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An Illustration of Thermodynamic Modeling and Economic Analysis of Inlet Air Chilling System for Gas Turbine Power Augmentation

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Authors' contributions

This work was carried out in collaboration between all authors. Author HSA designed the study, wrote the protocol, managed literature searches and wrote the first draft of the manuscript. Author JS managed the analyses of the study and performed the simulation. Author ID managed the analyses of the study and literature searches. All authors read and approved the final manuscript.

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ABSTRACT

Inlet air chilling system has been widely used in recent years as a method of the gas turbine (GT) power augmentation in power plants. In this paper, thermodynamic modeling of inlet chilling system had been studied in order to improve the performance of the two units of gas turbines (GT1 & GT2) in Pesanggaran power plant, in Bali, Indonesia. The study was focused on power enhancement, specific fuel consumption (SFC), heat rate and thermal efficiency and conducted on three cooling conditions i.e. 17°C, 15°C and 13°C. Moreover, the economic analysis was conducted to determine the capacity cost, operating cost and payback period due to the investment cost of the system. Based on the simulation results, the lower the chilling temperature, the higher the power

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enhancement and the parasitic load on the system. The maximum power enhancements for GT1 and GT2 were at the lowest chilling temperature 13°C, i.e. 6.57% and 6.61%. Based on the evaluation of the economic aspects, capacity costs should be ideally balanced with a decrease in the production cost after the application of the chilling system, however, the power cost was not economical because it couldn't compensate the investment cost and it had a very long payback period.

Keywords: Power augmentation; gas turbine; inlet chilling; power plant.

1. INTRODUCTION

Most countries in the world use gas turbines for generating electricity because they can be started and stopped quickly to meet energy demand at both peak and base load conditions [1]. These engines are composed of a compressor that supplies air at high pressure to combustor which provides flue gas at high pressure and temperature turbine. Since they are the constant-volume engine, their power output is directly proportional and limited by the air mass flow rate which depends on the temperature and relative humidity of the ambient air [2]. The power output of gas turbine can decrease significantly with increase in ambient temperature. High ambient air temperature affects the power output of a gas turbine in two ways: as the temperature of the air increases, the air density decreases and, consequently, the air mass flow decreases. The reduced air mass flow directly causes the gas turbine to produce less power output [3]. In hot days when the air is less dense, power output can fall off. A rise of 1°C temperature of inlet air can decrease the power output of 1% [4].

Compressor inlet air cooling is one of the most well-known methods to improve gas turbine power plant performance [5]. Many of research works were done by conditioning the ambient air at the compressor inlet in order to increase the power of the gas turbine. In general, inlet air cooling technology in the gas turbine system can be classified into three methods, i.e. evaporative cooling, fogging and chilling [6]. Evaporative cooling had been applied by Jaber et al. [7] and Carmona [8] which the power improvement had been improved by 5% and 8% respectively. Then Sanaye et al. [9] and Ehyae et al. [10] show that the fogging system appears to be capable of boosting the power generated by 7.51% and 7% respectively. In a chilling, Suneetha et al. [11] reported power enhancement of up to 12.47%, while Popli et al. [12] successfully achieved the highest increased power of up to 23.2%. Najjar et al. [13] had tried to combine the evaporative cooling system with chilling and they have successfully increased the gas turbine power output of up to a maximum of 15%. From all three cooling methods, it seems that chilling is the most effective cooling system for power augmentation of gas turbine because it can reduce the inlet condition of the gas turbine to the desired temperature, generally in ISO conditions or slightly lower during all times of operation (15°C). However, all the results of these studies can't be used as a reference, because the research was conducted on the different climatic conditions and gas turbine types. For example, Hilman et al. [14] was conducted a study of gas turbine power enhancement using evaporative cooling in Indonesia. However, the result was not significant compared with the results obtained in the Middle East and Africa. Also from the economic aspect, the investment cost was not economical.

In this study, power augmentation analysis was conducted on simple cycle gas turbine using inlet chilling system with three cooling conditions at compressor inlet i.e. 17°C, 15°C and 13°C. In addition, the important parameters such as specific fuel consumption (SFC), heat rate and thermal efficiency was performed in order to analyze the economical aspect of the cooling system on the gas turbine. The case study was conducted in Pesanggaran power plant Units 1 and 2 which has a base load capacity of 20 MW and 42 MW. The plant is operated by Indonesia Power located in southern Bali Island, Indonesia. The economic analysis was performed by determining the capacity cost, operation cost and payback period of the investment cost due to the addition chilling system on each gas turbines.

2. MATERIALS AND METHODS

2.1 Inlet Chilling System

Inlet chilling system is composed a cooling coil that is installed in the gas turbine inlet. Cooling fluid is pumped through the coil as the gas turbine inlet air flows through the coil. A source of the chilled water is required. The chilling system can be an electrically driven chiller(s) or absorption chiller(s) [2]. The chilling system will require a way to reject heat, which, in most cases is a wet cooling tower. The wet cooling tower uses the vaporization of water to the atmosphere to take away the heat of the cycle driver and the cooling effect [15]. The schematic of inlet chilling system can be seen in Fig. 1.

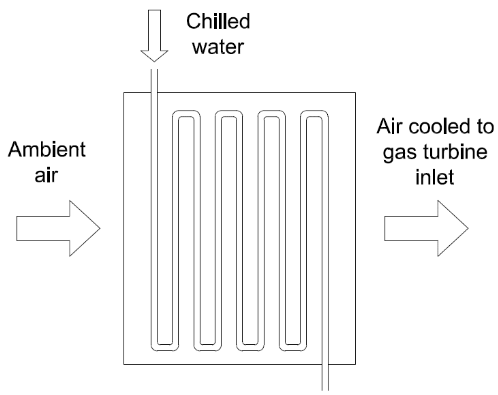


Fig. 1. Schematic of inlet chilling system at the inlet of gas turbine compressor [2]

Fig. 2 shows a schematic diagram of a simple cycle gas turbine cycle with inlet chilling system. The gas turbine performance will be evaluated with the proposed cooling technique.

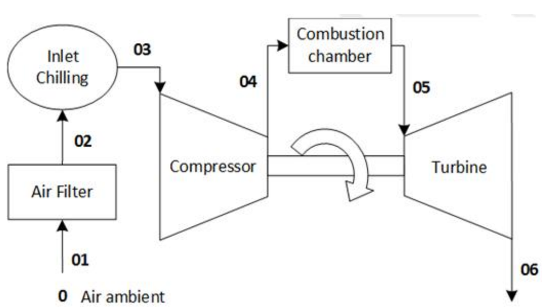


Fig. 2. Schematic diagram of the gas turbine cycle with the inlet chilling system

The extent of the evaporation and inlet temperature is inversely proportional to the percent humidity in the air stream, but in the chilling system, it can be set to the desired temperature, generally in ISO conditions (15°C) or slightly lower during all times of operation [15]. A temperature decreasing at the compressor inlet can be shown on the saturation process at the psychometric chart in Fig. 3.

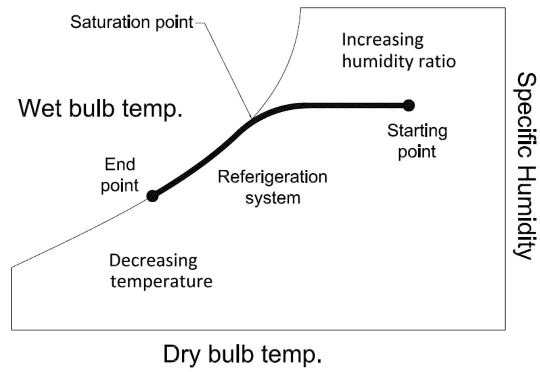


Fig. 3. Saturation process in the psychometric chart

According to Santos et al. [2], the inlet air temperature after the cooling process in Fig. 2 can be calculated as:

$$T_{03} = Tb_{02} - \varepsilon(Tb_{02} - Tw_{02}) \quad (1)$$

where Tb_{02} is the dry-bulb temperature, Tw_{02} is the wet- bulb temperature and ε is the cooling effectiveness.

The cooling load due to the application of inlet chilling system, can be calculated by:

$$\dot{Q}_{CL} = \dot{m}_a C_{pa,avg}(T_{02} - T_{03}) \quad (2)$$

where $\dot{m}_a$ is the mass air flow rate, and $C_{pa,avg}$ is the specific heat of dry air at constant pressure, determined as a function of the average temperature across the evaporative system.

The following are a thermodynamic calculation for open simple cycle gas turbine. The air and combustion products are assumed to behave as ideal gasses. The pressure of the air leaving the compressor (P_{04}) is calculated as:

$$P_{04} = r.P_{03} \quad (3)$$

where *r* is the compression ratio.

Assuming an ideal gas for state 04, the total temperature of the fluid leaving the compressor can be evaluated using ideal gas relations:

$$T_{04} = \frac{T_{03}}{\eta_c}\left[\left(\frac{P_{04}}{P_{03}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] + T_{03} \quad (4)$$

where η_c is the compressor efficiency and γ is the specific heat ratio.

The compressor work is calculated from the mass flow rate and enthalpy change across the compressor as follows:

$$\dot{W}_C = \dot{m}_a C_{pa}(T_{04} - T_{03}) \tag{5}$$

where $\dot{m}_a$ is the air mass flow rate and C_{pa} is the specific heat capacity of air at constant pressure. The turbine inlet pressure (P_{05}) can be calculated as:

$$P_{05} = P_{04}(1 - \Delta P_{cc}) \tag{6}$$

where P_{05} is the turbine entry level pressure, P_{04} is the combustion chamber inlet pressure, and ΔP_{cc} is pressure drop across the combustion chamber.

The heat delivered by the combustion chamber is determined from energy balance:

$$\dot{Q}_{in} = C_{pg}(T_{05} - T_{04}) \tag{7}$$

where C_{pg} is the specific heat capacity of combustion products.

By knowing the fuel gas lower heating value (LHV), the mass flow rate of fuel is computed as:

$$\dot{m}_f = \frac{\dot{Q}_{in}}{LHV \times \eta_{com}} \tag{8}$$

where η_{com} is combustor efficiency.

The exhaust temperature of the gas that leaving the turbine can be written as:

$$T_{06} = T_{05}\left\{1 - \eta_T\left[1 - \left(\left(\frac{P_{05}}{P_{06}}\right)^{\frac{1-\gamma g}{\gamma g}}\right)\right]\right\} \tag{9}$$

where η_T is the turbine isentropic efficiency and P_{06} is the ambient pressure. Hence, the turbine power is equal to:

$$\dot{W}_T = \dot{m}_g c_{pg}(T_{05} - T_{06}) \tag{10}$$

where $\dot{m}_g$ is the total mass flow rate of flue gas. It is composed of fuel and air mass flow rate and it is given by

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{11}$$

The net power obtained from the gas turbine is given by:

$$\dot{W}_{Net} = \dot{W}_T - \dot{W}_c \tag{12}$$

The specific fuel consumption compares the ratio of the fuel used by an engine to a characteristic power such as the amount of power the engine produces. This is a very important economic criterion. The specific fuel consumption (SFC) is determined by equation:

$$SFC = \frac{3600.\dot{m}_f}{\dot{W}_{Net}} \tag{13}$$

Another important gas turbine parameter is the heat rate (HR) of a gas turbine cycle is determined by:

$$HR = SFC \times LHV \tag{14}$$

Therefore, the thermal efficiency of the gas turbine is calculated as:

$$\eta_{th} = \frac{\dot{W}_{Net}}{\dot{m}_f.LHV} \tag{15}$$

2.2 Existing Performance Data

Table 1 summarizes the three gas turbines, manufacturer, years of commissioning and design capacity that are located at the Pesanggaran site.

The existing performance of the three gas turbines is based on the latest plant performance test results which are shown in Table 2.

2.3 Evaluation Methodology

The power augmentation and performance evaluation of the two units of the gas turbine in Pesanggaran site due to the addition of inlet chilling system was modeled using GTPro, a power cycle thermodynamic computer modeling

Table 1. Pesanggaran gas turbine installation summary

GT	OEM	Model	Year	Capacity [MW]	T_{amb} (°C)
1	GE	MS-500-L	1993	20.10 (B) 23.05 (P)	30
2	Westinghouse / Siemens	CW-251-B11	1994	42.07	27

Table 2. Gas turbine performance summary

GT	Last capacity tests date	Test results (MW)	Aux. loads (kW)	SFC (l/kWh)	T_{amb} (°C)
1	15/01/14 (7 pm-8 pm)	16.30	220	0.422	30
2	06/01/14 (7 pm-8 pm)	39.80	161	0.339	30

software program. The study is based on three scenarios with the target cooling the compressor inlet each for each gas turbine is 17°C, 15°C, and 13°C. The first step of evaluation is to replicate the New and Clean (N&C) performance of each gas turbine at the design ambient conditions when the unit was installed. Site-specific conditions are entered such as generator voltage, line voltage, site-specific fuel composition, GT starter mechanism, and others.

Using GTPro, the model is run at the ambient conditions of the latest performance test provided by the plant. Degradation factor can be calculated by comparing the corrected performance divided by the N&C performance. Then each GT is modeled in GTPro with each of the performance improvements under consideration. The analysis is focused on power output, fuel consumption, heat rate, and thermal efficiency. To perform the economic analysis due to the addition the fogging system in a gas turbine, GTPro capital cost estimating segment of the computer program (PEACE) was utilized to estimate the capacity cost, operating cost and payback period due to the investment cost of the evaporative cooling system on the gas turbine. It is assumed to use the default cost multipliers for Indonesia provided in GTPro for commodities, equipment, labor and materials which can be seen in Table 3.

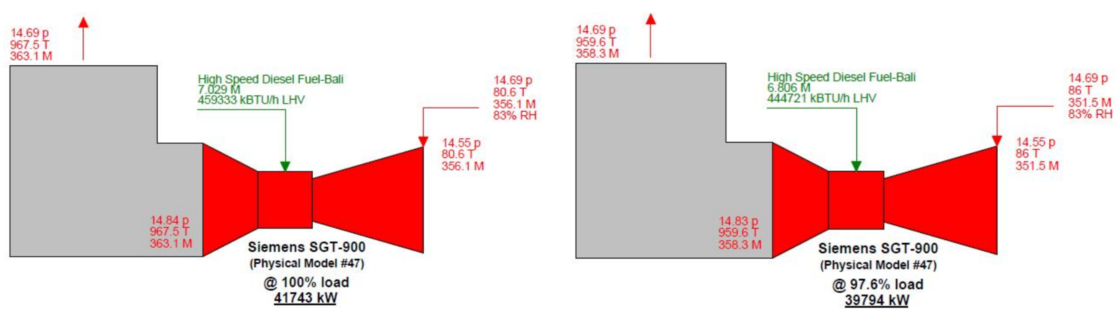
Table 3. GT Pro cost factors recommended for Indonesia [15]

Category	Factor for Indonesia
Specialized Equipment	1.05
Other Equipment	0.75
Commodities	0.65
Labor	0.54

3. RESULTS AND DISCUSSION

Based on the evaluation methodology, the initial step in the simulation is to analyze the effect of operating hours to the performance of New and Clean (N&C) condition, therefore, the performances before and after the addition of inlet chilling system can be corrected by the degradation factor. The simulation results for the N&C performance of gas turbine unit 2 (GT2) can be seen in Fig. 4. Based on these results, the degradation factor for the GT2 can be estimated at 2.4%. Degradation factors then are taken into account in the simulation of new GT performance by adding the inlet chilling system.

The simulation results for GT2 with the addition of chilling system and the cooling conditions at 17°C, 15°C, and 13°C can be seen respectively in Fig. 5, Fig. 6 and Fig. 7. The condition of the ambient air is used as a reference in the analysis



(a) (b)

Fig. 4. GTPro simulation result for (a) New and Clean (N&C) condition and (b the latest performance test

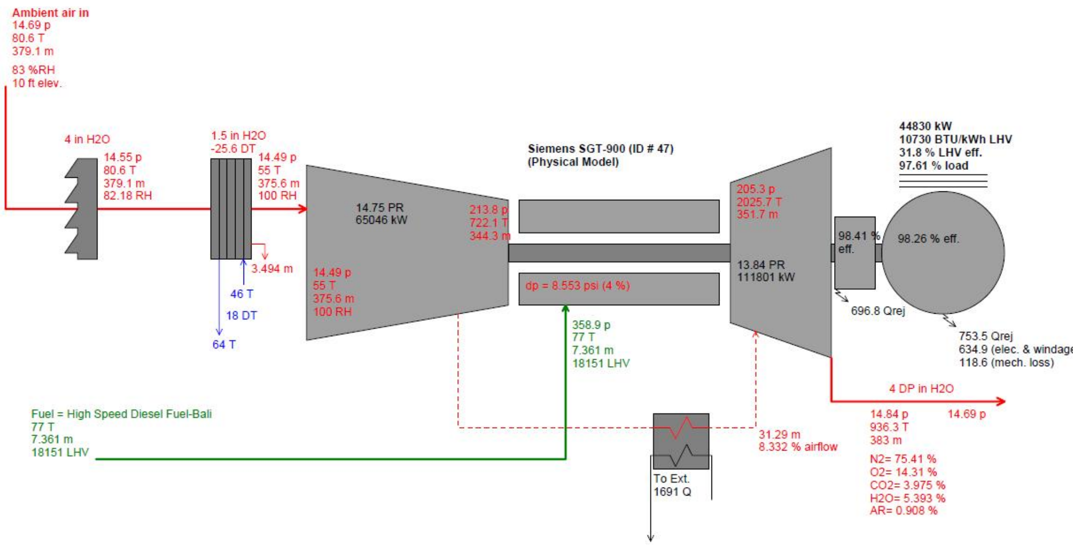


Fig. 5. Simulation result of new GT performance with degradation factor and inlet chilling at 13°C

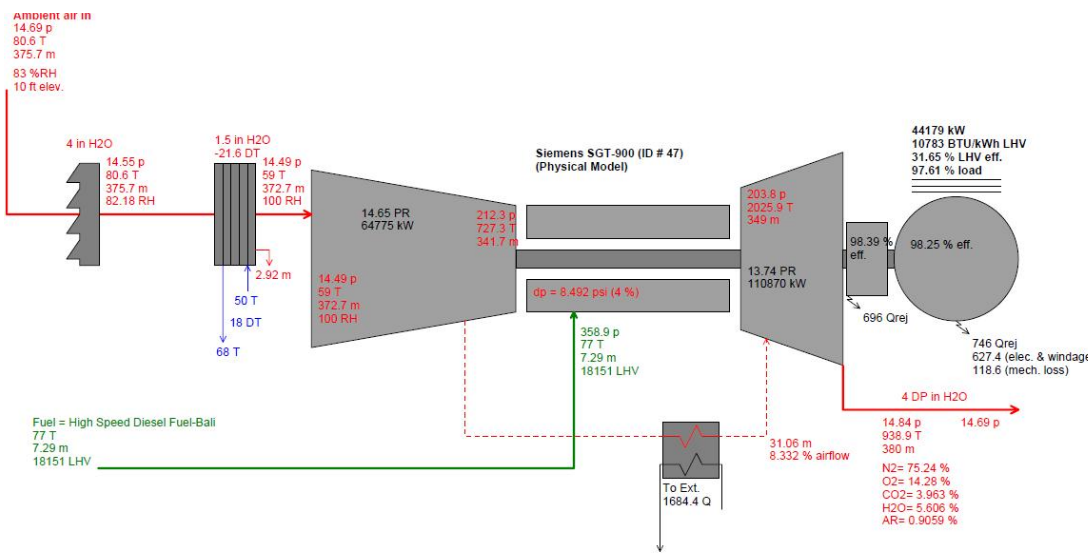


Fig. 6. Simulation result of new GT performance with degradation factor and inlet chilling at 15°C

is the condition on the latest performance test with temperature 80.6°F (27°C) and RH 83%. Based on the simulation results for the three cooling loads, it appears that the relative humidity of the three systems are at 100% RH or saturation line, where the temperature decreasing at the compressor inlet can be achieved in accordance with the predetermined targets, i.e. 63°F (17°C), 59°F (15°C) to 55°F (13°C).

The temperature decreasing has a direct impact on improving the performance of the gas turbine, but the lower the temperature at the compressor inlet, the higher the parasitic load on the system which will affect the net power of the gas turbine. Summary of performance improvement for GT1 and GT2 from the existing performance in each cooling conditions can be seen in Table 4.

Additional net power generated from the difference between after and before chilling conditions can be shown in Fig. 8(a). Each gas turbine power increases was in line with the declining of the cooling inlet of the gas turbine. For GT1, the increased power seems nearly linear which the maximum power was 6.57% at chilling temperature 13°C, however for the GT2, The increased power between the chilling temperature 15°C and 13°C are slightly different, because the ratio between the power generated and the parasitic load on the chilling 13°C is lower than the 15°C. The very important economic criterion is the specific fuel consumption (SFC) that shows the ratio of the fuel used by an engine to a characteristic power. Incremental SFC for GT1 and GT2 are calculated by the difference between before and after chilling conditions which are shown in Fig. 8(b). Incremental SFC for both gas turbines seems decreased with nearly linear where the maximum is at chilling temperatures of 13°C. Therefore, the fuel consumption per unit of power generated on both gas turbines after chilling tends to be lower than the existing condition. It also applies to the value of incremental heat rate because it has the similar functions with SFC (Fig. 8(c)). Then the difference between the thermal efficiency of the gas turbine after and before chilling can be shown in Fig. 8 (d). In both gas turbines, the efficiency tends to improve linearly with the maximum for GT1 and GT2 is at the chilling temperature of 13°C and respectively is 1.31% and 0.79%.

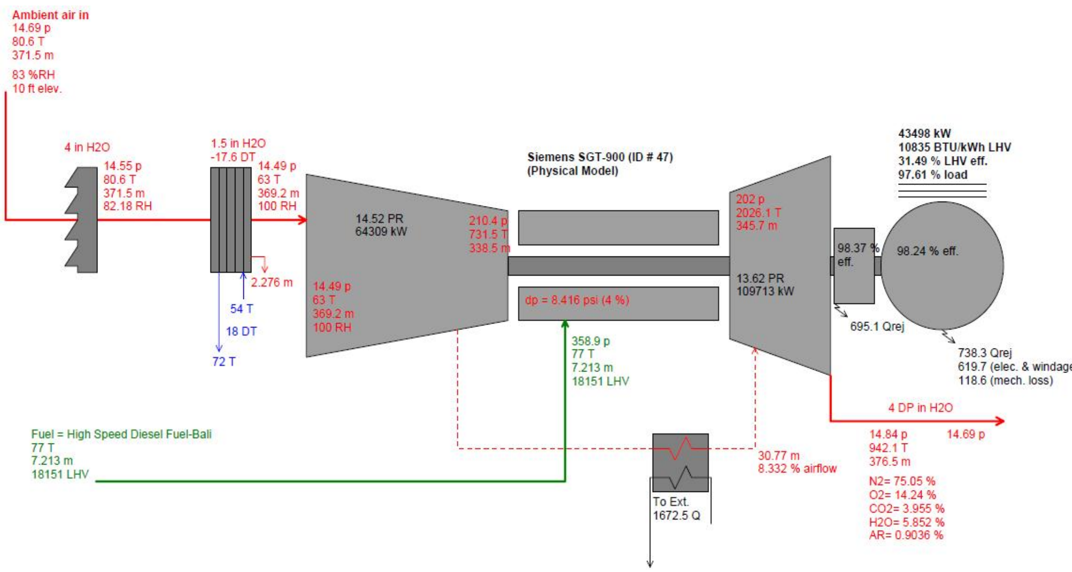
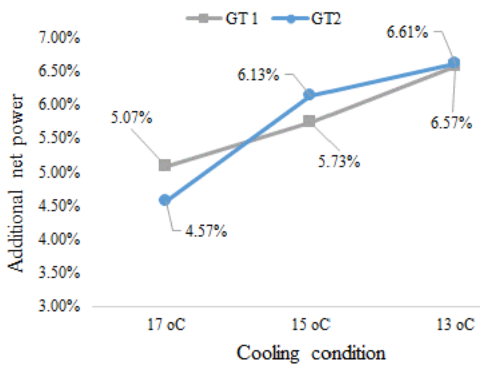


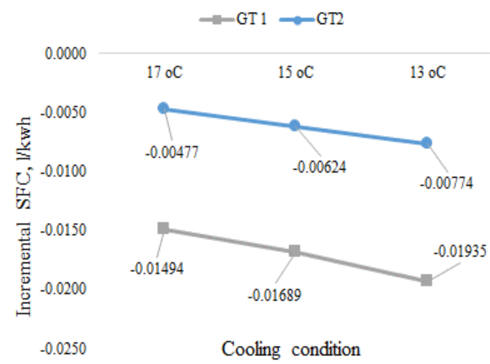
Fig. 7. Simulation result of new GT performance with degradation factor and inlet chilling at 17°C

Table 4. Performance improvement results for the three unit of the gas turbine (GT) in Pesanggaran plant

Parameter	Cooling T. 17°C		Cooling T. 15°C		Cooling T. 13°C	
	GT 1	GT 2	GT 1	GT 2	GT 1	GT 2
Base power, kW	16,203	40,743	16,203	40,743	16,203	40,743
New power, kW	17,991	43,498	18,271	44,179	18,579	44,830
New parasitic load, kW	966	894	1140	940	1311	1394
Additional net power, kW	822	1,861	928	2,496	1,065	2,693
Base heat rate, BTU/kWh	13695.0	11004.0	13695.0	11004.0	13695.0	11004.0
New heat rate, BTU/kWh	13166.0	10835.0	13097.0	10783.0	13010.0	10730.0
Incremental heat rate, BTU/kWh	-529.0	-169.000	-598.000	-221.000	-685.000	-274.000
Base specific fuel consumption, liter/kWh	0.387	0.311	0.387	0.311	0.387	0.311
New specific fuel consumption, liter/kWh	0.372	0.306	0.370	0.305	0.367	0.303
Incremental specific fuel consumption, liter/kWh	-0.01494	-0.00477	-0.01689	-0.00624	-0.01935	-0.00774
Base thermal efficiency	24.92%	31.01%	24.92%	31.01%	24.92%	31.01%
New thermal efficiency	25.92%	31.49%	26.05%	31.65%	26.23%	31.80%
Incremental thermal efficiency	1.00%	0.48%	1.13%	0.64%	1.31%	0.79%



(a)



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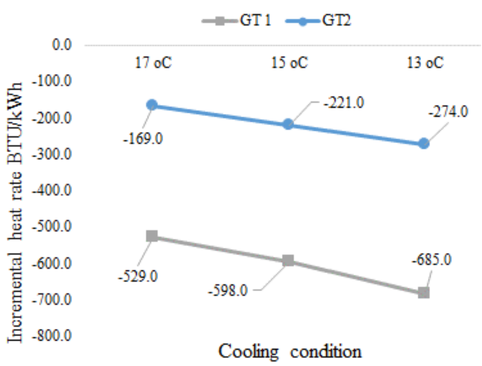

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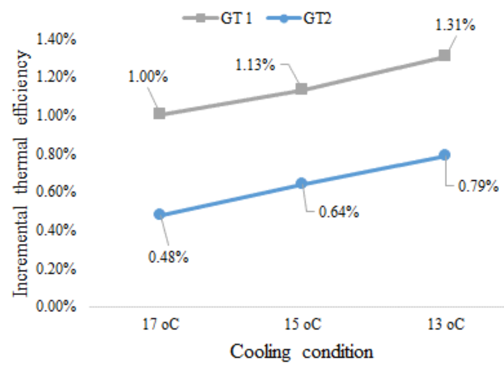

(d)

Fig. 8. Performance improvement results of the two unit of the gas turbine (GT) in Pesanggaran plant: (a) additional net power, (b) incremental SFC, (c) incremental heat rate, and (d) incremental thermal efficiency

The application of inlet chilling system on gas turbine has the impact to the addition of investment or capacity cost. The calculation results of capacity cost using PEACE due to the addition of inlet chilling system can be shown in Table 5.

The investment costs are estimated based on the costs required for a new plant with the addition of chilling system reduced by the cost of the base plant. If it divided by the additional energy generated from the addition of chilling system, the incremental capacity cost for both gas turbines can be seen in Fig. 9(a). In GT1, the ratio between investment costs of the chilling system with the energy generated tends to decrease though not significant, however in GT2 incremental capacity cost at chilling temperatures of 15°C tend to be smaller than the temperature of 13°C. Because the ratio between the power generated and the parasitic load on the chilling 13°C is lower than the 15°C.

The incremental capacity cost should ideally be balanced by a decrease in production costs due to the addition of inlet chilling system, therefore, the payback period of the investment can be estimated. Assuming the gas turbine operated for 2000 hours in a year and fuel costs of $ 0.88 per liter [16], the annual production cost can be shown in Table 6. The ratio between the cost of fuel and energy produced can generate average power cost in one year. Then the difference in fuel consumption between the existing conditions (base) and after the application of chilling system (new) can be expressed as the incremental average power cost in USD/kWh per year which is shown in 9(b). For both gas turbines, there is the average power cost reduction in one year, however, it is not significant. If the maximum reduction of annual production costs on GT1 and GT 2 is compared to the investment cost of the cooling system, the payback period of GT1 and GT 2 respectively are 82.5 and 92.93 years. Therefore, the application of inlet chilling system on both gas turbines in Pesanggaran plant is not economical because it can't compensate the investment cost and it has a very long payback period.

Table 5. Capacity cost evaluation results for the three unit of the gas turbine (GT) in Pesanggaran plant

Parameter	Cooling T. 17°C		Cooling T. 15°C		Cooling T. 13°C	
	GT 1	GT 2	GT 1	GT 2	GT 1	GT 2
Additional net power, kW	822	1861	928	2496	1065	2693
Cost base plant, kUSD	22,232	33,379	22,232	33,379	22,232	33,379
Cost new plant, kUSD	24,700	35,990	24,958	36,385	25,224	36,788
Estimated installed cost of cooling system, kUSD	2,468	2,611	2,726	3,006	2,992	3,409
Capacity cost of cooling system, USD/kWh	3,002.4	1,403.0	2,937.5	1,204.3	2,809.4	1,265.9

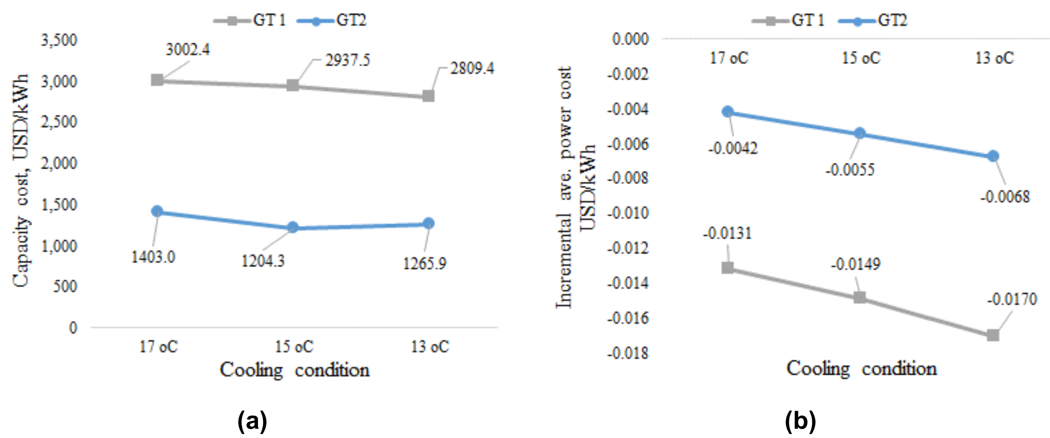

Fig. 9. Economic analysis results of the two unit of the gas turbine (GT) in Pesanggaran plant: (a) incremental average power cost and (b) incremental capacity cost

Table 6. Operating cost evaluation results for the three unit of gas turbine (GT) in Pesanggaran plant

Parameter	Cooling T. 17°C		Cooling T. 15°C		Cooling T. 13°C	
	GT 1	GT 2	GT 1	GT2	GT 1	GT 2
Base Electric Energy Generated, MWh	32406.0	81486.0	32406.0	81486.0	32406.0	81486.0
New Electric Energy Generated, MWh	34050.0	85208.0	34262.0	86478.0	34536.0	86872.0
Base Annual fuel consumption, kliter	12535.9	25328.1	12535.9	25328.1	12535.9	25328.1
New Annual fuel consumption, kliter	12663.1	26078.3	12675.2	26339.9	12691.7	26329.9
Base Annual Fuel Cost, kUSD	11031.6	22288.8	11031.6	22288.8	11031.6	22288.8
New Annual Fuel Cost, kUSD	11143.5	22948.9	11154.2	23179.2	11168.7	23170.3
Base Average power cost, USD/kWh	0.3404	0.2735	0.3404	0.2735	0.3404	0.2735
New Average power cost, USD/kWh	0.3273	0.2693	0.3256	0.2680	0.3234	0.2667
Incremental Ave. power cost, USD/kWh	-0.0131	-0.0042	-0.0149	-0.0055	-0.0170	-0.0068

4. CONCLUSION

Based on simulation results using GTPro, the lower the temperature at the compressor inlet, the higher the power enhancement and the parasitic load on the system. The maximum power enhancements for GT1 and GT2 were at the lowest chilling temperature 13°C, i.e. 6.57% and 6.61%. The obtained results are still far away and not very significant when compared to previous studies. It applies also to the SFC, heat rate and thermal efficiency where the influence of inlet chilling system doesn't have a significant impact on the performance improvement of the gas turbine. It could be due to the climatic conditions in Pesanggaran plant which the existing condition of inlet turbine temperature was not as hot as the previous studies, so the increased power is not too significant.

Furthermore, the economic calculation is analyzed based on capacity and production costs using PEACE. Based on the simulation results, there is the average power cost reduction for both gas turbines in one year, however, it is not significant. If the maximum reduction of annual production costs on GT1 and GT 2 is compared to the investment cost of the cooling system, the payback period of GT1 and GT 2 respectively are 82.5 and 92.93 years. Therefore, the application of inlet chilling system on both gas turbines in Pesanggaran plant is not economical because it can't compensate the investment cost and it has a very long payback period.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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