International Journal of Scholarly Research in
Engineering and Technology
Journal homepage: https://srrjournals.com/ijsret/
ISSN: 2961-3280 (Online)

(Research Article)

Techno-economic comparison of simple and modified gas turbine cycles

Efemena Efekemo *, Ebigenibo. G. Saturday and Joseph. C. Ofodu

Department of Mechanical Engineering, University of Port Harcourt, PMB 5323, Choba, Port Harcourt, Rivers State, Nigeria.

International Journal of Scholarly Research in Engineering and Technology, 2025, 05(01), 001-018

Publication history: Received on 26 November 2024; revised on 08 January 2025; accepted on 10 January 2025

Article DOI: https://doi.org/10.56781/ijsret.2025.5.1.0024

Abstract

This study investigates the integration of various waste heat recovery cycles as bottoming cycles to enhance the efficiency of gas turbine power plants. The primary focus is on the combined Gas Turbine, air bottoming cycle -Kalina cycle-organic Rankine cycle (GT-ABC-KC-ORC) configuration, which was compared to a simple cycle gas turbine (GT-KC) configuration. The analysis revealed that the GT-ABC-KC-ORC model achieved a superior overall plant efficiency of 53%, yielding an energy output of 238,573.9 kW—an increase of 42.5% over traditional systems. Additionally, this configuration produced the lowest exhaust temperatures, minimizing thermal pollution. The economic assessment indicated that the GT-ABC-KC-ORC model not only had the lowest cost of electricity generation but also required less investment in equipment compared to the GT-KC model. The cost analysis showed that upgrading a simple cycle gas turbine (GT) to the GT-ABC-KC-ORC would require an additional investment of 131.3% of the GT's cost, compared to 185.6% for the GT-KC upgrade. These findings underscore the potential of the GT-ABC-KC-ORC configuration as a cost-effective and efficient solution for enhancing power generation through waste heat recovery. Further research is recommended to explore different combinations of bottoming cycles to optimize waste heat recovery systems.

Keywords: Kalina Cycle; Economic Analysis; Waste heat; Modified cycles; LCOE; Efficiency

1. Introduction

Gas turbines is a type of internal combustion engine which is developed from the Brayton cycle. A simple cycle gas turbine is a common configuration in gas turbines for power generation, aircraft and other applications. It is made up of three major components; air compressor, combustion chamber and a turbine. A simple cycle gas turbine is open cycle because it draws air from the atmosphere and after the air has completed the cycle it releases it back into the atmosphere. In a simple gas turbine process, fresh air is drawn into the compressor component where it gets compressed to a higher pressure before entering the combustion chamber. At the combustion chamber, compressed air is combined with fuel and ignited for combustion to occur. The hot gases released after combustion exits the combustion chamber and channeled into the turbine. The hot gases impinge on the turbine blades and rotates it before it is released through to the environment through the exhaust.

Gas turbines operating in simple cycle configuration usually have efficiency of about 27% in older machines to 35% in newer more efficient machines. According to Espanani *et al,* [1], Improvement of gas turbine efficiency can be achieved by increasing the air inlet cooling, waste heat recovery and energy saving methods. Waste heat recovery methods require modification of the gas turbine cycle to capture waste heat. Alfellag [2] conducted a parametric study on an intercooled, reheated, and regenerative gas turbine power plant, examining the relationships between technical parameters and plant performance. The study found that increasing the effectiveness of the regenerator and intercooler, as well as improving turbine and compressor efficiency, enhanced the thermal efficiency of the plant. Optimal efficiency was achieved at a compression ratio of 2.2, after which efficiency declined with further increases.

* Corresponding author: E Efekemo

Various approaches to improving plant efficiency were explored. Increasing inlet temperatures requires enhanced thermal barrier coatings to withstand higher temperatures. Gas turbine modifications, including the heat recovery method, also improve efficiency. Combined heat and power (CHP) plants exemplify this, utilizing exhaust temperatures of around 500°C, which constitute about 60% of the total energy output of open cycle gas turbines [3].

Bolland and Stadaas [4] investigated different configurations, including water and steam injection and recuperation, to determine the most efficient setup for a combined cycle gas turbine plant. They compared a simple cycle, steam-injected cycle, and dual recuperated intercooled and after-cooled steam-injected cycle. The steam-injected cycle showed higher efficiency due to reduced cycle heat rejection and high specific work, though efficiency gains were limited by the pinch point in the exhaust gas heat recovery process. The DRIASI cycle, featuring steam and water injection, was found to be highly efficient, particularly for smaller turbines. For medium and large applications, the combined cycle offered the highest efficiency. Evaporation cycles, such as the humid air turbine (HAT) or integrated gasification humid air turbine (IGHAT), use water injection into the compressor exit, leveraging the heat to evaporate the water and create a single-phase mixture that is further heated before combustion. Chemical recuperation uses exhaust heat to produce hydrogen-rich fuel from methane, enhancing the fuel efficiency.

Bottoming cycles, like the combined cycle and Kalina cycle, utilize exhaust heat from a gas turbine to power an additional independent system. The combined cycle, known for its high thermal efficiency (over 55%), flexibility, and lower emissions, uses exhaust gases to generate steam for a steam turbine. The Kalina cycle, utilizing a mixture of water and ammonia, claims even higher efficiency at 58.8% and lower construction costs compared to conventional combined cycles [5]. These methods highlight the potential for significant efficiency improvements and cost savings in power plants.

Habib et al. [6] conducted a study on the use of a reverse Brayton cycle, combined with an intercooler, regenerative reheat, and evaporative cooling, to reduce the intake air temperature for a gas turbine. In this design, air passes through a refrigeration cycle (reverse Brayton cycle) and an evaporator air cooler before entering the combustion chamber, which helps to lower the inlet air temperature, increasing the mass flow rate and overall power output. The study found that lowering the temperature from 299K to 287K resulted in a 2.18% increase in power output and a reduction in NOx emissions from 1000ppm to 100ppm. Elwekeel and Abdala [7] explored mist cooling, where inlet air is cooled below ambient temperature, and steam injection in gas turbines to enhance efficiency and reduce NOx emissions. Their findings indicated that using a heat recovery steam generator (HRSG) and demineralized water supply, steam injection increased net work output by 22% in simple cycles and 14% in intercooled cycles, while also reducing specific fuel consumption and NOx emissions.

Kayadelen and Ust [8] compared simple, intercooled, steam-injected, and intercooled steam-injected gas turbine cycles, revealing that steam injection boosts net work output and decreases specific fuel consumption and NOx emissions. Mohaptra and Sanjay [9] evaluated different inlet air cooling methods, including vapor compression and evaporative cooling, finding that vapor compression improved efficiency and work output more significantly. Bassily [10] examined the effects of evaporative inlet cooling and after cooling on recuperated gas turbine cycles, concluding that these configurations could significantly enhance power and efficiency.

Finally, Shukla and Singh [11] evaluated the performance of gas turbines with steam injection and evaporative inlet cooling, demonstrating that these methods are effective in improving efficiency, particularly in hot climates, by increasing the density of the intake air and the mass flow rate of the working fluid. According to Bahrami *et al.,* [12] single shaft gas turbines are very sensitive to frequency drops because the sudden drop in load and frequency causes a reduction in mass flow rate of air flowing into the turbine. Steam injected gas turbine cycle performs better in sudden large drops of frequency due to the amount of available power and high specific heat capacity of the working fluid. Further studies carried out by Chakartegui [13] with respect to turbine inlet air cooling showed that a gas turbine may lose its power output up to about 7% with an increase of air temperature by 15^{o}C. Cooling the air inlet temperature is one way of increasing the performance of a gas turbine. Another method of improving the efficiency of a gas turbine is by increasing the turbine inlet temperature above 800^{o}C. To achieve this, there must be improvement of the cooling of the turbine buckets to avoid melting of the hot section parts of the gas turbine. This metallurgical restriction must be overcome by cooling to achieve a higher turbine inlet temperature.

A research on a gas turbine integrated with solid oxide fuel cell was conducted by Haseli *et al.,* [14]. They combined a gas turbine with a high temperature solid oxide fuel cell (GT-SOFC) to ascertain the performance and irreversibility's within the system. In this configuration, compressed air is passed through the recuperator where it gains heat which increases the total efficiency. Then it exits the recuperator into the SOFC along with natural gas to produce direct current electricity. Heat is also produced in the SOFC during the process of electricity generation before it enters the combustion chamber. Previous works have shown that the efficiency of this system can achieve up to 60%. Their study showed that the GT-SOFC achieved a thermal efficiency as much as 27.8% more than simple cycle gas turbine. Efficiency of the cycle studied was at 60.55%. SOFC is increasingly being studied as a hybrid with gas turbine to produce cleaner energy.

Heat recovery of an aero derivative gas turbine which is used as heat source in a combined cycle configuration was studied by Carcasci and Winchler, [15]. Organic Rankine Cycle was further analyzed using different organic fluids as its working fluids. Working fluids are a critical component of a thermodynamic system. They have thermodynamic properties which make them suitable for the system which they are applied. In a Organic Rankine Cycle, the working fluid converts thermal energy into mechanical energy by changing phases, or heat of compression/expansion. Working fluids have different boiling points. For their study, toluene, benzene, cyclopentane and cyclohexane were tested. For the research, two models were developed; waste heat was recovered from the exhaust of the gas turbine while in the second configuration, waste heat was recovered from an intercooler in the gas turbine. The result from their study showed that using benzene and cyclohexane, an efficiency of 54.4% can be reached for the combined plant while toluene and cyclopentane did not yield good results. for the configuration using the recovery heat from the intercooler, the efficiency increased by 2.2 percent. The temperature of heat source is a determinant of the of working fluid in an organic Rankine cycle. Herath *et al.,* [16] studied the performance of applying seven different working fluids in an organic rankine cycle for waste heat recovery. The fluids studied are Benzene, R-134a, Ethanol, R-245fa, Methanol, Acetone and Propane (R-290). They concluded from their results that Methanol and Benzene performed more efficiently in an ORC system when compared with the others.

An analysis of a combined gas turbine and Maisotsenko cycle model developed by Saghafifar and Gadalla, [17] to evaluate the efficiency of the proposed configuration successfully resulted in significant efficiency improvement. The proposed configuration consisted of a gas turbine at the top of the cycle and Maisotsenko gas turbine at the bottom cycle. The exhaust from the top gas turbine cycle was used to generate humidity and to heat up the air in the bottoming cycle. The proposed model included using an air saturator to replace the heat exchanger. Overall, an analysis of the efficiency resulted in an efficiency improvement of 3.7%.

A model of a gas turbine modified with air bottoming cycle was analyzed by Alklaibi *et al.,* [18] to compare the efficiency ratios with modified gas turbine and combined cycles. All the modified gas turbine cycles were equipped with intercooler and reheater. Their research showed that using more than one bottoming cycle is affected by the pressure ratio. From the studies, maximum efficiencies occurred when the pressure ratio was 3 for the bottom cycle and 11 for the top cycle. Also the study concluded that the use of an intercooler and reheater increased the GT-ABC efficiency up to a certain value of its pressure ratio. Korobitsyn, [19] studied the application of air bottoming cycle to upgrade simple cycle gas turbines installed in offshore for different other industrial applications. The application of ABC in combined heat and power plants whose output included hot air benefited the most. This modification had the advantages of significant fuel savings and also a payback period of between 3 – 4 years.

Pierobon and Haglind [20] studied ABC for offshore applications. In their work, a SGT-500 gas turbine was modeled various configurations of an air bottoming cycle (ABC). The applied theory of power maximization and multi-objective optimization for economic optimization of their model. The optimal pressure ratio was found to be 2.80 when applying the theory of power maximization.

Kalina cycle can be used as a bottom cycle in a combined cycle configuration. It has the advantage of having different configurations which can be used in various applications, however due the capacity to work with low grade heat sources, it is applied mostly for geothermal power generation. Kalina cycle has been used by Wang *et al.,* [21] in recovery of heat from a compressor intercooler. The system modeled was also compared with an organic Rankine cycle used to recover the waste heart intercooler. In their study, natural gas is compressed in two stages with an intercooler in between. Natural gas is first pumped into the first compressor where it is compressed then it passes through an intercooler where it is cooled before it is passed on to a second compressor. The heat released in the intercooler is then extracted by the binary fluid of either the Kalina cycle or the vapour compressor. The study concluded that the net power consumption shown in the optimization results was lower in the Kalina cycle when compared with the ORC.

Kalina and Leibowitz [22] also carried out a similar study to compare the performance of a Kalina bottom cycle and a triple pressure steam plant. The gas turbine cycle contained 3 units of gas turbines and a Kalina combined cycle arrangement. The Kalina cycle has three turbines: the high-pressure turbine, a reheating process before entry into the intermediate pressure turbine then a re-cooling process before entry into the low-pressure turbine. The recuperator heats up the vapour from the intermediate pressure turbine. Kalina cycle applied a binary fluid (ammonia water) as its working fluid due to its distinct properties. The condenser in the Kalina cycle uses a vapour absorption principle whereby the binary fluid goes through the process of separation then absorption, condensation and recombines before it enters the boiler. The results of their study showed a 16 – 32% increment in the bottom cycle power output compared to triple pressure steam cycle. They also concluded that there are no technological barriers in the application of Kalina cycle because the temperatures required are within the range found in power plants.

Oyedepo et al, [23] in their study revealed that approximately 72% of global primary energy consumption is lost post-conversion, with 63% of waste heat streams occurring at temperatures below 100°C. The largest share of this waste heat comes from electricity generation, followed by transportation and the manufacturing industry. The study suggests that a significant portion of waste heat can be recovered using sustainable technologies, contributing to sustainable energy development. Efficient utilization of waste heat can help slow the depletion of fossil fuels and reduce toxic emissions, supporting the transition to renewable energy sources. Their study concludes that WHR technologies are highly effective for sustainable energy development, offering economic and environmental benefits by reducing primary energy demand and CO2 emissions. There is an abundant supply of both fossil fuels and waste heat resources that can meet global energy demands while supporting sustainable development. However, the primary challenges are business and regulatory, rather than technical and there is an urgent need for policies that promote WHR technologies, which typically have short payback periods and require minimal maintenance. Comprehensive research into the optimal energy utilization in various regions is recommended to better understand the longevity of fossil fuel reserves and the impact on ecosystem stability, further highlighting the need for WHR technologies.

Farhat et al, [24] study concluded that water is the best working fluid for high-temperature Rankine cycles, while organic fluids like R245fa and ammonia perform well in specific conditions but have economic and operational limitations. Thermoelectric generators (TEGs) show potential for converting heat into electricity, with their efficiency influenced by the material properties and temperature differences. In residential systems, heat recovery from hot exhaust gases and drain water systems are promising applications. In industrial settings, WHR technologies are used in processes such as food waste boilers and coal combustion. Pilodia et al [25] discussed the considerable amount of low-temperature waste heat that is often not utilized and dissipates into the atmosphere. To tackle this inefficiency, the Kalina cycle was introduced, which efficiently captures this waste energy, offering a significant improvement over the traditional Rankine cycle. The Kalina cycle utilizes an ammonia-water binary mixture as its working fluid, operating effectively at temperatures below 200°C. This cycle provides a 20-40% enhancement in thermal efficiency compared to conventional Rankine cycle-based waste heat power plants, making it a crucial technology for energy recovery. Over the past decade, numerous Kalina Cycle power plants have been established to harness electricity from low-grade heat sources, including industrial waste heat, waste incineration, and geothermal springs. These installations have met or surpassed performance expectations, demonstrating a level of efficiency that other systems cannot achieve when extracting energy from low-grade heat. Ensuring reliable operation and high availability requires good control of water chemistry, particularly maintaining high pH levels to minimize the risk of corrosion. The potential for further deployment of the Kalina Cycle in industries like iron-steel and cement is actively being pursued, underscoring its importance in efficient energy utilization. [26] investigated how the ammonia fraction and turbine inlet pressure affected the cycle's performance. Their findings indicated that for a specific turbine inlet pressure, there was an optimal ammonia fraction that maximized cycle efficiency. However, achieving maximum cycle efficiency did not always align with the most favorable operating conditions for the system. Additionally, factors such as the utilization of the working fluid, cooling water, heat resource, and heat exchanger area per unit power produced were critical. For the conditions studied, an optimal range of operating pressure and ammonia fraction could be identified that led to the best overall cycle performance. Overall, the KCS11 demonstrated better performance at moderate pressures compared to the ORC.

2. Methodology

To carry out this study, three different gas turbine models would be developed; a model of a simple cycle gas turbine was from an existing installed gas turbine (GT), two modified gas turbine cycle were modeled from the GT to improve the efficiency of the installed gas turbine Gas turbine with high temperature Kalina Cycle (GT-KC) and Gas turbine cycle with Kalina Cycle, Air Bottoming Cycle and Organic Rankine Cycle (GT-ABC-KC-ORC). The different models provide a means to compare the simple cycle and modified cycles to compare the thermodynamic and economic analysis. The power plant used in developing the model is a 167.4MW Alstom GT13E2 gas turbine located rivers IPP in Afam rivers state. Engineering equation solver (EES) was used in modelling the simple and modified gas turbine cycles.

A simple cycle gas turbine (GT) is composed of three major components i.e. compressor, combustion chamber and turbine. A simple cycle is configured as an open cycle gas turbine which draws fresh air into the system at state 1 and compresses to a higher pressure at state 2 before entering the combustion chamber (CC). At the combustion chamber, compressed air is combined with fuel (natural gas) and ignited for combustion to occur. The hot gases released after combustion exit the combustion chamber at state 3 it is channeled through the turbine. The hot gases impinge on the turbine blades as it forces its way out of the system into the environment. Figure 1 describes a simple cycle gas turbine and Figure 2 shows the T-s diagram.

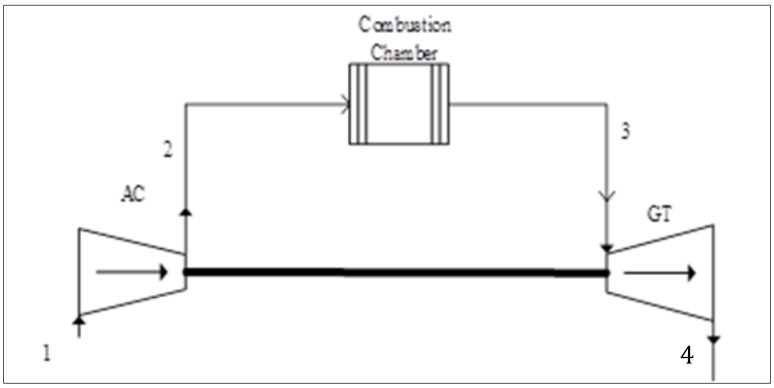

Figure 1 Schematic of simple cycle gas turbine

Kalina cycle is type of thermodynamic power cycle which extract heat and converts the heat into work. It is a modification of Rankine cycle, which uses a binary fluid instead of steam as its working fluid. Binary fluids are uniquely composed of two different mixable fluids having different concentrations. A binary fluid is unique because the two fluids in the mixture have different boiling and dew points. The uniqueness of the binary solution allows its application in power cycles requiring low heat sources. There are different types binary fluids but Ammonia-water is commonly used in Kalina Cycle. Ammonia water mixture is a non-azeotropic mixtures characterized by changes in its boiling points for different compositions of the fluids in the mixture. Ammonia water solution is suitable for Kalina cycle because it absorbs heat energy from low heat sources. Ammonia has a lower boiling point which means it vaporizes faster and can be superheated at lower temperatures compared to water. Kalina cycle is designed to extract low temperature heat from waste heat sources like geothermal wells. There are different configurations of Kalina cycle suited for different heat sources. High temperature Kalina cycle is a modified Kalina cycle configuration which is adapted to higher temperature heat sources like solar, geothermal sources etc. High temperature Kalina cycle is similar to Rankine cycle, in the sense that it converts waste heat from gas turbine into work. It comprises of 3 pressure levels, an economizer, an evaporator, and a super heater.

In the GT-KC model, air is introduced to the compressor at state 1 and compressed through the multiple stages of the axial compressor and passes into a combustion chamber at state 2 at a higher pressure. Inside the combustion chamber, natural gas is mixed in a lean ratio with the high pressure air before it is ignited using spark plugs. As the products of combustion exit the combustion chamber at state 3, the hot gases as they go through the turbine stages expand and energy is extracted. The hot gases expand through the turbine, losing heat and pressure before exiting at point 4. The now cooler combustion products are channeled through the heat recovery section where heat energy is extracted to be used by Kalina cycle. The exhaust gases pass through the super heater, evaporator, and economizer in the heat recovery vapour generator section before it exits to the atmosphere at a much lower rate. The working fluid with concentration x enters the heat recovery section at state 17 and absorbs heat energy. At the exit of the superheat (state 20), the superheated vapour enters the turbine where it impinges on the turbine blades transmitting energy from the hot, high-pressure vapour as it expands into rotating mechanical energy. The vapour exits the turbine at state 21 before entering the low-pressure heat exchanger. At state 21, the working fluid is in vapour state and still has a high amount of heat energy. This is extracted in a low-pressure regenerator before it enters the mixer chamber at state 22 where it is mixed with a weak solution of the working fluid (low mass fraction of ammonia). The weak solution exits from the separator at state 9 and passes through economizer 1 where is releases some of its heat and the it enters a throttle at state 10 to reduce the pressure before entering the mixing chamber at state 11. The mixing chamber helps to overcome the difficulty of condensation at the condenser. The difficulty arises because at the exit of the turbine, the concentration of ammonia is still too high and as such would not condense at the absorber. It is therefore mixed in the mixing chamber to reduce the mass fraction of ammonia in the vapour before entering the absorber at state 23.

At the absorber, water at ambient temperature serves as the heat sink and absorbs more heat from the working fluid. Since the mass fraction of ammonia in the working fluid is low, the fluid condenses to liquid before it is extracted by a condensate feed pump (CFP) at state 24. The CFP increases the pressure to the cycle intermediate pressure. At the exit of the condensate feed pump at state 25, the fluid split into two; 26 and 27. Working fluid at state 27 is then passed through the economizer 1 back through the low-pressure regenerator at state 28 to gain heat from the LP regenerator before it enters the separator at state 8. The second stream enters the mixer at state 26 mixes with the vapour exiting the separator at state 13 before it enters the condenser at state 14. At the condenser exit (state 15), the working fluid enters a pressure booster pump (PBP) at state 15 where the pressure of working fluid is increased further to a higher pressure at 16 before it passes through the economizer 2 and enters the high temperature economizer 3 (HT.EC3), then it is passed through the evaporator and super heater which converts the working fluid into a high pressure and high temperature vapour at state 20. At state 9, the solution is a weak solution with a low mass fraction of ammonia. This weak solution is pass through economizer 1 to loss some of its temperature then its pressure is reduced in a throttling device at state 11 where it mixes with the high temperature vapour exiting the turbine to reduce the ammonia concentration. Figure 3 provides a schematic showing a modified gas turbine with high temperature Kalina cycle.

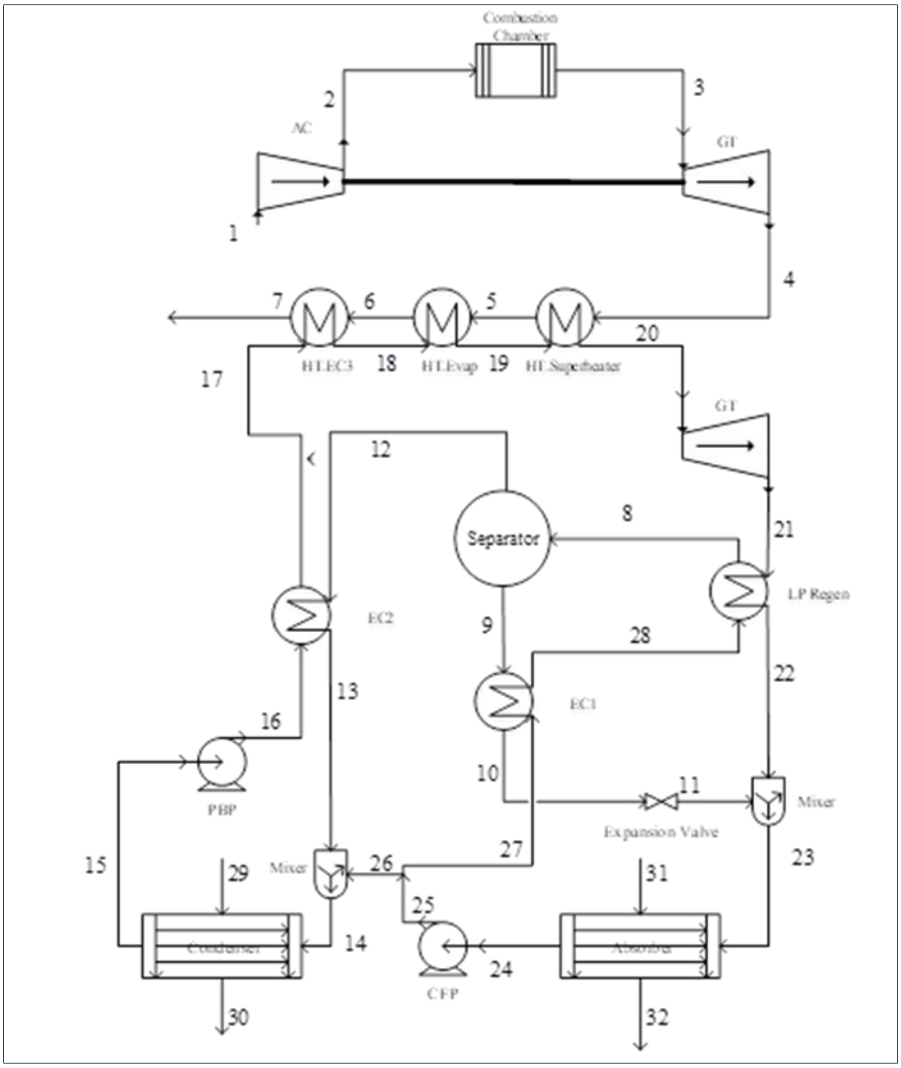

Figure 2 Gas turbine cycle with high temperature Kalina Cycle (GT-KC)

The temperature at the exhaust gases at the gas turbine operating in an open cycle is sufficient to provide heat for multiple bottom cycles. In our model, an air bottoming cycle (ABC), Kalina cycle (KC) and two organic Rankine cycles(ORC). The temperature of exhaust gases directly supplies heat to the ABC and an ORC (1) at the downstream of the ABC. The temperature at the exhaust of the ABC is sufficient to provide heat for a KC and an ORC at the downstream of the KC. A combination of bottom cycles will lead to an increase to the efficiency of the modified cycle and ensure the heat released to the atmosphere is very minimal. Figure 4 shows a schematic description of the modified Gas turbine cycle (GT-ABC-KC-ORC).

Combustion products from the gas turbine exhaust with high temperature enters the heat exchanger at state 4. Heat is transferred to the high-pressure air leaving the second stage compressor of the ABC (state 10). The compressed air in the air bottoming cycle exits the heat exchanged after heat is transferred to it at state 11. The high temperature and high pressure air is channeled into the turbine and energy is extracted from it. After work is extracted from the hot and high-pressure air, it exits the turbine section at a lower temperature and atmospheric pressure. The temperature at the exhaust of the air bottoming cycle is still high enough to produce work. It is then channeled through an evaporator and super heater where further heat is extracted to be used in a Kalina cycle and exits the evaporator at state 14. The vapour from Kalina cycle exits the separator at state 25 before flowing into the super heater to be super-heater. It exits the super heater at state 16 entering the Kalina cycle turbine for work to be done. The exit of the Kalina turbine has low-pressure vapour enters a mixing chamber at state 17 to be diluted by the weak solution exiting the separator at state 26. The weak solution exiting the separator is high in temperature and pressure. It enters the high temperature heat exchanger at state 26, then flows through the high pressure heat exchanger (HP.Hx) before entering a throttle valve at state 27 to reduce the pressure before entering the mixer at state 28. The diluted working fluid exits the mixer at state 18 before it enters low pressure heat exchanger. At state 19, the diluted mixture enters the condenser where heat is extracted from it by ambient cooling water before flowing into the BFP (boiler feed pump) at state 20. The fluid condenses to liquid in the condenser then gets pumped to increase the pressure from state 20 to 21. The mixture is then passed through the low-pressure, high-pressure heat exchangers and an evaporator before it enters the separator at state 24. The working fluid at this state point is known as a strong solution.

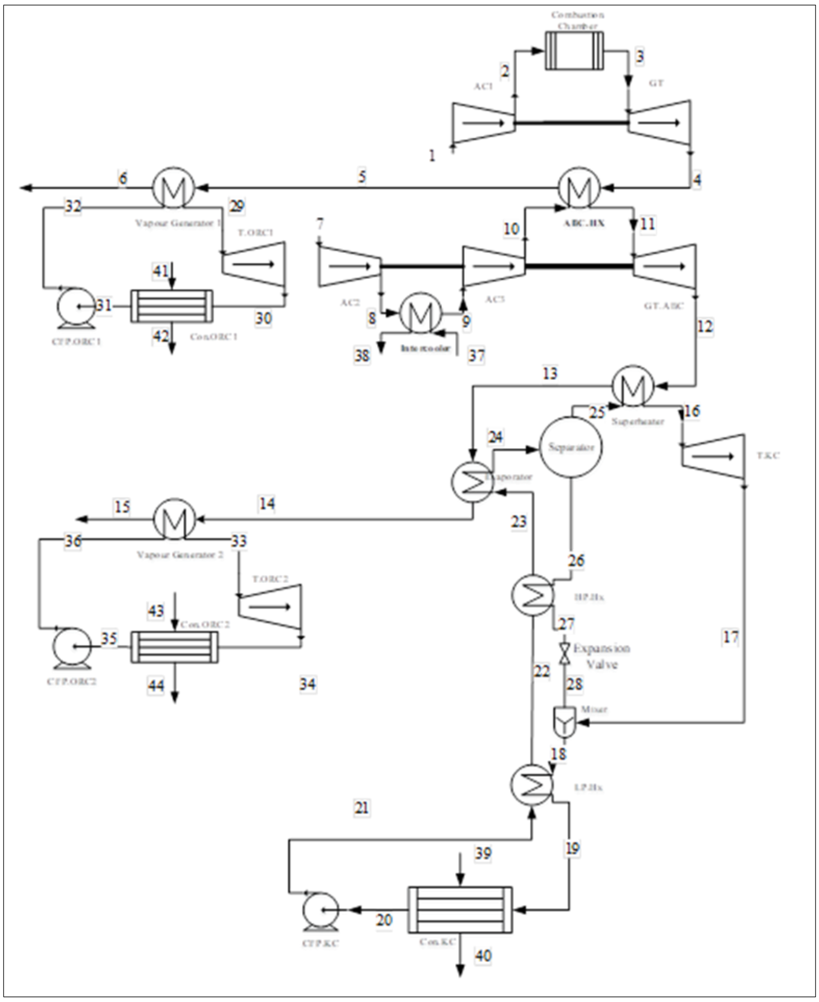

Figure 3 Gas turbine cycle with air bottoming cycle, Kalina cycle and organic Rankine cycle (GT-ABC-KC-ORC)

At the exit of the evaporator (state 14), the air still has sufficient heat for recovery in another bottom cycle. To extract this heat, an Organic Rankine Cycle with a working fluid (R134a) that have a saturation temperature around the temperature at the hot stream at the entry heat recovery exchanger. The hot air flows through vapour generator 2 and exits at state 15 in a much lower temperature. At vapour generator 2, high pressure refrigerant (R134a) enters at state 36 and gains heat before it flows into the ORC2 turbine for work. At state 34, the refrigerant exits the turbine and flows into a condenser (Con.ORC2) where it is cooled. At the exit of the condenser, the fluid is pumped to increase its pressure from state 35 to 36 before entering the vapour generator 2.

At state 5, the temperature is still sufficient for further extraction of heat. To recover this heat, the high temperature gas is directed through vapour generator 1 to transfer the heat into the high-pressure working fluid in an organic Rankine cycle. At vapour generator 1, high pressure refrigerant (R123) enters at state 32 and gains heat before it flows into the ORC1 turbine for work. At the state 30, the refrigerant exits the turbine and flows into a condenser (Con.ORC2) where is cooled. At the exit of the condenser, the fluid is pumped to increase its pressure from state 31 to 32 before entering the vapour generator 1.

It is important to state the assumptions used in carrying out the model for both top and bottom cycles in this configuration. The assumptions in modelling this modified cycle are as follows:

- The system will operate in steady state and flow.
- Combustion of fuel in the combustion chamber is 100% complete.
- Kinetic and potential energy losses are neglected.
- Fuel and air are ideal gases.
- The percentage composition of air is 79% N_2 and 21% O_2.
- Pressure drop at the combustion chamber is 5%.
- Air compressors and gas turbines operate in adiabatic conditions.
- Effectiveness of Heat Exchangers is 0.9.
- Pinch point for Heat Exchanger is 10K.
- Saturated liquid is at exit of the condenser.
- The throttling process is assumed to be isenthalpic.
- Outlets of the Separator are saturated liquid and vapour.
- Separator completely separates liquid and vapour.
- All devices are adiabatic.
- Kinetic and potential energy changes are neglected.

The energy balance equation is applied on each component of the modified gas turbine. Energy balance equation is expressed as Cengel and Boles [27];

$$\dot{E}(kJ) = \dot{Q}_{in} + \dot{W}_{in} + \sum{}_{in} \dot{m}(h + \frac{v^2}{2} + gz)_{in} = \dot{Q}_{out} + \dot{W}_{out} + \sum{}_{out} \dot{m}(h + \frac{v^2}{2} + gz)_{out} \text{..........(1)}$$

Where $\dot{E}(kJ)$ is the total energy, $\dot{W}$ is the work done, $\dot{Q}$ is rate of heat transfer, h is specific enthalpy, v is velocity and gz is component comprising gravity and height.

The operating parameters at different state points are modeled using Equation (2) and (3). The equations are the expressions relating to the temperature and pressures of the compressor and turbine in a Brayton cycle [28];

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \text{..........} \qquad (2)$$

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} \text{..........} \qquad (3)$$

Where P and T represents pressure and temperature at different points in the Brayton cycle, γ is the ratio of specific heat capacities.

Equation (4) represents the change in pressure that occurs within the combustion chamber.

$$P_e = P_i(1 - \Delta P_{cc}) \text{..........} \qquad (4)$$

Where P_e and P_i are the pressure at the exit and inlet of the combustion chamber, ΔP_{cc} is percentage pressure drop at the combustion chamber.

All components are assumed to be operating in steady state as such the steady state equations apply. Kinetic and potential energy variations are assumed to be insignificant. The energy balance equation is expressed in Equation (5)

$$\sum\dot{Q} + \sum\dot{m}_i h_i = \sum\dot{W} + \sum\dot{m}_o h_o \text{..........(5)}$$

Energy balance equation is applied to each component of the gas turbine cycle. Equation (6) is the expression for compressor work.

$$\dot{W}_{ac} = \dot{m}_a(h_2 - h_1) = \dot{m}_a c_{p,a}(T_2 - T_1) \quad \text{..............}(6)$$

Heat addition to the combustion chamber is represented by $\dot{Q}_{cc}$ in Equation (7).

$$\dot{Q}_{cc} = \dot{m}_f H_f \eta_{cc} = \dot{m}_{fg} h_3 - \dot{m}_a h_2 \text{..........}(7)$$

Where H_f is lower calorific value of fuel used by the gas turbine, η_{cc} is the combustion efficiency. Mass flow rate of the flue gas is made up of fuel and air.

$$\dot{m}_{fg} = \dot{m}_f + \dot{m}_g \text{..........} \qquad (8)$$

Turbine work is $\dot{W}_T$ and expressed in the Equation (3.28)

$$\dot{W}_T = \dot{m}_{fg}(h_3 - h_4) = \dot{m}_{fg} c_{p,g}(T_3 - T_4) \text{..........} \qquad (9)$$

Equation (10) expresses the effectiveness of ABC heat exchanger.

$$\epsilon_{hx} = \frac{q}{q_{max}} \text{..........} \qquad (10)$$

For a heat capacitance of the hot fluid more than the cold fluid in a counter flow heat exchanger, the pinch point would be located at the entry of the hot fluid and the exit of the cold fluid [29].

$$T_{e,c} = T_{i,h} - \Delta T_{pinch} \text{..........} (11)$$

Where $T_{e,c}$ represents hot fluid temperature exiting the heat exchanger while $T_{i,h}$ represents inlet temperature of hot fluid. ΔT_{pinch}represents the pinch point temperature in the heat exchanger.

The heat recovery exchangers in this cycle are split in two; evaporator and super heater. The evaporator heats the mixture before entry into the separator while the super heater heats the vapour at the exit of the separator. Equation (12) and (13) expresses the heat recovered in the process.

$$\dot{Q}_{total} = \dot{Q}_{eve} + \dot{Q}_{super} \text{..........} \qquad (12)$$

Where $\dot{Q}_{eve}$ is heat extracted in evaporator, $\dot{Q}_{super}$ is heat extracted at the super heater and $\dot{Q}_{total}$is the total heat extracted in the process.

Mass balance in the separation process for the model is given by equation (13) to (15):

$$\dot{m}_1 = \dot{m}_2 + \dot{m}_3 \text{..........} \qquad (13)$$

$$\dot{m}_1 x_1 = \dot{m}_2 x_2 + \dot{m}_3 x_3 \text{..........} \qquad (14)$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 \text{..........} \qquad (15)$$

Lever rule is used to determine the dryness and wetness fraction. Where $\dot{m}_1$, $\dot{m}_2$ and $\dot{m}_3$ represents the mass flow rates entering and exiting the separator.

The dryness fraction is derived by substituting $\dot{m}_{25}$ in equation (13). Equation (16) and (17) applies the lever rule to solve for the dryness fraction DF.

$$\dot{m}_1(x_1 - x_2) = \dot{m}_3(x_3 - x_2) \text{..........}(16)$$

$$DF_b = \frac{\dot{m}_3}{\dot{m}_1} = \frac{x_1 - x_2}{x_3 - x_2} \text{..........} \qquad (17)$$

Where $\dot{m}_i$ is the mass flow rate at states *i*, h_i is the enthalpy at states *i*.

Equations (18) is used to calculate the pump work.

$$W_p = v_{20}(p_o - p_i)\text{...........} \quad (18)$$

$$\epsilon_{vap.gen} = \frac{T_{hi}-T_{ho}}{T_{hi}-T_{ci}}\text{...........} \quad (19)$$

Where $\epsilon_{vap.gen}$ represents the effectiveness of the vapour generator in ORC, T_{hi}, is the temperature of the hot stream at the inlet of vapour generator and T_{ci} is the cold stream into the vapour generator, while T_{ho} is temperature of hot stream at the exit of the vapour generator.

The work done by the turbine in the organic Rankine cycles downstream of the gas turbine and the Kalina bottoming cycle is expressed by the Equation (23)

$$W_t = \dot{m}\,(h_i - h_o) * \eta_t\text{...........} \quad (20)$$

Where $W_{t.orc\ i}$ is the turbine work in the ORC, *m* represent mass flowrate and *h* represent the enthalpy at different points *i* in the cycle. η_t represents the efficiency of the turbine for ORC.

The feed pump exit condition can be calculated by using equation for isentropic efficiency.

$$h_j = h_i - \eta_{cfp}\left(h_i - h'_j\right)\text{...........} \quad (21)$$

LCOE model is applied in this economic analysis and expressed in equation (22) [30];

$$LCOE = \frac{TLCC}{\left\{\sum\left[\frac{Qn}{(1+d)^n}\right]_{n=1}^{N}\right\}}\text{...........} \quad (22)$$

Where *LCOE* represents levelized cost of energy. Q_n represents output energy, *TLCC* is total life cycle cost, discount rate is *d* and N is the analysis period.

Equation (23) calculates the total life cycle cost and (24) the cost incurred within a period respectively.

$$TLCC = \sum_{n=1}^{N}\left[\frac{C_n}{(1+d)^n}\right]\text{...........}(23)$$

$$C_n = I_n + O\&M + F_n\text{...........}(24)$$

Where C_n is the cost within the period which includes the investment cost I_n, fuel cost F_n, *O&M* is operation and maintenance cost and other costs associated with the project.

Purchase equipment costs used to calculate for both GT and ABC are expressed by equations (25) to (28) [31], [32], [33], [34];

$$PEC_{AC} = 44.71 m_a r_{p,AC} ln\left(r_{p,AC}\right) * \frac{1}{0.95-\eta_{AC}}\text{...........}(25)$$

$$PEC_{CC} = \frac{28.98 m_a}{0.995-\frac{p_{out}}{p_{in}}} * \left(1 + e^{0.015(T_{out}-1540)}\right)\text{...........}(26)$$

$$PEC_{GT} = \frac{479.34 * m_{fg}}{0.93-\eta_{GT}} ln\, ln\left(r_{p,GT}\right) * (1 + e^{0.036T_{in}-54.4}) \quad \text{...........}(27)$$

$$PEC_{HX1} = 4122\left(\frac{\dot{m}_{fg}c_{p,i}(T_i-T_o)}{18\Delta T_{LM,HE}}\right)^{0.6}\text{...........} \quad (28)$$

Equation (39) estimates the heat recovery exchanger cost in air bottoming cycle and vapour generators.

Cost functions for the Kalina cycle equipment are defined by equations (29) to (33) [35], [36].

$$PEC_{KT} = 4405.W_t^{0.7} \text{..........}(29)$$

$$PEC_{KP} = 1120.W_p^{0.8} \text{..........}(30)$$

$$PEC_{HX-Kalina} = 2143.A_{HX}^{0.514} \qquad \text{..........}(31)$$

$$PEC_{Cd} = 1397.A_{,cd}^{0.89} \text{..........}(32)$$

$$PEC_{Sep} = 280.3.m_{sep}^{0.67} \text{..........}(33)$$

Where equation (32) is estimates the cost of condenser and recuperator (assuming they are shell and tube types).

Logarithmic Mean Temperature Difference method (LMTD) can be used to estimate the Area of the heat exchanger. The LMTD is expressed by equations (34)

$$Q = UA\Delta T_{lm} \text{..........} \qquad (34)$$

Q is the heat exchanged in the heat exchanger, U represents heat transfer coefficient and ΔT_{lm} is the LMTD which is further expressed in equation (35);

$$\Delta T_{lm} = \frac{(T_{h1}-T_{c2})-(T_{h2}-T_{c1})}{ln\left(\frac{T_{h1}-T_{c2}}{T_{h2}-T_{c1}}\right)} \text{..........}(35)$$

Subscripts h represent hot and c cold while 1 and 2 represent in and out respectively.

To estimate the heat exchanger areas in the Kalina Cycle, data not yet available would be required to estimate the value of U. To overcome this, U is assumed as 1.0 KW/m^{2}K to estimate the values of the recuperator. While 0.9 and 1.1 KW/m^{2}K are estimated for the cost of the vaporizer, condenser and absorber.

The specific fuel consumption for the overall cycle is determined by equation (36) [37]

$$SFC = \frac{\dot{m}_f}{\dot{W}_{overall}} \text{..........}(36)$$

Annual gas turbine fuel consumption can be calculated by application of equation (37)

$$Annual\ Fuel\ Consumed = \text{SFC} * E_{Annual} \qquad \text{........}(37)$$

Where Annual fuel consumed is calculate from the E_{Annual} representing the energy generated by the gas turbine model and SFC specific fuel consumed. To find the annual energy generated E_{annual}, equation (38) is applied.

$$E_{Annual} = 8760 * \dot{W}_{overall} \text{..........}(38)$$

3. Results and Discussions

Operating data obtained from an operating Alstom GT13E2 gas turbine was obtained to develop a gas turbine model. The Table 1 shows Iso-operating parameters for the existing simple cycle gas turbine power plant being studied (GT). Two modified gas turbine models Gas Turbine with Kalina Bottoming Cycle (GT-KC) and Gas turbine with Air Bottoming Cycle, Kalina cycle and Organic Rankine Cycle (GT-ABC-KC-ORC) were developed from the existing gas turbine model GT. Thermodynamic and economic analysis was carried out on both the simple cycle gas turbine model and modified gas turbine cycles to determine the most efficient and economical using the methodologies earlier discussed in this study. To measure the effectiveness of the proposed gas turbine models some parameters would need to take and compared with the existing simple cycle gas turbine GT. The parameters are efficiency, overall work output, purchase equipment cost, levelized cost of electricity, specific fuel consumption and annual energy generated. Table 2 to 3 presents the result from thermodynamic analysis carried out on the simple cycle and modified gas turbine models. Some assumptions were required to successfully carry out an economic analysis using the Levelized cost of electricity (LCOE) method. These assumptions are listed in table 4. The results obtained from economic analysis for the different gas turbine models are shown in Figures 5 to 8.

Using the annualized forecasted electricity demand, an estimate of the fuel consumed annually by installed gas turbines in Nigeria was calculated by applying the specific fuel consumption and this was compared with the results if the proposed gas turbine models were implemented a on each gas turbine model developed. This was compared with the simple cycle gas turbine (GT) to show the annual saving in natural gas consumed by each of the models. The raw data for thermodynamic analysis and economic analysis carried out is available in Appendix A and B. Appendix C contains the raw data for specific fuel consumption for each model.

Table 1 Iso-Operating Parameters of the Simple Cycle Gas Turbine

S/N	Description	Symbol	Value	Unit
1	Compressor Work (GT)	W_{AC}	214,218.44	kW
2	Turbine Work (GT)	W_{GT}	381,618.44	kW
3	Heat Added (GT)	Q_{CC}	448,524.32	kW
4	Exhaust Temperature GT	T_4	524.00	^oC
5	Mass flow rate of GT	m_{fg}	532.00	kg/s
6	Network output	W_{net}	167,400.00	kW
7	Efficiency of Gas Turbine Cycle	η_{gt}	37.3%	-

Table 2 Results of thermodynamic analysis of simple cycle gas turbine model

S/N	Description	Symbol	Value	Unit
1.	Compressor Ratio	r_p	14.0	-
2.	Work Output	W_{net}	167400	kW
3.	Heat Rate	H.R.	9780	Btu/kWh
4.	Efficiency	-	36.1%	-
5.	Exhaust temperature (GT)	T_4	524.00	^oC
6.	Exhaust Mass Flow Rate	$\dot{m}_{fg}$	532	kg/s
7.	Ambient Temperature	T_1	288	K
8.	Ambient Pressure	P_1	1.0133	bar
9.	Isentropic Efficiency - Compressor	η_c	0.85	-
10.	Isentropic Efficiency – Turbine	η_t	0.88	-
11.	Specific Heat Capacity – Air	c_{pa}	1.005	kJ/kgK
12.	Specific heat capacity –gas	c_{pg}	1.026	kJ/kgK
13.	Specific heat ratio – Air	γ	1.40	-
14.	Specific heat capacity – flue gas	c_{pa}	1.33	-
15.	Mechanical Efficiency – Turbine	-	0.96	-
16.	Generator Efficiency	-	0.97	-

Table 3 Results of thermodynamic analysis of Gas Turbine with Kalina Cycle (GT-KC)

S/N	Description	Symbol	Value	Unit
1.	Compressor Work	W_{c1}	214,008.87	kW
2.	Turbine Work (GT)	W_{t1}	381,408.87	kW
3.	Heat Added	Q_{in}	449,165.69	kW
4.	Exhaust Temperature GT	T_{gt}	524.00	^{o}C
5.	Condensate Extraction Pump	W_{p1}	345.80	kW
6.	Boiler Feed Pump	W_{p2}	2000.00	kW
7.	Turbine Work (KC)	W_{kc1}	57,039.00	kW
8.	Exhaust temperature KC	T_{kc}	86.37	^{o}C
9.	Network output	W_{net}	222,093.20	kW
10.	Efficiency of Gas Turbine Cycle	η_{gt} $\eta_{overall}$	37%	-
11.	Efficiency of Kalina Cycle	η_{kc}	24%	-
12.	Overall Efficiency of Cycle (Brayton – Kalina Cycle)	$\eta_{overall}$	50%	-

Table 4 Results of thermodynamic Analysis of simple cycle gas turbine with air bottoming cycle, Kalina cycle and organic Rankine cycle (GT-ABC-KC-ORC)

S/N	Description	Symbol	Value	Unit
1	Compressor Work (GT)	W_{c1}	214,218.44	kW
2	Turbine Work (GT)	W_{t1}	381,618.44	kW
3	Heat Added (GT)	Q_{in}	448,524.32	kW
4	Exhaust Temperature GT	$T_{exh\text{-}gt}$	524.00	^{o}C
5	Mass flow rate of GT	$m.gt$	532.00	kg/s
6	Compressor Work 1 (ABC)	W_{bc1}	48,926.00	kW
7	Compressor Work 2 (ABC)	W_{bc1}	49,765.00	kW
8	Turbine Work (ABC)	W_{tc1}	128,222.00	kW
9	mass flow rate of ABC	$m.abc$	473.22	kg/s
10	Exhaust temperature (ABC)	$T_{exh\text{-}abc}$	246.00	^{o}C
11	Pump Work (KC)	W_{p2}	592.60	kW
12	Exhaust Temperature KC	$T_{exh\text{-}kc}$	149.30	^{o}C
13	Turbine Work (KC)	W_{kc1}	13,168.00	kW
14	Exhaust temperature (downstream of ABC)	T	192.60	^{o}C
15	Pump work (ORC1)	$W_{cfp.orc1}$	618.30	kW
16	Pump work (ORC2)	$W_{cfp.orc2}$	616.40	kW
17	Turbine Work (ORC1)	$W_{t.orc1}$	18,810.00	kW
18	Turbine Work (ORC2)	$W_{t.orc2}$	9,771.00	kW

19	Net Work Output	W_{net}	236,852.70	kW
20	Efficiency of Gas Turbine Cycle	η_{gt}	37%	-
21	Efficiency of ABC	η_{abc}	25%	-
22	Efficiency of Kalina Cycle	η_{kc}	27%	-
23	Efficiency of ORC1	η_{orc1}	25%	-
24	Efficiency of ORC2	η_{orc2}	16%	-
25	Overall Efficiency of Cycle (GT-ABC-KC-ORC)	$\eta_{overall}$	53%	-

Table 5 Economic Factors for Power Plant Models

S/N	Description of Economic Factor	Total	Unit
1	Operation and maintenance cost (10%)	10%	-
2	Operation and maintenance growth rate (2%)	2%	-
3	Availability factor	90%	-
4	Project lifespan	25	years
5	Generator Efficiency	97%	-
6	Discount rate	10%	-

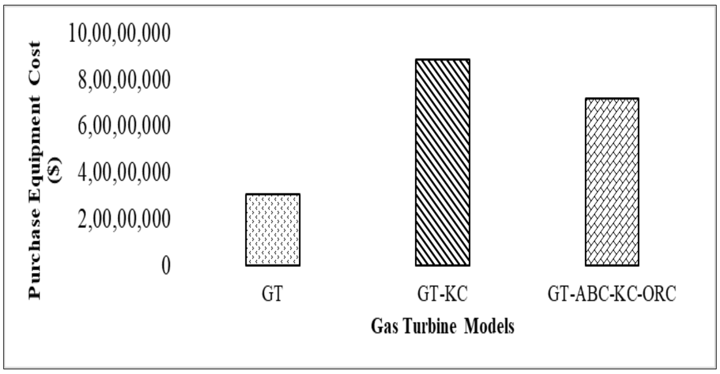

Figure 4 Purchase equipment cost for different gas turbine models

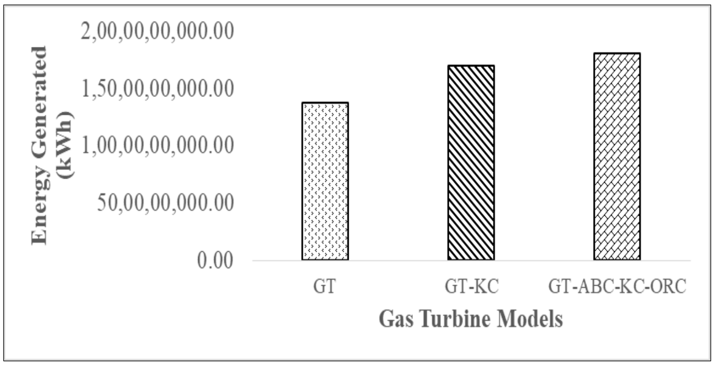

Figure 5 Annual energy generated in MWh for the different gas turbine models

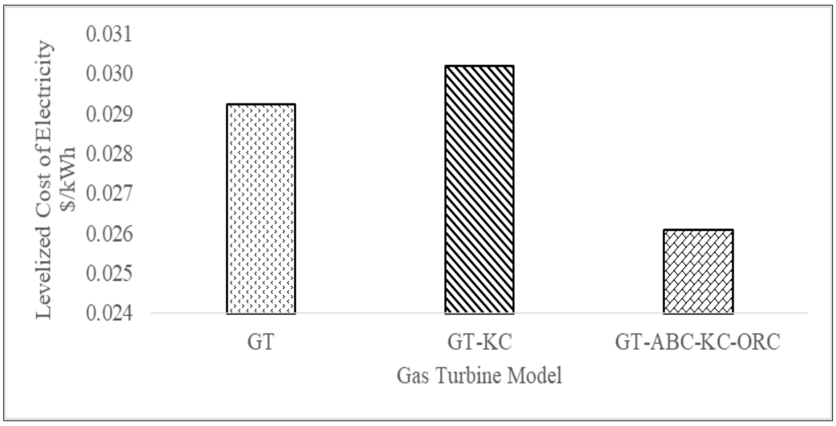

Figure 6 Levelized Cost of Electricity for each gas turbine model

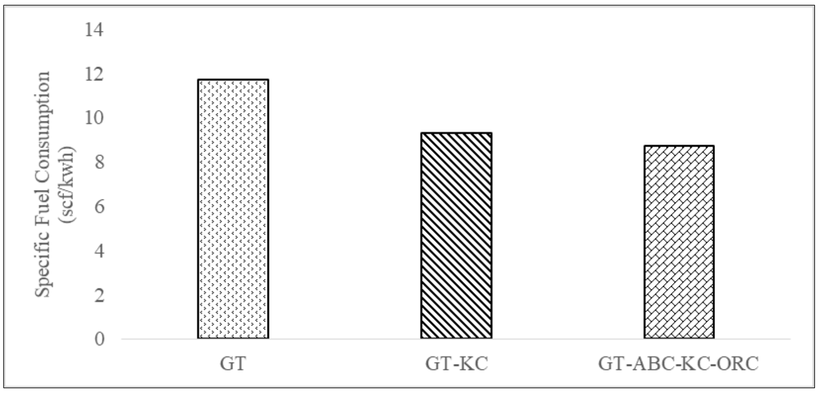

Figure 7 Specific fuel consumption for each gas turbine model

Thermodynamic and economic analysis carried out on the different models yielded a promising result for the two gas turbine models developed. As displayed in tables 1 - 3 which shows the result of the thermodynamic analysis carried out on the simple cycle gas turbine GT and the two modified models GT-KC and GT-ABC-KC-ORC, the efficiency of both modified models are high than the simple cycle gas turbine. Also there is significant drop in the exhaust temperature of the modified models compared to the simple cycle gas turbine model.

Figure 5 shows the result from the economic analysis for the three models studied. GT-KC shows a higher purchasing cost followed by GT-ABC-KC-ORC and finally GT. The purchase equipment cost (PEC) gives an idea of the amount required for manufacturing each of the models. Figure 6 shows the annual energy generated by each of the models. From the histogram, GT-ABC-KC-ORC produced the highest energy annually followed by GT-KC and finally GT. Figure 7 is a plot of the cost of electricity generated by the different gas turbine models in the study. In contrast to the annual energy generated, GT-ABC-KC-ORC generated electricity at the least cost. This is followed by GT and finally GT-KC. For specific fuel consumption of each of the models, Figure 8 shows a comparison between the different gas turbine models on their specific fuel consumption. GT-ABC-KC-ORC used less fuel to generate the same amount of electricity compared to GT-KC and GT.

4. Discussion of Findings

Firstly, a model of an existing simple cycle gas turbine was carried out. Then simple cycle gas turbine was modified with the aim of improving its efficiency. Two different models were developed: A Gas turbine modification with Kalina cycle (GT-KC) and gas turbine modified with air bottoming cycle, Kalina Cycle and Organic Rankine cycle (GT-ABC-KC-ORC). The Kalina cycle in GT-KC is designed for high temperature applications. In addition to improving the gas turbine efficiency, the other advantages of improving the gas turbine efficiency is to meet environmental sustainability goals, producing cheaper electricity etc.

In the GT-KC model, the Kalina cycle extracted heat from high temperature exhaust gas and converted the heat to generate additional work. While the GT-ABC-KC-ORC introduced an ABC to extract heat from the exhaust gas. To further improve the efficiency of the GT-ABC-KC-ORC cycle, a Kalina cycle and an organic Rankine cycle were introduced to extract waste heat from the Air bottoming cycle and the gas turbine.

Results from the thermodynamic analysis carried out on GT-KC revealed that modifying of the gas turbine lead to an increase of 13.9% in efficiency compared to the GT. The GT-KC model resulted in an efficiency of 50% while the GT was 36.3%. The GT-KC model resulted in a power output of 224,439.00KW compared to the initial output from the GT 167,400.00KW. In total, the GT-KC has a 34.1% higher work output than the GT. When considering environmental sustainability, the GT-KC model efficiently converted the waste heat from the GT and reduced the temperature of flue gas from 524^{o}C to 86.37^{o}C. This significant drop in temperature reduces the environmental pollution which associated with gas turbine operation. Economic analysis carried out on the GT-KC using Levelized cost of electricity method showed that electricity was generated at a cost of 0.030$/kWh which is equivalent to 45.75N/kWh using a parallel exchange rate of 1525N/$ as a conversion rate. When compared to the cost of generating electricity in a simple cycle gas turbine which produced electricity at 0.029$/kWh (44.225N/kWh). This indicates a slightly higher cost per kilowatt-hour for the modified GT-KC model. This can be attributed to the cost associated with including a high temperature Kalina cycle. This cost can is shown in figure 4.9. The purchase equipment cost for GT-KC was the highest for the three models. Kalina cycle is composed of a lot of components compared to the simple cycle gas turbine.

The result from the thermodynamic and economic analysis on GT-ABC-KC-ORC showed a much more prospect. The overall efficiency of the GT-ABC-KC-ORC was 53%, representing a 16.9% increase in efficiency when compared with the GT model. The efficiency of the GT-ABC-KC-ORC model was also higher than the GT-KC model. The power output increased by 71,173.90kW representing 42.5% increment to the GT bringing the overall output to 238,573.90kW. Exhaust temperatures for the various micro systems in the configuration were 48.44^{o}C and 46.49^{o}C which are significantly lower than the exhaust of the GT-KC and the GT. Considering environmental sustainability, the GT-ABC-KC-ORC was the model with the least heat lost into the environment as pollution. Economic analysis carried out on the model resulted in electricity generation at a rate of 0.026$/kWh (39.63N/kWh). This shows that the GT-ABC-KC-ORC model produced the cheapest electricity among the three models. Also, the economic analysis result revealed quite interestingly that despite the combination of different bottoming cycles to improve the efficiency of the model, the GT-ABC-KC-ORC model resulted in the least purchasing equipment cost. This can directly be attributed to the modular approach as against the high temperature Kalina Cycle included in GT-KC.

5. Conclusion

The study reached the following conclusions:

- The GT-ABC-KC-ORC configuration achieved a superior overall plant efficiency of 53% and an energy output of 238,573.9 kW, representing a 42.5% increase in energy production. This model also maintained a lower exhaust temperature of 48.44°C and 46.49°C. While both modified gas turbine cycles significantly improved efficiency, the GT-ABC-KC-ORC model demonstrated the highest efficiency.
- An economic analysis revealed that the GT-ABC-KC-ORC model had the lowest cost of electricity generation compared to the GT-KC model, making it a more cost-effective option.
- The purchase equipment cost for the three modified models indicated that the GT-ABC-KC-ORC model would be less expensive to acquire than the GT-KC model. This cost advantage is attributed to the strategic combination of multiple bottoming cycles, carefully selected based on their suitability for waste heat recovery, and the simplicity of the ORC and ABC bottoming cycles, which have fewer components.
- The analysis of purchase equipment costs showed that upgrading a simple cycle gas turbine (GT) to the GT-ABC-KC-ORC configuration would require an additional investment of 131.3% of the GT's cost, whereas upgrading to the GT-KC configuration would require 185.6% of the GT's cost.

In conclusion, our study demonstrates that integrating multiple waste heat recovery (WHR) cycles as bottoming cycles with a gas turbine results in higher overall efficiency compared to utilizing a single bottoming cycle. This multi-cycle approach not only improves the thermal efficiency of the system but also significantly reduces the capital expenditure associated with purchasing equipment and the operational costs for generating each unit of electricity. Additionally, the GT-ABC-KC-ORC model, which combines gas turbine, air bottoming cycles (ABC), Kalina Cycle (KC), and Organic Rankine Cycle (ORC), exhibited the lowest exhaust temperature, thus minimizing thermal pollution and environmental impact.

The findings suggest that the strategic combination of different WHR technologies can enhance the economic and environmental performance of power generation plants. However, further research is recommended to explore various configurations and combinations of bottoming cycles. Future studies should focus on optimizing these systems for different operational conditions and heat sources, potentially involving advanced simulations and experimental validations. Additionally, investigating the long-term reliability, maintenance requirements, and lifecycle costs of these combined systems will be crucial for developing cost-effective and sustainable solutions for waste heat recovery in power generation. This research could also explore the integration of renewable energy sources with WHR systems to further enhance efficiency and reduce greenhouse gas emissions.

Compliance with ethical standards

Disclosure of conflict of interest

No conflict of interest to be disclosed.

Statement of informed consent

Informed consent was obtained from all individual participants included in the study.

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