Performance evaluation of a combined cycle power plant integrated with organic Rankine cycle and absorption refrigeration system

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Cogent Engineering (2018), 5: 1451426
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Abstract: This study presents the performance evaluation of a combined gas- and steam- turbine cycle power plant (CCPP) integrated with organic Rankine cycle (ORC) and absorption refrigeration cycle (ARC). The attached ORC and ARC units are powered with the flue gas exhaust heat from the CCPP. The evaluation was conducted by performing energy, exergy and environmental sustainability index analysis of the integrated power plant (IPP) and its components. Based on the operating data of an existing CCPP operating in the tropical rain forest region of Nigeria, results of the analysis showed that by utilizing exhaust heat of the CCPP to power an ORC, using R113 as the working fluid, extra 7.5 MW of electricity was generated and by powering an ARC to cool inlet air streams to 15°C in the gas turbine plants, additional 51.1 MW of electricity was generated. The overall effect of integrating ORC and ARC to the CCPP showed that the net power output of the integrated power plant was increased by 9.1%, thermal and exergy efficiencies by 8.7% and 8.8%, respectively, while the total exergy destruction rate and specific fuel consumption reduced by 13.3% and 8.4% respectively. Sustainability index increased by 8.4% which means that the integrated plant has greater environmental sustainability potential over the combined cycle plant.

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PUBLIC INTEREST STATEMENT

With the increasing global demand for electric power, it is imperative to sustain efforts at improving power generation technologies with high energy conversion efficiencies. The combined cycle power plant (CCPP), though, an improvement over the single cycle power plant could be discharging usable low grade waste heat. The exhaust heat from existing power plants could be used to drive energy production units such as organic Rankine cycle (ORC) for electricity and absorption refrigeration cycle (ARC) for cooling. In this study, the exhaust heat from an active CCPP was used to drive an ORC and an ARC. Extra 7.5 MW of electricity was generated from the ORC while the ARC was used to cool the inlet air streams to the compressor of CCPP. By cooling inlet air to 15°C, extra 51.0 MW of electricity was generated from the CCPP. The overall plant efficiency improved while the rate of heat release into atmosphere decreased.
1. Introduction

Global warming and fossil fuel depletion issues have escalated the need and importance of integrating several energy production units to utilize a common primary energy input. In thermal power plants like gas turbine plant, fossil fuels are used as the primary energy input to generate electricity while the exhaust flue gases can be utilized to drive other thermal cycles like steam Rankine cycle, organic Rankine cycle, absorption refrigeration cycle, desalination cycle, heating units etc. Integrated multi energy generation is an emerging area with growing interest.

Oko and Njoku (2017) presented the performance analysis of an integrated gas-, steam- and organic fluid-cycle thermal power plant (IPP). The integrated power plant is based on the utilization of exhaust waste heat from an existing 650 MW combined (gas and steam) power plant (CCPP) to power an organic Rankine cycle (ORC) unit. The results showed that an extra 12.4 MW of electric power is generated from the waste heat fired ORC unit. The exergy and energy efficiencies of the IPP improved over that of the existing CCPP by 1.95 and 1.93%, respectively.

Mohan, Dahal, Kumar, Martin, and Kayal (2014) developed and analyzed an integrated energy system based on the utilization of exhaust waste heat from an existing natural gas fired turbine power plant for the simultaneous production of electricity from a steam Rankine cycle; clean water from a desalination cycle and cooling from an absorption refrigeration cycle. The results of the techno-economic analysis showed that for the integrated trigeneration system, energy efficiency was about 85%, normalized CO₂ emission per MWh was reduced by 51.5% (compared to the existing plant) and cumulative net present value of $66 million over the project life time and a payback period of 1.38 years were obtained. The simulation results also showed that the performance of the thermal cycles were affected by the ambient air intake temperature.

Ahmadi, Dincer, and Rosen (2013) conducted energy, exergy, exergoeconomics, exergoenvironmental and multi-objective optimization of an integrated multigeneration energy system based on the utilization of exhaust waste heat from a natural gas fired turbine power plant to drive a steam turbine cycle, an absorption chiller, an organic Rankine cycle / ejector refrigeration cycle, a domestic heater and a proton exchange membrane electrolyzer, for the simultaneous production of power, heating, cooling, hot water and hydrogen. Ahmadi, Rosen, and Dincer (2012) carried out exergy-based optimization of an integrated multigeneration energy system comprised of a gas turbine cycle as the prime mover, a steam turbine cycle, and an absorption refrigeration cycle and water heater. The authors applied a multi-objective evolutionary based optimization to find the best design parameters of the system considering exergy efficiency and total cost of the system as two objective functions.

Khaliq (2009) conducted exergy analysis of a gas turbine based trigeneration system for the simultaneous production of power, process steam for heating and cooling in an integrated gas turbine,-heat recovery steam generator and absorption refrigeration system. Performance of integrated system is simulated for different turbine inlet temperatures and pressure ratios, process heat pressures and evaporator temperatures of absorption refrigeration system. Ghaebi, Amidpour, Karimkashi, and Rezayan (2011) performed detailed thermodynamic analysis of a similar gas turbine based trigeneration system and further conducted thermo economic analysis to determine the cost of the system.

Inlet air cooling has been widely studied and established as means of improving the performance of gas turbine power plants (in both simple and combined cycle configurations) operating in arid and...
tropical weather conditions (Mohanty & Paloso, 1995). Jonsson and Yan (2005) concluded that using chillers for inlet air cooling can increase gas turbine power output by 15–20% and efficiency by 1–2% (i.e. if gas turbine exhaust gas thermal energy is recovered). Dawoud, Zurigat, and Bortmany (2005) compared inlet air cooling technologies with respect to their effectiveness in power boosting of small-size gas-turbine power plants used in two locations at Marmul and Fahud in Oman. Their findings show that fogging cooling is accompanied with 11.4% more power output in comparison with evaporative cooling in both locations, while lithium bromide-water absorption cooling offers 40% and 55% more power output than fogging cooling at Fahud and Marmul, respectively. Ameri and Hejazi (2004) in their study reported an improvement of 11.3% in gas turbine power output by cooling the intake air with an absorption refrigeration system. Boonnasa, Namprakai, and Muangnapoh (2006) reported that the addition of absorption chiller to a combined cycle power plant for inlet air cooling, could increase the power output of a gas turbine (GT) by about 10.6% and the combined cycle power plant by around 6.24% annually. Ehyaei, Hakimzadeh, Enadi, and Ahmadi (2012) studied the effect of using absorption chiller for inlet air cooling in gas turbine power plants for two regions in Iran, namely Tabas with hot–dry and Bushehr with hot–humid climate conditions. The net power output, first and second law efficiencies, environmental and electrical costs for GT power plant with inlet air cooler are calculated for the two mentioned regions, respectively. Results showed that using absorption chiller increased the output power by 11.5 and 10.3%, the second law efficiency is increased to 22.9 and 29.4% and cost of electricity production decreased by about 5.04 and 2.97%, for Tabas and Bushehr cities, respectively. Mohapatra (2014) compared the impact of integrating vapor compression and vapor absorption cooling system to a combined cycle power plant (CCPP) for inlet air cooling. Their study showed that the vapor compression inlet air cooling improved the CCPP specific power output by 9.02% compared to 6.09% obtained with the vapor absorption cooling. However to operate the vapor compression system, power is extracted from the gas turbine output.

For combined cycle power plant operating with air cooled condenser, ambient air conditions has direct impact on the performance of the gas turbine cycle and the steam turbine cycle respectively. A drawback of air-cooled condensers (ACC) is that their performance can decline as ambient temperatures increase and result in loss of steam turbine power output. Increased ambient temperature reduces the heat transfer (heat rejection) rate during steam condensation leading to rise in turbine back pressure. As the turbine back pressure increases, the output of the steam turbine decreases (Ramani, Rupeshkumar, Amitesh Paul, Anjana, & Saparia, 2011). Since air cooled condensers operate with the ambient dry bulb temperature as the theoretical minimum attainable temperature, their efficiency can drop by about 10% when ambient temperatures rise (Gadhamshetty, Nirmalakhandan, Myint, & Ricketts, 2006; Nirmalakhandan, Gadhamshetty, & Mummaneni, 2008).

Chuang and Sue (2005) presented the results of a performance test conducted on an active CCPP with air-cooled condenser operating in Taiwan. The results shows that the CCPP can produce more power output when operating at a lower ambient temperature (or lower condenser pressure) and for each 1°C drop in ambient temperature, the power output of this CCPP increased by 0.6% and efficiency improved by 0.1%.

In recent times, a lot of studies has been conducted on the utilization of exhaust gas heat from thermal power plants to power organic Rankine cycle plants for further power generation. Ahmadi, Dincer, and Rosen (2012) simulated a trigeneration system where the exhaust gas heat of a gas turbine plant is used to power an organic Rankine cycle (ORC), a single-effect absorption chiller and a domestic water heater for generation of electricity, heating and cooling. The results showed that the exergy efficiency of the trigeneration system is higher than that of a typical combined heat and power cycle or a gas turbine cycle. Also the rate of CO₂ emission was found to be comparatively lower. Guo, Du, Yang, and Yang (2015) studied the effects of three types of working fluids on the performance of organic Rankine cycle (ORC) powered by the exhaust gas heat of a boiler in a 240 MW coal-fired power plant. The temperature of the exhaust flue gas is given to be between 120 and 130°C. From the result of the analysis, there exists no optimal working fluid among the studied fluids for all performance indicators (thermal efficiency, heat exchanger area, mass flow and volumetric...
flow etc.). An appropriate working fluid should be chosen by taking both investment cost and power generating benefits into account. The evaluation of the costs/benefits ratio indicated that the payback period of the proposed ORC plant was within 4.5 years.

In practice, the basic subcritical ORC configuration (without regeneration), in which saturated or slightly superheated vapor is expanded in a turbine, is often used for waste heat to power applications (Le, Kheiri, Feidt, & Pelloux-Prayer, 2014). Although introduction of regeneration increases the cycle efficiency of ORC, this is not justified in waste heat to power applications, for which the power output should be maximized instead of the cycle efficiency (Dai, Wang, & Gao, 2009; Quoilin, Declaye, Tchanche, & Lemort, 2011). Studies show that isentropic organic fluids are most suitable for low temperature waste heat applications (Dai et al., 2009) in order to avoid liquid droplet impingement in the turbine blades during the expansion process (Chen, Yogi Goswami, & Stefanakos, 2010).

In this study, the performance evaluation of an integrated power plant with three different power cycles (gas turbine cycle, air cooled steam turbine cycle and organic Rankine cycle) and a waste heat driven absorption refrigeration unit is conducted. The proposed integrated power plant is modeled by attaching an organic Rankine cycle plant (ORC) and absorption refrigeration unit (ARC) to an existing combined gas and steam cycle power plant operating in the tropical rain forest region of Nigeria. The main objectives of the paper are:

- To perform energy, exergy and environmental sustainability analysis of the integrated system comprising gas turbine plants, steam turbine plant, organic Rankine cycle and absorption refrigeration unit.
- To investigate the performance effects of cooling the inlet air streams of the gas turbine plants using the waste heat driven absorption refrigeration unit.
- To evaluate the combined performance effects of integrating the ORC and ARC to the CCPP.

The results and simulations were generated using the MATLAB and Engineering Equation (EES) software.

2. Problem formulation and solution methods
The proposed integrated power plant configuration and operating description are presented in this section. The mathematical equations for the performance analysis are also presented.

2.1 System description
The base case in this study is an existing 650 MW combined cycle power plant with three natural gas fired gas turbine (GT) units; three dual pressure, forced circulation heat recovery steam generators (HRSG); a dual pressure steam turbine (ST) unit; air cooled condenser unit and two feed water pumps—high and low pressures.

Figure 1 illustrates the proposed inlet air cooled integrated power plant. An organic Rankine cycle power plant (ORC) and an absorption refrigeration unit (ARC) are integrated to the existing combined cycle power plant (CCPP). The ARC system used to cool the inlet air streams to the organic Rankine cycle, the gas turbine cycle, and the steam turbine cycle. The ORC and the ARC are powered with the exhaust gas heat from the CCPP.

In the combined cycle power plant, inlet air at the ambient temperature (state 1) is compressed by the air compressor (AC) to state 2 before entering the combustion chamber (CC) where it mixes with the natural gas from the fuel supply system to produce hot flue gases, which exit the CC and enters the gas turbine (GT). The flue gases expand in the GT from state 3 to state 4, producing power for driving the compressor and for conversion into electricity. The exhaust flue gases at state 4 pass through the heat recovery steam generator (HRSG) where high and low pressure feed water streams are heated to states 7 and 8, respectively, as the flue gases exit the HPHRSG and LPHRSG at states 5
and 6 respectively. The superheated steam from the HPHRSG at state 7 expands in the high pressure turbine (HPST) to state 8, and mixes with the superheated steam from the LPHRSG at state 9 to form a homogenous steam mixture at state 10 before expanding in the low pressure steam turbine (LPST) to state 11; The mechanical power from the steam turbines is converted into electrical power in the electrical generator 2 (el.Gen.2). The exit wet steam is condensed in the steam condenser (SC) to saturated liquid water at state 12 before pumped by the low pressure feed water pump (LPFWP) to state 13. One part of this low pressure feed water ($\alpha m$) is fed into the LPHRSG, while the remaining part ((1-$\alpha$)m) is pumped by the high pressure water pump (HPFWP) to state 14 and fed into the HPRSG for the dual-pressure steam turbine cyclic process to continue repeating.

Flue gases leaving the LPHRSG at state 6 flow through an organic Rankine cycle power plant (ORC). Here, an appropriately chosen organic fluid is evaporated in the organic liquid evaporator (OLE) to a saturated vapor at state 15 by the exhaust flue gases. The organic vapor is expanded in the organic vapor turbine (OVT) to state 16 to produce additional electric power in the electric generator (el.Gen.3). The wet organic fluid is condensed in the organic vapor condenser (OVC) to state 17 and subsequently pumped by the organic liquid feed pump (OLFP) to state 18, and fed in to the OLE to continue repeating the cyclic process. The phase diagram of the combined gas and steam turbine power plants are shown as Figure 2 while the phase diagram of the organic Rankine cycle is shown in Figure 3.

At point 19, the exhaust flue gases exiting the OLE units is used to power the LiBr-H$_2$O-absorption refrigeration unit (ARC) and then discharged at state 30. The ARC consists of a generator, regenerator, absorber, air-cooled condenser, and evaporator. Dilute LiBr-H$_2$O solution in the absorber is pumped from state 21 through the regenerator to the generator at 23. In the generator, the diluted solution is heated directly by the exhaust gases. A portion of water is boiled off the LiBr-H$_2$O solution. The water vapor flows to the condenser while the concentrated LiBr-H$_2$O solution at state 24 returns via the regenerator to state 25 and throttle valve (STv) to the absorber at 26. The boiled-off water vapor at 27 is condensed as it passes through the condenser. The resulting two-phase refrigerant is throttled (RTv) to low pressure and temperature to state 29 before entering the evaporator where it...
is evaporated gaining heat from the circulating water. Refrigerant vapor from the evaporator at state 20 flows back to the absorber where, it is absorbed by the concentrated solution from the generator (26). The resulting diluted solution (21) is pumped to the generator to complete the cycle.

Chilled water is circulated from the evaporator (state 31) through an air cooler to cool inlet ambient air (from state O to state 1). During this heat exchange process in the air cooler, the temperature of the inlet air decreases and exits the air cooler at state 1 and is drawn in to the compressors of the gas turbine power plants. This cyclic process continues repeating. The psychrometric processes of the compressor inlet air cooling is shown in figure 4.

2.2 Assumptions
The following assumptions were made in performing the energy and exergy analysis of the gas turbine cycle, the steam turbine cycle, the organic Rankine cycle and the LiBr-H₂O absorption refrigeration cycle:
• Mass and energy flow through the plant is in steady state.
• Changes in kinetic and potential forms of energy are negligibly small.
• All gases (air and flue gases) behave like ideal gases.
• Heat losses and mechanical losses were neglected.
• Air and water enter the system at ambient temperature and pressure.
• The thermodynamic equations are derived for the major thermodynamic devices by treating them as control volumes.
• The mass flow rates and thermodynamic characteristics of the three gas turbine units are identical, so that only one is analyzed and the extensive outputs are multiplied by a factor of 3.
• Solution leaving the absorber and the generator are assumed to be saturated and in equilibrium conditions at their operating temperatures and concentrations.
• The refrigerant states leaving the condenser and evaporator are also assumed to be saturated.

2.3. Thermodynamic analysis and system modelling

The thermodynamic assessment of the integrated gas-, steam-, and organic fluid-cycle turbine power plant. The major units undergo different thermodynamic processes as represented on the T – s diagram shown in Figures 2 and 3.

2.3.1. The gas turbine cycle

The conventional gas turbine plant runs on the Brayton cycle where both the compression and expansion processes take place in rotating machinery. Using Figure 2, the thermodynamic models for the major units are derived as follows: (Ahmadi & Dincer, 2011; Alhazmy & Najjar, 2004; Ibrahim & Rahman, 2012; Kaviri, Jaafar, & Lazim, 2012; Mansouri, Ahmadi, Ganjeh, & Jaafar, 2012; Nag, 2013; Tiwari, Hasan, & Islam, 2013; Zaki, Jassim, & Alhazmy, 2011).

(i) Air compressor

The power required to drive the compressor is given as

\[ W_{AC} = \dot{m}_a (h_2 - h_1) = \dot{m}_a c_p \Delta T \]

where \( \dot{m}_a \) (kg/s) is the mass flow rate of the air through the compressor; \( W_{AC} \) (kW) is the ideal compression power; \( c_p \) (kJ/kg·K) is the isobaric specific heat capacity of the air; \( h_1, h_2 \) (kJ/kg) are the specific enthalpies at entry and exit of the compressor, respectively; \( \Delta T \) (K) are the temperatures of air at entry and exit of compressor, respectively.
The actual compressor power is given as

\[ W_{o,AC} = \frac{W_{i,AC}}{\eta_{s,AC}} \]  

(2a)

where the isentropic efficiency of the compressor, \( \eta_{s,AC} \), is given as

\[ \eta_{s,AC} = 1 - \left( 0.04 + \frac{(P_2/P_1) - 1}{150} \right) \]  

(2b)

where \( P_1, P_2 \) (kPa) are the compressor inlet and exit air pressures, respectively.

The rate of exergy destroyed across the compressor, \( \dot{E}_{x,AC}(\text{kW}) \), is given as

\[ \dot{E}_{x,AC} = \dot{m}_a T_0 \left( c_{pa} \ln \frac{T_2}{T_1} - R_a \ln \frac{P_2}{P_1} \right) \]  

(3a)

where \( R_a \) (kJ/kg.K) is the specific gas constant of air, and \( T_0(K) \) is the environmental dead state absolute temperature.

The specific heat capacity of the air is given as

\[ c_{pa} = 1.0189134 \times 10^3 - 0.3783636 T + 1.9843397 \times 10^{-2} T^2 + 4.2399242 \times 10^{-5} T^3 - 3.7632 \times 10^{-10} T^4 \]  

(3b)

(ii) Combustion chamber

The rate of heat addition in the combustion chamber is given as (Zaki et al., 2011)

\[ \dot{Q}_{cc} = \dot{m}_f \left[ \eta_{cc} \text{LHV} + SH_f \right] = \dot{m}_f \left[ \eta_{cc} \text{LHV} + c_{pf} (T_f - T_i) \right] \]  

(4a)

where \( \dot{m}_f \) (kg/s) is the fuel mass flow rate; \( \eta_{cc} \) is the combustion efficiency and accounts for the incomplete combustion and heat losses in the combustion chamber; \( \text{LHV} \) (kJ/kg) is the lower heating value of the fuel gas at the initial temperature; \( SH_f \) (kJ/kg) is the increase in sensible enthalpy of the fuel gas due to preheating before entry to combustion chamber; \( T_f \) (K) is the fuel gas initial temperature before preheating; \( T_i \) (K) is the temperature of gas after preheating /entry to combustion chamber; \( c_{pf} \) (kJ/kg.K) is the specific heat capacity of the fuel (natural gas):

\[ c_{pf} = \sum_{i=1}^{n} y_i c_{pf,i} / M_i \]  

(4b)

where \( y_i, M_i \) (kg/kmol.K) and \( c_{pf,i} \) (kJ/kmol.K) are the mass fraction, the molar mass and the isobaric specific heat capacity of the \( i^{th} \) gas component, respectively.

This can be determined from the natural gas composition analysis as follows:

\[ c_{pf} = a + b T_{a,v} + c T_{a,v}^2 + d T_{a,v}^3 \]  

(4c)

where \( a, b, c, d \) are coefficients which can be obtained from standard tables given in Cengel and Boles (2011); \( T_{a,v} = \frac{T_f + T_i}{2} \) (K) is the average temperature of the natural gas.

The rate of exergy destroyed in the combustion chamber, \( \dot{E}_{x,cc}(\text{kW}) \), is given as (Nag, 2013; Tiwari et al., 2013)
where \( \dot{m}_g(= \dot{m}_f + \dot{m}_a) \) (kg/s) is the mass flow rate of the flue gas stream, \( R_g \) (kJ/kg.K) is the flue gas constant; \( T_0 \Delta S \) is the rate of exergy loss in the combustion or reaction in kW.

\[ T_0 \Delta S = \dot{m}_f \text{LHV (}\varnothing - 1) \]  

\( \varnothing \) is the ratio of fuel chemical exergy to the net heating value \( \varnothing = 1.06 \) for natural gas.

The specific heat capacity of the flue gases, \( c_{pg} \) (kJ/kg.K), is given as

\[ c_{pg} = 0.991615 + 6.99703 \times 10^{-5}T + 2.7129 \times 10^{-7}T^2 - 1.22442 \times 10^{-10}T^3 \]  

The fuel exergy, \( \Delta \dot{G}_f \) (kW), is given as

\[ \Delta \dot{G}_f = \varnothing \dot{Q}_{cc} \]  

(iii) Gas turbine

The ideal and actual turbine powers are given respectively, as

\[ W_{s,GT} = \dot{m}_g (h_3 - h_a) = \dot{m}_g c_{pg} (T_3 - T_4) \]  

and

\[ W_{a,GT} = W_{s,GT} \eta_{s,GT} \]  

The turbine isentropic efficiency, \( \eta_{s,GT} \) is given as (Alhazmy & Najjar, 2004)

\[ \eta_{s,GT} = 1 - \left( 0.03 + \frac{(P_2/P_1) - 1}{180} \right) \]  

\( T_4, T_1 \) are turbine inlet and exit flue gas temperatures respectively.

The rate of exergy destruction, \( \dot{E}_{x,GT} \) (kW) in the turbine is given as

\[ \dot{E}_{x,GT} = \dot{m}_g T_0 \left( c_{pg} \frac{I_4}{T_3} - R_g \frac{P_4}{P_3} \right) \]  

The rate of heat transfer associated with the turbine exit flue gas is given as

\[ \dot{Q}_{GT,exit} = \dot{m}_g c_{pg} (T_4 - T_1) = (1 - \eta_{GT,C}) \dot{Q}_{cc} \]  

(iv) Integral characteristics of the gas turbine cycle

The gas turbine net power output, \( W_{net,GT} \), and cycle thermal efficiency \( \eta_{GT} \) are respectively, given as

\[ W_{net,GT} = W_{a,GT} - W_{a,AC} \]  

and
The total rate of exergy destroyed and exergy efficiency of the gas turbine plant are respectively, given as

\[ \dot{E}_{d,GTC} = \dot{E}_{d,AC} + \dot{E}_{d,CC} + \dot{E}_{d,GT} \]  

\[ \eta_{II,GTC} = \frac{W_{\text{net},GTC}}{\dot{Q}_{cc}} = 1 - \frac{\dot{E}_{d,GTC}}{\dot{Q}_{cc}} \]  

2.3.2 The steam turbine cycle

The steam turbine power plant operates on the Rankine cycle. The major thermodynamic devices of the steam turbine plant are, the HRSGs, the steam turbine, the condenser and the feed water pumps. Following Figure 2, thermodynamic models for the major devices are given as follows:

(i) Heat recovery steam generators (HRSGs)

The dual pressure steam cycle has high and low pressure HRSGs (HPHRSG and LPHRSG), the individual and total heat transfer rates are given respectively as

\[ \dot{Q}_{\text{HPHRSG}} = (1 - \alpha) \dot{m}(h_7 - h_{14}) = \dot{m}_g c_{pg}(T_4 - T_5) \]  

\[ \dot{Q}_{\text{LPHRSG}} = \alpha \dot{m}(h_9 - h_{13}) = \dot{m}_g c_{pg}(T_5 - T_6) \]  

and

\[ \dot{Q}_{\text{HRSG}} = \dot{Q}_{\text{HPHRSG}} + \dot{Q}_{\text{LPHRSG}} = \dot{m}_g c_{pg}(T_4 - T_6) = \dot{m}[(1 - \alpha)(h_7 - h_{14}) + \alpha(h_9 - h_{13})] \]  

where \( \dot{Q}_{\text{HRSG}} \) (kW) is the heat transfer rate in the HRSG from the gas turbine exit flue gases to the feed water streams; \( \dot{Q}_{\text{HPHRSG}} \) (kW) is the heat transfer rate in the HPHRSG; \( \dot{m}_g \) is the mass flow rate of high pressure feed water flowing through the HPHRSG; \( \dot{Q}_{\text{LPHRSG}} \) is the heat transfer rate in the LPHRSG, \( \dot{m}_p = \alpha \dot{m} \) is the mass flow rate of water flowing through the LPHRSG; \( \alpha \) is the mass fraction of steam circulating in the low pressure circuit.

\[ \alpha = \frac{\dot{m}_p}{\dot{m}} = \frac{\dot{m}_p}{\dot{m}_{LP} + \dot{m}_w} \]  

The rates of exergy input, \( \dot{E}_{X,\text{HRSG}} \) and exergy destruction, \( \dot{E}_{d,\text{HRSG}} \) in the HRSGs are respectively

\[ \dot{E}_{X,\text{HRSG}} = \dot{m}_g c_{pg} \left[ T_4 - T_0 - T_0 \ln \left( \frac{T_4}{T_0} \right) \right] \]  

\[ \dot{E}_{d,\text{HRSG}} = T_0 \left[ (1 - \alpha) \dot{m}(s_7 - s_{14}) + \alpha \dot{m}(s_9 - s_{13}) - \left( \frac{\dot{m}_g c_{pg} \ln \frac{T_5}{T_4} - \dot{m}_g R_g \ln \frac{P_6}{P_4}}{\frac{T_5}{T_4} - \dot{m}_g R_g \ln \frac{P_6}{P_4}} \right) \right] \]  

where (kJ/kg.K) is the specific entropy at state \( i \).

(ii) Steam turbine

The total power output from the low and high pressure steam turbines is given by:
\[ W_{st} = n_{s,ST} [(1 - \alpha)\dot{m}(h_f - h_g) + \dot{m}(h_{10} - h_{11})] \]  

where \( n_{s,ST}(-) \) is the steam turbine isentropic efficiency.

The rate of exergy destruction in the steam turbine (HPST and LPST) is given as

\[ \dot{E}_{x,ST} = \dot{E}_{x,HPST} + \dot{E}_{x,LPST} = \dot{m}T_0 [(1 - \alpha)(s_g - s_f)(s_{11} - s_{10})] \]  

(iii) Air cooled steam condenser

Following the work of O’Donovan and Grimes (2014) the air cooled condenser analysis is performed. The analysis is based upon the assumption that only isothermal heat rejection occurs during condensation, the amount of heat rejected to condense steam to liquid water is totally absorbed by the cooling air, with sensible heat rejection (sub cooling) neglected. The energy balance across the ACC:

\[ Q_{ACC} = \dot{m}(h_{11} - h_{12}) = \dot{m}_{acc} c_{pa}(T_{35} - T_{34}) \]  

\[ m_{acc}, c_{out}, T_{34}, T_{35} \] are the mass flow rate of the ambient air inlet to ACC, specific heat of the air, inlet air temperature and the air exit temperature from the ACC respectively. The air exit temperature can be determined as

\[ T_{35} = \frac{\dot{m}(h_{11} - h_{12}) + \dot{m}_{acc} C_{pa} T_{34}}{\dot{m}_{acc} C_{pa}} \]  

The effectiveness of a cross-flow air-cooled heat exchanger is defined by;

\[ \varepsilon = \frac{T_{35} - T_{34}}{T_{11} - T_{34}} \]  

\( T_{11}(K) \) is the steam temperature at the condensing pressure (turbine back pressure)

\[ T_{11} = \frac{T_{35} - T_{34}}{\varepsilon} + T_{34} \]  

The quality of the turbine exhaust steam in to the condenser is given as

\[ X = \frac{h_{11} - h_{12}}{h_{11g} - h_{12}} \]  

\( h_{11g}(kJ/kg) \) is the specific enthalpy of saturated vapor at the condensing pressure.

The rates of exergy destruction in the condenser is given as

\[ \dot{E}_{x,ACC} = \dot{m}T_0 (s_{11} - s_{12} + \frac{h_{11} - h_{12}}{T_{AC}}) \]  

where \( T_{AC}(K) \) is the thermodynamic mean temperature of the cooling air given by (Oko, 2012)

\[ T_{AC} = \frac{(T_{a, out} - T_{a, in})}{\ln(T_{a, out}/T_{a, in})} \]  

(iv) Feed water pumps

The power consumption of the high and low feed water pumps is given as:
\[
W_{\text{FWP}} = \frac{m \left[ (h_{13} - h_{12}) + (1 - \alpha) (h_{14} - h_{13}) \right]}{\eta_{s,\text{FWP}}} \approx \frac{m \nu_{12} \left[ (P_{13} - P_{12}) + (1 - \alpha) (P_{14} - P_{13}) \right]}{\eta_{s,\text{FWP}}} 
\]
where \( \eta_{s,\text{FWP}}(-) \) is the high and low feed water pump isentropic efficiency.

The rate of exergy destruction in the high and low feed water pumps is given as

\[
E_{x_{\text{FWP}}} = E_{x_{\text{HPFWP}}} + E_{x_{\text{LPFWP}}} = m T_0 \left[ (s_{12} - s_{13}) + (1 - \alpha) (s_{13} - s_{14}) \right] 
\]

2.3.3 Overall performance characteristics of the combined cycle power plant

The overall combined cycle power output, thermal and exergy efficiencies are given respectively as:

\[
\eta_{I,\text{CCPP}} = \frac{W_{\text{CCPP}}}{Q_{cc}} 
\]
\[
\eta_{II,\text{CCPP}} = \frac{W_{\text{CCPP}}}{\Delta G_o} 
\]

\[
\Delta W_{\text{gain,STC}} = W_{\text{net,STC}} (\text{ACC cooling}) - W_{\text{net,STC}} 
\]

2.3.4 Thermodynamic analysis of ORC unit

The evaporation temperature when the maximum net power output is reached is known as the optimal evaporation temperature, \( T_{15} \), which can be determined by using the simplified formula (He et al., 2012)

\[
T_{15} = \xi T^* = \xi \sqrt{T_{12} (T_6 - \Delta T_e)} 
\]
\( \xi = 0.999 + 0.00041(T_6 - T_{18})/\left(h_{l9}/q^*\right) \)  

(27b)

\( \xi \) is a correction factor, \( h_{l9} \) is the latent heat of the organic fluid at the temperature \( T^* \) and \( q^* \) (kJ/kg) is the heat the working fluid absorbs from the condenser temperature to \( T^* \). Hence

\( q^* = c_{pw} (T^* - T_{17}) \)  

(27c)

\( c_{pw} \) (kJ/kg. K) is the specific heat capacity of the working fluid.

The mass flow rate, \( m_w \) (kg/s), of the organic working fluid is given as

\[
m_w = m_g c_{pg} \frac{(T_6 - T_{15} - \Delta T_e)}{h_{l9}}
\]

(28)

The rate of heat transfer in the organic liquid evaporator (OLE) is given as

\[
Q_{OLE} = m_w c_{pg} (T_6 - T_{19}) = m_w (h_{15} - h_{18})
\]

(29a)

The rate of exergy destruction in the OLE is given as

\[
Ex_{d,OLE} = m_w T_0 \left[ (s_{15} - s_{18}) - \left( \frac{h_{15} - h_{18}}{T_H} \right) \right]
\]

(29b)

\( T_H \) is the thermodynamic temperature of the heat source, \( T_H = (T_6 - T_{19})/\ln(T_6/T_{19}) \).

The organic vapor turbine (OVT) power output and exergy destruction rate are given respectively, as:

\[
W_{OVT} = m_w \eta_e \eta_G (h_{15} - h_{16})
\]

(30a) and

\[
Ex_{d,OVT} = m_w T_0 (s_{15} - s_{16})
\]

(30b)

where \( \eta_e \) and \( \eta_G \) OVT isentropic efficiency and the electric generator efficiency respectively.

The power consumed and the exergy destruction rate in the organic liquid feed pump (OLFP) are respectively given as

\[
W_{OLFP} = \frac{m_w (h_{18} - h_{17})}{\eta_p} = \frac{m_w V_w (P_{18} - P_{17})}{\eta_p}
\]

(31a) and

\[
Ex_{d,OLFP} = m_w T_0 (s_{18} - s_{17})
\]

(31b)

where \( V_w \) (m\(^3\)/kg) is the specific volume of the organic fluid and

The rates of heat rejection and exergy destruction in the condenser are respectively

\[
Q_{OVC} = m_w (h_{16} - h_{17}) = m_{co} c_{pco} (T_{41} - T_{40})
\]

(31c) and
\[
\dot{E}_{X_{\text{d,OVC}}} = m_g T_0 \left[ (s_{16} - s_{17}) - \left( \frac{h_{16} - h_{17}}{T_L} \right) \right]
\]

where \(T_{41}, T_{40}(K)\) and \(m_{cg}\) (kg/s) are the inlet and exit temperatures and the mass flow rate of the organic fluid condenser cooling air, \(T_L(K)\) is the thermodynamic mean temperature of the cooling air.

The total rates exergy destruction and exergy input rate in the ORC are respectively, given as

\[
\dot{E}_{X_{\text{d,ORC}}} = \dot{E}_{X_{\text{d,OLE}}} + \dot{E}_{X_{\text{d,OVT}}} + \dot{E}_{X_{\text{d,OFP}}} + \dot{E}_{X_{\text{d,OVC}}}
\]

and

\[
\dot{E}_{X_{\text{in,OVC}}} = m_g c_{pg} \left[ T_6 - T_{19} - T_0 \ln \left( \frac{T_6}{T_{19}} \right) \right]
\]

The ORC net power output, thermal and exergy efficiencies can be obtained respectively as

\[
W_{\text{ORC}} = W_{\text{OVT}} - W_{\text{OLFP}}
\]

\[
\eta_{I,\text{ORC}} = \frac{W_{\text{ORC}}}{Q_{\text{OLE}}}
\]

\[
\eta_{II,\text{ORC}} = \frac{W_{\text{ORC}}}{E_{X_{\text{in,ORC}}}}
\]

2.3.5. Selection of organic working fluid for the ORC unit
R-113 was selected as the working fluid for the ORC due to its good qualities and suitable thermophysical properties as indicated in works of Mago, Chamra, Srinivasan, and Somayaji (2008) and Safarian and Aramoun (2015).

2.4 Thermodynamic analysis of the LiBr-H_2O absorption refrigeration unit
Following some previous works (Dincer & Ratlamwala, 2016; Dincer, Rosen, & Ahmadi, 2018; Kaushik & Arora, 2009; Muhsin & Kaynakli, 2007; Popli, Rodgers, & Eveloy, 2013; Touaibi, Feidt, Vasilescu, & Tahar Abbes, 2013), the ARC is analyzed based on Figures 1 and 2 as follows:

(i) Refrigerant generator

Energy balance across the generator is given as

\[
\dot{Q}_{\text{gen}} = m_{rf} h_{27} + m_{ws} h_{24} - m_{ss} h_{23} = km_g c_{pg}(T_{19} - T_{30})
\]

where \(k\) is the number of turbines in the gas turbine power unit, in this case, \(k = 3\); \(m_{rf}, m_{ws}, m_{ss}\) and \(m_g\) (kg/s) are the mass flow rates of the refrigerant (water vapor evaporated in the generator), weak solution, strong solution, and flue gases, respectively.

The rate of exergy destruction in the refrigerant generator is given as

\[
\dot{E}_{X_{\text{d,gen}}} = \left( 1 - \frac{T_1}{T_9} \right) \dot{Q}_{\text{gen}} - (\dot{E}_{X_{24}} - \dot{E}_{X_{23}} + \dot{E}_{X_{27}})
\]

(ii) Refrigerant condenser
The rate of heat rejected in the condenser where saturated water vapor (refrigerant) from the generator is cooled to saturated liquid at the generator pressure is given as:

\[ Q_{\text{cond}} = \dot{m}_f (h_{27} - h_{28}) \] (35a)

The rate of exergy destruction in the condenser is given as:

\[ \dot{E}_{X,\text{cond}} = (\dot{E}_{x_{27}} - \dot{E}_{x_{28}}) - \left( 1 - \frac{T}{T_{\text{cond}}} \right) \dot{Q}_{\text{cond}} \] (35b)

where \( T_{\text{cond}} \) (K) is the temperature in the refrigerant condenser.

(iii) Throttle valves

The pressure of liquid refrigerant at generator pressure is reduced isenthalpically to the evaporator pressure, \( h_{28} = h_{29} \) and \( h_{25} = h_{26} \); so that the rates of exergy destruction in the solution and refrigerant throttle valves are, respectively,

\[ \dot{E}_{X_{\text{STV}}} = T_0 \dot{m}_{ws} (s_{26} - s_{25}) \] (36a)

and

\[ \dot{E}_{X_{\text{RTV}}} = T_0 \dot{m}_f (s_{29} - s_{28}) \] (36b)

(iv) Refrigerant evaporator

The liquid refrigerant at low pressure is evaporated by the circulating chilled water used for inlet air cooling in the air cooling coil (ACC), and becomes saturated vapor. The refrigeration (or cooling) load, \( Q_{\text{cl}} \) (kW), is given as

\[ Q_{\text{evap}} = \dot{m}_f (h_{20} - h_{29}) = Q_{\text{cl}} \] (37a)

The rate of exergy destruction in the evaporator is given as

\[ \dot{E}_{X_{\text{Evap}}} = \left( \frac{T}{T_0} - 1 \right) \dot{Q}_{\text{evap}} - (\dot{E}_{x_{20}} - \dot{E}_{x_{29}}) \] (37b)

(v) Refrigerant absorber

In the absorber, the weak solution from the generator which has been throttled to the absorber pressure readily absorbs the saturated refrigerant vapor from the evaporator to become a strong solution. This exothermic process releases some heat, which is removed by the cooling fluid flowing through the absorber. The rate of heat rejection from the absorber is given as

\[ Q_{\text{abs}} = (\dot{m}_f h_{20} + \dot{m}_{ws} h_{26}) - (\dot{m}_{ss} h_{21}) \] (38a)

where \( \dot{m}_{abs} \) (kg/s) is the mass flow rate of the cooling fluid in the absorber.

The rate of exergy destruction in the absorber is given as

\[ \dot{E}_{X_{\text{abs}}} = (\dot{E}_{x_{21}} - \dot{E}_{x_{26}} - \dot{E}_{x_{20}}) - \left( 1 - \frac{T}{T_{\text{abs}}} \right) \dot{Q}_{\text{abs}} \] (38b)

where \( T_{\text{abs}} \) (K) is the temperature in the absorber.
(vi) Solution pump

The pump transfers the strong LiBr-H₂O solution from the absorber to the generator, and its power consumption, which is usually negligibly small, is given as

\[ W_{sp} = m_{ss}(h_{22} - h_{21}) \]  

The rate of exergy destruction in the solution pump is given as

\[ Ex_{D,sp} = T_0 m_{ss}(s_{22} - s_{21}) \]  

(vii) Regenerator

The regenerator heats the strong solution from the absorber on its way to the generator and cools the weak solution returning from the generator to the absorber. The regenerator heat and exergy destruction rates are, respectively

\[ Q_{reg} = m_{ws}(h_{24} - h_{25}) = m_{ss}(h_{23} - h_{22}) \]  

and

\[ Ex_{D,reg} = (Ex_{24} - Ex_{25}) - (Ex_{23} - Ex_{22}) \]  

(viii) Coefficient of performance of the absorption refrigeration cycle:

\[ COP = \frac{Q_{Evap}}{Q_{gen} + W_{sp}} \]  

(ix) Exergy efficiency of the absorption refrigeration cycle:

\[ \eta_{II, ARS} = \frac{Q_{Evap} \left( \frac{T_e}{T_c} - 1 \right)}{Q_{gen} \left( 1 - \frac{T_e}{T_g} \right) + W_{sp}} \]  

where \( T_e \) and \( T_g \) (K) are the temperatures of the refrigerant in the evaporator and the logarithmic mean temperature of the waste flue gases in the refrigerant generator, respectively

2.5 Inlet air cooling load analysis

The total cooling load at the air cooler of the refrigeration system is the inlet air cooling load to the gas turbine plants. It comprises the heat removed to reduce the ambient air temperature from its initial ambient condition to the desired cooled state, i.e. the sensible heat of air and the heat required to condense the moisture contained in the air (the latent heat). Thus total inlet air cooling load or the refrigerating capacity, \( Q_{CL} \) (kW) is the summation of both the sensible \( (\dot{Q}_s) \) and the latent \( (\dot{Q}_L) \) cooling loads:

\[ Q_{CL} = \dot{Q}_S + \dot{Q}_L = Q_{Evap} \]  

The sensible cooling load can be calculated as (Dawoud et al., 2005)

\[ \dot{Q}_S = \frac{V_a c_{pa}}{V_a} (T_0 - T_1) \]
\( V_a \) (m\(^3\)/s) is volume flow rate of air at the ISO conditions normally based on actual data acquired from the plant; \( T_o \) (K) is dry bulb temperature of the ambient air and \( T_1 \) (K) is cooled air temperature at state 1. \( \nu_a \) (m\(^3\)/kg) is the specific volume of the humid air per kilogram of dry air.

The specific volume of humid air per kilogram of dry air can be obtained from (Dawoud et al., 2005)

\[
\nu_a = (0.287 + \omega_o 0.462) \frac{T}{P_{atm}}
\]

where \( T \) (K) is the dry bulb temperature; \( P_{atm} \) (kPa) is the atmospheric pressure; \( \omega_o \) (kg\(_{wv}\)/kg\(_{da}\)) is the specific humidity of the air which is given by (Oko & Diemuodeke, 2010)

\[
\omega_o = \frac{0.622 \cdot RH_o}{(P_{atm} - RH_o)} P_{wv}
\]

\( P_{wv} \) is the saturation water vapor pressure at the given dry bulb temperature, \( T \). \( RH_o \) is the relative humidity of the air.

The latent cooling load \( Q_L \) is given as

\[
\dot{Q}_L = \dot{V}_a \nu_a \left[ \omega_o \left( c_{pv} T_o + h_{fg} \right) - \omega_1 \left( c_{pv} T_1 + h_{fg} \right) - (\omega_o - \omega_1) c_{pw} T_1 \right]
\]

where \( c_{pv} \) (kJ/kg.K) is the isobaric heat capacity of water vapor in the humid air, \( h_{fg} \) (kJ/kg) is the air latent heat of evaporation of water at 0°C; \( \omega_1 \) (kg/kg.da) is the specific humidity of the air at the desired compressor inlet temperature; \( c_{pw} \) (kJ/kg.K) is the isobaric specific heat capacity of liquid water.

The mass flow rate of chilled water circulating from the evaporator through the air cooler is given as:

\[
\dot{m}_{cw} = \frac{\dot{Q}_{CL}}{c_{pw} \left( T_{33} - T_{31} \right)}
\]

where \( T_{31} \) is the chilled water supply temperature and \( T_{33} \) is the water return temperature.

The pump power required for chilled water circulation (\( W_{cw} \)) is given as:

\[
W_{cw} = \frac{\dot{m}_{cw} \Delta P}{\rho}
\]

\( \rho \) (kg/m\(^3\)) is the density of water and \( \Delta P \) (kPa) is the pressure loss to be overcome by the pump.

2.4 **Overall integrated power plant performance**

The total power output of the inlet air cooled integrated gas-, steam-, and organic fluid- cycle power plant is then given as

\[
W_{iac,IPP} = (3 \times W_{net,GTC}) + W_{net,STC} + W_{net, ORC} - W_{cw}
\]

The thermal efficiency becomes
The Second law efficiency is given as

\[ \eta_{II,IPP} = \frac{W_{loc,IPP}}{3Q_{cc}} \]  

(44b)

The Sustainability index is given as

\[ S.I = \frac{1}{1 - \eta_{II}} \]  

(45)

2.7 Sustainability index
To reduce the effect of global warming, the rate of fossil fuel depletion and increase environmental sustainability by lengthening the lives of fuel resources, it is important to utilize fuel resources like natural gas very efficiently through waste heat recovery efforts. The relationship between exergy efficiency and sustainability has been proposed (Ahmadi, Dincer, et al., 2012; Oyedepo, Fagbenle, Adefila, & Alam, 2015) and can be derived as

\[ S.I = \frac{1}{1 - \eta_{II}} \]

where \( S.I \) is the sustainability index (−) and \( \eta_{II} \) is exergy efficiency.

3. Results and discussion
Using the models given in Section 2, the results obtained from the analysis are presented and discussed here

3.1 Data used for the generation of the results
The operating parametric values of the active combined cycle power plant (CCPP) and other thermodynamic specifications of used in this study are presented in Table 1.

3.2 Validation of models for the existing combined cycle power plant analysis
The methodology and derived models used for the analysis were validated by comparing computed results with measured operating values of some key parameters of the active combined cycle plant as shown in Table 2. The error margins were within the acceptable range for power plant applications.

3.3 Thermodynamic characteristics of the integrated gas-, steam-, organic fluid- cycle power and absorption refrigeration plant
The performance parameters of the various power plant cycles are tabulated in Table 3. With a fuel energy input of 1,333 MW and fuel exergy input of 1,374 MW at the combustion chamber, the results of the analysis shows that by utilizing the flue gas waste heat of the combined gas- and steam- turbine cycle power plant (CCPP) to power an ORC unit, extra 7.5 MW of electricity was generated and by further powering an absorption refrigeration system to cool the inlet air streams to 15°C in the gas turbine plants, additional 51.1 MW of electricity was generated. In general, it can be seen that the combined effect of integrating the ORC and absorption refrigeration cycle (ARC) to the CCPP showed that the net power output of the integrated power plant (GTC + STC + ORC + ARC) was increased by 9.1%, thermal and exergy efficiencies by 8.7 and 8.8%, respectively, while the total exergy destruction rate and specific fuel consumption reduced by 13.3 and 8.4% respectively. Sustainability index increased from 1.88 in the existing CCPP to 2.04 in the integrated power plant, an increment of 8.4%. The high sustainability index value is an indication of the degree of environmental sustainability potential of the integration power plant.

The exergy analysis of the existing combined cycle power plant indicated that the highest rate of exergy destruction, about 59%, occurs in the combustion chamber of GT plant as shown in Figure 5. This is due to the irreversibilities associated with the highly exothermic combustion reaction and heat transfer across the large temperature differences between the reactants-fuel and air. The results closely agree with the work of Ersayin and Ozgener (2015).
For the ORC plant, the exergy destruction rate in the units are shown in Figure 6 with the highest rate of exergy destruction occurring in the organic liquid evaporator followed by the organic vapor condenser, due to the irreversibilities associated with heat transfer processes (with change of phase) occurring in this devices. A similar trend can be found in the work of Guo et al. (2015).

For the ARC, the exergy destruction rate in the units are shown in Figure 7 with the highest rate of exergy destruction occurring in the evaporator followed by the absorber. This may be due to the irreversibilities associated with the phase change and high temperature difference between the refrigerated water and evaporator space in the evaporator, and the mixing and heat transfer processes occurring in the absorber. A similar trend was observed in the work of Dincer et al. (2018) for single effect absorption refrigeration chiller.

The thermodynamic characteristics of the absorption refrigeration system are presented in Table 4. The low COP of the absorption refrigeration cycle under consideration is due to its high temperature difference between generator $T_{gen}$ (K) and evaporator temperature, $T_e$ (K) as may readily be verified from the equivalent Carnot COP, $(T_{gen} - T_e)/(T_{gen} (T_0 - T_e))$.

### 3.4 Sensitivity testing of some key parameters

Figure 8 shows the effects of cooling the compressor inlet air temperature on the cumulative net power output, thermal and exergy efficiencies of the integrated gas-, steam- and organic fluid-power plant. It was observed that as the air temperature increased the net power output, thermal and exergy efficiencies of the integrated plant decreased. This is because as the inlet air temperature to gas turbine cycle rises, its net power output drops (Boonnasa et al., 2006; Hosseini, Beshkani, & Soltani, 2007; Mohapatra, 2014; Singh, 2016).

Figure 9 shows the influence of compressor inlet air temperature on the specific fuel consumption and the sustainability index of the integrated gas-, steam- and organic fluid-power plant. It was observed that as the air temperature increased the specific fuel consumption of integrated plant decreased while the environmental sustainability index increased. The inverse relationship between the plant specific fuel consumption and sustainability index is a reflection of the effective utilization of the fuel energy supplied to the integrated power plant.

The impact of variations in the combustion efficiency of the gas turbine cycle (which is the prime mover of the integrated power plant) on the overall net power output, specific fuel consumption and HRSG stack exit temperature of the integrated plant is shown in Figure 10. It can be seen that the net power output of the integrated power plant increased while the specific consumption decreased with increase in combustion efficiency. Also the temperature of the flue gases ($T_7$) exiting the heat recovery steam generator (HRSG) increased thus leading to increase in the available input energy to the adjoining ORC and ARC units. Therefore, improving the combustion efficiency will improve the overall performance of the integrated plant. Similar trends were observed in the works of Oko and Njoku (2017).

Figure 11 shows the effect of variations in the compressor pressure ratio of the GTC on the net power output, thermal and exergy efficiencies, and the total exergy destruction rate of the integrated plant. It is observed that the pressure ratio increased with net power output and plant efficiencies up to certain optimum value, beyond which, further increase in pressure ratio resulted in drop of net power output and plant efficiencies. However, the total exergy destruction rate decreased in a reverse manner following the rise in the net power output and plant efficiencies. This is because from lower pressure ratios, increasing the compressor pressure ratio leads to increase in the compressor exit air temperature in to combustion chamber and subsequent increase in the gas turbine inlet temperature (GTIT) and hence, increase in the turbine expansion work and plant efficiencies. However increasing the pressure ratio beyond a certain optimum limit leads to increase in the compression work and reduction in the net power output and plant efficiencies. Similar trends
Table 1. Input data for the analysis of the integrated power plant

| Plant unit | Parameter | Symbol | Units | Value |
|------------|-----------|--------|-------|-------|
| **Operating data from the CCPP** | **Inlet temperature** | $T_{oi}$ | °C | 30.9 |
| | **Relative humidity of air** | $R_{H}$ | % | 70 |
| | **Inlet pressure** | $P_{i}$ | kPa | 99 |
| | **Volume flow rate($\times 3$)** | $V_{i}$ | m³/s | 1287 |
| **Combustion chamber** | **Outlet pressure** | $P_{o}$ | kPa | 1380 |
| | **Fuel inlet temperature** | $T_{fi}$ | °C | 60.2 |
| | **Fuel inlet pressure** | $P_{f}$ | kPa | 2650 |
| | **Fuel mass flow rate($\times 3$)** | $m_{f}$ | kg/s | 25.83 |
| **Gas turbine** | **Fuel lower heating value** | $LHV$ | kJ/kg | 52,580 |
| | **Outlet temperature** | $T_{o}$ | °C | 531 |
| | **Net power output ($\times 3$)** | $W_{net, gas}$ | MW | 447 |
| **HRSG** | **Flue gas mass flow rate($\times 3$)** | $m_{f}$ | kg/s | 1509.3 |
| **Steam turbine** | **Net power output** | $W_{net, ST}$ | MW | 202 |
| | **Inlet steam high pressure (HP)** | $H_{PST}$ | kPa | 10,020 |
| | **Inlet steam low pressure (LP)** | $L_{PST}$ | kPa | 537 |
| | **Mass flow rate of HP steam** | $m_{HP}$ | kg/s | 175.2 |
| | **Mass flow rate of LP steam** | $m_{LP}$ | kg/s | 54.72 |
| | **Inlet temperature of HP steam** | $T_{i}$ | °C | 512 |
| **Air cooled condenser** | **Inlet temperature of LP steam** | $T_{i}$ | °C | 257.2 |
| | **Outlet water temperature** | $T_{w}$ | °C | 60.7 |
| | **Outlet water pressure** | $P_{w}$ | kPa | 21 |
| | **Air mass flow rate** | $m_{w}$ | kg/s | 160,680 |
| **Other thermodynamic specifications** | **Combustion efficiency** | $\varepsilon$ | - | 0.98 |
| | **Flue gas constant** | $R_g$ | kJ/kg.K | 0.285 |
| | **Steam turbine isentropic efficiency** | $\eta_{ST}$ | - | 0.82 |
| | **Feed pump isentropic efficiency** | $\eta_{FP}$ | - | 0.90 |
| | **Desired compressor inlet air temperature** | $T_{ci}$ | °C | 15 |
| | **Relative humidity of air at state 1** | $R_{H}$ | % | 100 |
| | **Density of chilled water** | $\rho$ | kg/m³ | 1000 |
| | **Pressure loss overcome by chilled water circulation pump** | $\Delta P$ | kPa | $9.81 \times 10^5$ |
| | **Chilled water supply temperature** | $T_{13}$ | °C | 5 |
| | **Chilled water return temperature** | $T_{13}$ | °C | 10 |
| | **Environmental reference temperature** | $T_{o}$ | °C | 25 |
| | **Environmental reference pressure** | $P_{o}$ | kPa | 101.325 |
| | **Pinch temperature difference in the ORC evaporator** | $\Delta T_{e}$ | °C | 10 |
| | **Condenser temperature** | $T_{16}$ | °C | 10 |
| | **ORC turbine isentropic efficiency** | $\eta_{T}$ | - | 0.87 |
| | **ORC pump isentropic efficiency** | $\eta_{P}$ | - | 0.85 |
| | **Electric generator efficiency** | $\eta_{G}$ | - | 0.90 |

Sources:

1. Afam VI Combined Cycle Gas Turbine Plant (CCGT) (2015).
2. Saravanamuttoo, Cohen, and Rogers (1996) and Nag (2013).
3. Tiwari et al. (2013).
4. Boonnoa et al. (2006) and Mohanty and Paloso (1995).
5. Wang, Zhou, Guo, and Wang (2012).
6. Quoilin, Dewallef, Lemort, Van Den Broek, and Declaye (2013).
7. Chacartegui, Sánchez, Muñoz, and Sánchez (2009).
were observed in the works of Ahmadi, Dincer, et al. (2012) and Oyedepo, Fagbenle, and Adefila (2017).

4. Conclusion
The combined cycle power plant technology, though, offers more efficiency and environmental sustainability over the conventional single cycle gas- and steam-turbine plants, for fossil fuels to power generation, could be discarding usable low grade waste heat which can be used to generate more energy products from low grade energy conversion units such as the organic Rankine cycle and the absorption refrigeration cycle.

This study presents the performance evaluation of a combined cycle power plant integrated with an organic Rankine cycle for extra power generation and an absorption refrigeration cycle for production of cooling.

The attached ORC and ARC units are powered with the flue gas exhaust heat from the CCPP. The evaluation was conducted by performing the energy, exergy and environmental sustainability index analysis of the integrated system and its components. Based on the operating data of an existing combined cycle power plant operating in the tropical rain forest region of Nigeria, the results of the analysis showed that by utilizing the flue gas waste heat of the combined gas- and steam-turbine cycle power plant to power an ORC unit, using R113 as the working fluid, extra 7.5 MW of electricity was generated and by further powering an absorption refrigeration system to cool the inlet air streams to 15°C in the gas turbine plants, additional 51.1 MW of electricity was generated. The overall effect of integrating the ORC and absorption refrigeration cycle (ARC) to the CCPP showed that the net power output of the integrated power plant (GTC + STC + ORC + ARC) was increased by 9.1%, thermal and exergy efficiencies by 8.7 and 8.8%, respectively, while the total exergy destruction rate and specific fuel consumption reduced by 13.3 and 8.4% respectively. Sustainability index increased by 8.4% which means that integrated plant has greater environmental sustainability potential over the combined cycle plant.

Results of exergy analysis show that the highest rate of exergy destruction in the CCPP occurred in the combustion chamber, while in the ORC plant, the highest rate of exergy destruction occurred in the organic liquid evaporator followed by the organic vapor condenser, and in the ARC, the highest rate of exergy destruction occurred in the evaporator followed by the absorber.

Parametric investigations revealed that as the compressor inlet air temperature reduced, the net power output, thermal and exergy efficiencies, and the environmental sustainability index of the integrated power plant increased while the plant total exergy destruction rate and specific fuel consumption decreased. It was also shown that the overall plant performance was significantly affected by variations in combustion efficiency and compressor pressure ratio which are major design parameters of the gas turbine cycle.

It is expected that this study will aid integrated power plant designers, policy makers and plant owners on the choice of possible performance enhancement modifications to existing combined cycle power plants. Utilization of exhaust flue gas to run more energy devices would lead to the reduction of the stack exhaust temperature and the possibility of water vapor condensation. However,
| Performance parameters          | Gas turbine plants (GTC) | Steam turbine plant (STC) | Combined cycle plant (GTC + STC) | Organic Rankine cycle (ORC) | Integrated power plant-IPP (GTC + STC + ORC) | IPP-w/cooling (GTC + STC + ORC + ARC) |
|--------------------------------|--------------------------|---------------------------|---------------------------------|-----------------------------|--------------------------------------------|---------------------------------------|
| Net power output (MW)          | 441.3                    | 200.2                     | 641.5                           | 7.5                         | 649.0                                      | 700.1                                 |
| Thermal efficiency (%)         | 33.1                     | 29.7                      | 48.3                            | 8.4                         | 48.7                                       | 52.5                                  |
| Exergy Efficiency (%)          | 32.1                     | 29.2                      | 46.9                            | 30.5                        | 47.2                                       | 51.0                                  |
| Total exergy destruction rate (MW) | 548.7                  | 153.7                     | 702.4                           | 9.2                         | 711.6                                      | 609.3                                 |
| Specific fuel consumption (kg/MWh) | 210.7                    | 144.9                     | 143.3                           | 132.8                       |                                            |                                       |
| Sustainability index           | 1.47                     | 1.41                      | 1.88                            | 1.44                        | 1.90                                       | 2.04                                  |
the authors suggest the installation of scrubbers to trap condensed liquids and prevent plume discharge into the atmosphere.
Table 4. LiBr-H\textsubscript{2}O absorption refrigeration system parameters

| Parameter                        | Value  |
|----------------------------------|--------|
| Generator temperature, $T_{\text{gen}}$ (°C) | 95.0   |
| Condenser temperature, $T_{\text{cond}}$ (°C) | 46.0   |
| Absorber temperature, $T_{\text{abs}}$ (°C) | 30.0   |
| Evaporator temperature, $T_{\text{e}}$ (°C) | 5.0    |
| Input thermal energy (MW)        | 74.6   |
| Total cooling load (MW)          | 44.4   |
| Total exergy destruction rate (MW) | 31.8   |
| Coefficient of performance, COP (-) | 0.60   |
| Exergy efficiency, $\eta_{\text{iARS}}$ (%) | 27.0   |

Figure 8. Effects of compressor inlet air temperature on the net power output, thermal and exergy efficiencies of the integrated power plant.

Figure 9. Influence of compressor inlet air temperature on the specific fuel consumption and sustainability index of the integrated plant.
Figure 10. Influence of combustion efficiency on the net power output, specific fuel consumption and HRSG stack exit temperature of the integrated power plant.

Figure 11. Compressor pressure ratio vs. net power output, efficiencies and total exergy destruction rate of the integrated plant.
Nomenclature

\( h \)  Specific enthalpy, kJ/kg
\( P \)  Pumping power, kW
\( W \)  Power Output, MW
\( W_{\text{net}} \)  Net power output, MW
\( c_p \)  Specific heat capacity at constant pressure, kJ/kg.K
\( m \)  Mass flow rate, kg/s
\( R \)  Gas constant, kJ/kg.K
\( S \)  Specific entropy, kJ/kg.K
\( P \)  Pressure, Bar
\( T \)  Temperature, °C
\( \text{S.I} \)  sustainability index
\( \text{SFC} \)  specific fuel consumption

Greek letters

\( \beta \)  Number of gas turbine units
\( \Delta \)  Change between states
\( \eta \)  Efficiency
\( \gamma \)  Specific heat ratio
\( \omega \)  Specific humidity, kg/kg.da
\( \xi \)  Concentration by mass of LiBr in LiBr-H\(_2\)O solution

Subscripts and superscripts

\( a \)  Air
\( a, v \)  Average
\( c \)  Compressor
\( cc \)  Combustion chamber
\( \text{cond} \)  Condenser
\( da \)  Dry air
\( f \)  Fuel
\( g \)  Flue gas
\( gt \)  Gas turbine
\( i \)  Constituent or component
\( p \)  Pump
\( s \)  Steam
\( st \)  Steam turbine
\( t \)  Turbine
\( \text{tot} \)  Total
\( w \)  Water
\( wv \)  Water vapor
Abbreviations

| Abbreviation | Description                        |
|--------------|------------------------------------|
| el.Gen       | Electric generator                 |
| AC           | Air compressor                     |
| ACC          | Air cooled condenser               |
| ARC          | Absorption refrigeration unit      |
| CC           | Combustion chamber                 |
| CCPP         | Combined cycle power plant         |
| COND         | Condenser                          |
| GT           | Gas turbine                        |
| HP           | High pressure                      |
| HPFWP        | High pressure feed water pump      |
| HPST         | High pressure steam turbine        |
| HPHRSG       | High pressure heat recovery steam generator |
| LHV          | Lower heating value                |
| LPFWP        | Low pressure feed water pump       |
| LPHRSG       | Low pressure heat recovery steam generator |
| LPST         | Low pressure steam turbine         |
| RH           | Relative humidity                  |
| SH           | Sensible enthalpy (or heat) change |
| SHf          | Sensible heat gain of fuel         |
| OLE          | Organic liquid evaporator          |
| OLFP         | Organic liquid feed pump           |
| ORC          | Organic Rankine cycle              |
| ORCPP        | Organic Rankine cycle power plant  |
| OVC          | Organic vapor condenser            |
| OVT          | Organic vapor turbine              |

Funding
The authors received no direct funding for this research.

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Citation information
Cite this article as: Performance evaluation of a combined cycle power plant integrated with organic Rankine cycle and absorption refrigeration system, I.H. Njoku, C.O.C. Oka & J.C. Ofodu, Cogent Engineering (2018), 5: 1451426.

Cover image
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