Optimum Performance Based on Specific Fuel Consumption of the Intercooled – Reheat Gas Turbine Combined Cycle

Ng.T.G. Tri, M.H. Nugraha, T. Hardianto, H. Riyanto
Department of Mechanical Engineering, Faculty of Mechanical and Aerospace Engineering,
Institut Teknologi Bandung, Jl. Ganesa 10, Bandung 40132, Indonesia
Email: giatri.hcmut@gmail.com ; hafilnugraha@hotmail.com

Abstract

Herein is presented the results of optimization to a combined cycle power generation in which the topping cycle is an intercooled-reheat Brayton cycle and the bottoming cycle is a Rankine cycle with a single-pressure-heat-recovery-steam-generator. The specific fuel consumption (SFC) of the combined cycle is considered as the parameter of power generation performance to be optimized. This study compares the improvement performance including overall efficiency, power output by the effect of turbine inlet temperature and compression ratio on the simple gas turbine combined cycle (GTCC) and the intercooled-reheat gas turbine combined cycle (IHGTCC). Realistic input data have been considered for this study. The results show that at constant of compression ratio of 16, comparing with variation of TIT, at value of TIT around 1050°C, the SFC of both cycles are almost the same while the higher overall efficiency by 5% along with increasing in power output by 111% for the IHGTCC. At constant of TIT of 1200°C, in the range of compression ratio, the results present that the IHGTCC raises 5% in overall efficiency and presenting impressive augmentation by almost 134% in power output with the same of the SFC of the simple GTCC at compression ratio of 28.

Keywords: Combined cycle performance, turbine inlet temperature, compression ratio, specific fuel comsumption

Introduction

A combined cycle couples two power cycle such that energy discharged by heat transfer from one cycle is used partly or wholly as the input for the other cycle. The first cycle used is gas turbine cycle and the second is steam turbine (ST) cycle. The two power cycles are coupled. High temperature gas from gas turbine cycle is used as an input heat in vapor cycle to produce steam. With the use of two working fluids, air (gases) and water (steam), this cycle utilize Brayton cycle in GT and Rankine cycle in ST.

A Gas Turbine Combined Cycle (GTCC) power plants having Brayton cycle based topping cycle and Rankine cycle as bottoming cycle. The waste exhaust gas temperature is decreased as it flow through the Heat Recovery Steam Generator (HRSG) which consist of superheater, evaporator and economizer. Then the HRSG uses as a steam generator which is supplied it for turbine expansion and produce electricity as the second power generation.

The gas turbine is one of the most satisfactory mechanical power producing engines in industrial world. Thermodynamic process such as compression, combustion and expansion are performed in this cycle with each components to perform the processes which are compressor, combustion chamber and turbine. The whole process in the cycle is called Brayton cycle.

The principle of gas turbine is the atmospheric air is continuously drawn into the compressor, where it is compressed to a high pressure [1]. The air then enters a combustion chamber, or combustor, where it is mixed with fuel and combustion occurs, resulting in combustion products at an elevated temperature. The combustion products which is high temperature gas will be expanded into the turbine that connected to the shaft of generator to produce work.

Two modifications of the basic gas turbine that increase the net work developed are multistage expansion with reheat and multistage compression intercooling. When used both of the modifications, it can increase the thermal efficiency. The GTCC which already modified in this way is called intercooled-reheat gas turbine combined cycle (IHGTCC). The performance of

GTCC depends on the individual performance of the gas and ST cycle [2].

The study related to GT thermodynamic analysis are used to measure performance of the GTCC by measuring several parameters with modification of the cycle into the IHGTCC. Modification of this method can increase the performance of combined cycle for the optimum of specific fuel consumption (SFC). Thus the aim of present study is to develop thermodynamic analysis to find optimum performance of the GTCC and IHGTCC by utilizing the effect of turbine inlet temperature and compression ratios.

Modelling of the simple Gas Turbine Combine Cycle (GTCC)

A schematic of the simple GTCC with bottoming cycle using a single pressure of HRSG without reheating is illustrated in Figure 1.

To enable burning of natural gas as a fuel, a combustion chamber needed for a combustion process for producing high temperature gas as it compressed by the air compressor before entering the combustion chamber. Next, the GT which is linked to shaft is expanded and producing electricity. The flue gas is flowing into HRSG in which occur the heat transfer process in superheater, evaporator and economizer. Electricity produced with the transmission of steam by the HRSG into the ST [3].
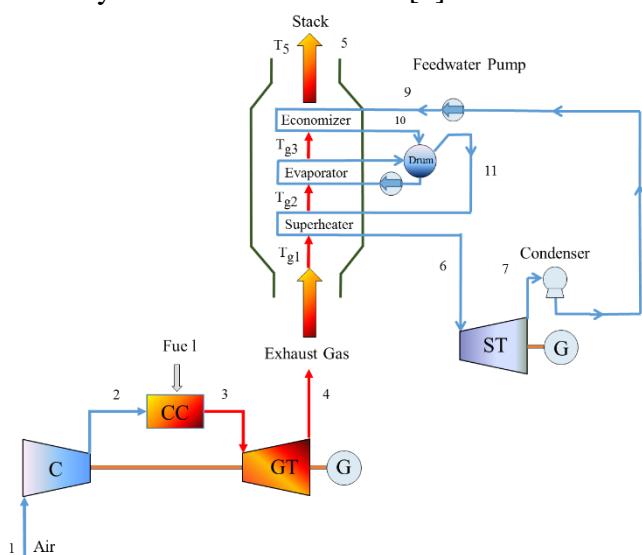

Figure 1. The simple GTCC with single pressure HRSG.

Gas Turbine Model

Four main components of gas turbine are compressor (C), combustion chamber (CC), gas turbine (GT) and generator. Figure 2 shows the schematic diagram of gas-turbine cycle without intercooled-reheat.
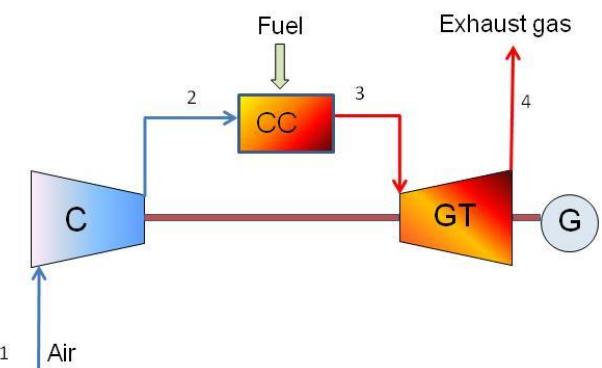

Figure 2. Schematic diagram of GT cycle.
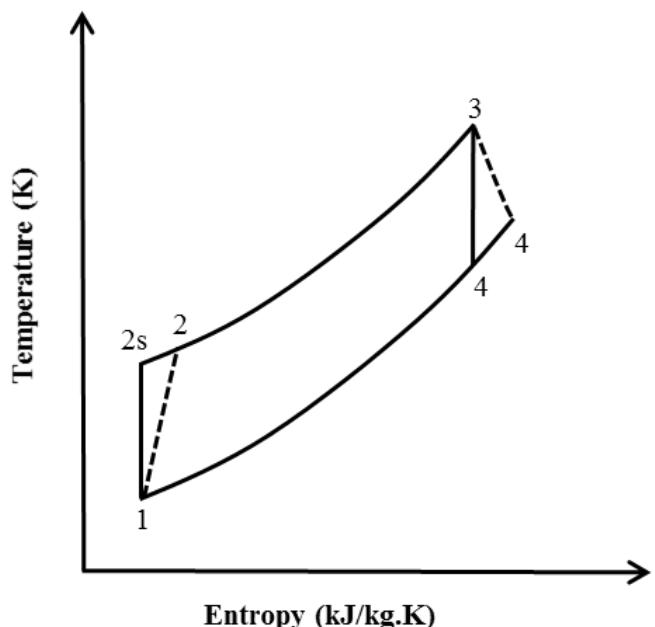
Figure 3. Temperature-entropy diagram of GT cycle

In accordance with the assumptions of an air-standard analysis, the temperature rise that would be achieved in the combustion process is brought about by a heat transfer to the working fluid from an external source and the working fluid are considered be air as an ideal gas. The air would be drawn into the compressor, transmitted into the combustion chamber in

order to combine with fuel for producing high temperature flue gas and finally expand in gas turbine to produce work.

In Eq. (1), the compression ratio is defined by the ratio of p_1 and p_2 (bar) which are the pressure of the air at the inlet and outlet sections of the compressor respectively:

$$r_p = \frac{p2}{p1} \qquad (1)$$

Eq. (2) presents the formula of the outlet temperature of the compressor:

$$T_2 = T_1 . \left[1 + \frac{1}{\eta_c}\left(r_p^{\frac{k-1}{k}} - 1\right)\right] \qquad (2)$$

Where T_1 and T_2 (°C) denote the temperature of the air at the inlet and outlet sections of the compressor respectively. k=1.4 is specific heat ratio and η_c is the isentropic efficiency of compressor.

Eq. (3) is used to calculate consumed power by the compressor:

$$W_c = c_{pa}(T_2 - T_1) \qquad (3)$$

Eq. (4) gives the specific heat (kJ/kg.K) of air according to the changing temperatures:

$$c_{pa}(T) = 1.048 - 1.83T + 9.45 \times 10^{-7}T^2$$
$$-5.49 \times 10^{-10}T^3 + 7.92 \times 10^{-14}T^4 \qquad (4)$$

Eq. (5) defines the specific heat of flue gas:

$$c_{pg}(T) = 1.8083 - 2.3127x10^{-3}T$$
$$+ 4.045x10^{-6}T^2 - 1.7363x10^{-9}T^3 \qquad (5)$$

Eq. (6) indicates the energy balance in combustion chamber :

$$\dot{m}_a C_{pa}T_2 + \dot{m}_f LHV + \dot{m}_f C_{pf}T_f = \dot{m}_g . C_{pg}.TIT \qquad (6)$$

Where $\dot{m}_a$, $\dot{m}_f$ and $\dot{m}_g$ are air, fuel and flue gas mass flow rate (kg/s) respectively, $\dot{m}_g = \dot{m}_a + \dot{m}_f$, LHV is low heating value (kJ/kg), C_{pf} is specific of fuel, T_f is temperature of the fuel, TIT=T_3 is turbine inlet temperature.

Eq. (7) represents for the fuel air ratio deriving from Eq. (6):

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{C_{pg} TIT - C_{pa} T_2}{LHV - C_{pg} TIT} \qquad (7)$$

Eq. (8) describes the exhaust gases temperature from the gas turbine:

$$T_4 = TIT . \left[1 - \eta_t \left(1 - \frac{1}{r_p^{\frac{k-1}{k}}}\right)\right] \qquad (8)$$

Eq. (9) shows the shaft work of turbine:
$$W_t = c_{pg}(TIT - T_4) \qquad (9)$$

Eq. (10) expresses the net work of GT :

$$W_{GT} = W_t - W_c = c_{pg} \times TIT \times \eta_t \left(1 - \frac{1}{r_p^{\frac{k-1}{k}}}\right) -$$
$$c_{pa} \times T_1 \left[1 + \frac{1}{\eta_c}\left(r_p^{\frac{k-1}{k}} - 1\right)\right] \qquad (10)$$

Eq. (11) shows the net power output of GT:
$$P_{GT} = \dot{m}_g \times W_{GT} \qquad (11)$$

Eq. (12) is used to calculate the specific fuel consumption:
$$SFC = \frac{3600 .f}{W_{GT}} \qquad (12)$$

Eq. (13) is formula for calculating the heat supplied:

$$Q_{add} = c_{pg}\left\{ TIT - T_1 \left[1 + \frac{1}{\eta_c}\left(r_p^{\frac{k-1}{k}} - 1\right)\right]\right\} \qquad (13)$$

Eventually, Eq. (14) depicts the GT thermal efficiency:

$$\eta_{GT} = \frac{W_{GT}}{Q_{add}}$$

$$= \frac{c_{pg} \times TIT \times \eta_t \left(1 - \frac{1}{r_p^{\frac{k-1}{k}}}\right) - c_{pa} \times T_1 \left[1 + \frac{1}{\eta_c}\left(r_p^{\frac{k-1}{k}} - 1\right)\right]}{c_{pg}\left\{ TIT - T_1 \left[1 + \frac{1}{\eta_c}\left(r_p^{\frac{k-1}{k}} - 1\right)\right]\right\}} \qquad (14)$$

Steam turbine model

The HRSG used as a steam generator that transmitted the steam into ST for produce work. After that the condensate fluid flow from ST into condenser with the cooling system by cooling tower mechanism. The last stage of this cycle is output from the condenser namely feed water is suctioned by the feed water pump to recirculate the water into HRSG. The cycle occurred is classified as a Rankine cycle with idealizations in ST and pump efficiencies. The solid line represent the idealizations and the dashed line as the actual process on temperature-entropy diagram illustrated by Figure 4.

For the simple GTCC with a single pressure of HRSG is commonly used. The temperature profile is divided by heat transfer in economizer, evaporator and the superheater as shown in Figure 5. Temperature of feed water that flow through HRSG increase gradually and then became superheated steam after heated in superheater. Conditions of GT exhaust gases, mass flow rate and the compositions are already known. For calculating the HRSG temperature,

the main parameters are pinch point T_{pp} and approach point T_{ap} that need to specify. The temperature of T_{g3} and T_{10} can be obtained depending on pinch point and approach. The following equations for obtaining the result has been introduced.
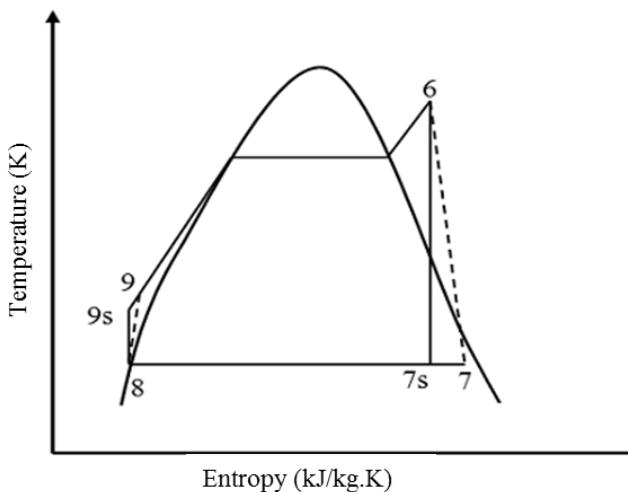
Figure 4. Temperature-entropy diagram of the ST
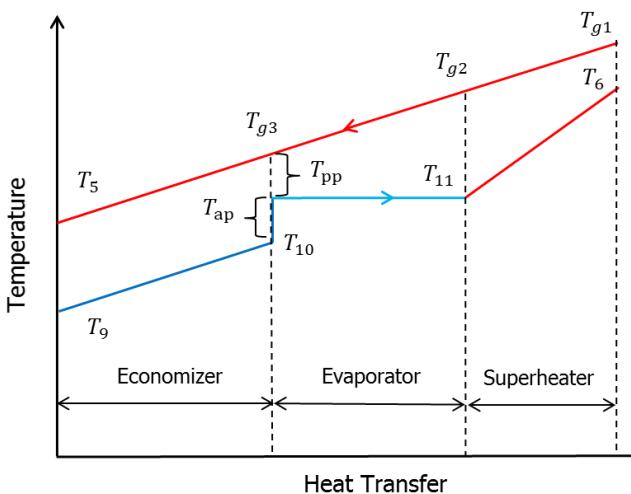
Figure 5. Heat transfer diagram for single pressure HRSG in the simple GTCC

Eq. (15) expresses temperature of the gas exitting from the evaporator:

$$T_{g3}=T_{11}+T_{pp} \qquad (15)$$

The saturation steam temperature leaving from the drum is denoted T_{11}

Eq. (16) presents the temperature of water entering the evaporator:

$$T_{10}=T_{11}+T_{ap} \qquad (16)$$

Eq. (17) is used for calculating the mass flow rate of the generation steam:

$$\dot{m}_{st}=\frac{\dot{m}_g(c_{pg1}T_{g1}-c_{pg5}T_5)}{h_6-h_{11}} \qquad (17)$$

Eq. (18) shows the heat available from the exhaust gas:

$$Q_a=\dot{m}_g c_{pg}(T_{g1}-T_5) \qquad (18)$$

Eq. (19) expresses work done of the ST:

$$W_{st}=\dot{m}_{st}(h_6-h_7) \qquad (19)$$

Eq. (20) presents the heat rejected from the condenser:

$$Q_{cond}=\dot{m}_{st}(h_7-h_8) \qquad (20)$$

The pump extracts the condensate from the condenser which is then elevated to the economizer pressure. Eq. (21) describes the corresponding work:

$$W_p=\dot{m}_{st}(h_9-h_8) \qquad (21)$$

Eq. (22) derives the ST power plant net work:

$$W_{ST}=W_{st}-W_p \qquad (22)$$

Eq. (23) expresses the net power output of ST:

$$P_{ST}=\dot{m}_{st}\times W_{GT} \qquad (23)$$

Eq. (24) shows the ST power plant efficiency:

$$\eta_{ST}=\frac{W_{ST}}{Q_a}=\frac{\dot{m}_{st}[(h_6-h_7)-(h_9-h_8)]}{\dot{m}_g c_{pg}(T_{g1}-T_5)} \qquad (24)$$

Finally, Eq. (25) is used to obtain the overall thermal efficiency of the GTCC power plant:

$$\eta_{CC}=\frac{W_{GT}+W_{ST}}{Q_{add}} \qquad (25)$$

Modelling of Intercooled-reheat Gas Turbine Combined Cycle (IHGTCC)

A schematic of the IHGTCC with bottoming cycle using a single pressure of HRSG is illustrated in Figure 6.
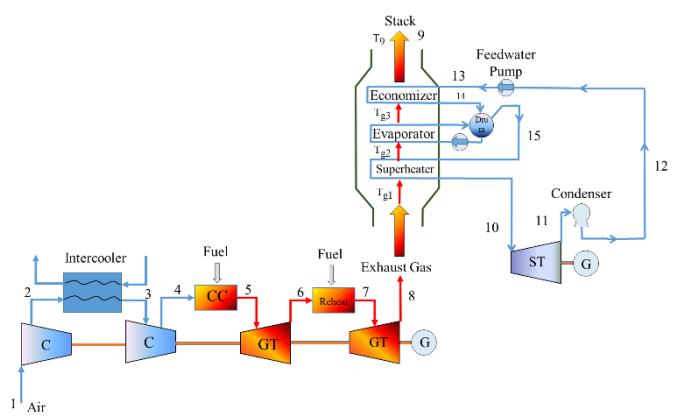
Figure 6. The IHGTCC with single pressure HRSG

The IHGTCC is modifications of GTCC that can produced higher net work output by combining the intercooling method with the reheat in one cycle of the GT in the GTCC.

The network output of a gas turbine can be increased by reducing the compressor work. This can be accomplished by multistage of compression by intercooling [4]. And also, the gas exiting the combustor contain sufficient air to support the combustion of additional fuel. For modification, gas turbine model can take advantage of the excess air by means of multistage turbine called reheater combustor. A schematic for intercooled – reheat gas turbine shown in Figure 7.

For reheat process, after expansion from state 5 to state 6 in the first turbine, the gas is reheated at a constant pressure from state 6 to state 7. The expansion is then completed in the second turbine in state 7 to state 8. The temperature of the exhaust gas is increased due to second combustion.

In Figure 8, shows temperature-entropy diagram for brayton cycle with modification by intercooled-reheat gas turbine.

Figure 7. The intercooled-reheat gas turbine model
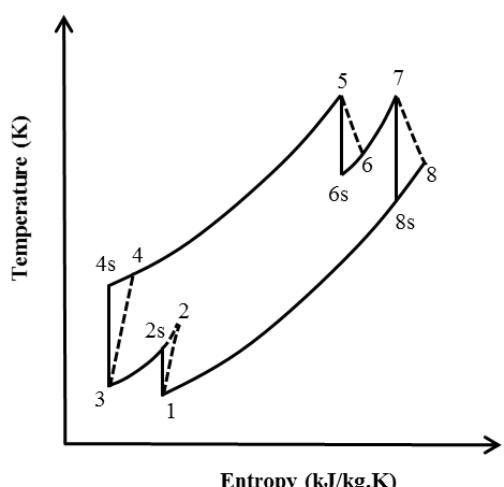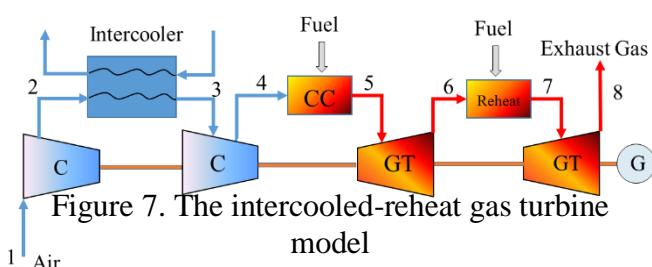
Figure 8. Temperature-entropy diagram of GT cycle with intercooled-reheat modifications

The following equations in intercooler and reheat are used to obtain the result.

Eq. (26) shows the effectiveness of intercooler:

$$x = \frac{T_2 - T_3}{T_2 - T_1} \tag{26}$$

Eq. (27) depicts the exhaust gases temperature of the high pressure turbine (HPT):

$$T_6 = TIT - \frac{c_{pa}}{c_{pg}} \times \left[\frac{1}{\eta_c}\left(r_p^{\frac{k-1}{k}} - 1\right) + (1-x)\frac{1}{\eta_c}\left(r_p^{\frac{k-1}{k}} - 1\right)^2 \right] \tag{27}$$

Eq. (28) presents the isentropic temperature exit from the HPT:

$$T_{6s} = TIT - \frac{TIT - T_6}{\eta_t} \tag{28}$$

Eq. (29) describes the isentropic temperature at the exit from low pressure turbine (LPT):

$$T_{8s} = \frac{TIT^2}{T_{6s} \times r_p^{\frac{2(k-1)}{k}}} \tag{29}$$

Eq. (30) shows the exhaust gases temperature from LPT:

$$T_8 = TIT - (TIT - T_{8s}) \tag{30}$$

Eq. (31) presents the net work of the IHGT cycle:

$$W_{IHGT} = c_{pg}(TIT - T_8) \tag{31}$$

Eq. (32) indicate net power output for the turbine:

$$P_{IHGT} = (\dot{m}_g + \dot{m}_{fH}) \times W_{IHGT} \tag{32}$$

Eq. (33) is used to estimate additional mass flow rate of fuel $\dot{m}_{fH}$:

$$\dot{m}_{fH} = \frac{c_{pg} \times \dot{m}_g (TIT - T_6)}{LHV - c_{pg} TIT} \tag{33}$$

Eq. (34) expresses heat supplied for the IHGT cycle:

$$Q_{add} = c_{pg}(TIT - T_4) + \frac{c_{pg}(\dot{m}_g + \dot{m}_{fH})}{\dot{m}_g} \tag{34}$$

Eq. (35) presents energy balance in the combustion chamber:

$$\dot{m}_a c_{pa} T_2 + \dot{m}_f LHV + \dot{m}_f c_{pf} T_f - \dot{m}_g c_{pg} TIT \tag{35}$$

Eq. (36) represents for the fuel air ratio deriving from Eq. (35):

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{C_{pg} TIT - C_{pa} T_2}{LHV - C_{pg} TIT} \tag{36}$$

Eq. (37) shows specific fuel consumption (SFC):

$$SFC = \frac{3600 \cdot f}{W_{IHGT}} \tag{37}$$

Eq. (38) can be used to find the overall efficiency:

$$\eta_{GT} = \frac{W_{IHGT}}{Q_{add}} \qquad (38)$$

The following input data and assumptions have been used in the simulation of the simple GTCC and the IHGTCC:

1. All the processes are steady state and steady flow.
2. Ambient conditions is given at pressure 1.013bar and the ambient temperature 15°C.
3. In the IHGTCC, the compression ratio is defined as ratio of p_4/p_1 and the variation in two pressures satisfy total compressor work is a minimum. Particular in Figure 7: $p_2=p_3=\sqrt{p_1 \cdot p_4}$
4. The fuel used in combustion chamber is natural gas.
5. In the IHGTCC, TIT is the same for combustion chamber and reheated of GT: $TIT=T_5=T_7$
6. In this paper, the TIT is referenced as firing temperature.
7. Isentropic efficiency of compressors, turbine and pump at 85%.
8. The last stage of turbine of GT expand into 1.3bar.
9. Steam turbine expands into 0.03bar.
10. The pinch point as the difference between the gas temperature leaving the evaporator and saturated steam temperature is given as 20°C.
11. The approach point as the difference between saturated steam temperature and water temperature leaving the economizer 20°C.
12. The pressure drops in the combustion chamber, HRSG and condenser are neglected.
13. Generators efficiency are given as 97.5%.

Results and discussion

As performed above thermal analysis, it is evident in this study that the variation of the TIT and compression ratio will affect the overall thermal efficiency, power output, SFC as well.

Turbine inlet temperature

Turbine inlet temperature determines the quality of steam exiting the turbine and the overall efficiency as a function of the TIT [5]. Figure 9 shows the effect of the TIT on performance of simple GTCC and IHGTCC at a constant value of compression ratio of 16.

In detail, Figure 9(a) illustrates the variation of the overall efficiency including GT and ST efficiency with respect to TIT. The higher TIT results in the rise of the efficiencies for simple

GTCC and IHGTCC because the combustion in gas turbine is more effective as well as the fuel-air mixture is more completely burnt (burning rate increase) when the TIT increase. Moreover, the IHGTCC has higher overall efficiency (from 41.3 to 50.4%) than that of simple GTCC (from 32.2 to 47.3%) in the range of the TIT from 900 to 1300°C. In GT cycle of the IHGTCC, starting from 1000°C of TIT, the higher ratio between overall heat consumed by combustion chamber and reheat chamber with total work done compare to that in GT cycle of the simple GTCC. This is main reason GT efficiency of the IHGTCC (peak at 28.5%) is less than that of the simple GTCC (highest at 31.5%). Contrary to ST cycle, with an increase of TIT, the greater temperature exiting GT go into HRSG of the IHGTCC leading to its higher ST efficiency (highest at 25.3%) compare to the simple GTCC (peak at 23.6%).

(a)

(b)

(c)
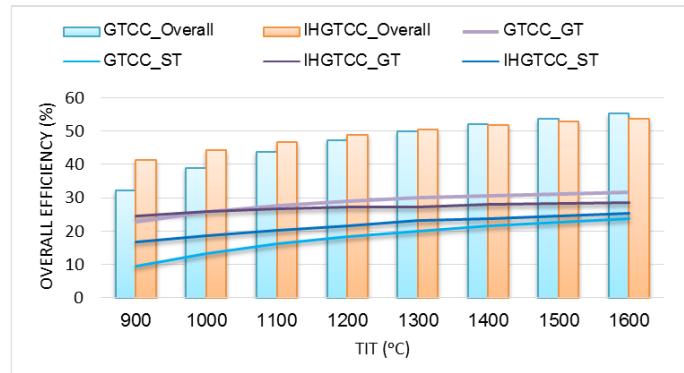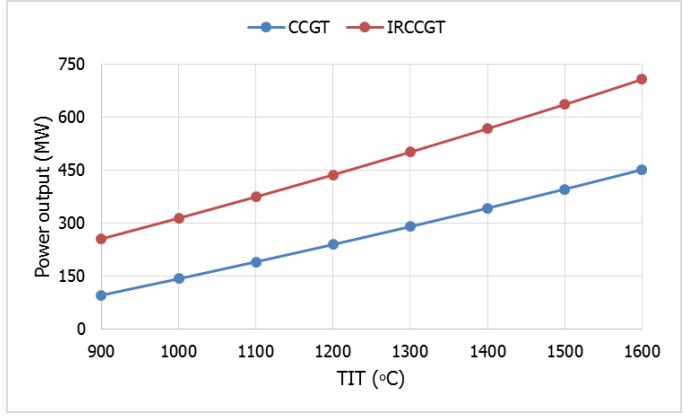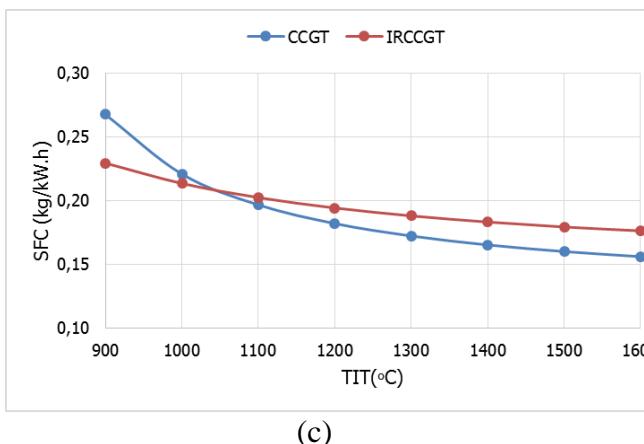
Figure 9. The performance at different TIT for the simple GTCC and IHGTCC

(a) Overall efficiency, (b) Power output, (c) SFC

Meanwhile, the power output at different TIT for simple GTCC and IHGTCC is shown in Figure 9(b). The power output sharply increases with increases in TIT because of effectiveness of combustion is enhanced and allowing to get higher work done in GT as well as improve the exiting temperature from gas turbine into HRSG resulting in the higher work done in ST too, therefore the power out increase. It is apparent that the IHGTCC use reheated combustion to gains thoroughly the available energy in flue gas exiting gas turbine as a result of extra power output can be obtained, thus IHGTCC produces more significantly power output between 255MW and 705MW compare to that of simple GTCC from 96MW to 451MW when TIT increase from 900 to 1600°C. Figure 9(c) depicts the relation of SFC versus the TIT. It is understandable that an increase of power output leads to decrease SFC when TIT raise. The optimum performance of the IHGTCC can be acquired at TIT equal 1050°C when the SFC of the IHGTCC (0.208 kg/kW.h) is almost the same with that of simple GTCC (0.207 kg/kW.h) while the higher overall efficiency by 5% (nearly 45.6%) along with increasing in power output by 111% (343MW) for the IHGTCC. In actual gas turbines, the TIT cannot be increased indefinitely because of temperature limitations of the materials [6]. Current-state-of-the-art gas turbines have firing temperatures (rotor inlet temperatures) that are limited to about 1320ºC [7]. Hence the optimum TIT at 1050°C is satisfactory.

Compression ratio

Figure 10 shows the graphical presentation of the effect of variation in compression ratio to performance of simple the GTCC and the IHGTCC at a constant ambient temperature of 15°C and turbine inlet temperature of 1200°C.

Figure 10(a) presents the influence of compression ratio on the overall efficiency of the simple GTCC and the IHGTCC. The increase in compression ratio means that increase in power output as a result of the overall efficiency increase [8]. However, if the compression ratio is too high, the compressor and turbine works increase but their difference, the GT work output and efficiency drop [9]. So that causing the decrease of power output and overall efficiency of combined cycle. As a result, the overall efficiency of the simple GTCC fractional increases from 47.4 to 47.5% from 10 until 12 of compression ratio, and then decreasing to 43.6% with increasing up to 35 of compression ratio. Thermal efficiency increased as there was a reduction in the losses due to the heat recovered from the flue gases in gas turbine [10]. The IHGTCC is known in lessening the energy losses in flue gas that exhaust from gas turbine. Consequently, the IHGTCC can extends the range of compression ratio (until 28) in that the overall efficiency (peak at 49.5%) is kept to improve with increase in compression ratio. It is evident that at constant turbine inlet temperature, with an increase compression ratio result in the reducing amount of fuel injected in GT and leading to diminish efficiency of ST cycle. The ST efficiency of the IHGTCC just somewhat decrease from 22.1% to 21.1% whereas that of the simple GTCC lessen from 21.7% down to 13%.

The performance of the simple GTCC and the IHGTCC are effected by the compression ratio showing in Figure 10(b). As a consequence of variation of the overall efficiency, the power output of simple GTCC diminish from 264 down to 176 MW with an increase of compression ratio whereas for the IHGTCC, the power output jump gradually start at 407 until approximately 450MW when compression ratio varies from 10 to 28 and then slightly reducing. Compression ratio increases, turbine work increases but with more increase in compression ratio compressor

work also increases that will results in decrease in work output of the GT [11]. As previously mentioned, the IHGTCC can extends the range of compression ratio in that the overall efficiency is kept to increase with increase in compression ratio. That explain the result of the higher power out of the IHGTCC compare that of the simple GTCC at a certain compression ratio.

In Figure 10(c), the SFC of the simple GTCC and the IHGTCC are plotted against compression

value of compression ratio is 28 as a result of optimum performance.
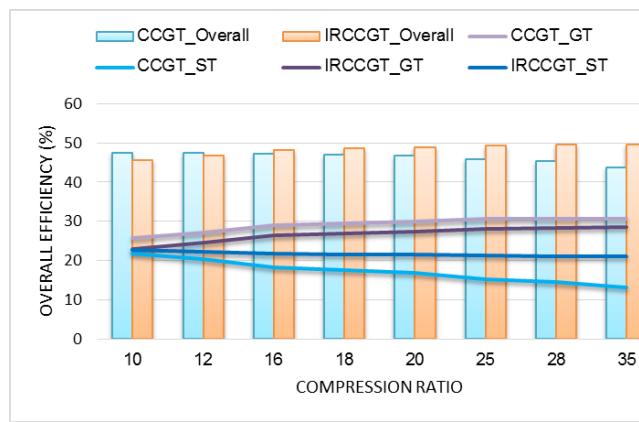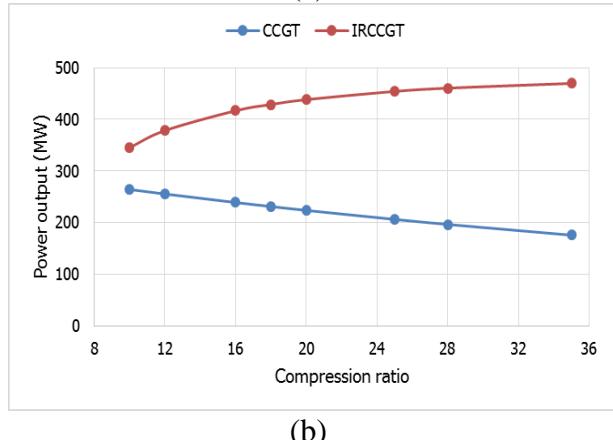
ratio. The SFC curves intersect at point 28 of compression ratio (0.192 kg/kW.h). According to compression ratio of 28, collating with the overall efficiency and power output, the IHGTCC gains 49.5% of overall efficiency and 450MW of power output while the simple GTCC achieves 45.3% and close to 197MW, correspondingly. It is justified that the optimal


(a)

(b)

(c)
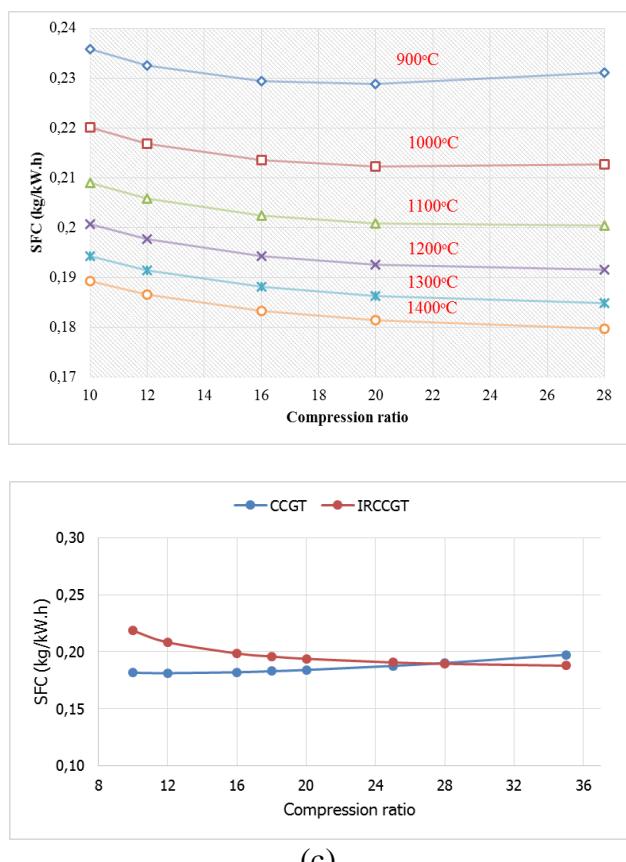
Figure 10. The performance at different compression ratio for the simple GTCC and IHGTCC (a) Overall efficiency, (b) Power output, (c) SFC
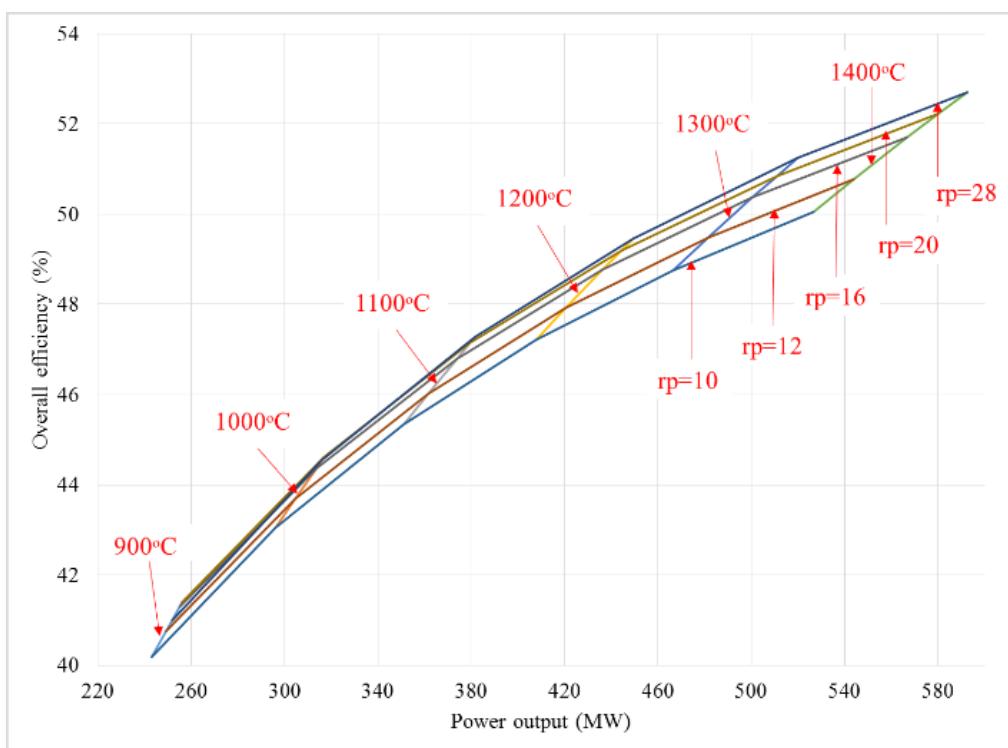
Figure 11. The performance at different TIT and compression ratio for IHGTCC

Figure 11 indicates variation of performance of the IRCCGT with respect to simultaneously compression ratio and turbine inlet temperature. It can be observed that the overall efficiency and power output of the IHGTCC are directly proportional while SFC is inversely proportional with TIT and compression ratio. But exception with TIT at 900 and 1000°C, the performance of the IHGTCC have tendency which are inversely proportional start from compression ratio of 28 and

Conclusion

This study focus on optimum performance in several operating condition based on optimal specific fuel consumption of the intercooled – reheat gas turbine combined cycle based which is developed form a simple GTCC (with the same input data) by using Cycle–Tempo software. Finding the optimal performance and still guarantee fuel economy by comparing rely on variation of both aforementioned combined cycles's performance that was strongly effected of turbine inlet temperature and compression ratio. This simulation illustrates proportional relation between the performance of the IHGTCC with TIT and compression ratio. Importantly, the simulation also propose the optimal results:

1. At constant value compression ratio of 16, the performance of the IHCCGT is the most improved at optimal turbine inlet temperature of 1050°C. The IHCCGT obtains 45.6% of overall efficiency and 343MW of power output at SFC of 0.207 kg/kW.h.
2. At constant value of turbine inlet temperature of 1200°C, the performance of the IHCCGT is the most enhanced at optimal compression ratio of 28. The IHCCGT obtains 49.5% of overall efficiency and 450MW of power output at SFC of 0.192 kg/kW.h.

Based on these optimal results, the IHGTCC will have choices which are favorable with initial operating and economic condition to get the best performance and economy

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