Numerical Study of Radiation Heat Transfer on Supplementary Firing with Gas Turbine Load Variations

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Abstract
In a combined cycle, it is rare to find the use of supplementary firing in Heat Recovery Steam Generator (HRSG). However, supplementary firing is actually responsible for 200%-400% steam production increase due to additional combustion process inside HRSG. This process evidently produced radiation heat transfer so it is appealing to conduct a further research on it. This study attempts to evaluate 3 dimensional radiation heat transfer in the industrial supplementary firing. Discrete Ordinates (DO) method and Weighted Sum-Gray Gases (WSGGM) model are combined to outline the radiation process, whilst non-premixed with Probability Density Function (PDF) Chemical Equilibrium model is used for combustion analysis. Evaluation is conducted using variations of load 50%, 75%, and 100% of Gas Turbine (GT) industrial datasheet. Validation is build upon load 100% of GT datasheet. Based on the evaluation, datasheet outcome shown that simulation possessed 2.90% and 5.28% deviation for temperature and pressure respectively. From wall heat flux radiation, this study found that load 50% of gas turbine variation had the best trend, are emit radiation (22413.50 W/m²) and absorb radiation (-11847.5 W/m²) compared to the other options available.

Keyword: Supplementary firing, Radiation, Load Gas Turbine,

1. Introduction
Combined cycle system applied to HRSG aims to improve the efficiency by utilizing exhaust gas from gas turbine. Which still having heat that can be used to heat the tube. There are some supplementary firing added in HRSG to increase the steam production to 200-400% [1]. Supplementary firing utilizes exhaust gas from gas turbine as oxidizer. The combustion types that occurred in supplementary firing is non-premixed. Then, the effect of this combustion is the radiation heat transfer.

Several previous studies already discuss about the importance and factors that increase the radiation heat transfer in combustion cases. Combustion cases [2] use some radiation model to know the experimental result of radiation heat transfer. In the industrial combustion, thermal radiation has important role when the temperature is very high. In addition, sometimes radiation transfer is often overlooked in the calculation process because it is considered too mathematically complex. Ignoring the radiation transfer especially in combustion system may lead to computation error [3]. In paper [2] shows the radiation heat transfer predominates on the furnace walls. The concentrations of combustion air will affect the radiation heat transfer. This is delivered on study [4] which analyzed the characteristic of combustion by varying the combustion air. Turbulence will also affect the combustion process. This is because of the influence of chemical reaction between air and fuel [5].

This study aims to know the radiation heat transfer on HRSG furnace wall using Computational Fluid Dynamic (CFD). There are 7 supplementary firing arranged vertically. Combustion air is obtained from gas turbine exhaust gas. There are 3 variations of load from GT that are, 50%, 75%, and 100%. Each load has different concentrations of air and inlet pressure. This will be discussed to see the effect of radiation on the furnace wall.

2. CFD model
2.1 Pressure Based and Solver
Simulation is conducted using equation Reynold-Averanged Navier Stokes (RANS) for the continuity (equation 1) and momentum (equation 2). Solver using double precisions pressure based
(segregated) for flowing liquid. Special discretization from scalar is made with second order upwind scheme and Pressure Staggering Option (PRESTO) for determine the pressure. These are selected because the fluid inside is assumed flowing with incompressible flow. Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) algorithm is used to pressure-velocity coupling [2,6,7]

Equation:
\[
\frac{\partial}{\partial x_j} (\rho u_i) = 0
\]
\[
\frac{\partial}{\partial x_j} (\rho u_i u_j) = - \frac{\partial \rho}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_k}{\partial x_k} \right) \right] + \frac{\partial}{\partial x_j} (-\rho u_i' u_j')
\]

2.2 Viscous k-ε Realizable

The value of k-ε realizable is better for non-premixed combustion cases. This is shown in experiments comparing k-ε standards and k-ε realizable [8] in oxy fuel combustion. In some combustion studies this turbulence used. Here is the k-ε Realizable equation:

\[
\frac{\partial (\rho \epsilon)}{\partial t} + \frac{\partial (\rho u_j \epsilon)}{\partial x_j} = \frac{\partial}{\partial x_j} \left\{ \mu \left( \frac{\partial \epsilon}{\partial x_j} + \frac{\partial \epsilon}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial \epsilon}{\partial x_k} \right) \right\} + \frac{\partial}{\partial x_j} \left( \frac{\epsilon}{\rho} \frac{\partial \rho}{\partial x_j} \right)
\]
\[
C_1 = \text{max} \left[ 0.43, \frac{\eta}{\eta + 5} \right]
\]
\[
\eta = \frac{\epsilon}{k}
\]
\[
S = \sqrt{\frac{\epsilon}{k}}
\]

Value \( \sigma_k = 1, \sigma_\epsilon = 1.2, C_2 = 1.9, C_{1t} = 1.44, \) and \( C_3k \) described with equation:

\[
C_{3k} = \tanh \left( \frac{\eta}{\eta + 5} \right)
\]

2.3 Radiative Transfer Equation

This model is used with specification from opaque wall inside the interior domain. The equation from this transfer radiation is for finite number from discrete solid angles [9]. The contribution from phase particle is shown in equation:

\[
\nabla I(\mathbf{s}) + (a + a_p + \sigma_t) I(\mathbf{s}) = a n^2 \frac{\epsilon \sigma_t^4}{\pi} + E_p + \frac{\sigma_p}{4\pi} + \int_0^{\Omega} I(\mathbf{s}) \Phi(\mathbf{s}, \Omega') d\Omega'
\]

Where the value of \( a, a_p \) are gas and particle from absorption coefficient, \( \sigma_t \) and \( \sigma_p \) are gas and particle from scattering coefficient, \( E_p \) is the equivalent from particle emission \( \sigma \) is Boltzmann constant, \( I \) is the intensity of radiation, \( T \) is the local temperature, \( \Phi \) is the scattering phase function and \( \Omega \) is solid angle. DO is used well at oxy fuel combustion in optical thickness[9,10,11]

2.4 Species non-premixed combustion

Non-premixed combustion is considered efficient because include some solution from transport equation for one or two scalar equation to be converted from equation for each species. Scalar conversion is mixture fraction that defined as thermo-chemical reaction of the fluid. Mixture fraction can be expressed with atomic mass fraction [9], as shown in equation

\[
f = \frac{Z_{i,\text{ox}} - Z_{i,\text{ox}}}{Z_{\text{fuel}} - Z_{i,\text{ox}}}
\]

This is the equation that shows the oxidizing agent from inlet flow and equation from fuel value of inlet flow. If the value of diffusion assumed to be same, the equation can be reduce/simplify to single equation for mixture fraction. (f). This equation is also suitable used in turbulent flow [9,12]. In this model, all thermo-chemical scalar (species fraction, density and temperature) is mixed to mixture
fraction. The form of this is PDF. This is used as a sub grid local fluctuations consideration among species and described by beta-function and PDF experiment process. [8,9]. Below is The Favre mean equation (density-averaged) mixture fraction:

\[
\frac{\partial}{\partial t} \langle \rho f \rangle + \nabla \cdot (\rho \vec{v} f) = \nabla \cdot \left[ \mu_t \nabla f \right] + S_m + S_{user} \tag{9}
\]

The value of \( S_m \) is mass transfer that happen from liquid phase or reaction from another phase to gas phase. If simplified, the equation become like this:

\[
\frac{\partial}{\partial t} \langle \rho f^{1/2} \rangle + \nabla \cdot (\rho \vec{v} f^{1/2}) = \nabla \cdot \left[ \frac{\mu_t}{\sigma_t} \nabla f^{1/2} \right] + C_g \mu_t \left( \nabla f \right)^2 - C_d \rho \frac{e}{k_f} f^{1/2} + S_{user} \tag{10}
\]

2.5 WSGGM (Weight Sum-Gray Gases Model)

WSGGM based is used to determine wall heat flux and radiative source terms to be expected [8,13]. An approach foo absorption from exhaust gas is need to be done to see spread inside the furnace. Here is the equation:

\[
\varepsilon = \sum_{i=0}^{l} \alpha_{e,i}(T) \left( 1 - e^{-k_i p s} \right) \tag{11}
\]

\( \alpha_{e,i} \) is the emissivity of the weighted factor for gray gases and quantity, \( k_i \) is the absorption coefficient of gray gases, \( p \) is the partial pressure of all the absorbed gases, \( s \) is path length [8].

3. Geometri dan Boundary Conditions

![Diagram of Furnace and Burner](image)

Figure 1: Furnace and burner are simplified in simulation.

The geometry of the simulation is a representation of the image in the data sheet. There are several forms in the furnace and a simplified burner. This is done to save computation time. However, in general it does not cause large error between the data sheet and the simulation.

There are 5 boundaries conditions that must be set to run the simulation. The inlet air has 2 boundaries with air inlet 2 is an additional of air inlet 1 form combustion air. The concentration of combustion air and the inlet pressure of each load are different. Air concentration and inlet pressure
can be seen in table 2. The hole nozzle is the way out of the fuel in table 2 and each variation load is set to have the same fuel concentration and inlet pressure. Wall is set with steel material. The outlet side has the same backflow temperature but different backflow pressure in table 2. This simulation has a tetrahedral mesh form. This mesh has 1334359 nodes and 7321245 elements.

Table 1. Boundary conditions input at simulation

| Boundary conditions | Nozzle burner | Air inlet 1 | Air inlet 2 | Wall furnace | Outlet |
|---------------------|---------------|-------------|-------------|--------------|--------|
| **Total pressure (Pa)** | 150.000 | Load 50% = 1716 | Load 50% = 1716 | - | Load 50% = 1500 |
|                     |               | Load 75% = 1814 | Load 75% = 1814 | Load 50% = 1500 | Load 75% = 1500 |
|                     |               | Load 100% = 2451 | Load 100% = 2451 | Load 100% = 2000 | Load 100% = 2000 |
| **Temperature (K)**  | 300 | Load 50% = 634 | Load 50% = 634 | 1000 | Load 50% = 1000 |
|                     |               | Load 75% = 682 | Load 75% = 682 | Load 75% = 1000 | Load 75% = 1000 |
|                     |               | Load 100% = 685 | Load 100% = 685 | Load 100% = 1000 | Load 100% = 1000 |
| **Tubulence model**  | Intensity and hydraulic diameter | Intensity and hydraulic diameter | Intensity and hydraulic diameter | - | Intensity and hydraulic diameter |
| **Hydraulic diameter (m)** | 0.01 | 1.67 | 0.185 | - | 6.49 |
| **Material**         | Mixture | Mixture | Mixture | steel | Mixture |

Table 2. Flue gas and fuel concentration inlet

| Flue gas concentration | 50% | 75% | 100% |
|------------------------|-----|-----|------|
| O2: 15,17%             | O2: 13,61 % | O2: 13,45% |
| N2: 73,46%             | N2: 72,92 % | N2: 72,56% |
| Ar: 0,88%              | Ar: 0,87 % | Ar: 0,87 % |
| CO2: 2,42%             | CO2: 3,17% | CO2: 3,24% |
| H2O: 8,07%             | H2O: 9,43% | H2O: 9,56% |

| Fuel concentration     |             |             |             |
|------------------------|-------------|-------------|-------------|
| CO2: 4,984%            |             |             |             |
| N2: 0,065%             |             |             |             |
| CH4: 86,039%           | CH4: 86,039%| CH4: 86,039%|
| C2H6: 3,974%           | C2H6: 3,974%| C2H6: 3,974%|
| C3H8: 2,808%           | C3H8: 2,808%| C3H8: 2,808%|
| IC4H10: 0,596%         | IC4H10: 0,596%| IC4H10: 0,596%|
| nC4H10: 0,710%         | nC4H10: 0,710%| nC4H10: 0,710%|
| IC5H12: 0,285%         | IC5H12: 0,285%| IC5H12: 0,285%|
| nC5H12: 0,186%         | nC5H12: 0,186%| nC5H12: 0,186%|
| C6H+: 0,353%           | C6H+: 0,353%| C6H+: 0,353%|

| Suppl. Firing air inlet temp. (K) | 634 | 682 | 685 |
|-----------------------------------|-----|-----|-----|
| **Gas pressure (pascal)**         | 150000 | 150000 | 150000 |

4. Result and Analysis

4.1 Validation model

Validation data use 2 parameters, are temperature and pressure. Those validation parameter use load 100%. From simulation result, point x=-3.75, y= 4.60, z=-1.26 used for temperature parameter.
Temperature read from the simulation is 1156.31 K while the value from data sheet is 1123 K, so the deviation values of temperature is 2.90%. The second validation use pressure at point x=-3.76, y=-3.83, z=-1.26, at that point the pressure value is 1894.39 Pascal, while the value from data sheet is 2000 Pascal, so the deviation value is 5.28%.

4.2 Numerical Result

There are 3 variation of loads from gas turbine. Each variation have different concentration and pressure. This affect the result of the combustion. At variation of load 50% have the highest concentration of O₂ (15.17%) which compared to other variations (75%=13.64% O₂, 100%=13.45% O₂). When the value of O₂ is high, the average reaction will also increase. This also shows the perfection of reaction between oxygen and fuel. This indication also shows at the increasing of temperature as a result of combustion reaction of CO₂ and H₂O [14]. The increase of temperature will affect to the radiation that received by the wall. At figure 2 shows at ranging are emitted (47849.558 W/m²) and absorbed (-53027.332 W/m²) radiation heat flux from the combustion. Dark color shows that it is closer to heat source, so the value will become smaller or negative.
The value of heat flux is the total from convection and radiation. From figure 3, load 50% has darkest color compared to load 75% and 100%. If we see from top side of the furnace, load increase makes the color become brighter. Load 100% has the biggest flue gas input pressure that is 2451 Pascal and highest input temperature (685 K). If we see from the data sheet, load 100% has the highest mass flow flue gas that is 129.7 kg/s. Load 75% and 50% each has 107.5 kg/s and 107.3 kg/s. Those show the convection value from load 100% is very high. Because the value of convection come from the heat flux minus the radiative heat flux. The value of radiative heat flux in load 100 % is smallest than others. On the other side, those could be more faster for cooling process inside the furnace. This is strengthened with low O$_2$ and high value of CO$_2$ and H$_2$O. Molecule CO$_2$ and H$_2$O will hamper the perfection of the combustion.
The first graphic shows the value of radiation wall that absorbed and emitted come back to another surface. The graphic above is taken at \( x=8.77, y=5.11, z=-1.18 \) as long as \( x \) which is at the center from above the furnace. Load 50\% shows the highest number of emit in (22413.5 W/m\(^2\)) and absorb radiation (-11847.50 W/m\(^2\)) the radiation of combustion in furnace. Even for the heat flux, load 50\% has the highest ranges, it means that total of the convection and radiation-emitting (37696 W/m\(^2\)) - absorbing (-2896.6 W/m\(^2\)) respectively.
5. Conclusion
This study aims to see the influence from variation of exhaust gas GT to radiation in the furnace. There are 3 loads, they are 50%, 75% and 100%. From the result of the analysis, this study conclude that load 50% emit radiation (22413.5 W/m²) and absorb (-11847.5 W/m²) higher than the other variations.

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References
[1] V. Ganapathy, Steam Generators and Waste Heat Boilers: For Process and Plant Engineers. 2014.
[2] M. A. Rajhi, R. Ben-Mansour, M. A. Habib, M. A. Nemitallah, and K. Andersson, “Evaluation of gas radiation models in CFD modeling of oxy-combustion,” Energy Convers. Manag., vol. 81, pp. 83–97, 2014.
[3] E. P. Keramida, H. H. Liakos, M. A. Founti, A. G. Boudouvis, and N. C. Markatos, “Radiative heat transfer in natural gas-fired furnaces,” Int. J. Heat Mass Transf., vol. 43, no. 10, pp. 1801–1809, 2000.
[4] K. Andersson and F. Johnsson, “Flame and radiation characteristics of gas-fired O2/CO2 combustion,” Fuel, vol. 86, no. 5–6, pp. 656–668, 2007.
[5] G. P. Merker, C. Schwarz, G. Stiesch, and F. Otto, Günter P. Merker · Christian Schwarz · Gunnar Stiesch · Frank Otto Simulating Combustion.
[6] R. Prieler, B. Mayr, M. Demuth, D. Spoljaric, and C. Hochenauer, “Application of the steady flamelet model on a lab-scale and an industrial furnace for different oxygen concentrations,” Energy, vol. 91, pp. 451–464, 2015.
[7] R. Prieler, B. Mayr, M. Demuth, B. Holleis, and C. Hochenauer, “Prediction of the heating characteristic of billets in a walking hearth type reheating furnace using CFD,” Int. J. Heat Mass Transf., vol. 92, pp. 675–688, 2016.
[8] M. Ghadamgahi, P. Ölund, T. Ekman, N. Andersson, and P. Jönsson, “A Comparative CFD Study on Simulating Flameless Oxy-Fuel Combustion in a Pilot-Scale Furnace,” J. Combust., vol. 2016, pp. 1–11, 2016.
[9] D. A. Granados, F. Chejne, J. M. Mejía, C. A. Gómez, A. Berrio, and W. J. Jurado, “Effect of flue gas recirculation during oxy-fuel combustion in a rotary cement kiln,” Energy, vol. 64, pp. 615–625, 2014.
[10] E. Karampinis, N. Nikolopoulos, A. Nikolopoulos, P. Grammelis, and E. Kakaras, “Numerical investigation Greek lignite/cardoon co-firing in a tangentially fired furnace,” Appl. Energy, vol. 97, pp. 514–524, 2012.
[11] H. B. Vuthaluru and R. Vuthaluru, “Control of ash related problems in a large scale tangentially fired boiler using CFD modelling,” Appl. Energy, vol. 87, no. 4, pp. 1418–1426, 2010.
[12] S. Septiawan, “Studi Numerik Karakteristik Pembakaran Natural Gas di dalam Boiler Furnace dengan Variasi Sudut Swirl Vanes pada Radially Stratified Flame Core Burners,” vol. 2, no. 1, 2013.
[13] R. Johansson, K. Andersson, B. Leckner, and H. Thunman, “Models for gaseous radiative heat transfer applied to oxy-fuel conditions in boilers,” Int. J. Heat Mass Transf., vol. 53, no. 1–3, pp. 220–230, 2010.
[14] M. A. Nemitallah and M. A. Habib, “Experimental and numerical investigations of an atmospheric diffusion oxy-combustion flame in a gas turbine model combustor,” Appl. Energy, vol. 111, pp. 401–415, 2013.