<tr>
<th>Diameter, mm</th>
<th>Diameter, mm</th>
<th>Diameter, mm</th>
<th>Diameter, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>9864</td>
<td>650</td>
<td>634.2</td>
<td>1.89</td>
</tr>
<tr>
<td>634.2</td>
<td>1250</td>
<td>348.9</td>
<td>2.97</td>
</tr>
<tr>
<td>1.89</td>
<td>348.9</td>
<td>2.97</td>
<td>1.65</td>
</tr>
</tbody>
</table>

Table 3

Comparison of the simulation results with Ramin power plant data.

<table>
<thead>
<tr>
<th>Outlet air mass flow rate, kg/s</th>
<th>Outlet gas temperature, K</th>
<th>Outlet air temperature, K</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>136.3</td>
<td>422.9</td>
<td>544.5</td>
<td>Numerical results</td>
</tr>
<tr>
<td>1.1%</td>
<td>7.5%</td>
<td>2%</td>
<td>Error%</td>
</tr>
<tr>
<td>138.0</td>
<td>393.4</td>
<td>534.3</td>
<td>Experimental results</td>
</tr>
</tbody>
</table>
speed, the hot area at the outlet section is increased which indicates a better heat transfer interaction.

The same pattern can be realized when viewed from the cold end, namely, by increasing the rotational speed, the temperature of the outlet air approached to temperature of the inlet gas.

Given the intermediating role of the matrix in the air preheater, it can be seen that by increasing the rotational speed, more energy is transferred between two fluids which led to more increase in the performance of the system. In fact, by increasing the rotational speed, the compact plates could not have the necessary time to reach to the temperature of the passing fluid. Therefore, the temperature difference between the fluid and the plates remain high which results better heat transfer interaction.

Fig. 12, shows the temperature distribution of the air preheater seen from side view of the cold end. As seen, by increasing the rotational speed of the matrix, a greater area of the matrix is exposed to the temperature increase which indicates a better heat transfer action. In addition, after 2 rpm speed, the top layer of the matrix leave the air passing channel without reaching to the inlet air temperature. This action is repeated conversely for the side view of hot channel.

Fig. 13 shows the effect of rotational speed of the matrix on the efficiency and temperatures outlet of fluids in the air preheater. As seen, by increasing the rotational speed the efficiency is greatly increased until certain limit and after that no significant increase is realized in the efficiency.

By increasing the rotational speed, the average outlet air temperature is increased while the average outlet gas temperature is decreased. The trend of the average air temperature is similar to that of the efficiency. As seen, after 4 rpm the difference between average temperatures is almost constant which is similar to the efficiency curve. Increasing the rotational speed increases the area available for heat transfer between hot and cold fluids for a fixed time interval which results an increase in the preheater efficiency.
On the other hand, it decreases the maximum temperature of the baskets when they are in the hot section and increases the minimum temperature of the baskets when they are in the cold section which results a decrease in the preheater efficiency. The balance between these two opposite effects determines the trend of efficiency curve.

The effect of rotational speed of the matrix on the maximum and minimum temperatures of the outlet fluids are shown in Fig. 14. Rotational speed has no significant effect on the minimum gas outlet temperature and the maximum air outlet temperature. The minimum gas outlet temperature is important due to the dew point formation in the preheater and matrix corrosion. Since rotational speed has no effect on the onset of fluid condensation and dew point, it can be concluded that the corrosion process could not be prevented by changing the rotation speed.

In the other side, the minimum outlet air temperature and the maximum outlet gas temperature are affected by the rotational speed, as seen in Fig. 14. The onset, where these effects are appeared, is approximately at 2 rpm when the matrix could not reach to the temperature of the inlet fluid in each channel. In addition, the minimum outlet air temperature and the maximum outlet gas temperature have the same trend to the average air and gas outlet temperatures, respectively.

The effect of speed on the minimum outlet air and the maximum outlet gas temperatures indicates that despite the rotational speed does not affect the onset of the condensation; it is influential on the exposed area of the condensation. The most corrosion is usually happened in the lower layer of the matrix when the cold matrix initially enters to the gas channel in the surfaces close to the perimeter.

5. The effect of air and gas flow rate

The other important parameter affecting the performance of a rotary preheater is the flow rate of fluids. Assuming the same flow rate for both fluids, the effect of flow rate on the efficiency and the...
Fig. 11. Effect of rotational speed on the temperature distribution seen from the hot end.

Fig. 12. Effect of rotational speed on the temperature distribution seen from side view.
average gas and air outlet temperature are shown in Fig. 15. By increasing the flow rate, the average air temperature is decreased while the average outlet gas temperature is increased, resulting a decrease in air preheater performance. This reduction is due to lack of matrix capacity for energy transfer.

6. The effect of material type

To study the effect of different plates’ material on the performance of air preheater, six different metals were used in the simulation as shown in Table 4.

The simulation result for air heater performance is presented in Fig. 16. As seen, stainless steel has the highest efficiency while copper show the least efficiency. Fig. 16 suggests that material with the least thermal diffusivity is appropriate for the air preheater considering only technical issue; however, the final selection of material depends on the many operational and economical parameters. It should be noted that the difference between efficiencies are not very significance.

7. The effect of inlet air temperature on the efficiency

To prevent the condensation on the matrix and formation of corrosive acids; one possible way is preheating the cold inlet air to reach a temperature between 65 and 85 °C by using a specific type of heat exchanger known as ‘clarifier’. Thermal energy for clarifier is supplied by steam coming from the turbine subsections. Although using clarifier results in lower power production in the power plant.
but from an economical point of view it has benefit for the power plant.

Based on the simulation results presented in Fig. 17, increasing inlet air temperature around 20 °C increases the average outlet gas temperature up to 13.9 °C and the average outlet air temperature around 6.1 °C. The exchanger efficiency is almost untouched and remains constant. Moreover, it could be shown that the minimum outlet gas temperature increases equal to the increase of air temperature.

The inlet gas temperature change has the same effect as the inlet air temperature on the average outlet gas and air temperatures. The simulation results presented in Fig. 18, shows the effect of inlet gas temperature on the average outlet gas and air temperatures. As seen, a 20 °C increase in the inlet gas temperature increases the average outlet gas and air temperature around 6.1 °C and 13.9 °C, respectively. This increase has no effect on the efficiency of rotary preheater.

8. Conclusion

Simulation of a rotary preheater was performed by considering the rotary matrix as a porous media and using FLUENT software. The results indicated that the isothermal lines within the matrix are almost linear except close to the center of the matrix. The matrix temperature is smoothly changed in the angular direction until approach to the inlet temperature of either air or hot gas. Existing of the separator plate changes the temperature distribution in the matrix especially close to the center. It was found that the rotational speed has significant effect on the preheater efficiency. Increasing the rotational speed, initially, increases the efficiency very rapidly up to a certain limit after that there is not any significant change. The effect of fluid flow rate on the performance of air heater was studied. It was shown that by increasing the air and gas flow rate the performance was decreased.

The rotational speed and fluid flow rates have no impact on the minimum outlet gas and the maximum outlet air temperatures. They are only effective in the minimum outlet air and the maximum outlet gas temperatures. By analyzing the materials used in the rotary preheater, it was found that material with low thermal diffusivity have better thermal efficiency. However, the material change had only a slight effect on the overall efficiency. The effects of air and gas inlet temperatures were also studied. The results showed that by increasing the inlet air temperature, the average temperature of the outlet gas increased 2.2 times more than the average temperature increase of the outlet air. This phenomenon helps in becoming far from the dew point temperature of the flow.

References


Nomenclature

\( c_p \): constant pressure specific heat (J kg\(^{-1}\) K\(^{-1}\))
\( D_h \): hydraulic diameter
\( E \): relative internal energy
\( F \): external body forces
\( h \): heat transfer coefficient (W m\(^{-2}\) K\(^{-1}\))
\( H \): relative total enthalpy
\( k \): thermal conductivity (W m\(^{-1}\) K\(^{-1}\))
\( K \): permeability (m\(^2\))
\( L \): porous layer thickness (m)
\( m \): mass flow rate of cold or hot fluid (kg s\(^{-1}\))
\( P \): pressure (pa)
\( q^0 \): heat production (W m\(^{-3}\))
\( Re \): Reynolds number
\( Sp \): Sparrow number
\( t \): time (s)
\( T \): temperature (K)

\( \Pi \): velocity of the moving frame relative to the inertial reference frame
\( v \): (u, v, w), seepage velocity (m s\(^{-1}\))
\( \pi \): absolute velocity
\( \Pi_r \): relative velocity
\( \Pi_t \): translational frame velocity
\( V \): fluid velocity (m s\(^{-1}\))

Greek letters

\( \omega \): angular velocity
\( \gamma \): porosity
\( \epsilon \): efficiency
\( \mu \): dynamic viscosity (N s m\(^{-2}\))
\( \rho \): density (kg m\(^{-3}\))
\( \tau \): relative stress tensor

Subscript

\( s \): solid
\( f \): fluid
\( m \): mixture
\( in \): outlet fluid
\( out \): inlet fluid