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EFFECT OF COMPRESSION RATIO ON THE PERFORMANCE OF DIFFERENT STRATEGIES FOR THE GAS TURBINE

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ABSTRACT

Finding clean and safe energy is the greatest challenge to meeting the requirements of a green environment. These requirements given way the long time governing authority of steam turbine (ST) in the world power generation, and gas turbine (GT) will replace it. It is therefore necessary to predict the characteristics of the GT and optimize its operating strategy by developing a simulation system. Several configurations of GT plants are proposed by thermal analysis. The integrated model and simulation code for exploiting the performance of GT plants have been developed using MATLAB code. The performance codes for heavy-duty GT power plants were validated with the Bassily model and the results have been satisfactory. Results of performance enhancing strategies show that higher power output occurs in the intercooler-reheat GT (IHGT) strategy (286 MW). At a compression ratio of 12.4 in the intercooler-regenerative-reheat GT (IRHGT) strategy, the highest (best) thermal efficiency obtained was 50.7%. In the IHGT, the lowest thermal efficiency obtained was 22% to 39.3% in the presence of a compression ratio between 3 and 30. The lowest fuel consumption (0.14 kg/kWh) occurred in the IRHGT strategy. Compression ratios are thus strongly influenced by the overall performance of the GT.

Keywords: Gas turbine; compression ratio; strategies; performance.

INTRODUCTION

A GT power plant is compact and light in order to improve its performance (Bassily, 2001). The stationary turbines for power generation do not have these constraints (Bassily, 2002). GT plants have been effectively working as heavy-duty plants (large scale) to generate electricity, where GTs presented a better generation system (Ibrahim, Rahman, & Abdalla, 2011a, 2011b; Poullikkas, 2005). Engineers have explored many strategies to apply to enhancing the performance of GT power plants. Different methods have been used to increase the net power of GT plants. Many studies (Ibrahim, Rahman, & Alla, 2010; Ibrahim & Rahman, 2012a, 2012b, 2012c; Rahman, Ibrahim, & Abdalla, 2011a; Rahman, Ibrahim, Kadirgama, Mamat, & Bakar, 2011b; Rahman et al., 2010) have presented a comprehensive design methodology to obtain the best performance of a GT plant. Parametric study of the effects of regenerating, intercooling, and reheating in a wide-range of existing GT is presented (Ibrahim & Rahman, 2012a, 2012b; Sayyaadi & Aminian, 2010). Applications of the proposed design steps have been implemented in three existing GT with wide-ranging design complexities. The performances of the modified GTs are presented and compared. A stage by stage analysis of the compressor and turbine sections of the modified GT is performed (Ibrahim et al., 2011a, 2011b; Ibrahim & Rahman, 2012a). All the modified GT plants were found to have improved performance, with a cycle efficiency increase of 9% to 26%, in comparison to their original values (Ibrahim et al., 2011b; Ibrahim & Rahman, 2012a; Sayyaadi & Aminian, 2010). The effects of the compression ratio, turbine inlet temperature and relative humidity, on the performance of the complex GT cycle were studied, through a computational thermodynamic analysis. The complex GT cycle consisted of the indirect intercooled reheat regenerative GT with inlet air affected by the indirect evaporative cooling and air evaporative aftercooling of the compressor discharge (Harvey & Kane, 1997; Ibrahim & Rahman, 2012a; Rahman et al., 2011b). The power output, and thermal efficiency varied significantly with the effect of the compression ratio, turbine inlet temperature and relative humidity. The objective of this study was to investigate the influence of the compression ratio on the performance of different strategies of the gas turbine.

METHODOLOGY

An important contribution of this study is to enhance the performance of GT and CCGT plants through the proposed integrated strategy, including all enhancing elements (modifications). These strategies provide incorporated simulation and optimization tools for effective operation parameters and ambient conditions for the performance of gas turbines, using all the enhancing elements, two-shafts, an intercooler, regeneration and reheating, as shown in Figure 1. These strategies aim to combine the different configurations of the enhancing elements to show their effects on the total performance of GT and CCGT power plants in ambient temperature and operation parameters and then to validate data from real GT and CCGT performances with the results from the performance enhancing strategies (PES).

Intercooler Gas Turbine

The work of an intercooler GT (IGT) cycle is as shown in Figure 1(a):

$$W_{Gnet} = C_{pg} TIT.\left[\frac{\eta_t}{\eta_m}\left(1-\frac{1}{\left(r_p^2\right)^{\frac{\gamma_g-1}{\gamma_g}}}\right)\right] - C_{pa}T_1\left(\frac{r_p^{\frac{\gamma_a-1}{\gamma_a}}-1}{\eta_m\eta_c}\right)\left[2+(1-x)\left(\frac{r_p^{\frac{\gamma_a-1}{\gamma_a}}-1}{\eta_c}\right)\right] \tag{1}$$

Eq. (2) expresses the heat supplied for the IGT cycle:

$$Q_{add} = C_{pg}\left[TIT - T_1 + T_2\left(\frac{r_p^{\frac{\gamma_a-1}{\gamma_a}}-1}{\eta_c}\right)\left((2-x)+(1-x)\left(\frac{r_p^{\frac{\gamma_a-1}{\gamma_a}}-1}{\eta_c}\right)\right)\right] \tag{2}$$

Regenerative Gas Turbine

From Figure 1(b), Eq. (3) expresses the network of the regenerative GT (RGT) cycle:

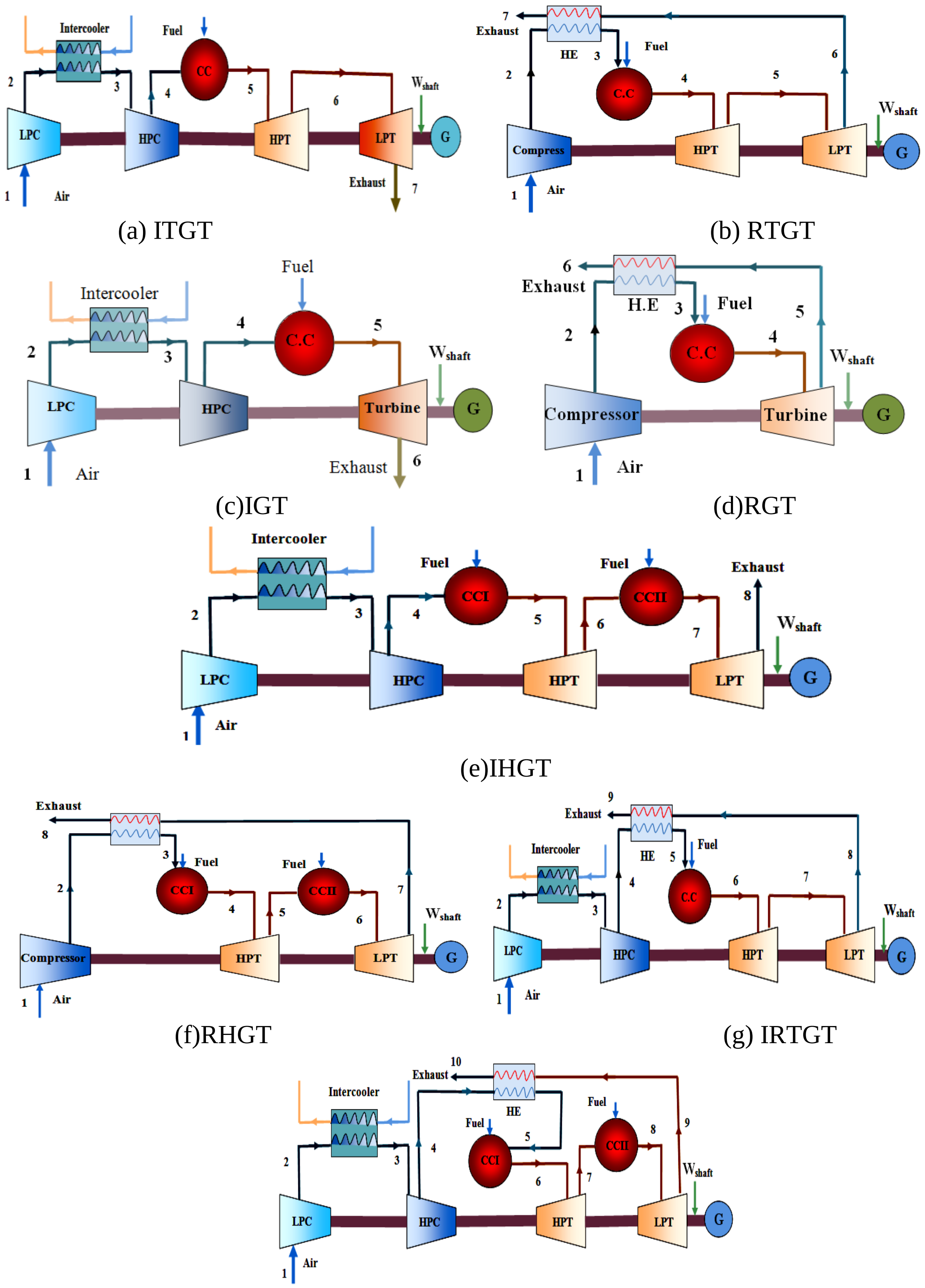

Figure 1. Configurations of gas turbine performance enhancing strategies.

$$W_{\mathrm{Gnet}} = \frac{C_{pg} \times TIT \times \eta_t}{\eta_m}\left(1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}}\right) - C_{pa} T_1 \left(\frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}}}{\eta_m \eta_c}\right) \tag{3}$$

Eq. (4) expresses the heat supplied for the RGT:

$$Q_{\mathrm{add}} = C_{pg} \times \left[TIT - T_1 (1 - \varepsilon) \times \left(1 + \frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c}\right) - \varepsilon \times TIT \times \left[1 - \eta_t \left(1 - \frac{1}{r_p^{\frac{\gamma_g - 1}{\gamma_g}}}\right)\right]\right] \tag{4}$$

Intercooler-Two-Shaft Gas Turbine

The work of an intercooler-two-shaft GT (ITGT) cycle is as shown in Figure 1(c):

$$W_{Gnet} = C_{pg}(T_6 - T_7) \tag{5}$$

Eq. (6) expresses the heat supplied for the (ITGT) cycle:

$$Q_{add} = C_{pg}\left[TIT - T_1 \times \left(1 + (2 - x)\left(\frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c}\right) + (1 - x)\left(\frac{r_p^{\frac{\gamma_a - 1}{\gamma_a}} - 1}{\eta_c}\right)\right)\right] \tag{6}$$

Regenerative-Two-Shaft Gas Turbine

From Figure 1(d), Eq. (7) expresses the network of the regenerative two-shaft GT (RTGT) cycle:

$$W_{Gnet} = C_{pg}(T_5 - T_6) \tag{7}$$

Eq. (8) expresses the heat supplied for the RTGT:

$$Q_{add} = C_{pg}(TIT - T_3) \tag{8}$$

Intercooler-Reheat Gas Turbine

From the energy balance for Figure 1(e), Eq. (9) expresses the network of the intercooler-reheat GT (IHGT) cycle:

$$W_{Gnet} = C_{pg}(TIT - T_8) \tag{9}$$

Eq. (10) expresses the heat supplied for the IHGT cycle:

$$Q_{add} = C_{pg}(TIT - T_4) + \frac{C_{pg}\left(\dot{m}_a + \dot{m}_f + \dot{m}_{fad}\right)}{\left(\dot{m}_a + \dot{m}_f\right)}(TIT - T_6) \tag{10}$$

Eq. (11) expresses the additional fuel burning in the second CC and is equal to $\dot{m}_{fad}$

$$\dot{m}_{fad} = \frac{C_{pg}\left(\dot{m}_a + \dot{m}_f\right)\left(TIT - T_6\right)}{\left(LHV \times \eta_{comb} - C_{pg} TIT\right)} \tag{11}$$

Regenerative-Reheat Gas Turbine

From the energy balance for Figure 1(f) Eq. (12) expresses the network of the regenerative-reheat GT (RHGT) cycle:

$$W_{Gnet} = C_{pg}\left(TIT - T_7\right) \tag{12}$$

Eq. (13) expresses the heat added for the RHGT cycle:

$$Q_{add} = C_{pg}\left(TIT - T_3\right) + C_{pg}\left(TIT - T_5\right) \tag{13}$$

Intercooler-Regenerative-Two-Shaft Gas Turbine

From the energy balance for Figure 1(g) Eq. (14) expresses the network of the intercooler- regenerative-two-shaft GT (IRTGT) cycle:

$$W_{Gnet} = C_{pg}\left(T_7 - T_8\right) \tag{14}$$

Eq. (15) expresses the heat added for the IRTGT cycle:

$$Q_{add} = C_{pg}\left(TIT - T_5\right) \tag{15}$$

Intercooler-Regenerative-Reheat Gas Turbine

From the energy balance for Figure 1(h) Eq. (16) expresses the network of the intercooler- regenerative-reheat GT (IRHGT) cycle:

$$W_{Gnet} = C_{pg}\left(T_8 - T_9\right) \tag{16}$$

Eq. (17) expresses the heat added for the IRHGT cycle:

$$Q_{add} = C_{pg}\left(TIT - T_5\right) + C_{pg}\left(TIT - T_7\right) \tag{33}$$

RESULTS AND DISCUSSION

To increase performance, a study of the effects of specific parameters and the development of plant models according to the behavior of the load of the GT plant was carried out. Study is required to understand the dependency of the important plant variables on the GT power output, exhaust gas temperature, fuel and air flow rate. This study presents the effects of the compression ratio on the performance of the GT plant. The energy balance is used to obtain the effects of the factors on the thermal efficiency, specific fuel consumption and power output. A comparison between the simulated thermal efficiency of the intercooler-regenerative-reheat GT and the Bassily model (Bassily, 2001), combined with the effects of the different values of the compression ratio and turbine inlet temperature, is presented in Figure 2. The compression ratio rises until it reaches 12.6 at a turbine inlet temperature of 1400K, and 15 at a turbine inlet temperature of 1700K, and then the thermal efficiency increases accordingly. The thermal efficiency starts to decrease with the increase in compression ratio, after reaching these compression ratios. In the regenerative strategy, thermal efficiency started to increase with the compression ratio after it had reached the value of 15. Thermal efficiency increased as there was a reduction in the losses due to the heat recovered from the flue gases (Dellenback, 2002). The regenerative model, when applied to reduce the exhaust temperature, proved to be inefficient after the compression ratio reached 15 (Elmegaard & Qvale, 2004; Ibrahim et al., 2011b; Rahman et al., 2010). There were also two values added in the comparison, such as, 1400 and 1700 K, for the turbine inlet temperature. The increase in the turbine inlet temperature leads to an increase in thermal efficiency. When compared to the Bassily model, the simulation results for this model were satisfactory (Bassily, 2001). A number of graphs represent the variation in the performance of the GT plants, and the effect of the compression ratio. The effect of the compression ratio on the temperature of the exhaust gases of the GT, for different configurations of performance enhancing strategies, is shown in Figure 3. The strategies that used the regenerative (RHGT, IRHGT, IRGT, IRTGT and RTGT) showed an increase in the exhaust temperature with an increase in the compression ratio, while the strategies that were without the regenerative (STG, ITGT and IHGT) showed a decrease in the temperature of exhaust gases from the GT with an increase in the compression ratio. This usually occurs when the regenerative strategies are applied to recover the energy from the exhaust gases at a low compression ratio (Canière, Willockx, Dick, & De Paepe, 2006; Ibrahim et al., 2011b; Kumar, 2010). Thus, the increase in the compression ratio leads to a decrease in the exhaust temperature.

When the compression ratio increased from 3 to 30 in the strategies with regenerative, the exhaust temperature increased by approximately 120 to 300K, owing to more heat being recovered from the exhaust gases in the regenerative heat exchanger at a low compression ratio (Dellenback, 2002; Harvey & Kane, 1997). When the compression ratio increased from 3 to 30 for other strategies, the exhaust temperature decreased by approximately 210 to 390K. It was also observed that the highest exhaust temperature occurred at IHGT, while the lowest exhaust temperatures occurred at IRGT. It also became evident that a significant variation in the exhaust temperatures was obtained at higher compression ratios, while minimum variations were obtained at lower compression ratios. The characteristic curves of all performance enhancing strategies at a turbine inlet temperature of 1450 K, constant ambient temperature of 288.15 K, and relative humidity of 60% are shown in Figure 4. As mentioned above, the thermal efficiency increased with the increase in compression ratio, after the value of the compression ratio reached 6.4 for the regenerative strategies (RHGT, IRHGT, IRGT and RTGT). The thermal efficiency decreased for an increase in the compression ratio, after the value of compression ratio reached 12.4 in the IRTGT strategy. The thermal efficiency increased with an increase in the compression ratio, for all the strategies without regenerative (STG, ITGT and IHGT). This was due to the low exhaust temperature as compared to other strategies, as can be seen in Figure 3.

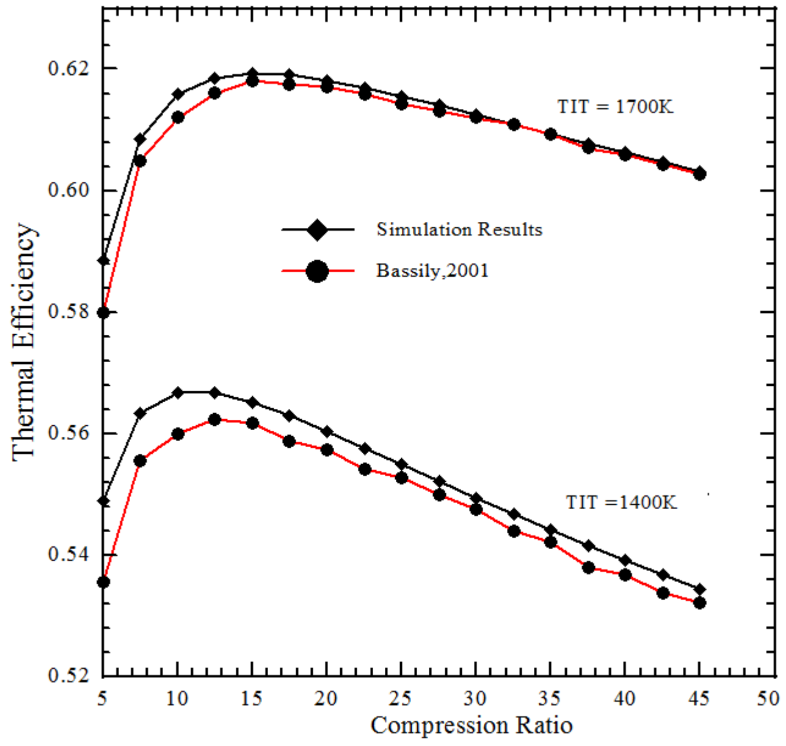

Figure 2. Comparison between simulated thermal efficiency of the intercooler-regenerative-reheat gas turbine and Bassily model with the effect of the compression ratio.

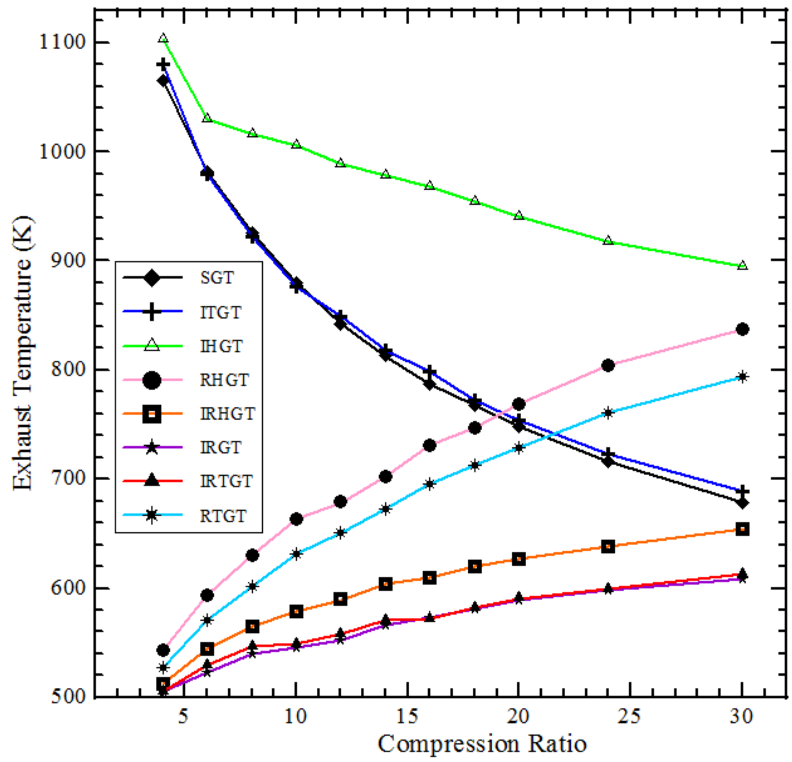

Figure 3. Effect of compression ratio on the exhaust temperature of the gas-turbine plants.

At a compression ratio of 12.4 in the IRHGT, the highest (best) thermal efficiency obtained was 50.7%, and in the IHGT, the lowest thermal efficiency of between 22% to 39.3% was obtained in the presence of a compression ratio between 3 and 30. Similarly, a noticeable variation in the thermal efficiency was observed at the lowest compression ratio, while an insignificant variation in the thermal efficiency was obtained at a low compression ratio for all the performance enhancing strategies of the GT. Compared to the previous models of Bassily (2002), a reasonable result was obtained.

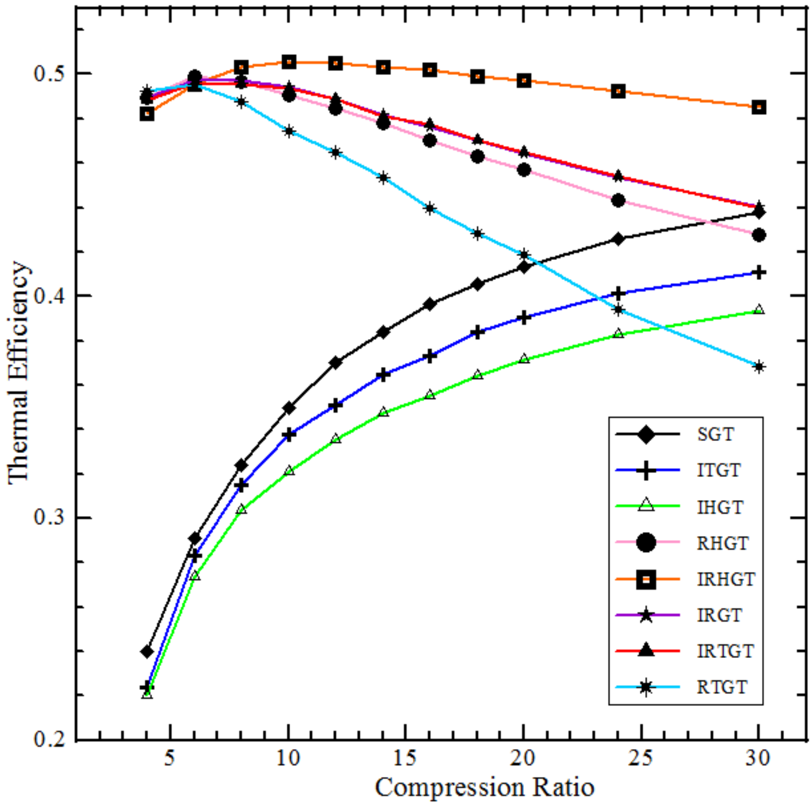

Figure 4. Effect of compression ratio on the thermal efficiency of the gas-turbine plants.

The variation in the effect of the compression ratio on the specific fuel consumption for different performance enhancing strategies of the GT cycle is shown in Figure 5. It can also be seen that the specific fuel consumption decreased with the increase of the compression ratio for the strategies without regenerative. The elevated temperatures at which the air exits the compressors and enters the combustion chamber of the GT are the main factors behind this (Dellenback, 2002; Kumar, 2010; Sayyaadi & Aminian, 2010). Specific fuel consumption was thus dropped in order to attain the required turbine inlet temperature. Specific fuel consumption increased with the increase in the compression ratio in the presence of the regenerative strategies. Under the influence of the compression ratio, a lower specific fuel consumption of 0.15 kg/kWh in the IRHGT strategy was obtained. This is as the heat from the exhaust was recovered by using the regenerative. This heat was used to increase the temperature of the air entering the combustion chamber (Canière et al., 2006; Ibrahim et al., 2011b; Poullikkas, 2005).

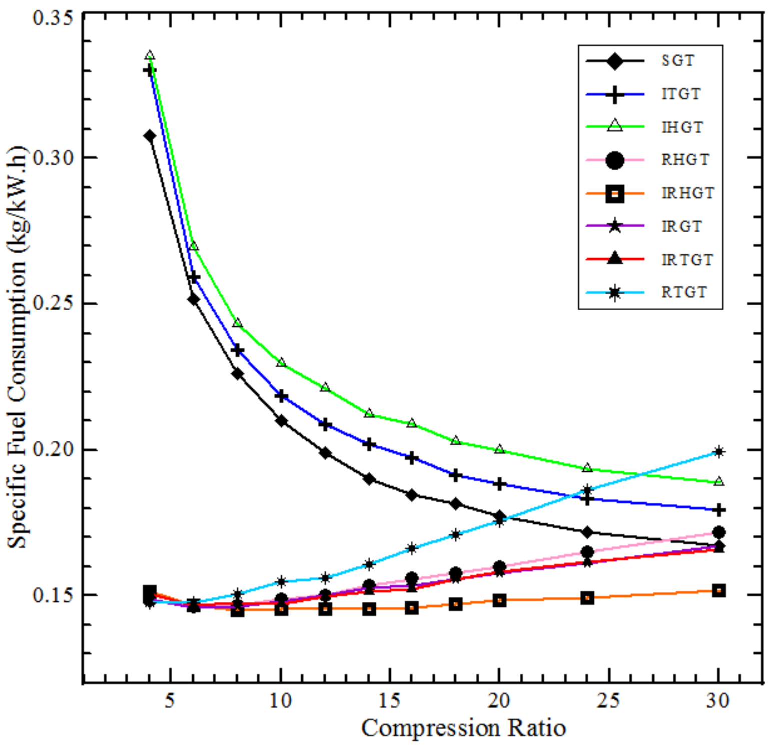

Figure 5. Variation of compression ratio on the specific fuel consumption of the gas-turbine plants.

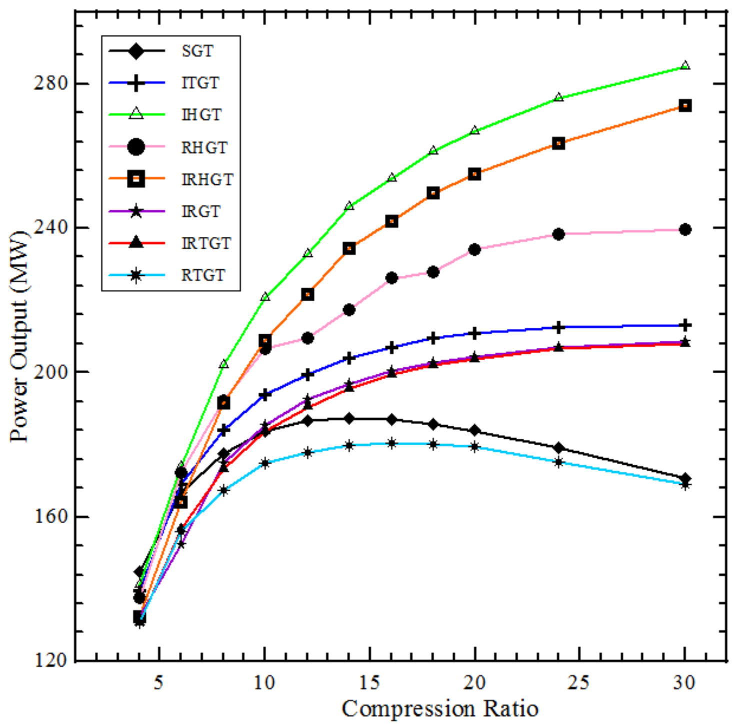

Figure 6. Effect of variation of the compression ratio on the power output of the gas-turbine plants.

When the compression ratio increased from 3 to 30, the higher value of the specific fuel consumption varied from 0.336 to 0.189kg/kWh in the IHGT. The high compression ratio seemed to have an impact on specific fuel consumption, while the effect of the low compression ratio was insignificant. Figure 6 shows the effect of the compression ratio on the power output of the GT plants for different performance enhancing strategies. Due to the mass flow rate through the GT, the power output also increases as the compression ratio increases. This results in an increase in the network as well as the power output. These conditions occurs in all the GT strategies, with the exception of SGT and RTGT, where after reaching the compression ratio of 15 there was a decrease in the power output with the increase in the compression ratio. In all the GT strategies, the lower compression ratios showed insignificant variations in power output, while it was significant for the high compression ratio. In addition, when the compression ratio increased from 3 to 30, the lower output strategy in the RTGT strategy increased from 132 to 169MW. On the other hand, in the IHGT strategy, there was higher output, increased from 144 to 286 MW after reaching a compression ratio from 3 to 30.

CONCLUSIONS

Complete models of the performance enhancing strategies for configurations of the GT power plant, with effects of the compression ratio, have been used in this thermodynamic study. The simulated modeling results are as follows:

i) The compression ratios are strongly influenced in the thermal efficiency of the GT power plant for all strategies.
ii) Peak GT thermal efficiency occurred in the intercooler-regenerative-reheat GT (IRHGT). Efficiency quoted range about 50.7%.
iii) The peak power output was about 286MW; it occurred in the intercooler-regenerative-reheat GT (IRHGT) at the higher compression ratio.

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