Numerical Study to Improve Fluid Flow Patterns in HRSG by using a Turning Vane and a Combination of using a Turning Vane with Transition Zone Geometry Modification

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Abstract. Since operated from 1995 until 2018, the total steam production of HRSG 1.1 has decreased by 6.34 % and exhaust gas outlet temperature increased by 40.86 °C. From previous studies on HRSG it can be assumed that flue gas flows at HRSG 1.1 has nonuniform velocity and temperature distribution. This research is carried out by modeling and simulating HRSG as porous media using Computational Fluid Dynamic (CFD) software. The simulation will be carried out on a 3D model with the steady-state condition, use energy model, a heat exchanger (HE) model, and standard k-ε as a turbulence viscous model. Energy absorption in HE will be modeled by an ungroup macro model with fixed inlet temperature and NTU model. Simulations in HRSG are implemented at a corner angle of 6 degrees (original), adding a turning vane on the inlet duct and combination adding a turning vane on inlet duct with 12 degrees corner angle on transition zone. Model validation was accomplished comparing simulation data to power plant data. The result shows adding a corner angle in the transition zone and installing a turning vane in the curve bend minimizes secondary flow in the transition zone. Heat absorption increases 0.008% and exhaust gas outlet temperature decrease 0.04% compared to the existing model.

1. Introduction
Block 1 Muara Karang combined cycle power plant is located in Jakarta, Indonesia. It has 506 MW total capacity, consisting of a 3 x 107 MW (GE MS9001E) gas power plant plus a steam turbine with a capacity of 185 MW. It began commercial operation in 1992 and supplies electrical energy in Jakarta. Block 1 Muara Karang combined cycle power plant is equipped with 3 vertical HRSG (Heat Recovery Steam Generator), operated open cycle, and combined cycle. HRSG serves to transfer heat energy from flue gas into the water or steam. Since operated from 1995 until 2018, the total steam production of HRSG 1.1 has decreased by 6.34 % thus reducing the production of electricity in the steam turbine. Exhaust gas outlet temperature of the LP economizer increased 40.86 °C (40.86 K).

Distribution of flue gas on vertical HRSG without perforate plate especially when passing superheater and evaporator were nonuniform [1]. The authors [2] conducted research on horizontal HRSG, it was found that flue gas velocity distribution in HRSG influenced by swirl angle and flue gas velocity distribution has the same trend with flue gas temperature distribution in HRSG. The results of research [3] stated that the modification of transition zone geometry can make the flow more uniform.
Changing the bottom angle from 20° to 14° makes the flow more uniform compared to the existing condition. Changing the corner angle from perpendicular to 30° makes the flow more uniform than modifying the bottom corners. The combination of bottom angle 14° and corner angle 30° produces a more uniform flow than the existing condition. The existing condition of HRSG Muara Karang combined cycle power plant has a corner angle of 6° and bottom angle 18°. [3] also stated that the increase in the uniformity of flue gas flow will increase heat transfer efficiency in heat exchangers. The authors [4] conducted research on heat exchanger (HE) modeling and simulating method using CFD software. Simulation will be carried out on 3D model with steady state condition, use energy model, heat exchanger (HE) model and standard k-ε and wall function as turbulence viscous model. In this study, HRSG 1.1 of Block 1 Muara Karang combined cycle power plant is modeled as porous media to get pressure drop. Energy absorption in HE will be modeled by an ungroup macro model with fix inlet temperature and NTU model.

Model in present this in figure 1. The 3D model in figure 2 is based on actual size, size or dimension of HRSG can be seen in table 4. Flue gas coming into ducting HRSG comes from gas turbine outlet is perpendicular to inlet duct. HRSG ducting inlet modeled as mass flow inlet and HRSG outlet modeled as pressure outlet. Properties flue gas is depend on temperature. Table 2 show flue gas composition at inlet HRSG. Combinations of corner angle and turning vane can be seen in table 3. There are 3 cases to be studied. The simulation will be carried out when the 99.6 MW loaded gas turbine.

2. Modeling and CFD Simulation
Modeling and simulating method using CFD was widely used for velocity distribution investigation. The authors [1] conducted research on flue gas flow in HRSG at a laboratory scale where HRSG's inflow is perpendicular to inlet plane. The authors [1] also performed numerical simulations on vertical HRSG and found that the standard k-ε turbulent was more suitable for laboratory scale research. The author [2] conducted research on horizontal HRSG. The author [2] performs a 3D numerical simulation in which heat exchanger (HE) is modeled as porous media to obtain pressure drop and heat transfer. The research was conducted with varying corner angle and bottom angle. Numerical simulation was conducted by modelling HE as porous media. The heat transfer in HE was modeled by the NTU model. Turbulent model used is RNG k-ε turbulent model. The authors [7] conducted research horizontal HRSG where heat exchanger was modeled as porous media.

This research is carried out by modeling and simulating HRSG as porous media using Computational Fluid Dynamic (CFD) software. Simulation will be carried out on 3D model with steady state condition, use energy model, heat exchanger (HE) model and standard k-ε and wall function as turbulence viscous model. Properties flue gas is depend on temperature. Table 2 show flue gas composition at inlet HRSG. Combinations of corner angle and turning vane can be seen in table 3. There are 3 cases to be studied. The simulation will be carried out when the 99.6 MW loaded gas turbine.
Table 1. Heat, porosity, 1/α and C₂ Coefficient.

| Number of Tubes | Diameter (mm) | Heat adsorbed (%) | Porosity (%) | 1/α | C₂ |
|-----------------|--------------|--------------------|--------------|-----|----|
| HP SH2          | 261          | 31.8               | 3.22         | 0.745 | 1,463.27 | 52.36 |
| HP SH1          | 348          | 31.8               | 14.36        | 0.748 | 37,344.13 | 47.36 |
| HP Evap         | 1392         | 31.8               | 35.30        | 0.801 | 206,131.42 | 32.69 |
| LP SH           | 564          | 38.0               | 2.10         | 0.732 | 237,249.10 | 23.39 |
| HP eco          | 1222         | 31.8               | 16.51        | 0.623 | 792,431.53 | 33.57 |
| LP Evap         | 1128         | 31.8               | 13.68        | 0.716 | 963,252.97 | 25.87 |
| LP Eco          | 940          | 31.8               | 14.83        | 0.736 | 1,262,814.42 | 24.42 |
| Wall Losses     |              |                    |              |      | 1.79       |      |

Table 2. Exhaust gas condition to HRSG.

| Value                                         | Value |
|-----------------------------------------------|-------|
| Flowrate of flue gas (kg/s)                   | 389.86|
| Gas turbine outlet flue gas temperature (K)   | 821.940|
| HRSG inlet flue gas temperature (K)          | 814.45|
| HRSG inlet flue gas Pressure HRSG (Pa)       | 16753 |
| HRSG outlet flue gas temperature (K)         | 377.65|
| Mass of O₂ in flue gas (%) at 1 atm & 15 °C  | 16.6  |
| Mass of CO₂ in flue gas (%) at 1 atm & 15 °C | 4.37  |
| Mass of H₂O in flue gas (%) at 1 atm & 15 °C | 4.95  |
| Mass of N₂ in flue gas (%) at 1 atm & 15 °C  | 74.08 |

Table 3. Variation of cases.

| Case | β  | γ    | Turning Vane | Note                                        |
|------|----|------|--------------|---------------------------------------------|
| 1    | 6° | 18°  |              | Reference                                   |
| 2    | 6° | 18°  | 1            | Install turning vane in curve bend          |
| 3    | 12°| 18°  | 1            | Change corner angle and install turning vane in curve bend |

Table 4. Dimension of HRSG.

|                      | Length (m) | Width (m) | Height (m) | Diameter (m) |
|----------------------|------------|-----------|------------|--------------|
| Inlet                | 3.353      | 5.768     |            |              |
| Duct length          | 53.06      |           |            |              |
| HP SH 2 (HP Superheater 2) | 16.2 | 7.05 | 0.202 |
| HP SH 1 (HP Superheater 1) | 16.2 | 7.05 | 0.272 |
| HP Evap (HP Evaporator) | 16.2 | 7.05 | 1.113 |
| LP SH (LP Superheater) | 16.2 | 7.05 | 0.338 |
| HP Eco (HP Economizer) | 16.2 | 7.05 | 0.913 |
3. Heat Exchanger Modelling

Heat exchangers are modeled as PM to model pressure drop on the flue gas side. The heat transfer between the flue gas and auxiliary flow is modeled by an ungroup macro heat exchanger. Heat absorption by each heat exchanger (HE) is calculated from the reliability run test data. Heat absorption by each HE becomes a heat reference in the NTU model with constant inlet temperature. The heat absorbed by every HE was calculated based on equation (1). The heat absorbed by each HE can be seen in table 1, which will be used as a reference in the simulation.

\[
Q_{\text{absorption}} = \dot{m}_f \cdot (h_{\text{out}} - h_{\text{in}}) \\
Q_{\text{absorption}} = \dot{m}_f \cdot C_{pf} \cdot (T_{\text{out}} - T_{\text{in}})
\]

Where \(Q_{\text{absorption}}\) is the heat absorbed by the fluid in HE, \(\dot{m}_f\) is the fluid mass flow rate, \(h_{\text{out}}\) is the enthalpy of fluid coming out of HE and \(h_{\text{in}}\) is the enthalpy of fluid entering HE. After we know the heat absorbed by every HE, we can calculate the value of the fluid specific heat capacity \(C_{pf}\) that is used as input in auxiliary fluid (water or steam side). Specific heat capacity of the fluid can be calculated using equation (2). Where \(C_{pf}\) is the specific heat capacity of the fluid, \(T_{\text{out}}\) is the temperature of the fluid coming out HE and \(T_{\text{in}}\) is the temperature of the fluid entering HE, respectively.

\[
\frac{NTU}{C_{f\text{min}}} = \frac{UA}{C_{f\text{min}}} \\
C_{f\text{min}} = \min \left[ (\dot{m}_f C_{pf})_{\text{primary fluid}}, (\dot{m}_f C_{pf})_{\text{auxiliary fluid}} \right] \\
C_{f\text{max}} = \max \left[ (\dot{m}_f C_{pf})_{\text{primary fluid}}, (\dot{m}_f C_{pf})_{\text{auxiliary fluid}} \right] \\
C_f = \frac{C_{f\text{min}}}{C_{f\text{max}}} \\
C_{f\text{r}} = \frac{1}{C_{f\text{r}}} \left[ \frac{1}{NTU} \arctan\left(\frac{1}{C_{f\text{r}} NTU}\right) \right]^{-0.22} \\
\varepsilon_{\text{HE}} = 1 - e^{\left(-C_{f\text{r}} NTU\right)^{0.78}} \\
Q_m = \varepsilon_{\text{HE}} C_{f\text{min}} (T_{\text{in auxiliary}} - T_{\text{in primary}})
\]
\[ Q_{\text{absorption}} = \sum_{i=1}^{n} Q_m \]  

(9)

NTU models cannot be used to model phase changes because the evaporation process takes place at a constant fluid temperature. So that the specific heat becomes infinite. However, based on field data, there was a slight increase in temperature, which inlet temperature was sub cooled, so \( C_{\text{pfl}} \) was calculated based on that data. From data heat absorption in HE (table 1), NTU is can be calculated with equation (3), where \( U \) is overall coefficient heat transfer, \( A \) is heat transfer area. Heat absorption in HE model with NTU was calculated with equation (8) and equation (9). Before calculating heat absorption, the effectiveness of HE (\( \epsilon_{\text{HE}} \)) and the ratio of \( C_{\text{pfl}} \) to \( C_{\text{pmax}} \) should be calculated by equation (7) and equation (6). \( Q_m \) is the amount of heat absorbed by the macro used in the HE model.

The pressure drop of flue gas (primary fluid) through HE is modeled by the addition of \( S \), source momentum to the standard fluid equation. Additional source Momentum (\( S \)) consists of viscous loss (Darcy law) and inertial loss. Additional source momentum equations can be written as follows:

\[ S_i = -\left( \frac{\mu_{fg}}{\alpha} v_i + C_2 \frac{1}{2} \rho_{fg} u_i v_i \right) \]  

(10)

\[ \Delta p_{fg} = -S_i \Delta n_{pm} \]  

(11)

Where \( \Delta p_{fg} \) is the flue gas pressure drop, \( \rho_{fg} \) is the flue gas density, \( \mu_{fg} \) is the flue gas dynamic viscosity, \( \Delta n_{pm} \) is the thickness of porous media, \( v \) is velocity, respectively. To obtain the coefficient of \( \alpha \) (permeability) and coefficient \( C_2 \) (inertial resistance), it is necessary some data flow velocity to flue gas pressure drop. Flue gas pressure drop through staggered fin tube heat exchanger type is calculated based on equation (12) [5]:

\[ E_u = 2 \frac{\Delta p_{fg}}{\rho_{fg} u_i n} \left( \frac{d^*}{d_i} \right)^{0.3} \frac{C_n}{Re_{di}^{0.25}} \]  

(12)

Where \( u \) is the maximum velocity, \( n \) is the number of rows, \( d_i \) is the hydraulic mean diameter, \( d^* \) is relative diameter, \( Re_{di} \) is Reynold number, \( C_n \) is correction factor based on row. \( C_n \) is a coefficient that affected by the number of HE rows (table 5). From equation (13) and equation (12) above, we get the coefficients \( \alpha \) and \( C_2 \) listed in table 1.

| Number of row | 1   | 2   | 3   | 4   | 5   | 6   | 7   | 8   | 9   | 10  |
|---------------|-----|-----|-----|-----|-----|-----|-----|-----|-----|-----|
| \( C_n \)     | 2   | 1.6 | 1.429 | 1.332 | 1.269 | 1.224 | 1.192 | 1.168 | 1.149 | 1.134 |

3.1 Validation

For validation, data of simulation result was compared with reliability run data. Error data of simulation result compared with actual data was -1.997 % on HRSG outlet temperature (figure 9). Grid independence test was carried out on 3 size models. The 1st model is 2,107,497 mesh, 2nd model is 2,986,162 mesh and 3rd model is 3,095,202 mesh, respectively. The 1st model gave -2.02 % HRSG outlet temperature error, the 2nd model gave -1.997 % HRSG outlet temperature error and the 3rd model gave -2.01 % HRSG outlet temperature error. Temperature error with a negative sign means the temperature is lower than the actual temperature. Thus, the 2nd model (2986162 mesh) was used as a model.

4. Result and Discussion

Based on figure 3, the secondary flow at the corner gradually decreases when the 1st case is compared with the 2nd and 3rd cases. In cases 1 and 3, there is a secondary flow at the bottom. Secondary flow reduced when installing turning vane as in cases 2 and 3, especially in positions \( z = 0.1 \) and \( z = 0.6 \). Thus, changing the corner angle of the transition zone from 6° to 12° and installing 1 turning vane in curve bend can improve the flow distribution more uniform.
Figure 3. Distribution of flue gas temperature in x-y plane HRSG at z = 2.6 (middle).

Based on figures 4, 5 and 6 by installing 1 turning vane in the existing model able to expand the energy absorption area. Change corner angle from 6° to 12° and install 1 turning vane can produce a wider area of heat absorption, seen with wider reds (more evenly distributed temperature). Secondary flow causes the temperature of the flue gas to decrease, which causes the temperature of the flue gas when he first passes HE to decrease. From table 6, case 1 shows that the flue gas temperature when entering HP SH 2 was lower than cases 2 and 3.

Figure 4. Distribution of flue gas temperature in x-y plane transition zone at z = 2.6 (middle).

Figure 5. Distribution of flue gas temperature in x-y plane transition zone at z = 0.1 (close to west wall).
Based on figure 7, the velocity distribution in case 3 is more uniform than case 1, the most uniform velocity distribution is case 2. So, adding a turning vane makes the velocity distribution more uniform. Based on figure 8, the addition of turning vane on the curve bend side can reduce secondary flow at the bottom and when combined with the addition of the corner angle, it can reduce the secondary flow in the transition zone. Thus, installing turning vane on the curve bend side is able to make the velocity distribution in HP superheater more uniform.

**Figure 6.** Distribution of flue gas temperature in x-y plane transition zone at z = 5.1 (close to east wall).

**Figure 7.** HP SH 2 and HP SH 1 inlet velocity distribution.

**Figure 8.** Distribution of flue gas velocity in x-y plane HRSG at z = 2.6.
Table 6. Flue gas temperature.

| Temperature (K)                | Case 1   | Case 2   | Case 3   | Actual  |
|-------------------------------|----------|----------|----------|---------|
| GT outlet                      | 821.94   | 821.94   | 821.94   | 821.94  |
| HRSG inlet                    | 814.13   | 814.60   | 814.34   | 814.45  |
| HP SH2 outlet                 | 801.91   | 802.23   | 802.22   |         |
| HP SH1 outlet                 | 742.01   | 741.75   | 741.89   |         |
| HP Evap outlet                | 585.47   | 585.29   | 585.41   |         |
| LP SH outlet                  | 576.80   | 576.66   | 576.74   |         |
| HP Eco outlet                 | 502.71   | 502.45   | 502.51   |         |
| LP Evap outlet                | 437.73   | 437.64   | 437.61   |         |
| LP Eco outlet                 | 370.11   | 370.01   | 369.98   | 377.65  |

Based on table 6, the lowest flue gas exit HRSG temperature was 369.98 K. The lowest flue gas temperature out of HRSG was in case 3. Based on table 7, the largest energy absorption in HE is in case 3, energy absorption was 193,134.5 kW. From table 8, not all auxiliary temperatures coming out of every HE in HRSG in cases 2 and 3 were greater than case 1. It shows that not all HE has increased heat absorption, but overall that in cases 2 and 3 resulted in heat absorption which was greater than case 1 (existing model).

![Figure 9. Flue gas temperature in case 1 and actual.](image-url)
Table 7. Heat absorption in HE.

|                     | Case 1     | Case 2     | Case 3     |
|---------------------|------------|------------|------------|
| HP SH2              | 6,093.3    | 6,083.4    | 6,127.5    |
| HP SH1              | 27,252.3   | 27,459.3   | 27,394.3   |
| HP Evap             | 68,306.3   | 68,273.2   | 68,269.9   |
| LP SH               | 3,981.9    | 3,989.7    | 3,986.0    |
| HP Eco              | 31,780.1   | 31,749.6   | 31,768.2   |
| LP Evap             | 26,996.2   | 26,991.6   | 27,033.2   |
| LP Eco              | 28,571.3   | 28,558.8   | 28,555.5   |
| Total               | 192,981.4  | 193,105.6  | 193,134.5  |

Table 8. Auxiliary flow temperature out HE.

|                     | Auxiliary outlet temperature (K) |
|---------------------|----------------------------------|
|                     | Case 1   | Case 2   | Case 3   |
| HP SH2              | 797.09   | 797.00   | 797.37   |
| HP SH1              | 745.37   | 746.64   | 746.24   |
| HP Evap             | 577.38   | 577.38   | 577.38   |
| LP SH               | 585.00   | 585.30   | 585.16   |
| HP Eco              | 569.46   | 569.32   | 569.41   |
| LP Evap             | 435.70   | 435.70   | 435.72   |
| LP Eco              | 426.31   | 426.26   | 426.25   |

Case 1, 2, 3, respectively produces a relatively equal pressure drop (figure 10). The biggest pressure drop in HRSG occurs in case 3 (1,339.49 Pascal). In case 2, when only adding to the turning vane, there is a decrease in the pressure drop in the ducting of 0.29 Pascal or the pressure drop is relatively the same as case 1. Modifying a corner angle from 6° to 12° and installing 1 turning vane resulted in an increase in ducting pressure drop of 5.36 Pa compared to the existing model (figure 11). The pressure drop in ducting in case 3 is 73.94 Pascal. The resulting increase in pressure drop is not significant.

Figure 10. HRSG pressure drop.

Figure 11. Ducting pressure drop.

Figure 12. ΔQ absorption of each case to case 1.
| Fuel              | CO₂ (kg/Tera Joule) |
|------------------|---------------------|
|                  | Default  | Upper       | Lower      |
| Natural gas      | 56100    | 58300       | 54300      |
| Residual Fuel Oil| 77400    | 78800       | 75500      |
| Diesel oil       | 74100    | 74800       | 72600      |

Based on figure 12, the difference in heat absorption in HE between case 3 and case 1 (reference) is 153.1 kW. Assuming that the capacity factor of power plant for 1 GT is 100% for a year, CO₂ emission reduction based on equation (13) [6] and table 9 is 270.8 tons CO₂/year.

\[ E_{\text{emission,GHG,fuel}} = Fuel \text{ Consumption}_{\text{fuel}} \times E_{\text{mission,Factor,GHG,fuel}} \] (13)

**5. Summary**

PM model makes the HE numerical simulation becomes simpler. The PM model saves time for numerical simulation. PM model and HE model can model the heat absorption of the flue gas to the auxiliary flow in the heat exchanger.

This study proves that the PM model is able to describe heat transfer in heat exchangers and can describe flue gas flow behavior in heat exchangers.

The most even distribution of velocity and temperature in the HP superheater inlet is obtained by placing the turning vane in the curve bend. Adding a corner angle in the transition zone and installing a turning vane in the curve bend minimizes secondary flow in the transition zone. Adding a corner angle and installing a turning vane can improve heat absorption efficiency in the heat exchanger. An increase in heat absorption is 0.08% and a decrease in exhaust gas outlet temperature of LP economizer temperature by 0.04% compared to the existing model.

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