Study of Non-condensable Gas, Fouling, and Plugging Effects on Condenser Performance Using NTU-Effectiveness Analytic Method and 2D Numeric Simulation

H A Tamimi$^{1,3}$, Prabowo$^{2,4}$

$^{1}$PT Pembangkitan Jawa Bali Services, Sidoarjo, Indonesia
$^{2}$Mechanical Engineering Department, Sepuluh Nopember Institute of Technology, Surabaya, Indonesia
$^{3}$hilman@ptpjb.com
$^{4}$prabowo@me.its.ac.id

Abstract. This paper was objected to determine the effect and how sensitive the variation of the percentage of non-condensable gas, fouling thickness and the amount of plugging to the performance of the condenser and the steam turbine power output. This study was conducted using the NTU-effectiveness analytical method, and specifically for the study of the effects of non-condensable gas combined with numerical simulations to obtain the value of the condensation heat transfer coefficient as an input for analytical calculations. The study was conducted with a variation of the percentage of non-condensable gas 0% (new and clean), 2%, 5%, and 8%, with fouling thickness 0 µm, 2 µm, 5 µm, and 8 µm, and the amount of plugging 0%, 5%, 10%, and 15%. 2D numerical simulation was used to determine the impact of non-condensable gas percentage variation on the value of the condensation heat transfer coefficient on the outside of the tube, both qualitatively and quantitatively along the circumference of the condenser tube. The results of this study were, the greater the percentage of non-condensable gas, the thicker the fouling layer, and the more amount of plugging, would have an impact on decreasing the net turbine work. Consequently for new and clean conditions, the percentage of non-condensable gas were 2%, 5%, 8%, fouling thickness 2 um, 5 um, 8 um, and the amount of plugging were 5%, 10%, 15%, resulting in net turbine work to 202.55 MW, 192.08 MW, 189.08 MW (non-condensable gas), 200.05 MW, 196.01 MW, 192.08 MW (fouling), 195.2 MW, 185.27 MW, and 174.77 MW (plugging). A financial loss simulation was also carried out due to the decline in electricity sales affected by the decreasing of power plant’s net turbine work.

1. Introduction

The function of a condenser in a steam power plant is to condense the exhaust steam from a steam turbine using a cooling medium in the form of circulating water from sea water or river water. The condensed water from the condenser is then used again as boiler fill water. Condenser performance is monitored by the value of NTU - Effectiveness, and the resulting pressure. The value of the pressure on this condenser will affect the performance of the steam turbine. The decrease in condenser performance can have an impact on the decreasing output power of steam power plant. Some of the
factors that cause a decrease in condenser performance are non-condensable gas, fouling and plugging. Non-condensable is the result of air leakage into the condenser or caused by the reduced ability of the air ejector. Inlet air leakage occurs because the condenser pressure is below 1 atmosphere, so if there is a leak in the connections or lines that come in contact with atmospheric air, then the air from outside will enter the condenser. Whereas the nature of air is non-condensable, so it will reduce heat transfer in the condenser tube. Fouling can occur on the inside of the tube due to the quality of circulating water sourced from seawater. The cause of the formation of fouling is usually due to the mud content in the accumulated sea water to form deposits. Sedimentation conditions at the plant's intake canal greatly affect the amount of sludge carried by sea water. Formation of the fouling layer will increase the thermal conduction resistance. But on the other hand, the fouling layer on the inside of the tube will reduce the cross-sectional area of the tube condenser and will increase the speed of seawater fluid in the tube. This enhancing in fluid velocity will increase the coefficient of heat transfer convection, and decrease the thermal resistance of the inner tube convection. These two opposites will be examined in this paper, to find out which factors are more dominant between increasing the thermal resistance conduction of material fouling or increasing the speed of fluid in sea water, on the effectiveness of overall heat transfer in the condenser.

Plugging in the condenser tube occurs due to blockage by organic and inorganic waste in the form of shells, oysters, plastic, and so on. The location of the plant affects a lot of inorganic waste. The power plant which is located near the city will certainly have a large amount of inorganic waste at its canal intake. Plugging can also be caused by a tube that has leaked and made a blockage so that sea water does not pollute the boiler fill water. Plugging the tube reduces the condenser heat transfer area. However, on the other hand it will also increase the speed of water fluid in the tube due to the reduced number of tubes that can be flowed by sea water fluid, thereby increasing the coefficient of heat transfer convection, and will reduce the thermal resistance of the inner tube convection. This enhancing in fluid velocity will increase the coefficient of heat transfer convection, and decrease the thermal resistance of the inner tube convection. These two opposites will be examined in this paper, to find out which factors are more dominant between increasing the thermal resistance of material conduction fouling or increasing the speed of seawater fluid, on the effectiveness of overall heat transfer in the condenser.

Plugging in the condenser tube occurs due to blockage by organic and inorganic waste in the form of shells, oysters, plastic, and so on. The location of the plant affects a lot of inorganic waste. The power plant which is located near the city will certainly have a large amount of inorganic waste at its canal intake. Plugging can also be caused by a tube that has leaked and made a blockage so that sea water does not pollute the boiler fill water. Plugging the tube reduces the condenser heat transfer area. However, on the other hand it will also increase the speed of water fluid in the tube due to the reduced number of tubes that can be flowed by sea water fluid, thereby increasing the coefficient of heat transfer convection, and reducing the thermal resistance of the inner tube convection. Similar to fouling, this paper will study which factors influence the overall heat transfer effectiveness.

Several previous studies have been conducted on non-condensable gas, fouling and plugging on the condenser. Studies conducted by Maneesh Punetha and Sameer Khandekar (2017) showed that the addition of non-condensable gas boundary layers can reduce the heat transfer coefficient of condensation due to the extra heat transfer resistance formed from non-condensable gas boundary layers, but no visualization of contours of the mass flux condensation results from the simulation was displayed [1]. Dehby A and Guentay S conducted a simulation using analytic method on a vertical condenser tube, and concluded that the higher the non-condensable gas inlet fraction, the lower the local Nusselt Number will decrease [2]. In this studies, condenser tube was assumed to be a flat plate. Studies conducted by Said M and Sami I (2016) showed that the combination of the effects of an increase in the temperature of seawater cooling fluid and fouling factor reduced the power output and thermal efficiency of a power plant [3]. Bilal A Q and Syed M Z (2016) also examined the impact of fouling on the power system [4]. Jianlan Li et al (2016) examined the online fouling monitoring method to determine when the most appropriate condenser tube cleaning was performed based on the
degree of fouling [5]. This research was economically studied by Michael E Walker et al (2012), which quantified the economic impact of fouling, varied with fouling rate, plant load, cooling water inlet temperature, and condenser cleaning [6].

In this paper, a study will be conducted to compare variations in the percentage of non-condensable gas, fouling thickness and the amount of plugging, so that we can find out how significant and sensitive the effect of these variations on overall heat transfer effectiveness, including the effect on the power output of the power plant and its financial impact.

2. Methodology

This study used data from the Gresik steam power plant with a capacity of 200 MW located in Gresik City, East Java, Indonesia. The methodology used in this study was the NTU - Effectiveness analytical method. NTU-Effectiveness method is a method to determine the effectiveness of heat exchangers in transferring a certain amount of heat. The NTU (Number of Transfer Unit) is a dimensionless parameter and is commonly used to analyze heat exchangers. Effectiveness (ε) is the ratio between the actual heat transfer rate in the heat exchanger with the maximum possible heat transfer rate. In the study of the effect of non-condensable gas, a combination of numerical simulations was performed to obtain the value of the condensation heat transfer coefficient, which was then used to analytically calculate the thermal resistance of the outer tube convection in the presence of non-condensable gas. To validate the value of the condensation heat transfer coefficient numerical simulation results, a comparison must be done with the value of the heat transfer coefficient results of analytical calculations on the percentage of non-condensable gas at 0%.

In this study, compared between the calculation of condenser performance in design conditions (new and clean) with the percentage conditions of non-condensable gas with variations of 2%, 5%, and 8%, fouling conditions with fouling thickness variation 2m, 5m, and 8μm and plugging conditions with variations in the amount of plugging of 5%, 10%, and 15%.

Analytical Calculation Algorithm For Condenser Performance Under Design Conditions

The analytical calculation algorithm for condenser performance under design conditions was as follows:

a) Thermodynamic Properties of exhaust turbine vapor fluid, including saturation temperature (Tsat), specific volume (v), latent heat (hfg), density (ρ), heat capacity (cp), and thermal conductivity (k), and seawater properties as condenser cooling media, including density (ρ), viscosity (μ), heat capacity (cp), thermal conductivity (k), Figures Prandtl (Pr).

b) Geometry of tube condenser, including: the inner diameter of the tube (Di), the outer diameter of the tube (Do), the effective length of the tube (L), the number of tubes (N).

c) Calculating the thermal resistance of the inner tube convection, including: calculation of the cross-sectional area of the tube perpendicular to flow (Ai), fluid velocity of circulating water (vi), Reynold figure (ReD), Nusselt figure (NuD), Convection Heat Transfer Coefficient (hi), and thermal resistance in the inner tube convection (Ri). In the fouling condition, there was an increase in the speed of circulating water fluid, so that the thermal resistance of the inner tube convection would drop.

\[
\text{Re}_D = \frac{\rho v D_i}{\mu} \quad (1)
\]

\[
\text{Nu}_D = 0.023 \text{Re}_D^{4/5} \text{Pr}^n \quad (2)
\]

Where \( n = 0.4 \) for heating (\( T_s > T_m \)).

\[
\text{h}_i = \frac{\text{Nu} \times k}{D_i} \quad (3)
\]

\[
R_i = \frac{1}{\text{h}_i A_i} \quad (4)
\]

Where \( A_i \) was the surface area of the inner heat transfer in each tube.
d) Calculates the resistance of the tube wall thermal conduction ($R_w$)

$$R_w = \frac{\ln \left(\frac{D_o}{D_i}\right)}{2\pi k_w L}$$

Where $k_w$ was the thermal conductivity of the tube wall material. The tube wall material in this case was JIS C6871T (aluminium brass).

e) Calculating the thermal resistance of the outer tube convection, including: calculation of the number of vertical tubes in the direction of condensation flow ($N_v$), Jacob Figures (Ja), latent heat modification ($h_{fg'}$), average condensation convection coefficient per tube ($\bar{h}_{D,N}$), thermal resistance convection outside the tube ($R_o$).

$$Ja = \frac{c_{pl}(T_{sat} - T_s)}{h_{fg'}}$$

Where $c_{pl}$ was the heat capacity of constant pressure in the liquid phase, $T_s$ was the surface temperature of the condenser tube wall.

$$h_{fg'} = h_{fg}(1 + 0.68Ja)$$

Where $h_{fg'}$ was the latent heat modification value.

$$\bar{h}_{D,N} = 0.729 \left[\frac{g\rho_l(\mu_l-\rho_v)h_{fg}(1 + 0.68Ja)}{N_{D}(T_{sat} - T_s)D_o}\right]^{1/4}$$

$$R_o = \frac{1}{\bar{h}_{D,N}A_o}$$

Where $A_o$ was the surface area of the heat transfer outside of each tube.

f) Calculating the total thermal resistance ($R_t$)

$$R_t = R_i + R_w + R_o$$

gh) Calculating Overall Heat Transfer Coefficient ($U_o$), NTU, and Effectiveness ($\varepsilon$).

$$U_o = \frac{1}{R_iA_o}$$

$$NTU = \frac{U_oA_oN_v}{C_{min}}$$

Where $C_{min}$ was the Minimum Heat Capacity.

$$\varepsilon = 1 - \exp(-NTU)$$

$\varepsilon$ was the thermal resistances of the outer tube convection.

h) Calculating condenser pressure and temperature, including: calculation of maximum heat ($q_{max}$), actual heat ($q_{act}$), difference in enthalpy condenser ($\Delta h_{cond}$), enthalpy of liquid phase condenser ($h_l$). The condenser pressure and temperature were obtained using a steam table with reference to the enthalpy value of the liquid phase condenser under conditions of fouling.

$$\Delta h_{cond} = \frac{q_{act}}{\dot{m}_h}$$

Where $\dot{m}_h$ was fluid flow steam.

$$h_f = h_{cond} - \Delta h_{cond}$$

The condenser pressure and temperature were obtained from the steam table based on the $h_l$ value.

i) Calculating the turbine net work ($W_{nett}$ turbine), to determine the derating amount of the generator.

$$W_{nett\ turbine} = \Sigma(\dot{m}_T \times h_T) - \Sigma(\dot{m}_oT \times h_oT)$$

Where $\dot{m}_T$ was the mass steam flow entering the turbine, $\dot{m}_oT$ was the mass steam flow coming out of the turbine, $h_T$ was the enthalpy of steam entering the turbine, and $h_oT$ was the enthalpy of steam coming out of the turbine.

2.1. Numerical Simulation Effects of Non-condensable Gas Variations on Heat Transfer Condensation Coefficients

Numerical simulations using computational fluid dynamic were used to obtain the value of the heat transfer coefficient on the outside of the tube / condensation convection coefficient. Furthermore, this coefficient value was used to analytically calculate the thermal resistance of the outer tube convection in the presence of non-condensable gas. The geometry used was 2D geometry of several tube condensers that represent the entire arrangement of the condenser tube. This simulation aimed to see
the effect phenomenon of non-condensable gas on the outside of the tube, while for the inner side flow of the tube (circulating water flow) was carried out using conjugate heat transfer analysis, namely the coefficient of heat transfer in the inner side of the tube was done from analytical calculations.

There were 3 phases defined in the modeling, namely: Phase 1 was steam turbine exhaust, phase 2 was non-condensable gas, and phase 3 was condensate water. Types of interactions between phases were defined as follows: (1) Mass transfer from the turbine exhaust vapor phase to condensate water, with a condensation mechanism. (2) Between the turbine exhaust vapor phase and non-condensable gas the interaction did not occur.

In modeling the percentage of non-condensable gas effected on the exhaust turbine exhaust fluid, the fairly critical step was in determining the properties of a mixture of water vapor and non-condensable gas, with several variations in the percentage of non-condensable gas. The percentage value of non-condensable gas, was determined by adjusting the volume fraction value in phase 2 fluid (non-condensable gas), while the phase 1 volume fraction value (steam turbine exhaust) would adjust its fraction value. In this simulation used 4 non-condensable gas fraction values, namely 0 (no NCG), 0.02, 0.05, and 0.08.

There were two types of output data generated by the simulation, namely:

1) Qualitative data: was a visual display data of the condensation heat transfer coefficient contour.
2) Quantitative data: was a data of the quantity value of the heat transfer condensation coefficient on the outside condenser tube. The data were compared between non-condensable gas conditions of 0%, 2%, 5%, and 8% along the radial tube condenser direction curve.

Furthermore, the value of the condensation heat transfer coefficient was used to analytically calculate the performance of the condenser under conditions of non-condensable gas. The analytical calculation algorithm was the same as the analytical method of condenser performance under design conditions, only there was an increase in the thermal resistance of the outer tube convection due to the presence of non-condensable gas ($R_{o,NCG'}$).

2.2. Analytical Methods of Fouling Thickness Variations Effect
The analytical calculation algorithm was the same as the analytical method of condenser performance under design conditions, except that there was an increase in thermal conduction fouling resistance ($R_{c,f}$) and an increase in circulating water ($v_f$) fluid speed due to the reduction in the tube cross-sectional area ($A_{i,f}$).

The fouling material in this case was dominated by mud material carried by circulating water fluid. So the total thermal resistance under fouling conditions could be calculated ($R_{t,f'}$).

$$R_{t,f'} = R_{i,f'} + R_{if} + R_{w} + R_{o}$$

2.3. Analytical Methods of Variation in Number of Plugging Effect
The analytical calculation algorithm was the same as the analytical method of condenser performance design conditions, just that the calculation of the circulating water fluid speed in conditions where there was plugging, using the number of tubes that did not experience plugging ($N_p'$).

Tables 1 and 2 contain data on steam power plants used in analytical calculations [9].

| Table 1. Condenser Tube Geometry Data on Gresik Steam Power Plant Unit 4 |
|------------------|------------------|------------------|
| **No.** | **Geometry** | **Value** |
| 1 | Condenser tube outer diameter ($D_o$) | 0.025 m |
| 2 | Condenser tube inner diameter ($D_i$) | 0.0225 m |
| 3 | Condenser tube effective length ($L$) | 8.909 m |
| 4 | Tube total amount ($N_t$) | 15,136 |
| 5 | Tube per pass amount ($N_p$) from 2 total pass | 7,568 |
Table 2. Circulation Water and Steam Turbine Exhaust Data on Gresik Steam Power Plant Unit 4

| No. | Operation Parameter       | Value      | Operation Parameter       | Value      |
|-----|---------------------------|------------|---------------------------|------------|
| 1   | Pressure                  | 0.087 bar (a) | Pressure                  | 1.994 bar (a) |
| 2   | Steam Fluid Temperature   | 316.1 K    | Average temperature of inlet and outlet condenser | 307.82 K |
| 3   | Flow                      | 1,532.69 m³/s | Flow                      | 5.878 m³/s |
| 4   | Condenser tube surface temperature | 307.82 K | Flow                      | 5.878 m³/s |
| 5   | Film Temperature          | 311.96     |                           |            |

3. Result

Based on numerical simulations of variations in the percentage of non-condensable gas, obtained the heat transfer condensation coefficient value of the tube outside, both qualitatively (the contour of the heat transfer coefficient convection visual display/wall function coefficient of heat transfer), or quantitatively along the circumference of the condenser tube.

![Figure 1. Coefficient Contour of Heat Transfer Condensation on Variation of Non-condensable Gas Percentage](image)
**Figure 2.** Coefficient Contour of Heat Transfer Condensation in Non-condensable Gas Variations 0% and 8%

Based on the heat transfer condensation coefficient contours on the variation of the percentage of non-condensable gas above, it could be seen that at 0% condition (there was no non-condensable gas), the heat transfer coefficient value was highest when compared to the condition of non-condensable gas. This indicated that the presence of non-condensable gas in the exhaust turbine vapor would reduce heat transfer in the condenser tube.

![Coefficient Contour of Heat Transfer Condensation](image)

**Figure 3.** Heat Transfer Coefficient Condensation Value Along the Tube Surrounding

The graph above illustrated the value of Condensation Heat Transfer Coefficient at points on the surface around the tube in the radial direction ($\Theta = 0^\circ - 180^\circ$). Based on these quantitative data, it could be seen that the greater the percentage of non-condensable gas in the turbine exhaust vapors, the Condensation Heat Transfer Coefficient value would be lower.

![Heat Transfer Coefficient Condensation Value Along the Tube Surrounding](image)
The Condensation Conditions at Every Point Around the Tube

The Condensation Heat Transfer Coefficient value at $\theta = 0^\circ$ (the highest point on the condenser tube) was the lowest, because at that point the turbine exhaust steam first came into contact with the condenser tube wall and had not yet been condensed (still one phase). At $\theta = 85^\circ$, a gap between the non-condensable gas percentage began to widen, indicating that the condensation process had begun. The highest Heat Transfer Coefficient Condensation value was at $\theta = 135^\circ$ because at that point, the turbine exhaust vapor had changed phase to liquid, and there was accumulation of condensate before descending to the condenser tube below it.

Furthermore, based on analytical calculations, the condenser performance values were obtained under design conditions (new and clean), non-condensable gas, fouling, and plugging as follows:

Table 3. Condenser Performance, Net Turbine Work, and Financial Impact on Conditional Variations

| No. | Condenser Condition | $U_0$ (W/m$^2$K) | NTU | $\varepsilon$ | $P_{\text{cond}}$ (bar) | $W_{\text{nett Turbine}}$ (MW) | Financial Loss (Rp / Year) |
|-----|---------------------|-------------------|-----|-------------|-----------------|--------------------------|--------------------------|
| 1   | New and Clean       | 2,306.6           | 1.013 | 0.637       | 0.09            | 202.55                   | -                        |
|   |     |      |      |      |      |      |
|---|-----|------|------|------|------|------|
| 2 | NCG 2% | 2,211.87 | 0.972 | 0.622 | 0.171 | 192.08 | 13.8 Billions |
| 3 | NCG 5% | 2,153.89 | 0.959 | 0.617 | 0.204 | 189.08 | 17.76 Billions |
| 4 | NCG 8% | 2,154.89 | 0.947 | 0.612 | 0.246 | 185.85 | 22.03 Billions |
| 5 | Fouling 2µm | 2,284.19 | 1.004 | 0.633 | 0.105 | 200.05 | 3.31 Billions |
| 6 | Fouling 5µm | 2,251.36 | 0.989 | 0.628 | 0.135 | 196.01 | 8.63 Billions |
| 7 | Fouling 8µm | 2,219.08 | 0.975 | 0.623 | 0.171 | 192.08 | 13.8 Billions |
| 8 | Plugging 5% | 2,619.18 | 0.984 | 0.626 | 0.142 | 195.2 | 9.69 Billions |
| 9 | Plugging 10% | 2,653.05 | 0.944 | 0.611 | 0.255 | 185.27 | 22.78 Billions |
| 10 | Plugging 15% | 2,687.78 | 0.903 | 0.595 | 0.455 | 174.77 | 36.63 Billions |

**Figure 5.** NTU vs Effectiveness on Non-condensable Gas Condition
Figures 5, 6, and 7 showed the reduction in NTU and effectiveness in the presence of non-condensable gas, fouling and plugging compared to design conditions (new and clean).
4. Conclusion
Based on numerical simulations and analytical calculations the effect of non-condensable gas on the side of the shell condenser, it was concluded that the greater the percentage of non-condensable gas, the condensation heat transfer coefficient would decrease, and the value of NTU and effectiveness also decrease.

Based on analytical calculations the effect of fouling thickness variation, it was concluded that the thicker fouling layer on the inside of the condenser tube, the condenser performance value, namely NTU and effectiveness decrease, it could be concluded that the effect of adding thermal conduction resistance due to fouling was more dominant than increasing fluid velocity circulating water in terms of influence on the effectiveness of heat transfer. This decrease in effectiveness had an impact on reducing the work of the turbine output. The variation of fouling thickness in the order of the unit turned out to have a significant reduction in turbine performance.

Based on analytical calculations the effect of variations in the amount of plugging, it was concluded that the more the amount of plugging in the condenser tube, the condenser performance value, namely NTU and effectiveness the more they decrease, it could be concluded that the effect of reducing the heat transfer area due to plugging was more dominant than increasing the speed of circulating fluid water in terms of influence on the effectiveness of heat transfer.

Because of the significant effect caused by the presence of non-condensable gas, fouling, and plugging on condenser, preventive actions need to be taken. Preventive actions are taken to prevent the decrease in condenser performance due to the presence of non-condensable gas in the turbine exhaust vapor, including the need for monitoring non-condensable gas parameters, if there is an indication of an increase in the percentage of non-condensable gas in the turbine exhaust vapor. An indication of the increase in non-condensable gas percentage can be monitored by dissolved oxygen values in condensate water condenser. Preventive actions need to be taken to prevent condenser performance degradation due to fouling, including monitoring the thickness of the fouling on the tube by routinely measuring the thickness of the fouling during overhaul inspection. Preventive action for plugging prevention also needs to be done, for example routine backwashing condenser or routine cleaning of tube sheets from rubbish that clogs the condenser tubes when the condenser experiences out service.

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