Computational investigation of fluid flow and heat transfer of an economizer by porous medium approach

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Abstract. Computation of fluid flow and heat transfer in an economizer is simulated by a porous medium approach, with plain tubes having a horizontal in-line arrangement and cross flow arrangement in a coal-fired thermal power plant. The economizer is a thermal mechanical device that captures waste heat from the thermal exhaust flue gasses through heat transfer surfaces to preheat boiler feed water. In order to evaluate the fluid flow and heat transfer on tubes, a numerical analysis on heat transfer performance is carried out on an 110 t/h MCR (Maximum continuous rating) boiler unit. In this study, thermal performance is investigated using the computational fluid dynamics (CFD) simulation using ANSYS FLUENT. The fouling factor ε and the overall heat transfer coefficient ψ are employed to evaluate the fluid flow and heat transfer. The model demands significant computational details for geometric modeling, grid generation, and numerical calculations to evaluate the thermal performance of an economizer. The simulation results show that the overall heat transfer coefficient 37.76 W/(m²K) and economizer coil side pressure drop of 0.2 (kg/cm²) are found to be conformity within the tolerable limits when compared with existing industrial economizer data.

1. Introduction

Due to the widespread utilization of heat exchangers in various industrial applications. Burning coal for electricity generation creates products of combustion which contain solid particles (fly-ash). These small solid particles are carried by the hot (flue) gas flow through the heat exchanger in a power utility boiler. The transported fly-ash can cause severe erosion to the heat exchanger boiler tubes, especially in the economizer. Such solid particles in suspension may also significantly influence the heat transfer characteristics of boiler heat exchangers [1].

Conventional thinking about how energy is extracted, converted, and consumed is being challenged by rising concerns about the environmental impacts of power generation on land, water, air quality, and climate. Currently, about half of our electricity is generated by coal-fired plants, which are responsible for more than 80% of carbon emissions from the power sector [2]. A comprehensive carbon management policy should both address ways to reduce emissions through a change in the fundamental energy supply structure which will take significant time and consecutively improve the performance of the existing infrastructure [2]. One such option to improve the existing energy infrastructure is to increase the thermal efficiency of existing coal-fired power plants. It has become important to improve the heat transfer characteristics of tubes in view of the energy situation [3].

The main equipment for recovering heat from exhaust flue gas is economizer in a coal-fired power plant. The economizer is a mechanical device used to preheat boiler feed water before it enters the boiler drum by utilizing the waste heat captured from boiler flue gasses [4]. The demand for reducing primary energy claims an improvement of the efficiency of heat exchangers, which contributes to the reduction of the CO₂-production [5].
There are plenty of techniques to increase heat transfer in tubular heat exchangers, such as rough surfaces, extended surfaces, surface vibration, fluid vibration, and so on. But porous media have a large contact surface with fluids, which can enhance the heat transfer process. A porous medium not only changes the condition of the flow field and makes the thermal boundary layer thinner, but also its thermal conductivity is usually higher than that of the studied fluid. Thus, adding a kind of highly conductive porous insert to a heat exchanger can effectively enhance its heat transfer characteristics and, consequently, we will be able to save the material used to construct the device and the space occupied by it [6]. The numerical investigation carried out in this study is a contribution to the research done for the heat transfer enhancement using porous media.

**Figure 1.** Economizer in a coal-fired Boiler (adapted from [7]).

Fig 1. shows the location of economizer in a coal-fired boiler. The economizer is typically the last water cooled heat transfer surface upstream of the air preheater. Flue gasses are the combustion exhaust gasses produced at power plants consist of mostly nitrogen, carbon dioxide, water vapor, soot carbon monoxide etc [8]. Frantisek et.al [9] investigated the flow and temperature fields in an economizer for the inline and staggered tube arrangement through numerical analysis. Patankar and Spalding [10] was perhaps the first one capable of calculating simultaneously the thermal fields in both fluids and in the tube metal and can be regarded as the framework from which successive contributions were developed.

1.1 Physical model, governing equations and numerical method

A schematic of the economizer is depicted in Fig 2. while the associated computational domain shown in Fig 3. some detailed geometric parameters for conducting simulations are tabulated in Table 1 and Table 2. The physical properties of flue gasses and feed water are tabulated in Table 3 and Table 4. As the flue gasses flows across the inline bare tube economizer is actually a quite complex interaction amid flow filed and obstacles. Hence some assumptions are made in the following to simplify the calculation:

1. Geometry is simplified into three different zones (Fluid, porous zone 1 and porous zone 2
2. All the water tubes are eliminated to reduce the complexity of the computational process.
3. All the heat lost from the flue gas is taken away by the water domain. This indicates that the losses in the models (heat loss to the surrounding atmosphere, thermal resistance due to fouling etc) are neglected.
4. Secondary fluid (water) is assumed to be stationary without any flow conditions being given. Water domain is modeled separate parallel to the flue gas domain (Dual Cell Method).
5. Viscous and inertial resistance for the porous zones is calculated based on the pressure drop recorded for the economizer.
6. Temperatures are fixed for the water domain based on calculating the heat released from the flue gas
**Figure 2.** Schematic diagram of multipass counter cross-flow economizer (own study)

**Figure 3.** Geometrical details of multipass counter cross-flow heat exchanger water flow circuit inside the heat exchanger $N_{\text{row}}=74$

**Table 1.** Detail geometric parameters of the economizer

| $d_i$ (mm) | $d_o$ (mm) | $P_L$ (mm) | $P_T$ (mm) | $n_t$ | $N_{\text{row}}$ | Tube material |
|------------|------------|------------|------------|-------|------------------|---------------|
| 55.3       | 50.8       | 95         | 135        | 20    | 35               | SA 210 GRA1   |

*Remarks:* $d_i =$ Tube inside diameter; $d_o =$ tube outside diameter; $P_L =$ longitudinal tube pitch; $P_T =$ transverse tube pitch; $n_t =$ number of tubes in row; $N_{\text{row}} =$ number of tube rows.

*Notes:* Tube layouts of the heat exchanger are the in-line layout. (SA 210 GRA1)
Table 2. Economizer specifications

| Parameters                        | Values                      |
|----------------------------------|-----------------------------|
| Type of economizer               | Plain tube, horizontal in-line |
| Type of flow                     | Multipass counter cross-flow |
| Location of the economizer       | Upstream of air preheater   |
| Total heating surface of an economizer | 1915 m²           |
| Economizer duty                  | 4.902Mmkcal/hr              |
| Total number of blocks           | 2                           |
| Economizer coil side pressure drop | 0.2 kg/cm²              |
| Economizer casing thickness      | 6 mm                        |

1.2 Data reduction and interpretation

1.2.1 Governing heat transfer equations

In this research, the effect of fly ash deposit on the heat exchanger performance is studied via thermal resistance of fouling. The following is the method for finding the thermal resistance due to fouling.

The heat transfer rate of a heat exchanger (Q) can be calculated from water side as

$$ Q = m_w (h_{w2} - h_{w1}) $$  \hspace{1cm} (1)

The heat transfer rate of the combustion gas is defined by

$$ Q = m_g c_{pg} (T_{g1} - T_{g2}) $$  \hspace{1cm} (2)

Heat conduction through the tube wall is known as

$$ Q = k A_{tot} \Delta T_{ln} $$  \hspace{1cm} (3)

Logarithmic mean temperature difference between input and output of the heat exchanger

$$ \Delta T_{ln} = \frac{(T_{g1} - T_{w2}) - (T_{g2} - T_{w1})}{ln(T_{g1} - T_{w2})/ln(T_{g2} - T_{w1})} $$  \hspace{1cm} (4)

The total thermal resistance can be calculated by

$$ \frac{1}{U A} = \frac{1}{h_i A_i} + \frac{\ln(d_o/d_i)}{2\pi k_i L} + \frac{1}{\eta_j h_o A_o} $$  \hspace{1cm} (5)

For multipass counter cross-flow with \((N_{row}=37\ or\ \alpha)\)

$$ \varepsilon_c = \frac{1-e^{-N_{row}U A(1-C_A)}}{1-C_A e^{-N_{row}U A(1-C_A)}} $$  \hspace{1cm} (6)

Basic law governing the flow of fluids through porous media is Darcy law

$$ Q = \frac{ka(p_b-p_a)}{\mu} $$  \hspace{1cm} (7)
Mass conservation
\[
\frac{d(\rho q)}{dx} + \frac{d(\rho q)}{dy} + \frac{d(\rho q)}{dz} = -\frac{d(\rho \phi)}{dt}
\]  
(8)

1.2.2 Governing equations of the fluid flow and heat transport

Continuity equation
\[
\frac{\partial \rho}{\partial t} - \frac{\partial (\rho u_i)}{\partial x_i} = 0
\]
(9)

Momentum equation
\[
\rho \left[ \frac{\partial u_j}{\partial t} + u_i \frac{\partial u_j}{\partial x_i} \right] = -\frac{\partial p}{\partial x_j} - \frac{\partial \tau_{ij}}{\partial x_i} + \rho g_j
\]
(10)

Where \( p \) is the static pressure, \( \tau \) is the stress tensor and \( \rho g \) is the gravitational body force.

\[
\tau = \mu \left( (\nabla v + \nabla v^T) - \frac{2}{3} \nabla v I \right)
\]
(11)

Where \( \mu \) is the molecular viscosity, \( I \) is the unit tensor and the second term on the right side is the effect of volume dilation.

Energy equation
\[
\rho \left[ \frac{\partial h}{\partial t} + u_i \frac{\partial h}{\partial x_i} \right] = \frac{\partial}{\partial x_i} \left[ \lambda \frac{\partial v}{\partial x_i} \right] - \tau_{ij} \frac{\partial u_j}{\partial x_i} + \frac{\partial p}{\partial t} + u_i \frac{\partial p}{\partial x_i}
\]
(12)

1.2.3 Porous medium formulation

Momentum equations for porous media

Porous media are modeled by the addition of a momentum source term to the standard fluid flow equations. The source term is composed of two parts: a viscous loss term and an inertial loss term.

\[
S_i = - \left( \sum D_{ij} \mu \frac{\partial v_j}{\partial x_i} + \sum C_{ij} \frac{1}{2} \rho |v| v_j \right)
\]
(13)

Where \( S_i \) is source term \( V \) is the magnitude of the velocity and \( D \) and \( C \) are matrices.

To recover the case of simple homogeneous porous media

\[
S_i = - \left( \frac{\mu}{\alpha} V_i + C_2 \frac{1}{2} \rho |v| v_j \right)
\]
(14)

Where \( \alpha \) is the permeability and \( C_2 \) is the inertial resistance.

Fluent allows the source term to be modeled as power law of the velocity magnitude

\[
S_i = -C_0 |V|^{C_1} = -C_0 |V|^{(C_1-1)} v_j
\]
(15)

Where \( C_0 \) and \( C_1 \) are user defined empirical coefficients.

### Table 3. Flue gasses specifications

| Parameters                                  | Values       |
|---------------------------------------------|--------------|
| Flue gas temperature entering economizer   | 370°C        |
| Flue gas temperature leaving economizer    | 259°C        |
| Flue gas pressure at economizer inlet      | 458.95 pa    |
| Flue gas pressure at economizer outlet     | 615.85 pa    |
| Flue gas velocity                          | 5.61 m/s     |
| Flue gas density                           | 1.31 kg/m³   |
| Flue gas mass flow in economizer           | 12280 kg/m³  |
### Table 4. Feed water specifications

| Parameters                              | Values                               |
|-----------------------------------------|--------------------------------------|
| Feed water temperature economizer inlet| 210°C                                |
| Feed water temperature economizer outlet| 260.9°C                              |
| Feed water pressure economizer inlet    | 135.1 kg/cm²                          |
| Feed water velocity inside the tubes    | 0.31 m/s                              |
| Total number of parallel paths on water side| 74                                   |
| Water flow to Economiser                | 110 t/h MCR (Maximum continuous rating) |

2. Methodology
The basic steps involved in solving any CFD problem are as follows:
- Identification of flow domain.
- Geometry Modeling.
- Grid generation.
- Specification of boundary conditions and initial conditions.
- Selection of solver parameters and convergence criteria.
- Results and post processing.

Figure 4. shows the overall view of the CFD approach.

2.1 Identification of flow domain
Before constructing modeling, it is required to understand the exact flow domain of an economizer. In the model, water flows through tubes and the surrounding space was flowing with flue gasses. Beside symmetry conditions, velocity inlet and pressure outlet were defined for both the zones. Convection and conduction heat transfer was considered and the suitable mesh was created for this purpose. A schematic of the economizer for multipass counter cross-flow inline bare tube arrangement is depicted Fig. 5.
2.2 Geometric modeling

The software that is used for generating the geometry and meshing is decided based on nature and complexity of the geometry. The most common and reliable economizer design is the bare tube, in-line horizontal multipass counter cross-flow economizer as shown in Figure 6. When a coal is fired, the fly ash (flue gasses) creates a high fouling and erosion environment. The advantage of the bare tube, the inline arrangement was it minimizes the erosion and trapping of ash. It also easiest geometry to kept clean by soot blowers. The computational domain of the inline bare tube economizer is shown in Fig 6 and the physical properties of porous media are tabulated in Table 5. and Fig 7.

Figure 5. Schematic diagrams of the economizer for multipass counter cross-flow inline bare tube arrangement (N_row=5)

Figure 6. Solid geometric model of an economizer with four viewports.

Table 5. Boundary conditions for porous media
| Parameters                  | Values                  |
|----------------------------|-------------------------|
| Porous media length        | 3m                      |
| Total heating surface area | 1915 m²                 |
| Volume of the porous zone  | 85.48 m³                |
| Overall heat transfer coefficient | 37.76 kcal/ m² hr deg C |

**Figure 7.** Geometric details of a model and Boundary conditions for porous zones 1 & 2

2.3 Grid generation.

Meshing is an integral part of the CFD analysis process. The mesh influences the accuracy, convergence, and speed of the solution. For the purpose of flow domain discretization, The economizer model considered for 3D grid generation. The mesh used in this study is shown in Fig 8. The grid dependency checked and the quality was 1, max skewness was 1E^-10, aspect ratio-1.114 and grid having 238896 cells. The grid generated with dual cell method.

**Figure 8.** Grid generation with dual cell method
2.4 Mesh details

| Parameters                  | Value | Limit |
|-----------------------------|-------|-------|
| Min orthogonal quality     | 1     | >0.2  |
| Max skewness                | $1 \times 10^{-10}$ | <0.8  |
| Aspect ratio                | 1.1142 | <70   |
| Number of cells             | 238896 |       |

2.5 Solver

To avoid excessive computational demands of full monolith modelling, this paper is an attempt to apply and demonstrate the porous medium approach for predicting the flow and heat transfer characteristics. The incompressible steady state form of governing equations for mass conservation, momentum balance, and energy conservation together with the equations for modelling turbulence are solved using Ansys Fluent 17.1. The pressure correlation approach using the SIMPLE algorithm is adopted.

3. Results and Discussions

Figure 9. Variation of contour pressure plots

Figures 9 and 10 shows the contours of the pressure of the flow from top to bottom. The variation of pressure across the porous zones 1 and 2 decreasing from 202.21 Pa to 5.45 Pa.

Figure 10. Pressure drop variation of flue gasses

Figure 11. Variation of contour temperature plot

Figure 12. Temperature drop variation of flue gasses
Figures 11 and 12 shows the contours of the temperature of the flow from top to bottom. The variation of temperature across the porous zones 1 and 2 decreasing from 651°K to 501°K, with a flue gas velocity of 5.61 m/s, the flow direction is from top to bottom.

4. Conclusions
The study numerically examines the geometric parameters of a multipass counter cross-flow economizer on the flue gas side performance of a 37-row bare tube inline arrangement economizer. The main purpose of this article was to show that CFD can be a helpful tool in analyzing the economizer. The gas is observed to be uniformly circulated at the heating surface. The predicted values of gas velocity, temperature, and pressure distributions through the heat transfer bundles indicate the acceptable distribution of flow in the heat transfer.

Numerical simulation has been conducted for a porosity range of 0.94. The results obtained show that the amount of heat transfer from the flue gasses to water from 651°K to 501°K with the heat transfer area of 1915 m² with a flue gas velocity of 5.61 m/s, and pressure variation from 202.21 Pa to 5.45 Pa. The forming of foulant is mostly on the front part of the bare tubes. The developed empirical model could be used to predict the thermal resistance due to the fouling. The simulation results show that the overall heat transfer coefficient 37.76 W/(m²K) and economizer coil side pressure drop of 0.2 (kg/cm²) are found to be conformity within the tolerable limits when compared with existing industrial economizer data.

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