British Journal of Applied Science & Technology
11(1): 1-12, 2015, Article no.BJAST.19184
ISSN: 2231-0843

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Performance Enhancement of Gas Turbine Power Plant via Wet Compression Technique and Its Effects on Economic Aspects

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Authors' contributions

This work was carried out in collaboration between all authors. Author HSA designed the study, wrote the protocol and wrote the first draft of the manuscript and managed literature searches. Authors 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.

Article Information

DOI: 10.9734/BJAST/2015/19184
Editor(s):
(1) Chang-Yu Sun, China University of Petroleum, China.
Reviewers:
(1) Anonymous, Universiti Teknologi Malaysia, Malaysia.
(2) Anonymous, Uttar Pradesh Technical University, Lucknow, India.
(3) Claudio Bruno, Thermo-Fluid-Systems Department, United Technologies Research Center, USA.
(4) Rodrigo Crespo Mosca, São Paulo University, Brazil.
Complete Peer review History: http://sciencedomain.org/review-history/10355

Original Research Article

Received 29th May 2015
Accepted 8th July 2015
Published 30th July 2015

ABSTRACT

The study is intended to improve the performance of gas turbine engines to meet both electrical power demand and peak load in the power plant and to analyze their economic aspects. Performance enhancement of gas turbine was analyzed using thermodynamic modelling and focusing on power enhancement, specific fuel consumption (SFC), heat rate and thermal efficiency due to the application of wet compression system. The study was conducted on the two gas turbine units (GT1 and GT2) in the Pesanggaran power plant, Southern Bali Island, Indonesia between December 2014 and March 2015. GTPro, a power cycle thermodynamic computer modeling software program, was utilized to conduct the performance evaluation of the application of wet compression system. The study was conducted based on three conditions, i.e. saturated fogging

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and wet compression at 1% and 2% over spray (OS). The capacity cost, operating cost, and payback period due to the investment cost of the system are determined using PEACE. Based on the results of evaluation, performance enhancement of gas turbines using wet compression system is more significant than with conventional fogging, with maximum incremental power reaching 23.09%. In addition, in terms of SFC, heat rate and thermal efficiency, the wet compression system is more economical than conventional fogging; therefore the application of the wet compression system on GT1 and GT2 can achieve the fastest payback in, respectively, 9 and 10 year. It can be concluded that the wet compression system is more effective and economical than conventional fogging for improving the performance of gas turbines. However, the experimental work to validate a thermodynamic modelling and the effects of adding water to gas turbine components needs be further investigated.

Keywords: Thermodynamic modeling; wet compression; gas turbine; power enhancement.

ABBREVIATIONS

GT	*Gas turbine*
OS	*Over spray*
OEM	*Original equipment manufacturer*
PEACE	*Plant Engineering and Cost Estimator*
N&C	*New and Clean*

NOMENCLATURES

T	*Temperature (°C)*
Tb	*Dry-bulb temperature (°C)*
Tw	*Wet-bulb temperature (°C)*
ε	*Cooling effectiveness (%)*
$\dot{Q}$	*Heat transfer rate (kW)*
C_p	*Specific heat at constant pressure (kJ/kg.°C)*
LHV	*Lower fuel heat value (kJ/kg)*
HR	*Heat rate (kJ/kWh)*
$\dot{m}$	*Mass flow rate (kg/s)*
P	*Pressure (Pa)*
ΔP	*Pressure drop (Pa)*
$\dot{W}$	*Power output (kW)*
γ	*Specific heat ratio*
$\dot{W}$	*Power output (kW)*
γ	*Specific heat ratio*
η	*Efficiency*
r	*Compressor pressure ratio*

SUBSCRIPTS

0	*Ambient air*
01, 02, 03, 04, 05, 06	*Point denoted in Fig. 3*
a	*Air*
avg	*Average*
c	*Compressor*
cc	*Combustion chamber*
com	*Combustor*
g	*Flue gas*
f	*Fuel*
in	*Input*
N	*Net*
th	*Thermal*

1. INTRODUCTION

Inlet air-cooling is considered the most cost-effective way to increase the power output as well as thermal efficiency of industrial gas turbines [1]. In general, the inlet air cooling technology on the gas turbine system can be classified into three categories: Evaporative cooling, fogging and chilling [2-4]. Jaber et al. [5] and Hosseini et al. [6] have been using evaporative cooling to improve the performance of gas turbine where the results shows that the system appears to be capable of boosting the generated power respectively are 5% and 13.3%. Later, Ehyae et al. [7] has reported the power enhancement of gas turbine using fogging technique by 7%. In chilling, Suneetha et al. [8] reported an increase in power of up to 12.47% while Popli et al. [9] successfully achieved of up to 28.57%. Although inlet chilling shows to be superior in all three systems for power augmentation, the system has several drawbacks, i.e., there is the permanent pressure drop in the inlet filter housing that permanently reduces the GT output the year around, regardless of the whether the inlet chilling system is operational or not even during part load operation. The inlet chilling system is generally very slow at responding, due to the thermal inertia of the warm water sitting in the piping before startup and it requires a wet cooling tower that uses the vaporization of water to the atmosphere to take away the heat of the cycle driver and the cooling effect.

In recent years, wet compression has gained in popularity among power augmentation techniques for gas turbines; this technology, which is also called over-fogging or overspray'' is considered the evolution of the concept of fogging [10-11]. Bracco et al. [10] reported the results of the increase of gas turbine power output of up to 14%, while Sanaye et al. [11] and Shepherd et al. [12] produced a power gain of each of up to 7.73% and 18% respectively. All of the cases applied the conditions of 2% over spray. In this paper, the study of power augmentation is performed on a simple cycle gas turbine using wet compression system. The performance enhancement of gas turbine was compared at three conditions, i.e. conventional saturated fogging and wet compression at 1% and 2% overspray. Case studies were conducted in Pesanggaran power plant Unit 1 and 2 that have a base load 20 MW and 42 MW, respectively. The plant is operated by Indonesia Power located in southern Bali Island, Indonesia. In addition, an economic analysis was conducted to determine the capacity cost, operating cost and pay-back period of the investment cost due to the addition of the wet compression system.

2. MATERIALS AND METHODS

2.1 Wet Compression System

In wet compression system, demineralized pressurized water is injected close to the inlet of the gas turbine and/or in various stages of the gas turbine compressor; the schematic of wet compression system can be seen in Fig. 1.

The water nozzles create an ultra-fine droplet size water mist. At the inlet some of the water evaporates quickly cooling the inlet air. The water injected above that needed to achieve saturation is drawn into the compressor blade path and evaporates during the various stages of compression. Wet compression has the advantage of providing the effect of inter-stages cooling as water evaporates in each stage. Evaporation of the water droplets in the compressor blade path causes the air temperature to drop and thereby reduces the power consumption of the compressor because less energy is required to compress cool air compared to warm air with the same mass. This translates into a decrease in turbine work because one-half to two-thirds of a turbine's output is typically used to drive the compressor. The result is more turbine power for the generation of electricity and improved gas turbine efficiency. A decrease in temperature at the compressor inlet can be shown on the saturation process in the psychrometric chart at Fig. 2.

However, water that does not completely evaporate impinges on the compressor blades and can easily cause pitting and premature damage to the compressor blades, particularly the leading edges. When an early stage of a compressor fails, the broken pieces go downstream resulting in a multi-million dollar repair bill and at least 8 – 12 weeks out of service. Therefore proper applications require the compressor blades and stators to be coated to protect the surfaces for the steam and water impact. Siemens uses an Advanced Compressor Coating on gas turbine hardware to minimize the damage of wet compression [13]. Considerable downtime would be required to coat the compressor components. The compressor rotor and stator components would have to be shipped off site to an authorized service shop to apply the advanced coating. This would results in a few weeks of downtime. Alternatively this work could be conducted during a scheduled major overhaul. Fig.. 3 shows a schematic diagram of a simple cycle gas turbine cycle with wet compression system. The gas turbine performance will be evaluated with the proposed cooling technique.

According to Santos et al. [3] and Oyedepo et al. [4], the inlet air temperature after the cooling process in Fig. 3 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 wet compression 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 thermodynamic calculation for the open simple cycle gas turbine. The air and combustion products are assumed to behave as ideal gases. 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:

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

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 which can be calculated using Table 1.

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. According to Ganjehkaviri et al. [14] ΔP_{cc} can be assumed of 0.05.

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 which can be calculated using Table 1.

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:

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

where η_t is the turbine isentropic efficiency and P_{06} is the ambient pressure.

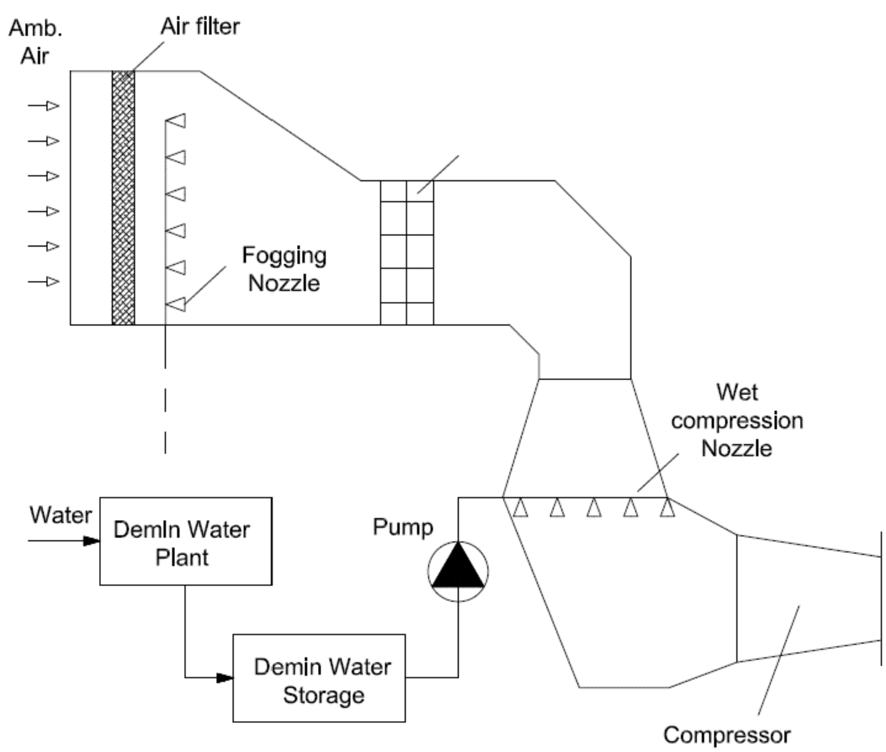


Fig. 1. Schematic of wet compression and fogging system at the inlet of gas turbine

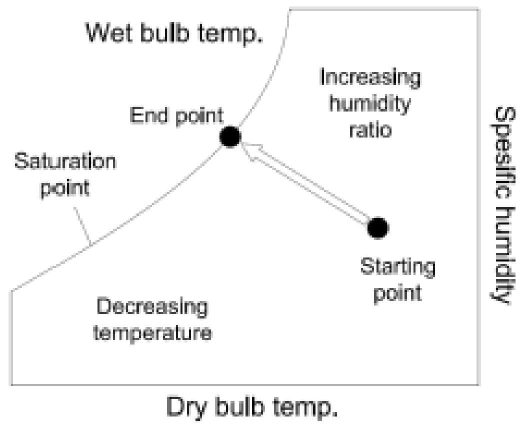


Fig. 2. Saturation process in the psychrometric chart

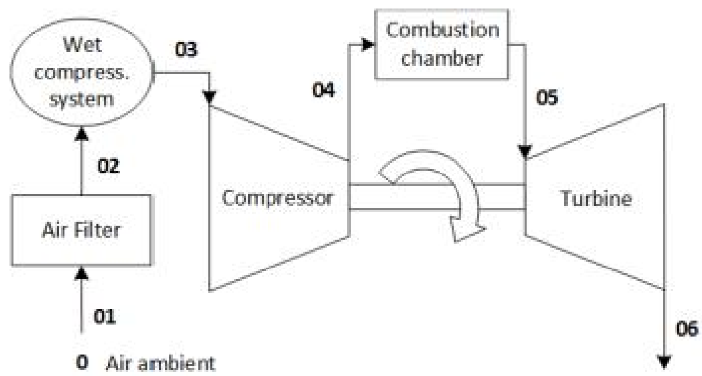


Fig. 3. Schematic diagram of the gas turbine cycle with cooling system

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:

$$\dot{W}_N = \dot{W}_t - \dot{W}_c \tag{12}$$

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:

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

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:

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

2.2 Existing Performance Data

Table 2 summarizes data of the two gas turbines, manufacturer, years of commissioning and design capacity that are located at the Pesanggaran site.

The existing performance of the two gas turbines is based on the latest plant performance test results which are shown in Table 3.

Table 1. Physical properties of the fluids [15]

Component	Property	
Air	Specific heat	$For\ 200\ K < T < 800\ K$ $C_{pa} = 1.0189134 \times 10^{3} - 1.3783636 \times 10^{-1}T + 1.9843397 \times 10^{-4}T^{2} + 4.2399242 \times 10^{-7}T^{3}$ $For\ 800\ K < T < 2200K$ $C_{pa} = 7.9865509 \times 10^{2} + 5.3392159 \times 10^{-1}T - 2.2881694 \times 10^{-4}T^{2} + 3.7420857 \times 10^{-7}T^{3}$
Flue gas	Specific heat	$C_{pg} = 1.0887572 \times 10^{3} - 1.4158834 \times 10^{-1}T - 1.9160159 \times 10^{-3}T^{2} - 1.2400934 \times 10^{-6}T^{3} + 3.0669459 \times 10^{-10}T^{4} - 2.6117109 \times 10^{-14}T^{5}$

Table 2. Pesanggaran gas turbine installation summary

GT	OEM	Model	Year	Capacity [MW]	T_{am} (°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 3. Gas turbine performance summary

GT	Last capacity test date	Test results (MW)	Aux. loads (kW)	T. amb. (°C)
1	15/01/14 (7pm-8pm)	16.30	220	30
2	06/01/14 (7pm-8pm)	39.80	161	30

2.3 Evaluation Methodology

GTPro, a power cycle thermodynamic computer modeling software program, was utilized to conduct the performance evaluation of the addition of wet compression system on both gas turbine unit in Pesanggaran site. The study was conducted based on three conditions, i.e. saturated fogging and wet compression at 1% and 2% over spray. 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 modelled in GTPro with each of the enhancements under consideration. For each enhancement, the parameters are power output, fuel consumption, heat rate, thermal efficiency and total installed cost of the GT are recorded. Using the GTPro capital cost estimating segment of the computer program (PEACE), an estimate of the cost to install the gas turbine in current year dollars is determined.

2.4 Assumption

In order to model this combined cycle power plant, some assumptions are made [14,16].

- All processes in this case were assumed as steady-state and steady-flow.
- The air and the gases resulting from combustion were considered as ideal gases.
- The energy variation and the kinetic and potential exergies were assumed negligible.
- The ambient conditions were assumed as identical to the conditions at the input to the compressor.
- Operating costs are only calculated based on the costs of fuel consumption, neglecting all other aspects.
- In the calculation of the annual operating cost, it is assumed that gas turbine operates for 2000 hours a year.
- The capacity costs of cooling system are calculated based on the ratio of the total investment cost to the power generated.

- The payback period of the investment cost is calculated from the difference between the benefits arising from the reduction of production costs before and after the application of cooling system.
- In the calculation of capacity cost, it is assumed to use the default cost multipliers for Indonesia provided in GTPro for commodities, equipment, labor and materials (see Table 4). Some initial costs require additional adjustments not modelled in GTPro including demineralized water plant and compressor coatings which will be added.

Table 4. GT pro cost factors recommended for Indonesia

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 performance of gas turbine (GT) was analyzed using GTPro simulation. The initial step in the simulation is to analyze the effect of operating hours to the performance declining of New and Clean (N&C) condition, so that the results of evaluation before and after the addition of wet compression system in the gas turbine can be corrected by the degradation factor. The simulation results for N&C performance of GT 2 can be seen in Fig. 4(a) while for the latest performance test in Fig. 4(b). Based on these results, the degradation factor for the GT2 can be estimated at 2.4%. This value is derived from the net amount of power based on the performance test divided by net power based on the N&C performance. Degradation factors then are taken into account in the simulation of new GT performance by adding the conventional inlet fogging and wet compression system. The simulation results of new performance for GT2 by entering the degradation factor and cooling system (saturated fogging, 1% OS, and 2% OS) can be seen in Figs. 5(a), (b), and (c). Based on the simulation results, summary of performance improvement for GT1 and GT2 from the existing performance in each cooling conditions can be seen in Table 5.

Additional net power generated from the difference between after and before cooling conditions can be shown in Fig. 6(a). The performance of each gas turbine has increased along with the level of saturation of the gas turbine inlet air. For GT1 and GT2, the performance enhancement due to the application of wet compression system is quite significant compared to saturated fogging, because of a significant increase in air mass on the condition of 1% and 2% OS. The maximum incremental power due to wet compression system for GT1 and GT2 occurred in 2% of the OS and are respectively, 25.42% and 23.09%. The advantages of wet compression system compared to the conventional fogging can be also seen in the aspect of fuel consumption that are shown in Fig. 6(b).

The incremental SFC in GT1 and GT2 at the wet compression system experienced a significant decrease compared to the fogging system, even though in GT 2, that uses a fogging system, there was a slight increase in incremental SFC compared to existing conditions. That shows that the wet compression system has a ratio of fuel consumption to power generated lower and is much more economical than conventional fogging. This applies also to the incremental heat rate because it has a function of the same parameters as SFC, as shown in Fig. 6(c). Another important parameter is the gas turbine thermal efficiency, where the incremental thermal efficiency is shown in Fig. 6(d). There is thermal efficiency improvement both in GT1 and GT2 using a wet compression system; however with a fogging system the calculated efficiency decreased for GT1, while increased slightly for GT2. The maximum enhancement of thermal efficiency in GT1 and GT2 with the assumed wet compression system is 2.20% and 1.92% respectively.

The cooling system application on the gas turbine has an impact to the additional costs, i.e. the costs of investment or capacity costs. The investment costs are estimated based on the costs required by the new plant with the addition of cooling system, demineralized water plant, and the compressor component recoating then reduced to the base cost of the plant. The estimated capacity costs due to the application of cooling system using PEACE can be shown in Table 6. From Fig. 7(a), the incremental capacity cost of the wet compression system is lower than conventional fogging and in line with the increasing of the overspray percentage. Because the ratio of the investment costs to the power generated on the wet compression system is better than fogging system. The additional capacity costs due to investment of cooling system should ideally be offset by gains from the difference in production costs between after and before the application of cooling system. If it is assumed that gas turbines operate for 2000 hours a year and fuel costs of $ 0.88 per liter [17], the annual production cost can be estimated and shown in Table 7 and the incremental average power cost from the existing condition are shown in Fig. 7(b).

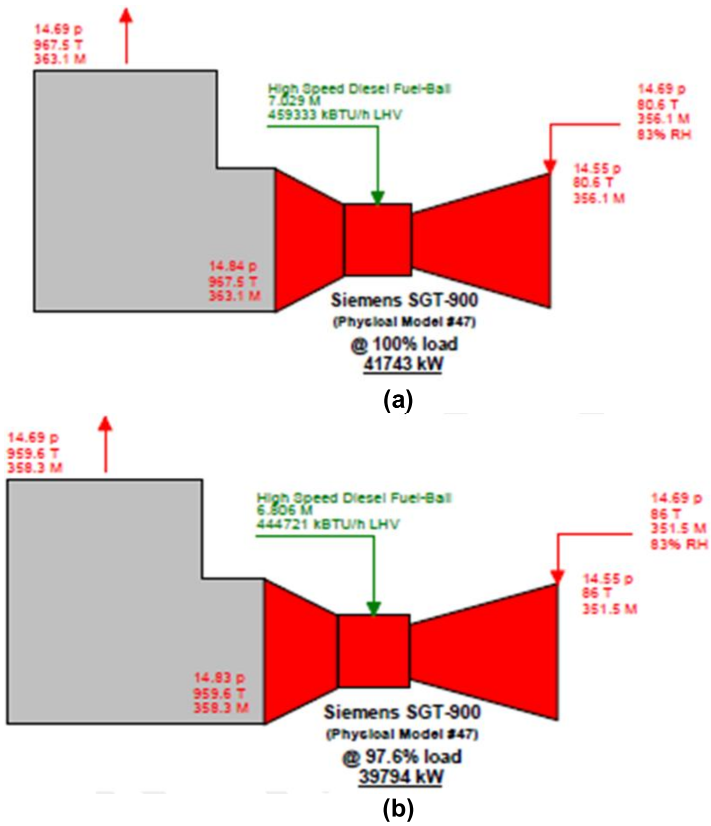
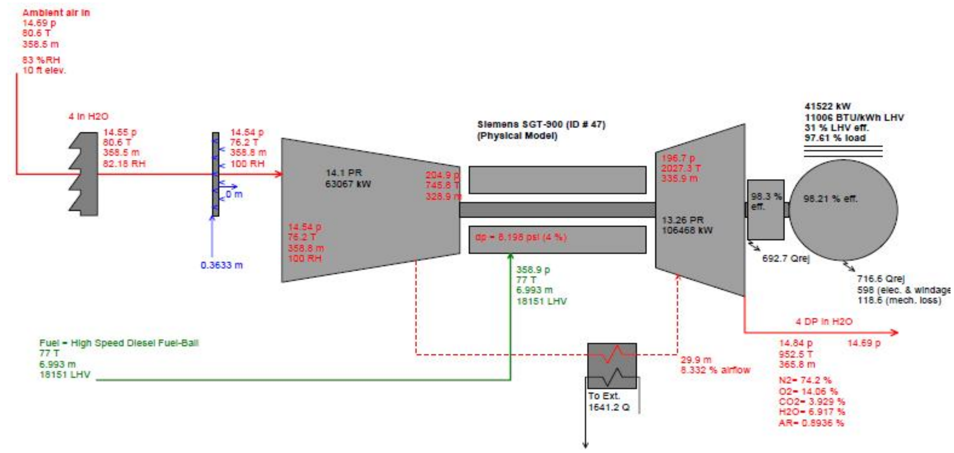


Fig. 4. GTPro simulation result for (a) New and Clean (N&C) condition and (b) The latest performance test

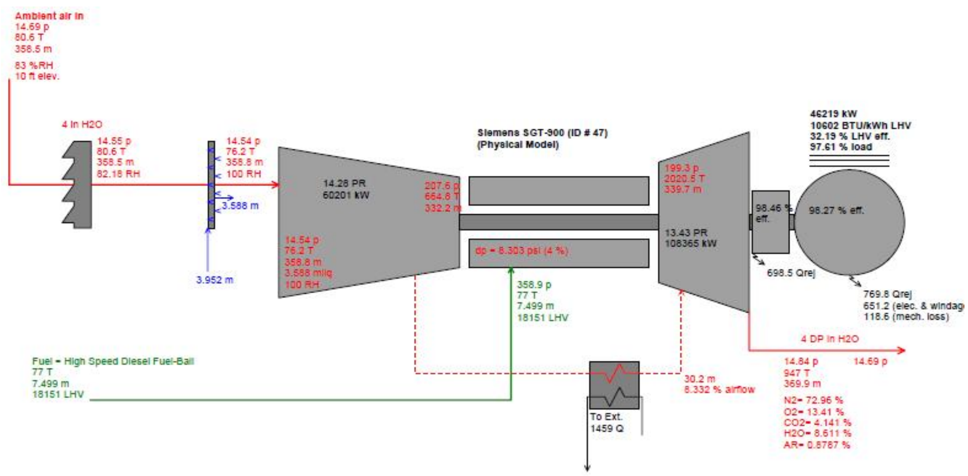
Table 5. Performance improvement results for the two units of gas turbine (GT) in Pesanggaran plant

Parameter	Saturated fogging		1% over spray		2% over spray	
	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	16,615	41,522	18,648	46,219	20,420	50,244
New parasitic load, kW	9	16	34	53	99	92
Additional net power, kW	403	763	2,411	5,423	4,118	9,409
Base heat rate, BTU/kWh	13695.0	11004.0	13695.0	11004.0	13695.0	11004.0
New heat rate, BTU/kWh	13543.0	11006.0	12934.0	10602.0	12583.0	10361.0
Increm. heat rate, BTU/kWh	-152.0	2.0	-761.0	-402.0	-1112.0	-643.0
Base SFC, l/kWh	0.387	0.311	0.387	0.311	0.387	0.311
New SFC, l/kWh	0.383	0.311	0.365	0.299	0.355	0.293
Incremental SFC, l/kWh	-0.0043	0.0001	-0.0215	-0.0114	-0.0314	-0.0182
Base thermal efficiency	24.92%	31.01%	24.92%	31.01%	24.92%	31.01%
New thermal efficiency	25.20%	31.00%	26.38%	32.19%	27.12%	32.93%
Incremental thermal efficiency	0.28%	-0.01%	1.46%	1.18%	2.20%	1.92%

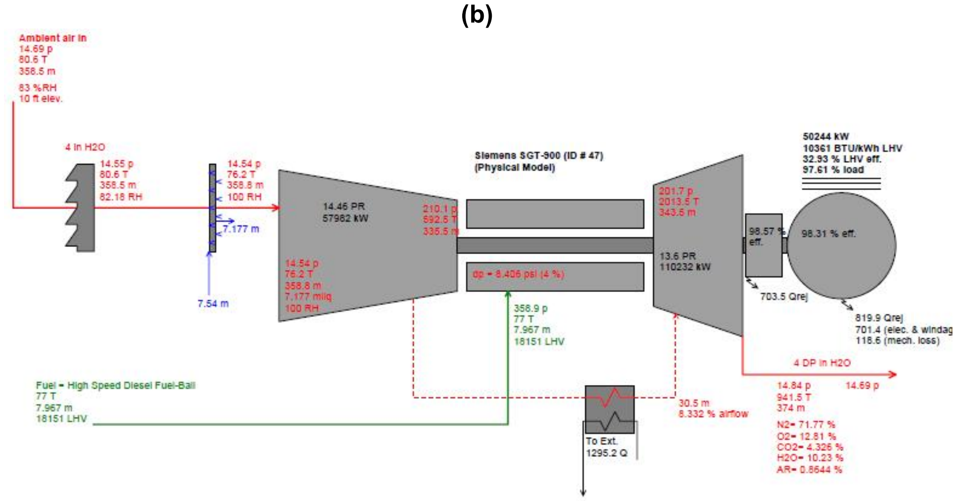


(a)

Fig. 5. Simulation result of new GT performance with degradation factor for (a) saturated fogging



(b)



(c)

Fig. 5. Simulation result of new GT performance with degradation factor for (b) 1% OS, and (c) 2% OS (continued)

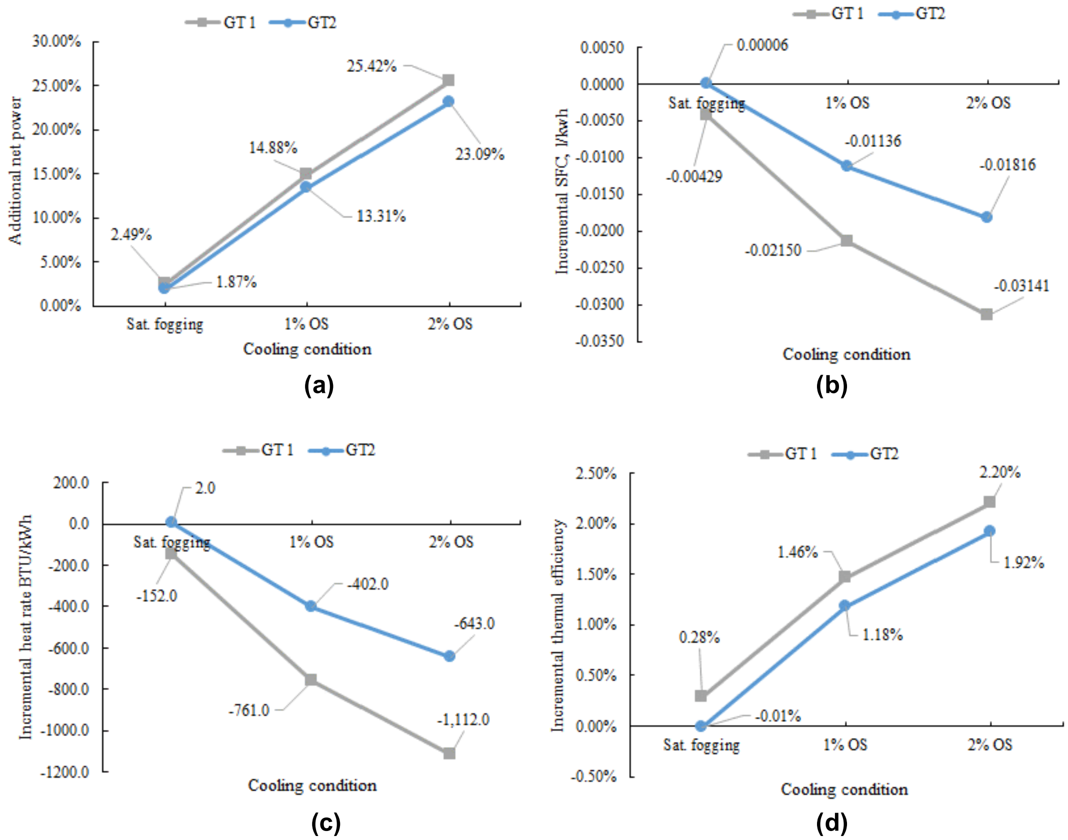


Fig. 6. Performance improvement for the two gas turbine (GT) units of the Pesanggaran plant: (a) additional net power, (b) incremental SFC, (c) incremental heat rate, and (d) incremental thermal efficiency

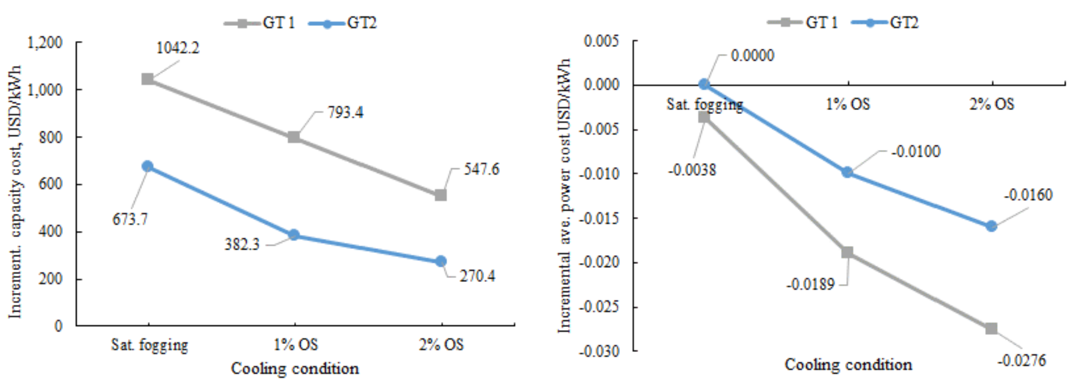
Table 6. Capacity cost evaluation results for the three gas turbine (GT) units in the Pesanggaran plant

Parameter	Saturated fogging		1% over spray		2% over spray	
	GT 1	GT 2	GT 1	GT 2	GT 1	GT 2
Additional net power, kW	403	763	2411	5423	4118	9409
Cost base plant, kUSD	22,232	33,379	22,232	33,379	22,232	33,379
Cost new plant, kUSD	22,577	33,793	23,040	34,426	23,295	34,761
Cost Demin/RO plant, kUSD	75	100	405	276	492	412
Compressor component recoating, kUSD	0	0	700	750	700	750
Estimated cost of cooling system, kUSD	420	514	1,213	1,323	1,555	1,794
Capacity cost of cooling system, USD/kWh	1,042.2	673.7	503.1	244.0	377.6	190.7

For GT1 and GT2, there is a decrease in production costs for both fogging and wet compression, unless the production cost in fogging system on GT2 is almost identical to the existing condition. From the reduction of the production cost, the payback period of the investment of the cooling system can be estimated by comparing against the cost of the investment system. Based on results, the fastest payback period using the wet compression system and 2% OS for GT1 and GT2 is ± 9 and 10 years respectively. For 1% OS, the payback period for both GT1 and GT2 is 20 years. Thus the payback period of a fogging system is very uneconomical when compared with the cost of the investment, far exceeding the operational life limits of both gas turbine units.

Table 7. Operating cost evaluation results for the three gas turbine (GT) units in the Pesanggaran plant

Parameter	Saturated fogging		1% over spray		2% over spray	
	GT 1	GT 2	GT 1	GT 2	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	33212.0	83012.0	37228.0	92332.0	40642.0	100304.0
Base annual fuel consumption, kliter	12535.9	25328.1	12535.9	25328.1	12535.9	25328.1
New annual fuel consumption, kliter	12705.1	25807.2	13601.0	27650.9	14445.4	29355.5
Base annual fuel cost, kUSD	11031.6	22288.8	11031.6	22288.8	11031.6	22288.8
New annual fuel cost, kUSD	11180.5	22710.3	11968.9	24332.8	12711.9	25832.8
Base average power cost, USD/kWh	0.3404	0.2735	0.3404	0.2735	0.3404	0.2735
New average power cost, USD/kWh	0.3366	0.2736	0.3215	0.2635	0.3128	0.2575
Incremental Ave. power cost, USD/kWh	-0.0038	0.0000	-0.0189	-0.0100	-0.0276	-0.0160



(a) (b)

Fig. 7. Economic analysis results of the two unit of gas turbine (GT) in Pesanggaran plant: (a) incremental capacity cost and (b) incremental average power cost

4. CONCLUSION

Based on the simulation results using GTPro, the performance enhancement due to the application of wet compression system is quite significant compared to saturated fogging. The maximum incremental power due to wet compression system for GT1 and GT2 occurred with 2% OS and are 25.42% and 23.09%, respectively. Based on specific fuel consumption and heat rate, a wet compression system has a ratio of fuel consumption to power generated lower (much more economical) than conventional fogging and therefore increases the thermal efficiency of the system. The maximum enhancements of thermal efficiency in GT1 and GT2 using wet compression system at 2% of OS are 2.20% and 1.92%, respectively. Furthermore economic aspects were analyzed in terms of capacity and production costs using PEACE. Based on the simulation results, the incremental capacity cost due to the investment of a wet compression system is lower than conventional fogging and in line with the increasing of the overspray percentage. If the maximum reduction of annual production costs on GT1 and GT 2 is compared to the investment cost of the cooling system, the fastest payback periods with a wet compression system were obtained with 2% OS and were 9 and 10 years for GT1 and GT2, respectively. It can be concluded that a wet compression system is more effective and economical than conventional fogging for improving the performance of gas turbines. However, experimental work to validate thermodynamic modelling and the effects of adding water to gas turbine components needs be further investigated.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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Peer-review history:
The peer review history for this paper can be accessed here:
http://sciencedomain.org/review-history/10355