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THERMAL EFFICIENCY ANALYSIS OF COMBUSTION TURBINE GENERATOR (821-G-101) PT TRANS PACIFIC PETROCHEMICAL INDOTAMA AFTER MAJOR INSPECTION

Moh. Arif Batutah [1) ✉], Achmad Safi'i [1)], Hadi Kusnanto [1)], Indah Kurniawati [2)]

[1)] **Prodi Teknik Mesin**
Fakultas Teknik, Universitas Muhammadiyah Surabaya
Jl. Raya Sutorejo No. 59 Surabaya, 60113
Email : arifbatutah@ft.um-surabaya.ac.id

[2)]**Prodi Teknik Elektro**
Fakultas Teknik, Universitas Muhammadiyah Surabaya
Jl. Raya Sutorejo No. 59 Surabaya, 60113

Abstract

The gas turbine is one of the most widely used equipment in oil and gas industries as an electric generator driving system. To maintain the reliability and performance of the gas turbine, a major inspection program is required. Thermal efficiency analysis is the parameters of success in major inspection activities. Thermal efficiency analysis is carried out by comparing data and operating parameters of gas turbine before and after the major inspection. The analysis aims to determine the actual efficiency of each component, and also the actual overall efficiency in the gas turbine. The study result is the actual thermal efficiency of a gas turbine after a major inspection has increased from 12.20% to 12.84% at a load of 6 MW, an increase from 15.76% to 16.77% at a load of 9 MW, and an increase of 18.59% to 20.16% at a load of 12 MW. Based on the actual performance comparison data with design data, it is known that the overall actual performance is still below the design performance, both heat efficiency, heat consumption and heat rate. This is caused by the operating lifetime.

Keywords: *Thermal Efficiency, Gas Turbine, Major Inspection*

Corresponding Author:
✉ **Moh Arif Batutah**
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1. INTRODUCTION

PT Trans-Pacific Petrochemical Indotama (PT TPPI) is a petrochemical company that produces petroleum / oil and gas and aromatics. In the production process, PT TPPI processes around 100,000 barrels of condensate / day. The need for electricity in an industry such as the PT TPPI refinery is very important as the main condition for the operation of the refinery. Therefore, the continuity and reliability of the electric power system is expected to be optimal in order to ensure smooth operation and production results of the highest quality. PT TPPI has two sources of electricity, namely electricity from the State Electricity Company (PLN) and electric power from PT TPPI's generating unit that uses Combustion Turbine Generator (CTG). The power source from PLN has a capacity of 10.2 MW, while electricity from PT TPPI's own generating unit is through 3 (three) CTGs, namely CTG 101, CTG 102 and CTG 103, each of which has a capacity of 21.5 MW. The electrical power requirement for PT TPPI's operations is around 27 MW, and for reliability reasons, the three CTG units operate with a load of around 9 MW each.

The working principle of CTG is to utilize hot gas from the combustion of fuel and air in the combustion chamber, then used to rotate the turbine that is similar to the compressor [1,2,3]. The gas turbine will rotate the generator so that it is able to produce electricity [4]. In

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terms of thermal efficiency, CTG units are classified as thermal units with the lowest efficiency, ranging from 20-40% [5]. The Brayton cycle is the basic principle of gas turbine operation [6,7]. The CTG unit at PT TPPI is included in the simple cycle. Flue gas from the CTG exhaust unit is utilized by the Heat Recovery Steam Generator (HRSG) to produce hot steam to meet steam needs in the process area.

Several gas turbine power plant efficiency analyses were carried out to determine its effectiveness such as Komarudin and Fauzi, 2017 [8], Bambang et al., 2017 [9]., Rafea et al., 2022 [10]. In the process, the gas turbine unit is composed of major components such as compressors, turbines, combustion chambers, and generators. The equipment if used continuously will decrease in performance and reliability, due to reduced service life (lifetime) so that this will be directly related to the efficiency of the gas turbine unit. Large energy losses in gas turbines can occur in one or more of the components of the gas turbine unit [11]. Therefore, to maintain the operation of the gas turbine unit in optimal and reliable condition, maintenance program actions are carried out, one of which is a major inspection (MI) [12].

One of the parameters that shows the success rate of MI activities in the CTG unit is that the performance and efficiency of the gas turbine system has increased compared to before the major inspection [13]. Based on the description above, the study will conduct an analysis to determine the performance of gas turbine components, thermal efficiency values, heat efficiency comparisons, heat consumption, heat reat and knowing the factors that affect the improvement of CTG 101 gas turbine performance at PT TPPI before and after the major inspection.

2. RESEARCH METHODS

This research method is qualitative where data is taken on the object and then analyzed based on the calculation of equation 1 – equation12, The specification data of the Combustion Turbine Generator unit at PT TPPI is as below.

General Data and Construction Features of Gas Turbine following:

Manufacturer	: GE Nuovo Pignone, Italy
Model series	: MS5001PA
Type	: Single shaft
Cycle	: Simple
Shaft rotation	: CW viewed from air outlet
Type operation	: Continuous
ISO rating	: 26300 KW
Speed	: 5100 rpm
Air compressor	: 17 stages, axial flow type, max tip speed 332 m/s, IGV (64 blades) and EGV (2 stages)
Turbine	: 2 Stages, axial type, max tip speed 438 m/s.
Combustor	: 10 units, annular and dual fuel type, max temp. Spark plug 2 unit, flame scanner
Fuel nozzle	: one per combustor, mix fuel

Figure 1. Unit CTG 101 at PT TPPI

The Brayton cycle is the ideal thermodynamic power cycle for gas turbines. In the Bryton cycle, each process state can be analyzed as follows: [14,15]:

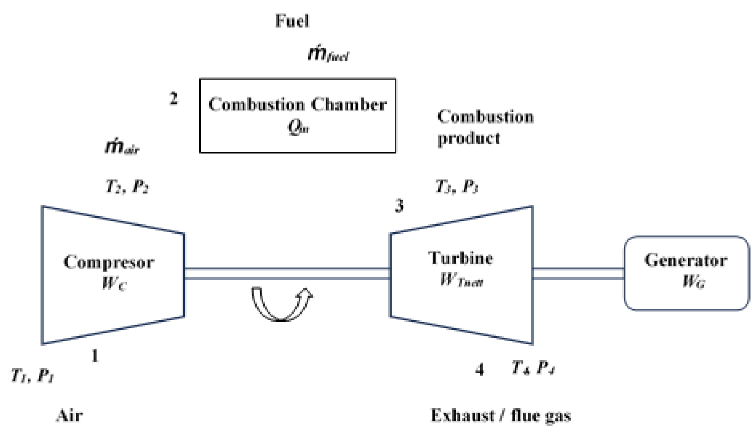

Figure 2. Brayton Cycle Flow Diagram at PT TPPI' CTG 101 Unit

a. Process 1→2 (isentropic compression)

The air is compressed by the compressor to a certain pressure. This process is not followed by a change in entropy, so it is called the isentropic process. Because processes 1-2 are isentropic processes like Equation 1 follows:

$$\frac{T_{2S}}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \tag{1}$$

where T_1 and T_{2S} are temperature in and out at compressor (K), P_1 and P_2 are pressure in and out of compressor (bar), and k is heat capacity ratio at compressor (1.4), the compressor works ($\acute{W}_c$) in the process as Equation 2 follows:

$$\acute{W}_c = \dot{m}_{air}\,(h_2 - h_1) \tag{2}$$

$\acute{m}_{air}$ is air mass flow rate (kg/s), h_1 and h_2 are air enthalpy in and out of compressor (kJ/kg)

b. Process 2→3 (isobaric combustion)

At process 2-3, compressed air enters the combustion chamber. Fuel is injected into the combustion chamber, and followed by the combustion process. Because of the constant pressure, this process is called isobaric. The heat energy produced (Q_{in}) by fuel and is obtained from Equation 3:

$$Q_{in} = \dot{m}_{fuel}\, x\, LHV \quad (3)$$

Or it can also be obtained by the following Equation 4,

$$Q_{in} = (\dot{m}_{air} + \dot{m}_{fuel})(h_3 - h_2) \quad (4)$$

where $\dot{m}_{fuel}$ is fuel mass flow rate (kg/s), *LHV* is low heating value of fuel, and h_3 is enthalpy combustion product (kJ/kg).

c. Process 3→4 (isentropic expansion)
Compressed air that has absorbed the heat from combustion, expansion occurs through the turbine. Since processes 3-4 are isentropic processes as equation 5, then:

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}} \quad (5)$$

where T_3 and T_4 are temperature in and out of turbine (*K*), P_3 and P_4 are pressure in and out of turbine (bar), and k is heat capacity ratio at turbine (1.33), for the work required turbine ($\dot{W}_T$), and h_4 is flue gas enthalpy (kJ/kg) as Equation 6 below:

$$\dot{W}_T = (\dot{m}_{air} + \dot{m}_{fuel})(h_3 - h_4) \quad (6)$$

d. Process 4→1 (Heat dissipation)
In the ideal Brayton cycle, the air coming out of this turbine still leaves a certain amount of heat energy (Q_{out}) as Equation 7 follows:

$$Q_{out} = (\dot{m}_{air} + \dot{m}_{fuel})(h_4 - h_1) \quad (7)$$

Air fuel ratio, Specific Fuel Consumption, and Gas Turbine Efficiency are obtained by Equation 8-12, respectively below [14,15],

1. Air fuel ratio (AFR)

$$\frac{A}{F} = \frac{\dot{m}_{air}}{\dot{m}_{fuel}} \quad (8)$$

2. Specific fuel consumption (SFC)

$$SFC = \frac{\dot{m}_{bb} \text{ x } 3600}{\dot{W}_{Tnett}} \quad (9)$$

3. Efficiency
   a. Compressor efficiency (η_c)

$$\eta_C = \frac{T_{2S} - T_1}{T_2 - T_1}\, x\, 100\% \quad (10)$$

   b. Turbine efficiency (η_T)

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4S}}\, x\, 100\% \quad (11)$$

   c. Thermal cycle efficiency (η_s)

$$\eta_S = \frac{\dot{W}_{Tnett}}{Q_{in}}\, x\, 100\% \quad (12)$$

3. RESULTS AND DISCUSSION

Gas turbine operation data is obtained from process trend data (PHD), field log sheet data, and DCS log sheet from the CTG 101 gas turbine unit. In this study, the variation in generator load at 6 MW (28% base load), 9 MW (42% base load) and 12 MW (56% base load), because the normal operation of this gas turbine and the electricity demand at the refinery are in that range.

3.1. Analysis and Discussion of CTG 101 Gas Turbine Performance

while the operational data of PT Trans-Pacific Petrochemical Indotama's CTG 101 unit before and after the major inspection as shown in **Table 1**.

Table 1. Operating condition CTG 101 before and after MI

Parameter	value before MI			value after MI			Unit
Power, W_{sp}	6.0	9.0	12.0	6.0	9.0	12.0	MW
Actual power, W_{act}	6.02	9.06	12.01	6.04	9.05	12.03	MW
compressor inlet temp., T_1	303.26	302.54	302.97	302.48	302.46	302.84	K
ambient pressure, P_{amb}	101.36	101.36	101.36	101.36	101.36	101.36	kPa
Pressure drop inlet filter, Δp_1	0.93	0.95	0.96	0.26	0.27	0.29	kPa
Compressor outlet temp., Δp_1	577.86	592.83	598.53	579.15	594.52	600.59	K
Compressor outlet pressure, P_2	704.53	770.72	794.51	740.72	810.89	836.10	kPa
Turbine exit temperature, T_4	644.18	661.89	703.97	648.07	665.96	708.46	K
Turbine outlet pressure, P_4	102.56	102.69	102.74	102.56	102.70	102.75	kPa
Fuel			Kerosene				-
Low heating value, LHV	46018	46018	46018	45705	45705	45705	kJ/kg
Fuel flow rate, $\dot{m}_{bb}$	1.121	1.289	1.439	1.077	1.218	1.338	kg/s
Generator efficiency, η_G	95.60	96.90	97.55	95.60	96.90	97.55	%

A. Comparison of CTG 101 Gas Turbine Performance before, after major inspection

1. Compressor Isentropy Efficiency Comparison

The increase in compressor efficiency after major inspection on generator load variations of 6 MW, 9 MW and 12 MW, respectively by 1.56%, 1.68% and 1.70%. The increase occurred due to an increase in the pressure ratio produced by the compressor up about 4-5% from before the major inspection in **Figure 3**. Dirty rotor blades and compressor stators (fouling) will reduce the air compression process on the stove due to increased pressure drop due to fluid friction losses. So that the pressure produced is not optimal and after cleaned during major inspection, the pressure ratio is increased. So that the efficiency of the compressor will rise because the isentropic temperature of the compressor (T2s) rises.

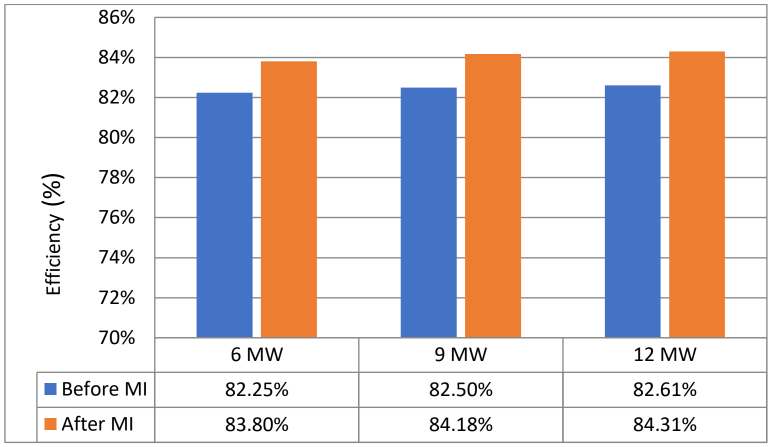

Figure 3. Compressor Efficiency Comparison of CTG 101

2. Turbine Icenttropy Efficiency Comparison

The increase is also caused by the increase in the pressure ratio in the compressor so that the turbine efficiency also increases. An increase in compressor outlet pressure (P2) results in an increase in temperature (T3) and pressure (P3) in combustion products entering the turbine, So that the enthalpy of the fluid (H3) also rises in **Figure 4**. The increase in efficiency in the turbine was also influenced by the replacement of the stage 2 bucket which was found to have suffered severe damage before a major inspection, as well as the replacement of stage 1 and 2 shrouds, so that the blade tip clearance returned to manufacturer standard

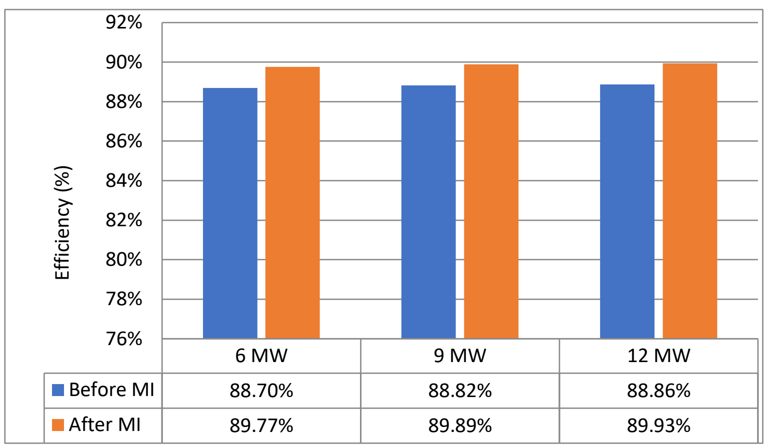

Figure 4. Turbine Efficiency Comparison of CTG 101

3. Cycle Efficiency Comparison

Increase in gas turbine cycle efficiency after major inspection on generator load variations of 6 MW, 9 MW and 12 MW, respectively by 0.54%, 1.01% and 1.57%. After a major inspection, the CTG 101 gas turbine experienced a decrease in fuel consumption even though the LHV value of fuel also decreased slightly. This results in the required heat (Qin) also falling at the same generator load output, so that the efficiency of the gas turbine cycle increases in **Figure 5**.

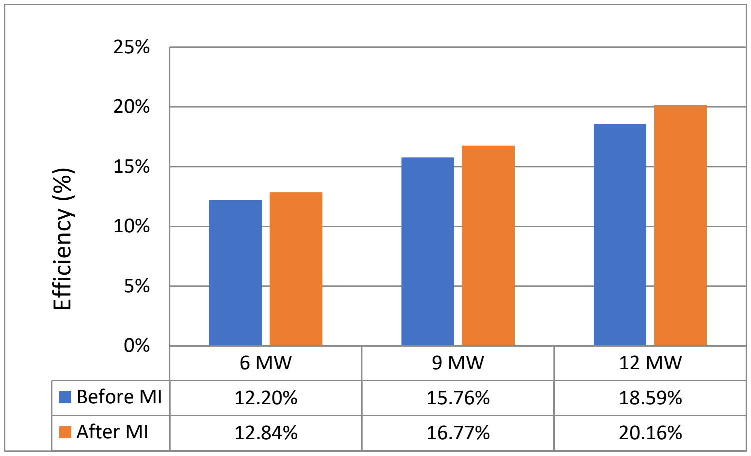

Figure 5. Cycle efficiency comparison of CTG 101

B. Comparison of CTG 101 Gas Turbine Performance with Design Data

The following are the values of heat efficiency, heat consumption and heat rate before and after a major inspection of PT Trans-Pacific Petrochemical Indotama's CTG 101 gas turbine as shown in **Table 2** below:

Table 2. Comparison of Actual Heat Efficiency CTG 101 with Data Design

Daya (MW)	*Heat Efficiency* (%)			*Heat Consumption* (MJ/h)			*Heat Rate* (MJ/kWh)		
	before MI	after MI	*Design*	before MI	after MI	*Design*	before MI	after MI	*Design*
6	12.20	12.84	14.91	185773	177129	143267	29.50	28.04	22.95
9	15.76	16.77	18.86	213602	200481	172773	22.85	21.47	18.45
12	18.59	20.16	21.20	238421	220224	205259	19.37	17.86	16.07

From the actual comparison data of performance with data design, it is known that the overall actual performance is still below the performance design, both heat efficiency, heat consumption and heat rate in **Figure 6**, **7**, and **8** respectively. This can be influenced by several factors, namely a decrease in generator efficiency, a decrease in compressor rotor performance, a decrease in the performance of auxiliary components. These three factors are caused by the lifetime that has reached 73185 running hours, so further studies need to be carried out on these components, In general, all the heat in the gas turbine generator process will not be absorbed or used all, because it will be lost convection along with the exhaust gas, conduction heat loss also occurs in the steam generator turbine components.

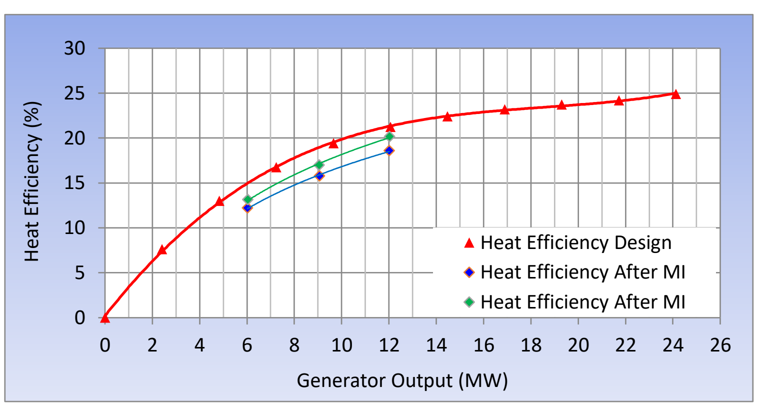

Figure 6. Comparison of Actual Heat Efficiency CTG 101 with Data Design

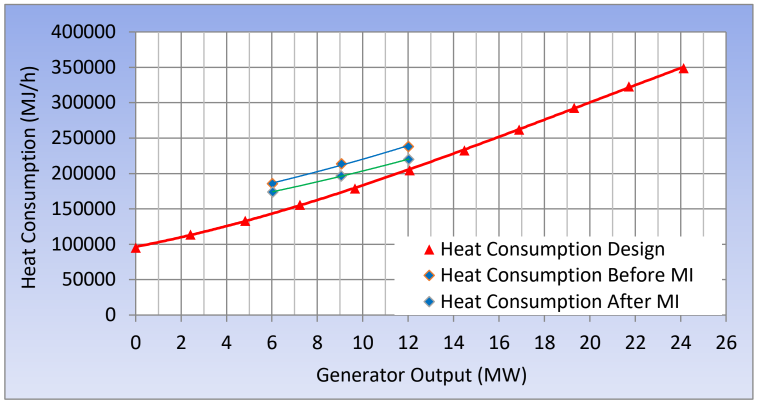

Figure 7. Comparison of Actual Heat Consumption CTG 101 with Data Design

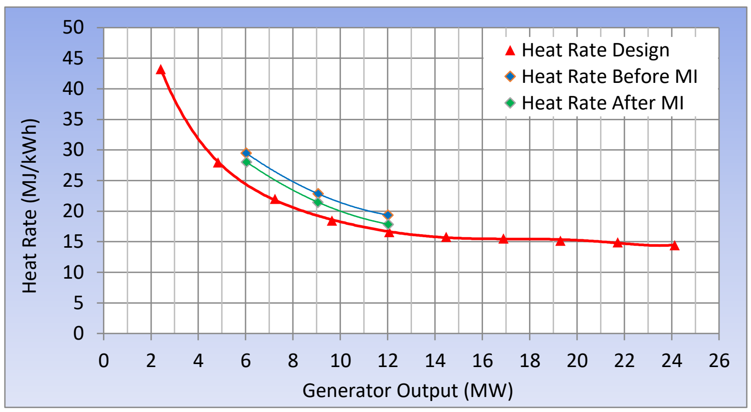

Figure 8. Comparison of Actual Heat Rate CTG 101 with Data Design

3.2 Factors Improving CTG 101 Gas Turbine Performance After MI

There are several factors that influence the overall performance increase in the CTG 101 gas turbine as shown in **Table 3** as follows:

Table 3. Pressure Drop Comparison of IAF CTG 101

Daya (MW)	*Pressure drop* IAF (inH_2O)		Rasio tekanan kompresor		*Exhaust Temperature Spread* (°C)		
	before MI	after MI	before MI	after MI	before MI	after MI	*Allowable Spread*
6	3.74	1.03	7.02	7.33	16	6	36
9	3.82	1.10	7.68	8.02	21	8	41
12	3.87	1.15	7.91	8.02	28	11	47

a. Replacement of Air Inlet Filter

Air inlet filter (IAF) is an ambient air filter that will enter the compressor from dirt / dust / foreign material. The inlet filter water **Figure 9** itself has a service life and will experience blocking/particle fusion when there is a lot of dirt/dust attached to the filter membrane. This can be seen from the pressure drop (Δp) parameter on the manometer attached to the water inlet filter [5,15].

b. Fouling cleaning in IGV blades, compressor rotors and stators

Dirty rotor blades and compressor stators (fouling) can reduce the air compression process on the stove due to increased pressure drop due to fluid friction losses. The increase in friction losses occurs due to friction between the fluid and the rotor blade surface and rough contact surface of the compressor due to dirt / fouling attached to the blade surface. Rotor blades and compressor stators **Figure 10** that have been cleaned during major inspection, hence the pressure ratio (P2/P1) increases [5,15].

c. Turbine Bucket Replacement

The stage 2 bucket turbine of **Figure 11** was found to have suffered severe damage during major inpection. Damage to the stage 2 turbine bucket is caused by friction (rubbing) with the seal plate on the exhaust plenum that is damaged first. Damage to the stage 2 turbine bucket is at the end of the bucket (blade tip) with a damage area of about 5-10%. This results in a decrease in the cross-sectional area of the bucket, so that the kinetic energy of the hot gas rate cannot be converted optimally to rotate the turbine [5,15].

d. Hot Part Turbine Replacement

Hot part turbine **Figure 12** replacement includes replacement of new parts nozzle stage 1, nozzle stage 2, shroud stage 1 and shroud stage 2. The replacement of the part affects the increase in efficiency on the turbine side, because the clearance returns to the manufacturer's standard. The clearance in question is blade tip clearance and seal strip clearance. Before a major inspection, there are several points whose clearance is beyond the manufacturer's tolerance due to wear and/or service life [5,15].

e. Fuel System Part Replacement

Replacement of parts in the fuel system of **Figure 13** includes replacement of fuel nozzle, fuel check valve and flow divider. During a major inspection, damage to the fuel nozzle was found, namely wear on the swirl tip plate and nozzle tip. Damage to the tip nozzle results in an incomplete atomization process between liquid fuel (kerosene) and atomizing air, resulting in a less complete combustion process. One indicator of complete and flat combustion in gas turbines is a flat turbine exit temperature or low exhaust temperature spread [5,15].

f. Combustion Chamber Part Replacement

Replacement parts in the combustion chamber of **Figure 14** include combustion liner, transition piece, crossfire tube. During a major inspection, damage to the combustion liner was found, namely wear on the hula seal. Damage to the hula seal results in hot gas losses due to a sealing system that is not optimal, thus resulting in the flow rate of hot gas from the combustion chamber to the turbine will be reduced due to internal recirculation of the combustion product to the dilution gap of water and cooling water [5,15].

Figure 9. IAF conditions before MI CTG 101

Figure 10. Fouling conditions on the Compressor Rotor Blade before MI CTG 101

Figure 11. Turbine Stage 2 Bucket Damage before MI CTG 101

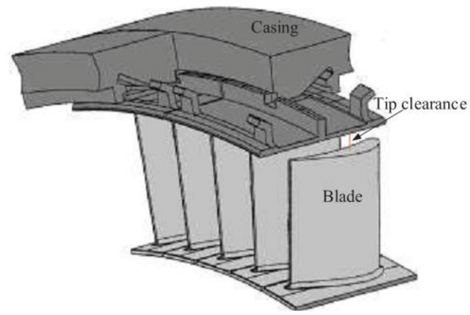

Figure 12. Blade Tip Clearance on Turbine

Figure 13. Fouling Conditions on the Fuel Nozzle Before MI CTG 101

Figure 14. Hula Seal Damage to Combustion Liner Before MI CTG 101

4. CONCLUSION

From the results of performance, efficiency and exergy analysis on the CTG 101 unit before and after the major inspection at PT Trans-Pacific Petrochemical Indotama can be concluded that:

The actual thermal efficiency of the gas turbine after major inspection increased from 12.20% to 12.84% at 6 MW load, increased from 15.76% to 16.77% at 9 MW load, and increased from 18.59% to 20.16% at 12 MW load. Comparison of actual performance with data design, it is known that overall actual performance is still below performance design, both heat efficiency, heat consumption and heat rate. This is due to service life. Factors that improve the performance of the CTG 101 gas turbine after major inspection, namely replacement of inlet filter water, cleaning of fouling (soot) in the IGV blade, rotor and stator blade, replacement of turbine bucket, replacement of hot turbine parts, replacement of fuel system parts and replacement of combustion chamber parts.

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