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Effect of Operating Parameters on the Performance of Combined Cycle Power Plant

A K Tiwari*, **M M Hasan and Mohd Islam**

Faculty of Engineering & Technology, Department of Mechanical Engineering, Jamia Millia Islamia, New Delhi, India

Abstract

This paper is intended to review the literature on research, development and projects related to gas turbine combined cycle. It focuses on summarizing several research investigations carried out by the author and associates, during the past years, in the field of gas turbine combined system. The performance of gas-steam combined cycle power plant depends on various operating parameters. The power output and efficiency both depends on operation of topping as well bottoming cycle but mainly depends on topping cycle which is Brayton cycle in this study. Besides the power output and efficiency there are different losses which occur in different components of plant. These are based on first and second law of thermodynamics. The second law approach (exergy analysis) gives better understanding of different losses and optimization of system for higher power output and efficiency. Hence the effect of different parameters on the performance of combined cycle is reviewed in this paper.

Introduction

The gas turbine engine is characterized by its relatively low capital cost compared with steam power plants. It has environmental advantages and short construction lead time. However, conventional industrial engines have lower efficiencies especially at part load. One of the technologies adopted nowadays for improvement is the combined cycle. Hence, it is expected that the combined cycle continues to gain acceptance throughout the world as a reliable, flexible and efficient base load power generation [1]. Combined-cycle systems utilizing the Brayton Cycle gas turbine and the Rankine Cycle steam system with air and water as working fluids achieve efficient, reliable, and economic power generation. Flexibility provided by these systems satisfies both utility-power generation and industrial-cogeneration applications. Current commercially available power- generation combined-cycle plants achieve net plant thermal efficiency typically in the 50–55% LHV range. Further development of gas turbine, high-temperature materials and hot gas path, metal surface cooling technology show promise for near-term future power generation combined-cycle systems capable of reaching 60% or greater plant thermal efficiency. Additional gas turbine technological development, as well as increases in steam-cycle pressure and temperature and steam-turbine stage-design enhancement, is expected to achieve further STAG™ combined-cycle efficiency improvement. Current General Electric STAG™ (trade name designation for the GE product line of combined- cycle systems) product line offerings, combined-cycle experience, and advanced system development are used to demonstrate the evolution of combined-cycle system technology [2].

Different Operating Parameters

The major operating parameters which influence the combined cycle performance are;

1. Turbine inlet temperature
2. Compressor pressure ratio
3. Pinch point
4. Ambient Temperature
5. Pressure levels

Besides these operating parameters there are several configurations developed over past years to improve the combined cycle performance have been also reviewed in this paper.

Effect of turbine inlet temperature

The power output of a gas turbine is a function of turbine inlet temperature. The turbine inlet temperature (TIT) plays an important role on the performance of combined cycle. The maximum value of TIT is fixed due to the metallurgical problem of turbine blade cooling. The results shown most recently by Kaviri et al. [3], that increases in the gas turbine inlet temperature decrease in the combustion chamber exergy destruction. The reason is due to the fact that this increase leads in the decrease of the entropy generation. And recently compared by Sanjay [4], the parameter that affects cycle performance most is the TIT (turbine inlet temperature). TIT should be kept on the higher side, because at lower values, the exergy destruction is higher. The summation of total exergy destruction of all components in percentage terms is lower (44.88%) at TIT of 1800 K as compared to that at TIT=1700 K. The sum total of rational efficiency of gas turbine and steam turbine is found to be higher (54.91%) at TIT=1800 K as compared to that at TIT=1700 K. The detailed analysis was done by and Khaliq [5] and Khaliq and Kaushik [6] reported that the exergy destruction in the combustion chamber increases with the cycle temperature ratio, and the second-law efficiency of the primary combustor behaves in reverse from the second-law analysis. Increasing the maximum cycle temperature gives a significant improvement in both efficiency and specific work-output. Further it is shown that the efficiency reduces rapidly with a reduction in TIT. Thermal efficiency equates to useful work output divided by heat input. Work output is found to decrease more rapidly than the reduction in heat input, and therefore, thermal efficiency decreases. Polyzakis et al. [7], shown that the power output

***Corresponding author:** A.K.Tiwari Faculty of Engg. & Tech. Department of Mechanical Engineering Jamia Millia Islamia University, New Delhi, India, E-mail: aktiwari_jmi@rediffmail.com

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decreases as TIT is reduced. Sanjay et al. [8], shows the effect of TIT on blade coolant requirement for various cooling means for an allowable blade surface temperature of 1123 K. As expected for all cooling means coolant requirement increases with increase in TIT. It shows clearly that if TIT is increased beyond 1700 K (temperature level in modern turbines), steam cooling is the best coolant option whereas, for air-cooling, ATC, followed by AFC are the next best options. The plant specific work first increases and then decreases with increase in TIT for all air-cooling means. This behavior may be explained by the fact that the net gain due to increase in plant specific work as a result of increasing TIT and decrease in plant specific work due to increasing pumping, cooling and mixing losses, is always positive and this net gain is higher at lower values of TIT. There exists an optimum turbine inlet temperature for a given compressor pressure ratio of combined cycle with air cooling means which has been found to be 1600 K. However, in the case of steam cooling there is no optimum value of turbine inlet temperature and its higher values continue to give better performance. Sanjay et al.[9], shows the effect of turbine inlet temperature (TIT) on coolant mass flow rate for different values of compressor pressure ratio and for a fixed value of maximum blade surface temperature and reheat pressure ratio of 2.8. The coolant (steam) flow requirement increases with increase in TIT expectedly. The quantity of coolant flow requirement in case of steam coolant is significantly less than that when compressor bled air is used. This is because of the fact that the specific heat of coolant steam is higher as compared to that of coolant air and its temperature is also lesser [10]. The second law efficiency of the waste heat recovery power generation system increases with exhaust gas inlet temperature. Higher gas inlet temperatures increases steam generation rate in HRSG, network output and availability of the gas. For same pinch point the entropy generation number increases with gas inlet temperature due to increased irreversibilities in HRSG. Yadav and Singh [11], shows cycle pressure ratio lie at about 18 for the maximum efficiency at the turbine inlet temperature of 1400K. While the maximum specific work output from the combined cycle is obtained at cycle pressure ratio of 16 and 1400 K TIT. It is obvious that the higher TIT is desirable for maximizing the efficiency and specific work output both. But the cycle pressure ratio should be selected for the maximum performance depending upon the cycle's operating states.

Effect of compressor pressure ratio

The temperature of air leaving the compressor and entering in combustion chamber depends upon the compressor pressure ratio. Currently Kaviri et al. [3] shows that increase in the compressor pressure ratio decrease the cost of exergy destruction. The reason is that by increase the compressor ratio, the outlet temperature increases. So, the temperature difference decreases. Since cost of exergy destruction is a direct function of exergy destruction. It leads to decrease in the cost of exergy destruction. As the compression ratio increases, the air exiting the compressors is hotter, therefore less fuel is required (lowering the air fuel ratio) to reach the desired turbine inlet temperature, for a fixed gas flow to the gas turbine. The work required in the compressor and the power output of the gas turbine, steadily increases with compression ratio, then cause decreases in the exhaust gases temperature. This lower gas temperature causes less steam to be produced in the HRSG, therefore lowering the outputs of the steam cycle. It is noticed that the total power output increases with compression ratio. However the variation of total power output is minor at lower compression ratio while it is significant at higher compression ratio for all gas turbine configurations. Ibrahim et al. [12], found overall thermal efficiency for regenerative gas turbine configuration very significance with lower compression ratio it reaches to 64.6% at compression ratio 6.4. But when the compression ratio increases the overall thermal efficiency of combined cycle with regenerative gas turbine configuration decreases, because, when compression ratio increases, temperature of the exhaust gases from the turbine decrease and temperature of air in outlet of compressor increases, so, the recovered thermal energy in regenerative heat exchanger falls until zero. The effect of compressor pressure ratio on second law efficiency of various components was reported by Khaliq [6], that the second-law efficiency of the adiabatic compressor increases with pressure ratio because the absolute values of the work input and exergy increase are both larger and the magnitude of exergy destruction in the adiabatic compressor increases with the increase in pressure ratio. The first-law efficiency of the adiabatic turbine increases with the increase in pressure ratio. The second-law efficiency decreases with the pressure ratio show that the first-law and second-law efficiencies of the combined cycle increases up to the pressure ratio of 32, and then they start decreasing with increases in the pressure ratio. But it is interesting to note that the second-law efficiency of the combined cycle is greater than the first-law efficiency for same pressure-ratio. If the pressure ratio is too low, then the gas-turbine cycle and combined-cycle efficiencies and their specific work-outputs drop, whereas the steam cycle work-output increases due to the high gas-turbine exhaust temperature. At an intermediate pressure-ratio, both the efficiency and specific work reaches at peak. If the pressure ratio is too high, the compressor and turbine works increase but their difference, the net gas-turbine work output drops. The absolute magnitude of exergy destroyed in both compressor and turbine increases as the logarithm of pressure ratio. The exergy destruction in the combustion chamber decreases with the pressure ratio the effect of increasing the pressure ratio and the cycle-temperature ratio on the first-law efficiency of the gas-turbine cycle. The increase in pressure ratio increases the overall thermal efficiency at a given maximum temperature [5]. Thermodynamic analysis indicates that exergy destruction in combustion chamber and HRSG is significantly affected by the pressure ratio and turbine inlet temperature, and not at all affected by pressure drop and evaporator temperature. It also indicates that maximum exergy is destroyed during the combustion and steam generation process; which represents over 80% of the total exergy destruction in the overall system. The exergy destruction in combustion chamber and heat recovery steam generator decreases significantly with the increase in pressure ratio but increases significantly with the increase in turbine inlet temperature. At a given TIT, pressure drop, process heat pressure, and evaporator temperature, the exergy destruction in compressor and turbine increases with the increase in pressure ratio. In the case of turbine blade cooling Sanjay et al. [8], reported that at any turbine inlet temperature, the plant specific work decreases linearly with increase of compressor pressure ratio for all cooling means. Therefore, it is desirable to select lower value of compressor pressure ratio for higher plant specific work for all cooling means. However, a compromise is to be made for optimum compressor pressure ratio due to different behavior for compressor pressure ratio and turbine inlet temperature. In his previous work Sanjay [9], shows of reheat pressure ratio (rp, hpt) on combined cycle plant efficiency and specific work for R3PR configuration at rp, c=30, TIT=1700 K and Tb=1123 K (in both combustors) for closed-loop-steam cooling (CLSC) is depicted. It shows that both plant efficiency and specific work increases. Yadav [11], reported that the cycle pressure ratio lie at about 18 for the maximum efficiency. While the maximum specific work output from the combined cycle is obtained at cycle pressure ratio of 16. But the cycle pressure ratio should be selected for the maximum performance depending upon the cycle's operating states.

Effect of pinch point

The pinch point is the minimum difference between the gas temperature leaving the evaporator section of the HRSG and the saturation temperature corresponding to the steam pressure in the evaporator section. The effect of pinch point is earlier reported by Kaviri et al. [3], that the increase of the pinch temperature of the heat recovery boiler, the cycle's exergy efficiency reduces, the reason is that the water section uses less of the energy of the output gases from the heat recovery boiler. Moreover, by increasing the temperature of the super heater, both the exergy loss in the heat recovery boiler and the cost of exergy loss are reduced. On the other hand, it is clear that by raising the pinch temperature, both the exergy loss and the cost of exergy loss are increased. Consequently the more specific result by Ataei and yoo [13], shows the Combined Pinch and Exergy Analysis (CPEA) of the hot and cold Composite Curves of the Rankine cycle and defines the energy and Exergy requirements. The basic assumption of the minimum approach temperature difference required for the Pinch Analysis is represented as a distinct exergy loss that increases the fuel requirement for power generation. The exergy composite curves put the focus on the opportunities for fuel conservation in the cycle. The application results of CPEA in the power plant showed that its fuel consumption could be reduced by 5.3% and the thermal cycle performance could be increased from 39.4 to 41.9%. In addition, the production drop problem in current power plants, due to the inefficiency of the cooling system, especially in warm seasons, could be eliminated. And in previous years Butcher and reddy [10], states that the second law efficiency for the waste heat power generation system decreases with pinch point. This is due to reduced steam generation and network output with increased pinch point values. With higher pinch point the temperature difference between gas and water/steam is also high resulting in high irreversibilities. The second law efficiency of the system is sensitive to pinch point and the pinch point should be low for better performance based on second law point of view. The second law efficiency for the waste heat power generation system decreases with pinch point. This is due to reduced steam generation and network output with increased pinch point values. With higher pinch point the temperature difference between gas and water/steam is also high resulting in high irreversibilities. The variation of first law efficiency with pinch point is not so much compared to second law efficiency. This clearly demonstrates that the second analysis provides more details on the performance of the power generation system based on quality point of view. The second law efficiency is higher as it provides a measure of how efficiently the system is using thermodynamic resources based on quality point of view. The first law efficiency is low because the majority of the heat input is carried away by the condenser. The second law efficiency decreases with higher pinch point as work output is reduced due to lower steam generation. Generally, lowering the pinch point results in an increase of total heat recovered in the evaporator section. However, lowering the pinch point will decrease the logarithmic mean temperature difference (LMTD) in the HRSG, requiring more heat transfer surface area. This significantly increases the equipment cost. Sue and chang [14], shows, if the pinch point of an HRSG is changed to 10, 20 or 30°C and the pressure of process steam/water is fixed, the total required heating surface of HRSG will be reduced, leading to a reduction in steam temperature. This change will affect the steam cycle thermal performance.

Effect of ambient temperature

The location of power plant plays a major role on its performance. The atmospheric air which enters the compressor gets hotter after compression and goes to combustion chamber. The effect of ambient temperature has been reported by several authors and more recently by Ashley et al. [15], that for every K rise in ambient temperature above ISO condition the Gas Turbine looses 0.1% in terms of thermal efficiency and 1.47 MW of its Gross (useful) Power Output. While in same year the author Ibrahim et al. [12], reported that when ambient temperature increase from 273K to 333K, the total power output increase about 7% for all configurations except the regenerative gas turbine. The overall thermal efficiency also decreases due to increases the losses of the exhaust gases. The overall thermal efficiency of the combined cycle obtained maximum value with regenerative gas turbine configuration about 62.8% at ambient temperature 273K and the minimum value of the overall thermal efficiency was about 53% for intercooler gas turbine configuration at ambient temperature 333K. The gas turbine combined cycle inlet air cooling was studied by Yang et al. [16], and the effect of RH and ambient temperature on gas turbine combined cycle inlet air cooling (GTCCIAC) efficiency ratio is shown. When relative humidity (RH) remains the same, GTCCIAC efficiency ratio increases with the rise of ambient air temperature (ta). It is worth pointing out that GTCCIAC efficiency decreases in the case of ta<15°C or so due to the inlet air temperature characteristics of GTCC. More precisely the result was shown by Polyzakis et al. [7], as the ambient temperature increases, the air density falls; hence, for a given TIT, the mass flow through the engine is reduced. As a consequence, the engine output power is lower. While the ambient temperature rises, the net power generated in the combined-cycle thermal-plant decreases in spite of the use of the maximum supplementary-firing. Arrieta and lora [17] registered that, with a gas temperature of 675°C after the supplementary firing, the net electric-power varies in the range from 640 to 540 MW when the ambient temperature varies between 0 and 35°C. While the ambient temperature rises, the combined cycle thermal plant's heat-rate increases (that is, the efficiency decreases), in spite of the use of the minimum supplementary- firing temperature. The heat rate is even greater when the maximum supplementary-firing temperature is used. Sanaye and tahani [18], shows the effects of evaporative cooling on gas turbine performance. Evaporative cooling process occurs in both compressor inlet duct (inlet fogging) and inside the compressor (wet compression). By predicting the reduction in compressor discharge air temperature, the modeling results were compared with the corresponding results reported in literature and an acceptable difference percent point was found in this comparison. Then, the effects of both evaporative cooling in inlet duct, and wet compression in compressor, on the power output, turbine exhaust temperature, and cycle efficiency of 16 models of gas turbines categorized in four A–D classes of power output, were investigated. The analysis for saturated inlet fogging as well as 1% and 2% overspray are reported and the prediction equations for the amount of actual increased net power output of various gas turbine nominal power output are proposed. Furthermore the change in values of physical parameters and moving the compressor operating point towards the surge line in compressor map was investigated in inlet fogging and wet compression processes. Sue and chuang [14], shows that the location of the power station plays an important role on its performance. The power output of a gas turbine increases as the inlet air temperature decreases.

Effect of pressure levels

The HRSG configuration and optimization was modeled by several authors in past years and the useful result was found towards the performance of combined cycle. The most recent was reported by Mansouri et al. [19]. In his work a double pressure and two triple pressures (with and without reheat) HRSG were modeled and the results were compared. The results show that the stack gas exergy and the exergy destruction due to heat transfer decrease with the increase in number of pressure levels of steam generation. Also, the increase of pressure levels of steam generation in HRSG increases the heat recovery from the flue gas and as a result, the energetic efficiency of the cycle increases. Furthermore, the exergy destruction rate of the cycle decreases with the increase in number of pressure levels of steam generation in HRSG. The increase of the number of pressure levels of steam generation increases the total and specific investment cost of the plant as 6% and 4%. The net present value (NPV) of the plant increases as 7% in the case of triple pressure reheat in comparison with the double pressure CCPP. Therefore, it is economically justifiable to increase the number of pressure level of steam generation in HRSG. The comparison of CC plants was presented by Woudstra et al. [20], with increasing number of pressure levels of steam generation in the HRSG shows that the efficiency gain of a triple pressure system in comparison with a single pressure system is caused by the reduction of the exergy loss of heat transfer in the HRSG as well as the lower exergy of the flue gasses discharged to the stack. The last effect is even more important than the reduction of exergy losses due to heat transfer as can be learned from the value diagrams of the HRSG's. In the case of the triple pressure system the remaining exergy losses of heat transfer and flue gas discharge together are about 5% of the fuel exergy. A further increase of the number of steam pressure levels in the HRSG does not seem to be really beneficial; it enables only a small reduction of the overall exergy loss of the plant. Srinivas [21], focused on a dual pressure reheat (DPRH) HRSG to maximize the heat recovery and hence performance of CC. Deaerator, an essential open feed water heater in steam bottoming cycle was located to enhance the efficiency and remove the dissolved gasses in feed water. Each of the heating section in HRSG is solved from the local flue gas condition with an aim of getting minimum possible temperature difference. For high performance, better conditions for compressor, HRSG sections, steam reheater and deaerator are developed. The CC system is optimized at a gas turbine inlet temperature of 1400°C due to the present available technology of modern gas turbine blade cooling systems.

Miscellaneous Configurations and Discussion

Besides the operating parameters discussed above there are several configurations reported by different authors in over the years which improve the combined cycle performance has been discussed here. Tica et al. [22], presented a method to transform a CCPP physical model designed for simulation in an optimization-oriented model, which can be further used with efficient algorithms to improve start-up performances. Chacartegui et al. [23], used different fuels of the referred type on the performance of a gas and steam combined cycle performance. Sipöcza et al. [24], reduces the costs of CO_2 capture. The gas turbine is utilizing exhaust gas recirculation (EGR) with a level of 40% and thereby the CO2 content in the gas turbine exhaust gas is increased to almost the double compared to conventional operating gas turbines. Godoy et al.[25], reported the optimal combined cycle gas turbine power plants characterized by minimum specific annual cost values. Haglind [26], studied two different gas turbine configurations, a two-shaft aero-derivative configuration and a single-shaft industrial configuration. The results suggest that by the use of variable geometry gas turbines, the combined cycle part-load performance can be improved. Franco [27], shows that even if the use of supercritical HRSG is not particularly convenient in the perspective of efficiency increase it can be a valid technical solution aiming to the development of medium size (50-120 MW) combined cycle power plants. Kaushik et al. [28], provided a detailed review of different studies on thermal power plants over the years and also throw light on the scope for further research and recommendations for improvement in the existing thermal power plants [29]. The latent heat of spent steam from a steam turbine and the heat extracted from the air during the compression process are used to heat liquefied natural gas (LNG) and generate electrical energy. It shows that the net electrical efficiency and the overall work output of the proposed combined cycle can be increased by 2.8% and 76.8 MW above those of the conventional combined cycle. Regulagadda et al. [30], performed a thermodynamic analysis of a subcritical boiler–turbine generator for a 32 MW coal-fired power plant. The exergy loss distribution indicates that boiler and turbine irreversibilities yield the highest exergy losses in the power plant. Aljundi [31], shows that the percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the boiler system (77%) followed by the turbine (13%), and then the forced draft fan condenser (9%). In addition, the calculated thermal efficiency based on the lower heating value of fuel was 26% while the exergy efficiency of the power cycle was 25%. Qiu and hayden [32], constitutes a "renewable biofuel" energy resource and energy from waste (EfW). This approach innovates by delivering better performance with respect to energy efficiency and CO_2 mitigation. Srinivas et al. [33], shows that the major exergetic loss that occurs in combustion chamber decreases with introduction of steam injection in to the combustion chamber. Koch et al. [34], deals with the application of an evolutionary algorithm to the minimization of the product cost of complex combined cycle power plants. Han et al. [35], proposed a novel combined cycle with synthetic utilization of coal and natural gas, in which the burning of coal provides thermal energy to the methane/steam reforming reaction. As a result, the overall thermal efficiency of the new cycle reaches 52.9%, as energy supply by methane is about twice as much as these of coal. A complex thermodynamic analysis of a steam power cycle with feed water heaters (fwh) is simplified with the introduction of a generalized concept for 'n' fwhs [36]. The exergetic loss in the boiler decreases with the addition of heaters. The results show that the exergetic loss associated with the combustion of fuel is more when compared to heat transfer. Poullikkas [37], introduced a technique to reduce the irreversibility of the steam generator of the combined cycle. The results indicated that the optimized combined-cycle is up to 1% higher in efficiency than the reduced-irreversibility combined-cycle, which is 2–2.5% higher in efficiency than the regularly-designed combined-cycle when compared for the same values of TIT and Kakaras et al. [38], used low-temperature waste heat or solar heat for the evaporation of injected water droplets in the compressed air entering the gas turbines combustion chamber. Poullikkas [37], recommended that the future use of combined cycle technology with natural gas as fuel. By the use of natural gas combined cycle, the CO_2 emissions would be significantly reduced. Recent advances in gas turbine development have led to wider usage of combined power plant for electrical power generation, and made it possible to reach a thermal efficiency of 55–60%. This was a result of introducing higher turbine inlet temperature (TIT) and other factors. However, this temperature is restricted by the metallurgical limit of turbine blades of about 800°C. Thus, need arises to design efficient cooling systems to cool the turbine components subjected to such high temperatures. The performance of a combined system with different cooling techniques in the high temperature section of the turbine is evaluated. Najjar et al. [39], developed a general model of the combined system and used to compare the performance relevant to the three main schemes of blade cooling, namely air-cooling, open-circuit steam cooling (OCSC) and closed-loop steam cooling (CLSC). Leo et al. [40], studied a new design of a combined-cycle gas turbine power plant

CCGT with sequential combustion that increases efficiency and power output in relation to conventional CCGT plants. Franco and Russo [41], optimized the heat recovery steam generator (HRSG) that approaches the 60%, with and increases of the heat surface and a decrease of the pinch-points [42]. The HRSG optimization is sufficient to obtain combined cycle plant efficiencies of the order of 60% Gas/steam combined cycle has already become a well-known and substantial technology for power generation due to its numerous advantages including high efficiency and low environmental emission. Many studies have been carried out for better performance and safe and reliable operation of combined-cycle power plants. A power plant is basically operated on its design conditions. However, it also operates on the so called off-design conditions due to the variation in a power load, process requirements, or operating mode. Therefore, the transient behavior of the system should be well-known for the safe operation and reliable control. Shin et al. [1], performed dynamic simulation to analyze the transient behavior of a combined-cycle power plant. Each component of the power plant system is mathematically modeled and then integrated into the unsteady form of conservation equations. Transient behavior was simulated when rapid changes and periodic oscillations of the gas turbine load are imposed. Time delay characteristic caused by the thermal and fluid damping is analyzed and overall time-response of the combined power plant system is shown. Bejan [43], begins with a review of the concept of irreversibility, entropy generation, or exergy destruction. Examples illustrate the accounting for exergy flows and accumulation in closed systems, open systems, heat transfer processes, and power and refrigeration plants. Wu [44], describe the use of intelligent computer software to obtain a sensitivity analysis for the combined cycle. Andriani et al. [45], carried out the analysis of a gas turbine with several stages of reheat for aeronautical applications. Heppenstall [46], describes and compares several power generation cycles which have been developed to take advantage of the gas turbine's thermodynamic characteristics. Emphasis has been given to systems involving heat recovery from the gas turbine's exhaust and these include the combined, Kalina, gas/gas recuperation, steam injection, evaporation and chemical recuperation cycles. Nag and Raha [47], evaluated the effects of pressure ratio and peak cycle temperature ratio of the gas cycle and the lower saturation pressure of the steam cycle on the overall performance of the combined plant. Here the working fluids work over a large temperature range, say from 1100°C to 550°C in the gas turbine and 550°C to the ambient temperature in the steam turbine, thus achieving an overall efficiency approaching 50%. Horlock [48], based on thermodynamic considerations, outlined more recent developments and future prospects of combined-cycle power plants. Polyzakis [49], carried out the first-law analysis of reheat industrial gas- turbines use in a combined cycle and suggested that the use of reheat is a good alternative for combined-cycle applications. El-Masri [50], analyzed the combined gas–steam plant, without reheat, from the thermodynamic point of-view. In his analysis, he singled out the parameters that most influence efficiency, and further reported that combined cycles exhibit a good performance if suitably designed, but if the highest gas-turbine temperatures are used, expensive fuel must be utilized. Wunsch [52], claimed that the efficiencies of combined gas–steam plants were more influenced by the gas-turbine parameters like maximum temperature and pressure ratio than by those for the steam cycle and also reported that the maximum combined-cycle efficiency was reached when the gas-turbine exhaust temperature is higher than the one corresponding to the maximum gas-turbine efficiency. Czermak and Wunsch [53], carried out the elementary thermodynamic analysis for a practicable Brown Boveri 125 MW combined gas–steam turbine power plant. The early development of the gas/steam turbine plant was described by Khaliq and Kaushik [6]. The performance analysis based on the first-law alone is inadequate and a more meaningful evaluation must include a second-law analysis. One reason that such an analysis has not gained much engineering use may be the additional complication of having to deal with the "combustion irreversibility", which introduced an added dimension to the analysis. Second-law analysis indicates the association of exergy destruction with combustion and heat-transfer processes and allows a thermodynamic evaluation of energy conservation in thermal power cycles.

Conclusions

The combination of the gas turbine Brayton Cycle and the steam power system Rankine Cycle complement each other to form efficient combined-cycles. The Brayton Cycle has high source temperature and rejects heat at a temperature that is conveniently used as the energy source for the Rankine Cycle. The most commonly used working fluids for combined cycles are air and steam. Other working fluids (organic fluids, potassium vapor, mercury vapor, and others) have been applied on a limited scale. Combined-cycle systems that utilize steam and air-working fluids have achieved widespread commercial application [2] due to:

- High thermal efficiency through application of two complementary thermodynamic cycles.
- Heat rejection from the Brayton Cycle (gas turbine) at a temperature that can be utilized in a simple and efficient manner.
- Working fluids (water and air) that is readily available, inexpensive, and non-toxic.

After going through the past review of research the effects of major operating parameters can be summarized as follows:

1. The turbine inlet temperature (TIT) significantly affects the performance of combined cycle. It should be kept on higher side for minimizing the exergy losses.
2. The compressor pressure ratio should be optimum for maximum performance of combined cycle.
3. The decrease in pinch point temperature the more heat transfer in steam bottoming cycle thus improving the combined cycle performance.
4. The ambient temperature also affects the combined cycle performance.
5. The increase in number of pressure levels improves the combined cycle performance.
6. There are miscellaneous configurations which also improves the combined cycle performance.

References

1. Shin JY, Jeon YJ, Maeng DJ, Kim JS, Ro ST (2002) Analysis of the dynamic characteristics of a combined-cycle power plant. Energy 27: 1085-1098.
2. Chase DL Combined-Cycle Development Evolution and Future. Schenectady, NY, USA.
3. Kaviri AG, Jaafar MNM, Lazim TM (2012) Modeling and multi-objective exergy based optimization of a combined cycle power plant using a genetic algorithm. Energ Convers Manage 58: 94-103.
4. Sanjay (2011) Investigation of effect of variation of cycle parameters on thermodynamic performance of gas-steam combined cycle. Energy 36: 157-167.

5. Khaliq A(2009) Exergy analysis of gas turbine trigeneration system for combined production of power heat and refrigeration. Int J Refrig 32: 534-545.

6. Khaliq A, Kaushik SC (2004) Second-law based thermodynamic analysis of Brayton/Rankine combined power cycle with reheat. Appl Energ 78: 179-197.

7. Polyzakis AL, Koroneos C, Xydis G (2008) Optimum gas turbine cycle for combined cycle power plant. Energ Convers Manage 49: 551-563.

8. Sanjay, Singh O, Prasad BN (2008) Influence of different means of turbine blade cooling on the thermodynamic performance of combined cycle. Appl Therm Eng 28: 2315-2326.

9. Sanjay Y, Singh O, Prasad BN (2007) Energy and exergy analysis of steam cooled reheat gas–steam combined cycle. Appl Therm Eng 27: 2779-2790.

10. Butcher CJ, Reddy BV (2007) Second law analysis of a waste heat recovery based power generation system. Int J Heat Mass Tran 50: 2355-2363.

11. Yadav JP, Singh O (2006) Thermodynamic Analysis of Air Cooled Simple Gas/ SteamCombined Cycle Plant. IE (I) Journal-MC 86.

12. Ibrahim TK, Rahman MM, Abdalla AN(2011) Optimum Gas Turbine Configuration for Improving the performance of Combined Cycle Power Plant. Procedia Engineering 15: 4216-4223.

13. Ataei A, Yoo CK (2010) Combined pinch and exergy analysis for energy efficiency optimization in a steam power plant. International Journal of the Physical Sciences 5: 1110-1123.

14. Sue DC, Chuang CC (2004) Engineering design and exergy analyses for combustion gas turbine based power generation system. Energy 29: 1183-1205.

15. Ashley De Sa, Zubaidy SA (2011) Gas turbine performance at varying ambient temperature. Appl Therm Eng 31: 2735-2739.

16. Yang C, Yang Z, Cai R (2009) Analytical method for evaluation of gas turbine inlet air cooling in combined cycle power plant. Applied Energy 86: 848-856.

17. Arrieta FRP, Lora EES (2005) Influence of ambient temperature on combined-cycle power-plant performance. Applied Energy 80: 261-272.

18. Sanaye S, Tahani M (2010) Analysis of gas turbine operating parameters with inlet fogging and wet compression processes. Appl Therm Eng 30: 234-244.

19. Mansouri MT, Ahmadi P, Kaviri AG, Jaafar MNM (2012) Exergetic and economic evaluation of the effect of HRSG configurations on the performance of combined cycle power plant. Energy Conversion and Management 58: 47-58.

20. Woudstra N, Woudstra T, Pirone A, Van der Stelt T (2010) Thermodynamic evaluation of combined cycle plants. Energy Conversion and Management 51: 1099-1110.

21. Srinivas T (2010) Thermodynamic modelling and optimization of a dual pressure reheat combined power cycle. Indian Academy of Sciences 35: 597-608.

22. Tica A, Gueguen H, Dumur D, Faille D, Davelaar F (2012) Design of a combined cycle power plant model for optimization. Applied Energy 98: 256-265.

23. Chacartegui R, Sanchez D, Munoz de Escalona JM, Sanchez T (2012) Gas and steam combined for low calorific syngas fuels utilisation. Applied Energy.

24. Sipöcza N, Tobiesen A, Assadi M (2011) Integrated modelling and simulation of a 400 MW NGCC power plant with CO2 capture. Energy Procedia 4: 1941-1948.

25. Godoy E, Benz SJ, Scenna NJ (2011) A strategy for the economic optimization of combined cycle gas turbine power plants by taking advantage of useful thermodynamic relationships. Appl Therm Eng 31: 852-871.

26. Haglind F (2011) Variable geometry gas turbines for improving the part-load performance of marine combined cycles- Combined cycle performance. Appl Therm Eng 31: 467-476.

27. Franco A (2011) Analysis of small size combined cycle plants based on the use of supercritical HRSG. Appl Therm Eng 31: 785-794.

28. Kaushik SC, SivaReddy V, Tyagi SK (2011) Energy and exergy analyses of thermal power plants: A review. Renewable and Sustainable Energy Reviews 15: 1857-1872.

29. Xiaojun Shi, Agnew B, Che D, Gao J (2010) Performance enhancement of conventional combined cycle power plant by inlet air cooling, inter-cooling and LNG cold energy utilization. Appl Therm Eng 30: 2003-2010.

30. Regulagadda P, Dincer I, Naterer GF (2010) Exergy analysis of a thermal power plant with measured boiler and turbine losses. Appl Therm Eng 30: 970-976.

31. Aljundi IH(2009) Energy and exergy analysis of a steam power plant in Jordan. Appl Therm Eng 29: 324-328.

32. Qiu K, Hayden ACS (2009) Performance analysis and modeling of energy from waste combined cycles. Appl Therm Eng 29: 3049-3055.

33. Srinivas T, Gupta AVSSKS, Reddy BV (2008) Sensitivity analysis of STIG based combined cycle with dual pressure HRSG. International Journal of Thermal Sciences 47: 1226-1234.

34. Koch C, Cziesla F, Tsatsaronis G (2007) Optimization of combined cycle power plants using evolutionary algorithms. Chemical Engineering and Processing 46: 1151-1159.

35. HanW, Jin H, Xu W (2007) A novel combined cycle with synthetic utilization of coal and natural gas. Energy 32: 1334-1342.

36. Srinivas T, Gupta AVSSKS, Reddy BV (2007) Generalized Thermodynamic Analysis of Steam Power Cycles with 'n' Number of Feedwater Heaters. Int J of Thermodynamics 10: 177-185.

37. Poullikkas A (2004) Parametric study for the penetration of combined cycle technologies into Cyprus power system. Appl Therm Eng 24: 1697-1707.

38. Kakaras E, Doukelis A, Leithner R, Aronis N (2004) Combined cycle power plant with integrated low temperature heat (LOTHECO). Appl Therm Eng 24: 1677-1686.

39. Najjar YSH, Alghamdi AS, Al-Beirutty MH (2004) Comparative performance of combined gas turbine systems under three different blade cooling schemes. Appl Therm Eng 24: 1919-1934.

40. Leo TJ, Perez-Grande I, Perez-del-Notario P (2003) Gas turbine turbocharged by a steam turbine: a gas turbine solution increasing combined power plant efficiency and power. Appl Therm Eng 23: 1913-1929.

41. Franco A, Russo A (2002) Combined cycle plant efficiency increase based on the optimization of the heat recovery steam generator operating parameters. International Journal of Thermal Sciences 41: 843-859.

42. Franco A, Casarosa C (2002) On some perspectives for increasing the efficiency of combined cycle power plants. Appl Therm Eng 22: 1501-1518.

43. Bejan A (2002) Fundamentals of exergy analysis, entropy generation minimization, and the generation of flow architecture. Int J Energy Res 26: 545-565.

44. Wu C (1999) Intelligent computer-aided sensitivity analysis of multi-stage Brayton/Rankine combined cycle. Energy Conv Mgmt 40: 215-232.

45. Andriani R, Ghezzi U, Anntoni LFG (1999) Jet engines with heat addition during expansion: a performance analysis paper 99-0744. AIAA.

46. Heppenstall T (1998) Advanced gas turbine cycles for power generation: a critical review. Appl Therm Eng 18: 837-846.

47. Nag PK, Raha D (1995) Thermodynamic analysis of coal based combined cycle power plant. Heat recovery system and CHP 12: 115-129.

48. Horlock JH (1995) Combined Power Plants-Past, Present, and Future. J Eng Gas Turbines Power 117: 608.

49. Polyzakis A (1995) Industrial gas-turbine for combined cycle plant. MSc thesis, Cranfield University.

50. El-Masri MA(1987) Exergy Analysis of Combined Cycles: Part 1-Air-Cooled Brayton-Cycle. Gas Turbines" Transactions of the ASME 109: 228.

51. Cerri G (1987) "Parametric Analysis of Combined Gas-Steam Cycles" Transactions of ASME 109: 46.

52. Wunsch A (1985) Highest efficiencies possible by converting gas-turbine plants into combined cycle plants. Brown Boveri Review10: 455-463.

53. Czermak H, Wunsch A (1982) The 125 MW combined cycle plant design features, plan performance and operating experience paper no. 82 GT-323]. ASME.