Implementation of Humid Air Turbine for Combined Cycle Power Plant

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Abstract

The humid air turbine is a regenerative power cycle using high-humidity air. This system improves the gas turbine's performance. This research investigates the performance improvement for utilizing humidified air ona combined cycle power plant. There are 3 modification models proposed for the modification. Model 1 is a combined cycle utilizing a humidifier. Model 2 is a combined cycle operating with low pressure steam injection at the combustor inlet and model 3 is a combined cycle operating with high pressure steam injection at the combustor inlet. The effects of pressure ratio, turbine inlet temperature, cold fluid recuperator outlet temperature and mass of steam injected on the thermal efficiency and the net power produced will be discussed. A performance comparison between the proposed models and the original combined cycle will be studied.The current thermal efficiency and net power produced bythe original combined cycle are 48.94% and 603.49 MW, respectively. The simulation of model 1 yields maximum thermal efficiency of 46.36% and net power produced of 696.45 MW for turbine inlet temperature 1070℃ and pressure ratio 16.Model 2 has thermal efficiency of 50.43% and net power produced of 573.33 MW for 5% injected steam at pressure ratio 20, and model 3yields thermal efficiency of 49.97% and net power produced of 570.76 MW for 5% injected steam at pressure ratio 18.

Keywords: Gas Turbine, Modification, Combined Cycle, Humidified Air, High Thermal Efficiency

1.Introduction

The world's electrical energy demand is increasing with the rate of 1.7% annually [1**]**.The source of electrical energy generation in the world is dominated by gas turbine which accounted for 48% of total source of electrical energy [2]. This domination of gas turbine in the electrical energy generation is caused by gas turbine's characteristics. Compared with coal fired steam turbine, gas turbine has a higher thermal efficiency, lower investment cost, shorter start-up time, lower electrical power production cost, and lower SOx and NOx emission [3].

Today gas turbine still relies on fossil fuel. The recent research shows that the hydrate gas which is abundant in the earth's crust can be utilized as the fuel for gas turbine [4]. If in the future this hydrate gas is widely used for gas turbine's fuel, then the source of electrical energy generation will still be dominated by gas turbine. On the other hand, modified combined cycle power plant using humidified air is an integrated solution to increase the gas turbine's performance. Combined cycle increases the thermal efficiency and net power produced by a gas turbine but the operation characteristics can still be improved further.

One of the way to increase the performance of the combined cycle is tohumidify the air entering combustor inlet in Brayton cycle using water. This effort can be achieved by humidying the air using humidifier or by injecting the steam from the Rankine cycle. The humidified power cycle has several advantages over non-humidified power cycle, which are higher specific net power produced, higher thermal efficiency, can be utilized to cogenerate electricity and process steam, and be able to deliver better performance for partial electrical load [5, 6, 7, 8, 9, 10]. This power cycle is suitable for small scale industry to high scale electrical power generation.

In Indonesia, the demand of electrical power is increasing with the rate of 7% annually [11]. Jakarta as the capital of Indonesia hold multiple roles as governmental centre, business, and commerce. The self-sufficient electrical energy for Jakarta is critical. One of the power plants supplying electricity for Jakarta and surrounding is PLTGU Tanjung Priok. In order to serve the increase in electrical energy demand, PLTGU Tanjung Priok must improve its performance. As a case study, this paper usesthe operation data of PLTGU Tanjung Priok for the basic model for the humid air modification.

2.Research Methodology

This paper covers the thermodynamics analysis of the modified combined cycle. To study the effect of the humidified air in combined cycle, the thermodynamics properties of working fluid are estimated using Helmholtz energy equation of state. This equation works for pure fluid and for the mixture fluid. The Helmholtz energy equation of state is described in the equation (1).

$$\frac{A}{RT} = \alpha(\tau,\delta) = \alpha^0(\tau,\delta) + \alpha^{\mathrm{r}}(\tau,\delta) \qquad (1)$$

The basic Helmholtz energy equation of state for the fluids used in the modified cycle refers to the papers presented in Table 1.

Table 1. Thermodynamic model of fluids

Fluids	Author	Year
Water	Wagner and Pruβ [12]	2002
Air	Lemmon & Jacobsen [13]	2000

The mixing of two different fluids or more is assumed to be following ideal mixing rule. The mixture fluids will be analysed using Helmholtz energy equation of state for mixture by modifying term $\alpha^0(\tau,\delta)$ and $\alpha^r(\tau,\delta)$ in equation (1) to be following equations (2) and (3) consecutively.

$$\alpha^0 = \alpha^0(T,V,\mathbf{x}) = \sum_{i=1}^{l} x_i \alpha_i^0(T,V) + \sum_{i=1}^{l} x_i \ln x_i \qquad (2)$$

$$\alpha^{\mathrm{r}}(\tau,\delta,\mathbf{x}) = \sum_{i=1}^{l} x_i \alpha_i^r(\tau,\delta) + \Delta\alpha^{\mathrm{r}}(\tau,\delta,\mathbf{x}) \qquad (3)$$

Inside the combustor, the air humidity doesn't interfere with the combustion stability [14], as long as the humidity ratio of the air entering combustor is below 20%. The air humidity ratio in this simulation doesn't reach 20%. The basic model equation of the combustion (unbalanced) is shown in the equation (4).

$$air + fuel + H_2\mathrm{O} \rightarrow CO_2 + H_2O + N_2 + Ar \qquad (4)$$

For the engineering calculation, air is assumed to be the mixture of nitrogen, oxygen, and argon with mole fraction of 0.7812, 0.2096, and 0.0092, respectively. The fuel used in this research is High Speed Diesel (HSD). It is a mixture of carbon and hydrogen with the mole fraction of 0.30584 and 0.69416consecutively [15]. In this research, the fuel mass flow rate is iterated to match the turbine inlet temperature for each variation in certain operation parameter.

The optimization performed in this research is not the rigorous mathematic approach. The term "optimization" in this research is closely related to vary certain operation parameters and choose the onesthat yields the highest performance. The optimized operation parametersin this research are pressure ratio, gas turbine inlet temperature, cold fluid recuperator outlet temperature, and the mass of high and low pressure steam injected into the Brayton cycle.

3.Object of Case Study

As the object of case study in this paper, the operation data of the modified cycle follows the operation data of PLTGU Tanjung Priok at base load operation. The operation data of PLTGU Tanjung Priok is shown in the Table 2 for gas turbine and Table 3 for steam turbine. This system consists of three gas turbines, a low pressure steam turbine, and a high pressure steam turbine.

Table 2. Operation data for gas turbine

No.	Operation parameters	Value
1	Inlet temperature	30℃
2	Inlet pressure	1bar
3	Inlet relative humidity	83%
4	Inlet air flow rate	1416 kg/s
5	Combustorinlethumidity ratio	20%
6	Fuel consumption	35.4 kg/s
7	HSD Low Heating Value (LHV)	42 MJ/kg
8	Combustion efficiency	95%
9	Gas turbine isentropic efficiency	87.05%
10	Compressor isentropic efficiency	90.2%
11	Recuperator effectiveness	95%
12	Pressure ratio	12
12	Turbine Inlet Temperature	1070℃

Table 3. Operation data for steam turbine

No.	Operation parameters	Value
1	HP steam temperature	482℃
2	HP steam pressure	60 bar
3	HP steam flow rate	167.7 kg/s
4	LP steam temperature	135℃
5	LP steam pressure	31 bar
6	LP steam flow rate	184.6 kg/s
7	Condenser pressure	0.085 bar
8	Condensat pump pressure	11 bar

Table 3. (Continued)

No.	Operation parameters	Value
9	Steam turbine isentropic efficiency	90.5%
10	Pump isentropic efficiency	85%

The thermodynamic analysis for PLTGU Tanjung Priok using data shown in Tables 2 and 3 yields thermal efficiency of 48.94% and net power produced of 603.49 MW.

4.Modification Model 1

Model 1 is a combined cycle utilizing a humidifier. This modification model is indicated in figure 1. As shown in the figure 1, there are additional components introduced in this modification model. These are aftercooler, humidifier, economizer, and recuperator. Compressed air is leaving compressor and then entering aftercooler evaporating make-up water. After that, the air is humidified inside humidifier. Inside humidifier, air is direct-contacted with water. Leaving the humidifier, the humidity ratio of air reaches 20%. The humid air enters recuperator gaining the heat from combustion gas. After that, the humid air enters combustor and reacts with fuel producing combustion gas. Air then enters recuperator and after that air enters economizer evaporating the make-up water for humidifying air. The temperature of combustion gas leaving economizer is so low that it cannot evaporate the high pressure water in the Rankine cycle.

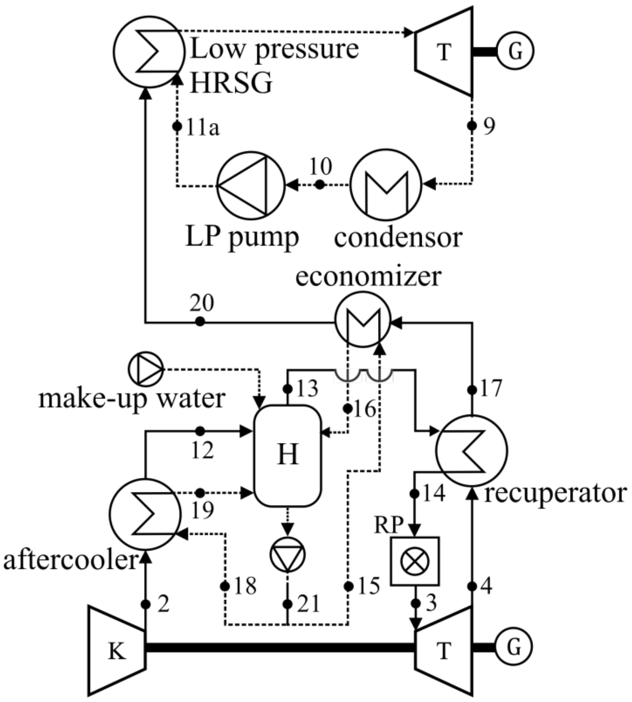

Figure1. Modification model 1.

5.Modification Model 2

The schematic of model 2 is shown in figure 2.The main major difference from the original combined cycle is that a portion of low pressure steam is injected into the combustor inlet. There will be an optimized condition for this power cycle at a certain amount of low pressure injected steam.

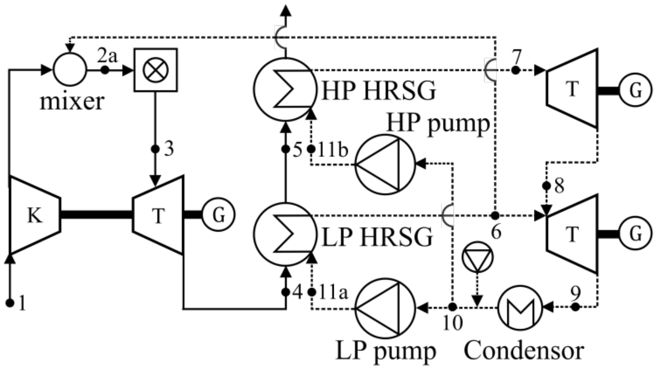

Figure 2. Modification model 2.

6.Modification Model 3

Model 3 is a combined cycle operating with a portion of high pressure steam generated by HRSG is extracted for the injection. The schematic for this power cycle is indicated in the figure 3.

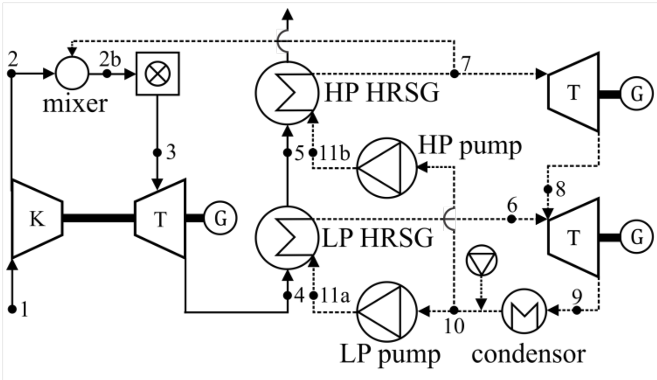

Figure 3. Modification model 3.

For this modification, there will be an optimized condition at a certain amount of high pressure steam injected.

7.Results and Discussions

The studied parameters for modification model 1 are cold fluid recuperator outlet temperature, pressure ratio, and turbine inlet temperature. Cold fluid is the humidified air leaving the humidifier and hot fluid is the combustion gas leaving gas turbine. For the model 2 and 3, the studied parameters are pressure ratio and mass of the injected steam.

7.1. Cold Fluid Temperature Leaving Recuperator

The effect of the cold fluid temperature leaving recuperator on the thermal efficiency is shown on figure 4.

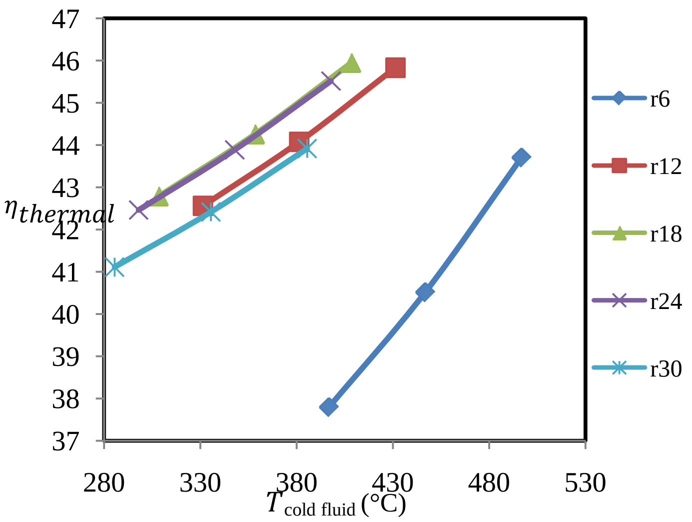

Figure 4. Effect of cold fluid temperature.

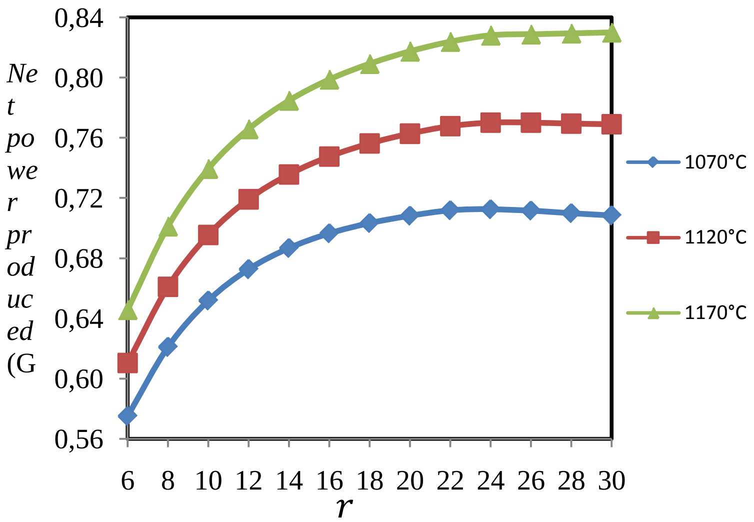

Figure 6. Net power produced by model 1.

Higher temperature of cold fluid leaving recuperator causes higher thermal efficiency. With the high temperature of cold fluid leaving recuperator, the mass flow rate of fuel entering combustor is reduced to achieve the same turbine inlet temperature. Therefore, the thermal efficiency finally increases.

7.2. Performanceof Model 1

Performance of model 1 is shown on the figure 5 and 6. As shown on these figures, at turbine inlet temperature 1070°C, maximum thermal efficiency is 46.36% with net power produced 696.45 MW at pressure ratio 16. At turbine inlet temperature 1120°C, maximum thermal efficiency is 47.72% with net power produced 746.49 MW at pressure ratio 16. At turbine inlet temperature 1170°C, maximum thermal efficiency is 49.36% with net power produced 809.22 MW at pressure ratio 18. Generally, higher the turbine inlet temperature, makes higher the thermal efficiency and greater net power produced. The optimized condition tends to shift up into higher pressure ratio as the turbine inlet temperature increased.

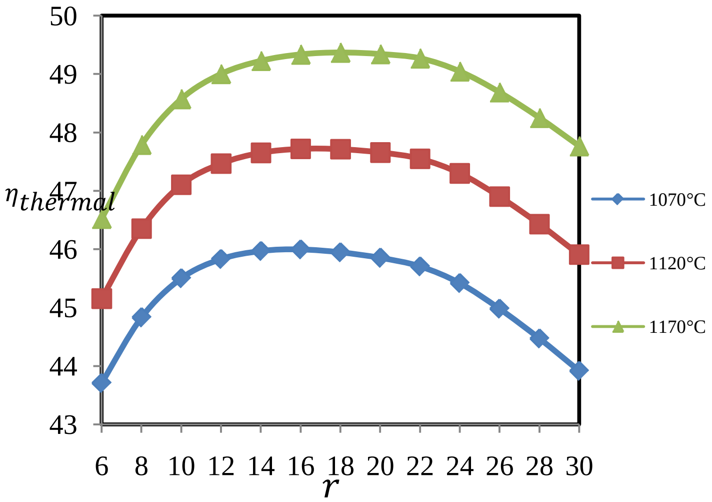

Figure 5. Thermal efficiency for model 1.

7.3. Performanceof Model 2

Performance of model 2 is affected by pressure ratio and the amount of steam injected into combustor inlet. The pressure ratio is varied from 6 up to 30. The amountof steam injected is denoted by calculation of $\frac{\dot{m}_v}{(\dot{m}_a+\dot{m}_v)} \times 100\%$ or in the other word, the percentage of steam injected mass flow rate to the total of mass flow rate of working fluid. The performance curve of model 2 is shownon figure 7.

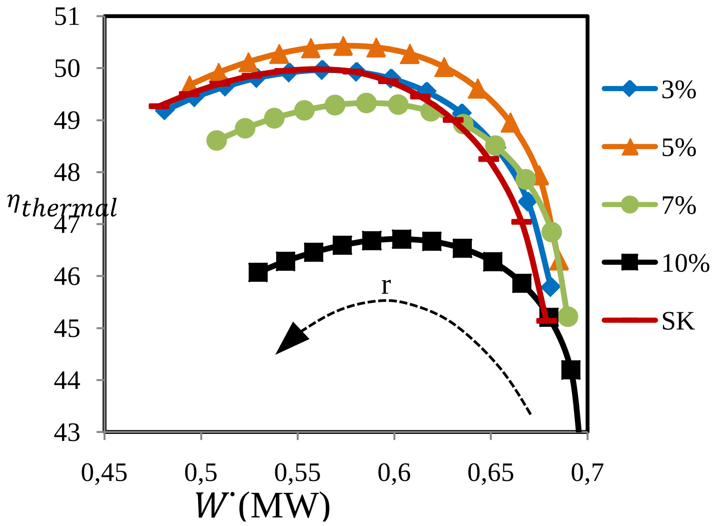

Figure 7. Performance of model 2.

The red curve is the performance curve of the PLTGU Tanjung Priok. As seen on the figure 7, the optimum condition for the combined cycle with low pressure steam injection is achieved at 5% of steam injection. At higher amount of steam injection, the performance reduces.

7.4. Performanceof Model 3

Performance of model 3 is shown on the figure 8. Unlike the previous model, the high pressure steam injection reduces the thermal efficiency of the orginal combined cycle performance. Still, the optimum condition in this model is achieved at 5% of high pressure steam injection.

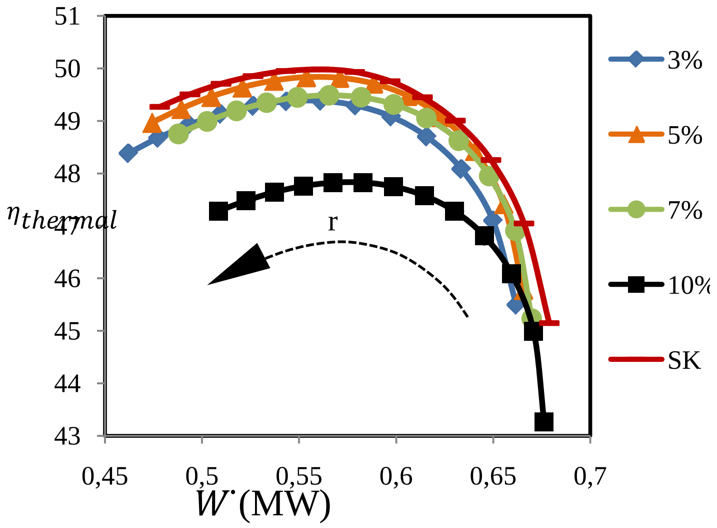

Figure 8. Performance of model 3.

The net power produced by high pressure steam turbine contributes a great portion for the total thermal efficiency. By injecting high pressure steam into the combustor inlet, the net power produced by high pressure steam turbine is reduced.Therefore, the thermal efficiency also reduces.

7.5. PerformanceComparison

Figure 8 shows the performance of the proposed models compared to the performance of the original combined cycle. For the same turbine inlet temperature (1070°C), the performance curve of model 2 is the highest among the proposed models. Note that model 1 produces more net power compared to the other models and the original combined cycle. In order to compete with the current thermal efficiency of the original combined cycle, the turbine inlet temperature of model 1 has to be increased to 1170°C.

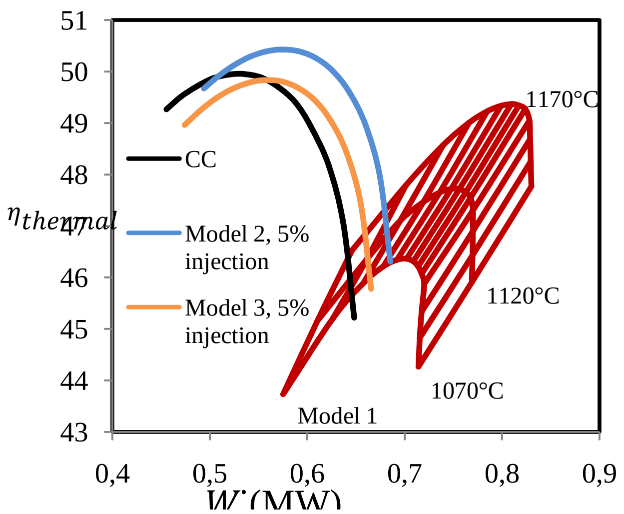

Figure 8. Performance comparison.

8. Conclusions

Having performed thermodynamic analysis on the case study and modification models, it can be summarized that the results are concluded into 6 conclusions:

1. Each modification model proposed has special characteristics affecting the performance.
2. Operation parameters affecting the performance of model 1 are pressure ratio, turbine inlet temperature, and cold fluid recuperator outlet temperature.
3. Operation parameters affecting the performance of models 2 and 3 are pressure ratio and mass of steam injected.
4. Implementation of optimized model 1 yields an increase in net power produced of 22.89%.
5. Implementation of optimized model 2 yields an increase in net power produced of 6.46% and thermal efficiency of 1.49%.
6. Implementation of optimized model 3 yields and increase in net power produced of 3.2% and thermal efficiency of 0.13%.

9. Nomenclature

$\dot{m}$	mass flow rate [kg·s^{-1}]
r	pressure ratio
T	temperature [°C]
η	efficiency
subscript	
a	air
v	steam

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