OPEN ACCESS
Saudi Journal of Engineering and Technology
Abbreviated Key Title: Saudi J Eng Technol
ISSN 2415-6272 (Print) |ISSN 2415-6264 (Online)
Scholars Middle East Publishers, Dubai, United Arab Emirates
Journal homepage: http://scholarsmepub.com/sjet/

Original Research Article

Comparative Exergo-Economic Analysis of Simple and Modified Gas Turbine Cycles

E. G. Saturday, E. Efekumo
Department of Mechanical Engineering, University of Port Harcourt, Rivers State, Nigeria

***Corresponding author:** Ebigenibo Genuine Saturday
| **Received:** 07.04.2019 | **Accepted:** 17.04.2019 | **Published:** 30.04.2019
DOI:10.21276/sjeat.2019.4.4.5

Abstract

Comparative exergo-economic analysis of a simple and modified cycle gas turbine power plants was carried out in this work, using a GE MS5001 gas turbine located at Trans Amadi in Rivers state, Nigeria as a case study. The simple cycle plant was modified to have an intercooler, regenerator and a reheater with one additional compressor and one additional turbine. Energy, exergy and exergo-economic analyses were carried out on both plants. The net output of the GE MS5001 gas turbine is 26.90MW, and its fuel is natural gas with heating value of 50MJ/kg. The simple gas turbine cycle has an efficiency of 30.4% while the modified gas turbine cycle has an output power and efficiency of 44.67MW and 48.0% respectively. The exergy efficiency of the simple cycle is 22.6% while that of the modified cycle is 43.0%. The highest exergy destruction rate was in the combustion chamber in both cycles and accounted for 90% of exergy destroyed in the simple gas turbine cycle and 43% and 36% in the combustion chamber and reheater of the modified cycle plant. The economic analysis carried out showed that the total purchase equipment cost was higher in the simple cycle plant because of the high pressure ratios the compressor and the turbine operate in the simple cycle compared to the modified cycle. The cost of producing electricity for the simple gas turbine cycle was 10.18$/GJ while that for the modified gas turbine cycle was between 7.3339$/GJ and 7.7969$/GJ. The cost of electricity from the modified cycle is thus cheaper. All the exergo-economic parameters favour the operation of the modified cycle.
Keywords: Exergy analysis, Exergo-economics analysis, modified cycle, purchase equipment cost, simple cycle Nomenclature.

INTRODUCTION

In the period of organic energy economy, animals were harnessed for their energy in carrying out work. As humans evolved and created more complex tools and developed bigger societies, also did the need for energy increased [1]. The development of steam engines was a turning point in the timeline of energy utilization which characterized the industrial revolution [2]. Energy remains the driving force of the world and it is one of the main ingredients required for growth and development in any country. Nigeria is a country richly blessed with abundant energy resources which include renewable and non- renewable energy resources. The natural gas deposit is the second largest in the whole of Africa after Algeria [3]. The high dependence on fossil fuel is due to its relatively low cost when compared to other energy resources [4]. Nigeria's electricity generation capacity is 4890MW which is not enough to meet the high demand of electricity in Nigeria estimated to be about 10,000MW [5] .

The thermal power plants in Nigeria mostly use gas turbines either in combined cycle or simple cycle to generate electricity using natural gas which is a fossil fuel as source of energy. The utilization of fossil fuels is associated with hidden costs which are not reflected in market prices [6]. The impact on the environment and health is very massive which has given rise to questions on sustainability of energy solutions. Though Natural gas is the cleanest fossil fuel, it produces NOx, sulfur, mercury and particulates. The high cost of energy is also a very important factor when the question of sustainability of energy resources which is one reason for the constant research on optimization of gas turbine for power generation [7]. The concern over cost of utilization of fossil fuels by thermal power plants has led gas turbine manufacturers to seek ways to improve on the efficiency of gas turbine combustion [8].

The need for sustainability of the energy solutions have introduced some factors which are; efficiency, cost effectiveness, effective utilization of the resources, better design and analysis, energy security and a better environment [9]. In recent times, exergy-based cost analysis and optimization of thermal systems are the main focus for the solution of finding the optimal energy consumption [10]. The process of improving the efficiency of a gas turbine considers the costs involved in employing the gas turbine and the energy available. The ratio of the thermodynamic loss rate to the capital cost is a significant parameter in evaluating plant performance [10]. An exergy-based cost analysis method is a means by which the costs of products and the irreversibilities generated in the conversion process can be determined. The process applies the cost partition criterion which is a function of the exergy content of the energy flow taking place in the process [11]. A knowledge of this cost leads to a more efficient performance of the system and a reduced cost of the product. The combination of exergy analysis and economical aspect of the process is referred to as exergo-economic analysis when the cost is in monetary terms and exergy analysis when the cost is in exergy terms [12]. This exergo-economic tool is applied to a simple cycle plant and a modified cycle plant derived from the simple cycle plant to ascertain which plant performs better and hence advice plant operators appropriately. This becomes imperative as gas turbine power plants in Nigeria operate on the simple cycle basis which consumes more fuel.

METHODOLOGY

A model of a simple gas turbine cycle and a modified gas turbine cycle with intercooling, reheat and regeneration were developed to carry out the energy, exergy and exergo-economic analysis. The simple cycle gas turbine cycle is a 25MW GE MS5001 engine located at Trans Amadi power station. The gas turbine power plant in view consists of a single shaft with an air inlet, compressor, combustion chamber (CC), a turbine, exhaust and support systems. The block diagram of the engine is shown in Figure-1.

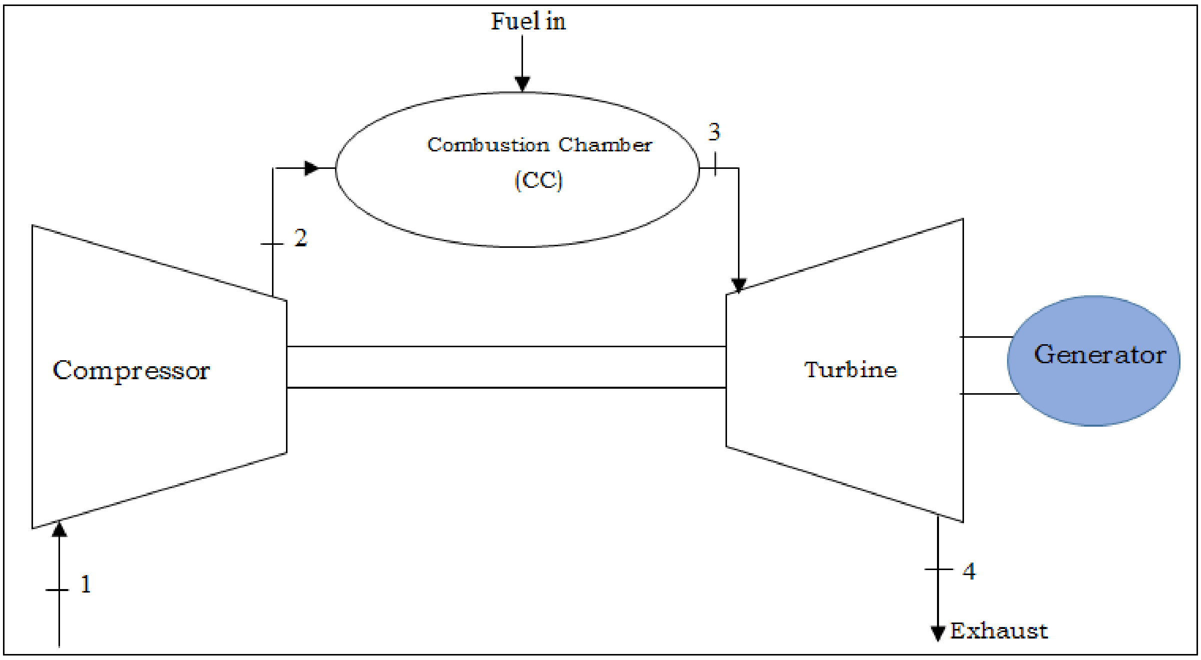


Fig-1: Block diagram of a simple gas turbine cycle

The modified gas turbine cycle with intercooling, reheating and regeneration achieves higher compression ratio by making use of two compressors and a heat exchanger (intercooler) located in between the two compressors. To further harvest the heat released at the exhaust, a regenerator is installed after the exhaust to raise the temperature of the compressed air leaving the second stage compressor before it enters the combustion chamber. The regenerator and the intercooler are both heat exchangers denoted as REG and INC respectively in this work. Block diagram of the modified gas turbine engine cycle is shown in Figure-2.

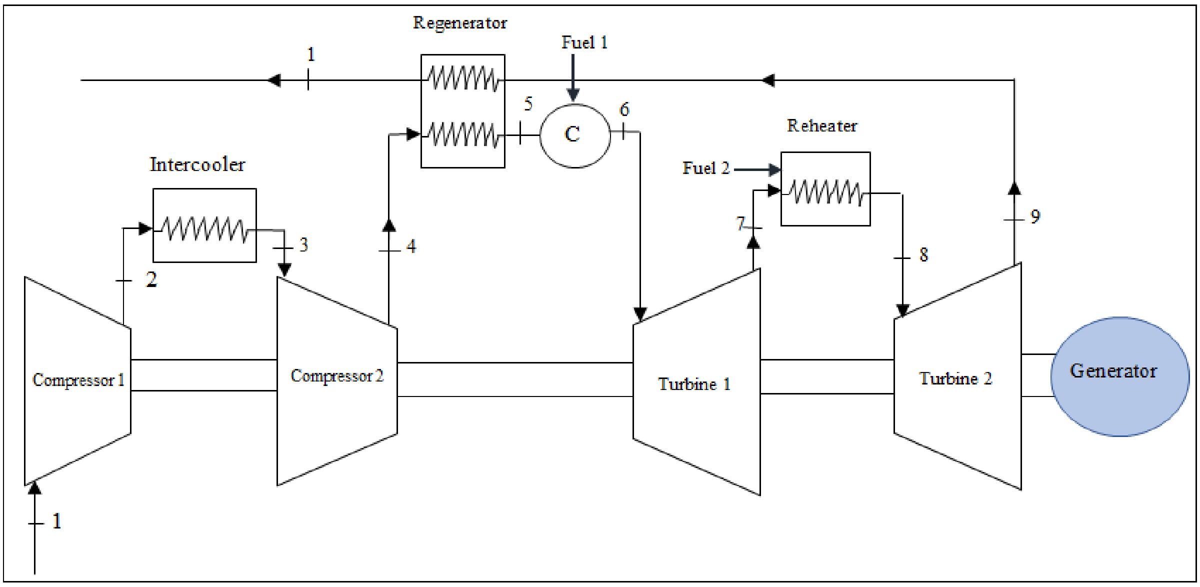


Fig-2: Block diagram of a modified gas turbine cycle with intercooling, regeneration and reheating

To carry out a successful system modeling, there are certain assumptions that are made. These include negligible kinetic and potential energy losses, steady state operations, complete combustion and fuel and air as ideal gases [13, 14]. Pressure drop in the CC and reheater is assumed as 5%.

Thermodynamic Analysis of a Gas Turbine Power Plant

The energy balance of the control volume is given as [15];

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

Where $\dot{E}(kJ)$ is the total energy in the system, $\dot{W}(kW)$ is the work done, $\dot{Q}(kW)$ is the heat transfer rate, $h\left(\frac{kJ}{kg}\right)$ is the specific enthalpy, $v(\frac{m}{s})$ is the velocity, $g\left(\frac{m}{s^2}\right)$ is the acceleration due to gravity and $z(m)$ is the height. The subscripts in Equation (1) describe the inlet (in) and outlet (out) energy respectively [16]. The compressor exit temperature $T_{AC,e}$ and turbine exit temperature $T_{GT,e}$ are obtained from Equations (2) and (3) respectively,

$$T_{AC,e} = T_{inlet}\left\{1 + \left(\frac{r_p^{\frac{1.4-1}{1.4}} - 1}{\eta_{AC}}\right)\right\} \quad .... (2)$$

$$T_{GT,e} = T_{TIT}\left\{1 - \eta_{GT}\left(1 - \left(\frac{P_3}{P_4}\right)^{\frac{1.4-1}{1.4}}\right)\right\} .... (3)$$

Where T_{inlet} is the compressor inlet temperature, T_{TIT} is the turbine inlet temperature, η_{AC} is the isentropic efficiency of the air compressor, η_{GT} is the isentropic efficiency of the gas turbine. The pressure at the exit of the combustion chamber $P_{e,cc}$ is expressed as [17];

$$P_{e,cc} = P_{i,cc}(1 - \Delta P_{cc}) \quad .... (4)$$

Where $P_{i,cc}$ is the inlet pressure to the combustion chamber and ΔP_{cc} = 5% is the percentage pressure drop across the combustion chamber.

Energy balance for the Simple Gas Turbine Cycle

Turbines and compressors are steady flow devices. A balance between the energy in the fluid entering into the compressor and that exiting the compressor will give the compressor work,

$$\dot{W}_{AC} = \dot{m}_a(h_2 - h_1) = \dot{m}_a c_{p,a}(T_2 - T_1) \quad .... (5)$$

Where $\dot{m}_a$(kg/s) is the mass flow rate of the air entering into the air compressor, T_1 is the compressor inlet temperature, $c_{p,a}$ is the specific heat capacity of air. The heat added into the combustion chamber $\dot{Q}_{CC}$ is,

$$\dot{Q}_{CC} = \dot{m}_f H_f(1 - \eta_{cc}) = \dot{m}_{fg}h_3 - \dot{m}_a h_2 \equiv \dot{m}_{fg}c_{p,g}T_3 - \dot{m}_a c_{p,a}T_2 \quad .... (6)$$

$$\dot{m}_{fg} = \dot{m}_f + \dot{m}_a \quad .... (7)$$

Where $c_{p,g}\left(\frac{kJ}{KgK}\right)$ is the specific heat capacity flue gas, η_{cc} is the efficiency of the combustion process, $\dot{m}_{fg}$ and $\dot{m}_f$ are the mass flow rate of the flue gas and fuel respectively. The work done by the turbine $\dot{W}_{GT}$ is,

$$\dot{W}_{GT} = \dot{m}_{fg}(h_3 - h_4) = \dot{m}_{fg}c_{p,g}(T_3 - T_4) \quad .... (8)$$

The first law efficiency for simple cycle, $\eta_{1,SC}$ is given as;

$$\eta_{1,SC} = \frac{(\dot{W}_{GT} - \dot{W}_{AC})}{\dot{Q}_{net,CC}} \quad .... (9)$$

Energy Balance in the Modified Gas Turbine Cycle

The work consumed by the compressors is expressed as,

$\dot{W}_{ACk} = \dot{m}_a(h_e - h_i) = \dot{m}_a c_p(T_e - T_i)$ …………………. (10)

Where the subscript *e* and *i* represents the exit and inlet which corresponds to the exit and entry points of the 1st and 2nd stage compressors; *k* = 1, 2.

The energy balance in the intercooler is expressed as,

$\dot{m}_a c_{p,a}\epsilon_{IC}(T_2 - T_3) - \dot{m}_w c_{p,w}(T_b - T_a) = 0$ ………………… (11)

Where T_b and T_a are the temperature of the cooling water out and into the intercooler respectively and ϵ_{IC} is the effectiveness of the intercooler. The effectiveness of a regenerator ϵ_{Reg} is similar to that of the intercooler and is expressed as,

$\epsilon_{Reg} = \frac{h_5 - h_4}{h_9 - h_4} = \frac{T_5 - T_4}{T_9 - T_4}$ …………………. (12)

The effectiveness of the regenerator will be assumed to be 0.85 [15].

The energy balance in the CC is expressed as,

$\dot{Q}_{CC} = \dot{m}_{f1}H_f(1 - \eta_{cc}) = \dot{m}_{fg}h_6 - \dot{m}_a h_5 \equiv \dot{m}_{fg}c_{p,g}T_6 - \dot{m}_a c_{p,a}T_5$ …………….. (13)

Where $\dot{Q}_{CC}$, is the heat added to the system, H_f is the heating value of the fuel and $\dot{m}_{f1}$ is the mass flow rate of fuel entering the combustion chamber.

The work done by the turbines $\dot{W}_{GTk}$ is expressed as,

$\dot{W}_{GTk} = \dot{m}_{fg,k}(h_i - h_e) \equiv \dot{m}_{fg,k}c_{p,g}(T_i - T_e)$ …………………. (14)

The subscripts *e* and *i* represent the exit and the inlet of the gas turbine respectively and *k* represents the 1st or 2nd gas turbine. $\dot{m}_{fg}$ is the mass flow rate of the flue gasses after the combustion chamber.

The energy balance in the reheater is,

$\dot{Q}_{RCC} = \dot{m}_{f2}H_f(1 - \eta_{RCC}) = \dot{m}_{fg}(h_8 - h_7) \equiv \dot{m}_{fg}c_{p,g}(T_8 - T_7)$ ……………… (15)

Where $\dot{Q}_{RCC}$ is the heat added in the reheater, $\dot{m}_{f2}$ is the mass flow rate of the fuel into the reheater.

The efficiency of the modified cycle $\eta_{1,MC}$ is given as;

$\eta_{1,MC} = \frac{(\dot{W}_{total,GT} - \dot{W}_{total,AC})}{\dot{Q}_{total,CC}}$ ……………………. (16)

Where $\dot{W}_{total,GT}$ is the total work output from the two turbines, $\dot{W}_{total,AC}$ is the total work consumed by the two compressors and $\dot{Q}_{total,CC}$ is the total heat added to the system at the combustion chamber and the reheater.

Exergy Analysis

Considering steady flow condition, the exergy balance for a control volume is given as [18];

$\dot{E}x_{in} - \dot{E}x_{out} - \dot{E}x_D = \Delta\dot{E}x$ ………………… (17)

$\sum\left(1 - \frac{T_o}{T_k}\right)\dot{Q}_k - \dot{W} - P_oV - T_o\dot{S}_{gen} = \Delta\dot{E}x$ ……………….. (18)

Where $\dot{E}x_{in}$ and $\dot{E}x_{out}(kW)$ are the exergy rates at the inlet and exit , $\Delta\dot{E}x$ is the stream exergy rate, $\dot{Q}_k(kW)$is the rate of heat transfer which occurs at the boundary of the control volume, $T_k(K)$ is the temperature at which the heat transfer takes place, $\dot{W}(kW)$ is the work transfer rate, $\dot{E}x_D$ is the rate of exergy destroyed in the system, $\dot{S}_{gen}$ is the entropy generated in the component, P_o is the ambient pressure and T_o is the ambient temperature.

$\dot{E}x_D = T_o\dot{S}_{gen}$……………………….. (19)

The exergy of the stream is divided into physical, chemical, kinetic and potential exergy denoted as $\dot{E}x_{ph}, \dot{E}x_{ch}, \dot{E}x_{ke}$ and $\dot{E}x_{pe}$ respectively.

$\dot{E}x = \dot{E}x_{ph} + \dot{E}x_{ch} + \dot{E}x_{ke} + \dot{E}x_{pe}$ ………………… (20)

The physical exergy $\dot{E}x_{ph}$ is expressed as

$\dot{E}x_{ph} = \dot{m}_k\{(h_k - h_o) - T_o(s_k - s_o)\}$……………………… (21)

The chemical exergy expression as;

$\dot{E}x_{ch} = \dot{m}_{fuel}\bar{h}_{oi} + \dot{m}_{fuel}R_{fuel}T_0\sum\alpha_i ln\,\alpha_i$ …………… (22)

$\bar{h}_{oi} = \sum\alpha_i h_{oi}$ ………………………………… (23)

Where $h_k\left(\frac{kJ}{kg}\right)$ and $h_o\left(\frac{kJ}{kg}\right)$ are the specific enthalpy, $s_k\left(\frac{kJ}{kgK}\right)$ and $s_o\left(\frac{kJ}{kgK}\right)$ are the specific entropy, $\overline{Ex}_{ch,i}(\frac{kJ}{kg})$ is the standard chemical exergy of the i-th component of the (fuel, air) and $x_i(-)$ is the mole fraction of the ith component which is applicable to mixture of gasses such as natural gas, i = 1,2,3,…, T_0(K) is the absolute temperature of the environment, R_{fuel} is the mean gas constant of the fuel, α_i(-) is the mole fraction, $\bar{h}_{oi}\left(\frac{kJ}{kmol}\right)$ is the mean heat of formation of the fuel, $h_{oi}\left(\frac{kJ}{kmol}\right)$ is the heat of formation of the fuel.

Chemical exergy for the fuel can be estimated as [17];

$ex_{ch,f}=\xi LHV_f$ …………………… (24)

Where LHV_f is the lower heating value of the fuel, ξ is the ratio of chemical exergy to LHV_f and for methane it is given as $\xi_{CH_4} = 1.06$.

Exergo-economic Analysis

The exergo-economic analysis procedure involves forming a set of simultaneous equations where the costs of the various streams are estimated. The costs of the equipment of the gas turbine system are first estimated followed by cost balance which leads to the formation of set of simultaneous equations.

Purchase Equipment Cost

Purchase equipment cost (Z_k) is the cost of purchase of the equipment which is required when carrying out an exergo-economic analysis. Economic models which are used to express the cost of the components as a function of their thermodynamic variables are presented in [16, 20, 21]. The PECs for the components of the gas turbine are given by Equations (28) to (31),

$$PEC_{AC} = \left(\frac{c_{1,1}\dot{m}_a}{c_{1,2}-\eta_{AC}}\right)\left(\frac{P_e}{P_i}\right) ln\left(\frac{P_e}{P_i}\right) \quad \text{.....................} (28)$$

Where $c_{1,1} = 75\$/(\frac{kg}{s})$, $c_{1,1} = 0.9$, P_e and $P_i(kPa)$ are the exit and inlet pressures of the air compressor respectively, Z_{AC} is the purchase cost of the air compressor.

$$PEC_{CC} = \left(\frac{c_{2,1}\dot{m}_a}{c_{2,2}-\frac{P_e}{P_i}}\right)\left[1+exp\left(c_{2,3}T_e - c_{2,4}\right)\right] \quad \text{…………………..} (29)$$

Where $c_{2,1} = 48.64\ \$/(\frac{kg}{s})$, $c_{2,2} = 0.995$, $c_{2,3} = 0.018K^{-1}$, $c_{2,4} = 26.4$, T_e is the exit temperature from the combustion chamber.

$$PEC_{GT} = \left(\frac{c_{3,1}\dot{m}_g}{c_{3,2}-\eta_{GT}}\right) ln\left(\frac{P_i}{P_e}\right)\left[1+exp\left(c_{3,3}T_i - c_{3,4}\right]\right. \quad \text{…………………} (30)$$

Where $c_{3,1} = 479.34\frac{\$}{\frac{kg}{s}}, c_{3,2} = 0.92, c_{3,3} = 0.036\ K^{-1}, c_{3,4} = 54.4$

The expression for PEC_{HE} is given as [13];

$$Z_{HE} = PEC_{HE} = 4.122\left(\frac{\dot{m}_{fg}c_{p,i}}{18\Delta T_{LM,HE}}\right)^{0.6} \quad \text{……………………} (31)$$

Where i=1,2 which represents the specific heat for the flue gas in the regenerator and the cooling water in the intercooler.

To convert the capital investment to cost per unit time, Equation (32) is applied;

$$\dot{Z}_k = \frac{\dot{C}_k.CRF\varphi}{N} \quad \text{…………………………………} (32)$$

Where φ(-) is the maintenance factor given as 1.06, N(hr) represents the operating hours per year which is 8000 in this work and CRF (-) is the capital recovery factor [22].

Cost Balance

The cost balance expresses the summation of the cost of the streams exiting the component and the streams entering the component plus the charges due to the capital investment of the component being analyzed ($\dot{Z}_k^{CI}$), and its operation and maintenance cost ($\dot{Z}_k^{O\&M}$). Mathematical relatiohsip is expressed as [13],

$$\sum_e \dot{C}_{e,k} + \dot{C}_{w,k} = \dot{C}_{q,k} + \sum_i \dot{C}_{i,k} + \dot{Z}_k \quad \text{……………………………………………} (33)$$

$$\sum_e (c_e\dot{E}_e)_k + c_{w,k}\dot{W}_k = c_{q,k}\dot{E}_{q,k} + \sum_i (c_i\dot{E}_i)_k + (\dot{Z}_k^{CI} + \dot{Z}_k^{O\&M}) \quad \text{…………………} (34)$$

The cost rates associated with the exiting and entering stream is;

$$\dot{C}_j = c_j\dot{E}_j \quad \text{………………………………} (35)$$

Where $\dot{C}_e, \dot{C}_w, \dot{C}_i, \dot{C}_q \left(\frac{\$}{s}\right)$ are the cost rates associated with the entering and exiting streams of power and heat transfer while $\dot{E}_e, \dot{E}_i, \dot{E}_q$ and $\dot{W}_k$ are the exergy rates of the exiting and entering streams of power and heat transfer, k represents the component while $c_i, c_e, c_w, c_q \left(\frac{\$}{kJ}\right)$ represents the average cost per unit exergy of the streams exiting and entering the component k.

The cost balance equations will be applied to both the simple and modified cycles. For components with n exiting streams, the rule is that n-1 auxiliary equations will be required for that component [13]. Using the fuel (F) principle and product (P) principle of the specific exergy costing method, SPECO [23], the cost balance equations are formulated.

Cost Balance Equations of the Simple Cycle

Air compressor

$$\dot{C}_2 = \dot{C}_1 + \dot{C}_6 + \dot{Z}_{AC} \quad \text{…………………}(36)$$

Where $\dot{C}_6$ represents the compressor work.

Combustion chamber

$$\dot{C}_3 = \dot{C}_2 + \dot{C}_5 + \dot{Z}_{cc} \quad \text{……………………..} (37)$$

$\dot{C}_5$ is the cost rate of the fuel (methane) which is obtained as [24];

$$\dot{C}_f = \dot{C}_5 = c_f\dot{m}_f \times LHV_f \times 3600 \quad \text{……………………………} (38)$$

Where the fuel cost per energy unit is c_f = 0.004$/MJ, $\dot{m}_f$ is the mass flow rate of fuel and LHV_f is the lower heating value of fuel.

Gas turbine

$$\dot{C}_3 + \dot{Z}_{GT} = \dot{C}_4 + \dot{C}_6 + \dot{C}_7 \quad \text{(39)}$$

Auxilliary Equations for Simple Cycle;

We assume the same unit cost of exergy for the net power exported from the system and the compressor work.

$$\frac{\dot{C}_5}{\dot{W}_{AC}} = \frac{\dot{C}_6}{\dot{W}_n} \quad \text{(40)}$$

Where $\dot{W}_n$ is the net power exported from the system to do work. The cost of air entering into the compressor is assumed to be zero;

$$\dot{C}_1 = 0 \quad \text{(41)}$$

$$\frac{\dot{C}_5}{\dot{Ex}_5} = \frac{\dot{C}_6}{\dot{Ex}_6} \quad \text{(42)}$$

Cost Balance Equations for the Modified Cycle

The cost balance equations for the modified cycle were generated for the components of the modified cycle gas turbine. To carry out the cost balance, net work from GT1, GT2 and compressor work for AC1 and AC2 were designated with subscript 15, 16, 17 and 18 respectively to aid the computation of the cost rates $\dot{C}_i$. The components are analyzed in the order of flow from state 1.

Air compressor 1

$$\dot{C}_1 + \dot{C}_{15} + \dot{Z}_{AC1} = \dot{C}_2 \quad \text{(43)}$$

Air compressor 2

$$\dot{C}_3 + \dot{C}_{16} + \dot{Z}_{AC2} = \dot{C}_4 \quad \text{(44)}$$

Intercooler

$$\dot{C}_2 + \dot{C}_{13} + \dot{Z}_{Int} = \dot{C}_3 + \dot{C}_{14} \quad \text{(45)}$$

Regenerator

$$\dot{C}_4 + \dot{C}_9 + \dot{Z}_{Reg} = \dot{C}_5 + \dot{C}_{10} \quad \text{(46)}$$

Combustion Chamber

$$\dot{C}_5 + \dot{C}_{11} + \dot{Z}_{cc} = \dot{C}_6 \quad \text{(47)}$$

Gas turbine 1

$$\dot{C}_6 + \dot{Z}_{GT1} = \dot{C}_7 + \dot{C}_{15} + \dot{C}_{17} \quad \text{(48)}$$

Reheater

$$\dot{C}_7 + \dot{C}_{12} + \dot{Z}_{RH} = \dot{C}_8 \quad \text{(49)}$$

Gas turbine 2

$$\dot{C}_8 + \dot{Z}_{GT2} = \dot{C}_9 + \dot{C}_{16} + \dot{C}_{18} \quad \text{(50)}$$

Auxilliary Equations for Modified Cycle

$$\dot{C}_1 = 0 \quad \text{(51)}$$

$$\frac{\dot{C}_2}{\dot{Ex}_2} = \frac{\dot{C}_3}{\dot{Ex}_3} \quad \text{(52)}$$

$$\dot{C}_{13} = 0 \quad \text{(53)}$$

Where $\dot{C}_1$ and $\dot{C}_{13}$ are the cost of air and water entering the compressor and intercooler respectively.

$$\frac{\dot{C}_9}{\dot{Ex}_9} = \frac{\dot{C}_{10}}{\dot{Ex}_{10}} \quad \text{(54)}$$

$$\dot{C}_{11} = c_f \dot{m}_{f1} \times LHV_f \times 3600 \quad \text{(55)}$$

$$\frac{\dot{C}_6}{\dot{Ex}_6} = \frac{\dot{C}_7}{\dot{Ex}_7} \quad \text{(56)}$$

$$\frac{\dot{C}_{15}}{\dot{W}_{AC1}} = \frac{\dot{C}_{17}}{\dot{W}_{n,1}} \quad \text{(57)}$$

$$\dot{C}_{12} = c_f \dot{m}_{f2} \times LHV_f \times 3600 \quad \text{(58)}$$

$$\frac{\dot{C}_8}{\dot{Ex}_8} = \frac{\dot{C}_9}{\dot{Ex}_9} \quad \text{(59)}$$

$$\frac{\dot{C}_{16}}{\dot{W}_{AC2}} = \frac{\dot{C}_{18}}{\dot{W}_{n,2}} \quad \text{(60)}$$

These equations put together to form a system of linear equations which can be represented in a matrix form as in Equation (61) to solve for the cost of the streams [25];

$$[\dot{E}_k] \times [c_k] = [\dot{Z}_k] \quad \text{(61)}$$

Where $[\dot{E}_k]$ represents the exergy rates which were obtained in the exergy analysis carried out previously, $[c_k]$ is the vector representing the exergetic costs and $[\dot{Z}_k]$ is the vector containing the costs of equipment. In the simple cycle, there are a total of 7 exergetic cost terms hence 7 equations could be formulated while in the modified there will be 18 equations as there are 18 exergetic cost terms. There formulated equations were solved in MATLAB.

Exergo-economic Variables

To evaluate thermal systems using exergo-economic analysis, certain quantities play major roles. These are, the average cost of fuel for the kth component ($\dot{c}_{f,k}$), the average cost of product for the kth component ($\dot{c}_{p,k}$), the cost rate of the exergy destroyed in each component ($\dot{C}_{D,k}$), and the exergo-economic factor f_k [26]. These parameters are defined as follows:

$$\dot{c}_{f,k} = \frac{\dot{C}_{f,k}}{\dot{E}_{f,k}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \ (62)$$

$$\dot{c}_{p,k} = \frac{\dot{C}_{p,k}}{\dot{E}_{p,k}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \ (63)$$

$$\dot{C}_{D,k} = \dot{c}_{f,k}\dot{E}_{D,k} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \ (64)$$

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + \dot{C}_{D,k}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \ (65)$$

Results and Discussion

The thermodynamic characteristics of the simple cycle gas turbine plant are presented in Table 1 while those of the modified cycle plant are in Table 2. The simple cycle gas turbine plant has a total work output of 26930kW and thermal efficiency of 30.4% while the modified gas turbine cycle has a net power output of 40,611.22kW and thermal efficiency of the 39.9%. This shows that the modified gas turbine cycle urns less fuel to produce a unit amount of electrical power. The exergetic efficiency showed the SGTC is 22.62% while that of the MGTC is 21.7%. The net work output, compressor work, turbine work and heat added to the modified cycle are higher than those of the simple cycle. Also, the exergetic efficiency of the modified is higher than that of the simple cycle.

Table-1: Thermodynamic characteristics of the simple gas turbine cycle

S/No	Parameter	Symbol	Unit	Value
1.	Net work	$\dot{W}_{net}$	kW	26,930.00
2.	Turbine Work	$\dot{W}_{GT}$	kW	67,038.67
3.	Compressor Work	$\dot{W}_{AC}$	kW	40,108.67
4.	Heat Added	$\dot{Q}_{in}$	kW	88,480.79
5.	Energy efficiency	$\eta_{I,SCGT}$	-	30.4%
6.	Exergy efficiency	$\eta_{II,SCGT}$	-	22.6%
7.	Back Work Ratio	BWR	-	59.8%
8.	Mass flow rate (fuel)	$\dot{m}_f$	kg/s	2.2205

Table-2: Thermodynamic analysis of the modified gas turbine cycle

S/No	Parameter	Symbol	Unit	Value
1.	Net work	$\dot{W}_{net}$	kW	44670.88
2.	Turbine Work	$\dot{W}_{GT,total}$	kW	78950.98
3.	Compressor Work	$\dot{W}_{AC,total}$	kW	34280.11
4.	Heat Added	$\dot{Q}_{in}$	kW	93334.03
5.	Energy efficiency	$\eta_{I,MCGT}$	-	48%
6.	Exergy efficiency	$\eta_{II,MCGT}$	-	43%
7.	Back Work Ratio	BWR	-	43.4%
8.	Mass flow rate of fuel (CC)	$\dot{m}_{f1}$	kg/s	1.078
9.	Mass flow rate of fuel (RCC)	$\dot{m}_{f2}$	kg/s	0.879

The result for the exergy analysis carried out on the simple gas turbine cycle is shown in the Table 3 and that of the MGTC is shown in Table-4. Exergy destruction rate between the components of the SGTC when compared with that of the MGTC showed that the highest exergy efficiency occurred at the turbine in both cycles; 95.40% in the SGTC and 97.00% in GT1 and GT2 in the MGTC. The exergy destruction rate on the other hand was highest in the combustion chamber of both the simple and the modified gas turbine cycle.

Table-3: Exergy analysis of the simple gas turbine cycle

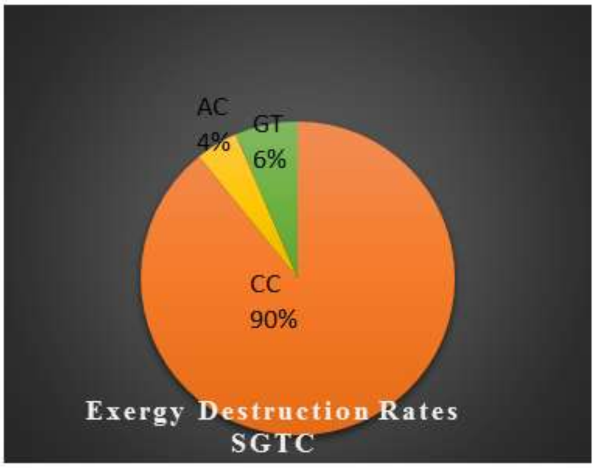
S/No	Component	Heat/Work rate $\dot{Q}_k/\dot{W}_k(kW)$	Exergy destroyed $\dot{Ex}_D(kW)$	Exergy efficiency $\eta_{II}(\%)$
1.	Air Compressor	40108.67	2959.08	92.60%
2.	Turbine	67038.67	4381.54	95.40%
3.	Combustion Chamber	88480.79	60342.99	61.40%

Table-4: Exergy analysis of the modified gas turbine cycle

S/No	Component	Symbol	Heat/Work rate $\dot{Q}_k/\dot{W}_k(kW)$	Exergy destroyed $\dot{Ex}_D(kW)$	Exergy efficiency $\eta_{II}(\%)$
1.	Air Compressor 1	AC1	16938.54	1904.39	89.00
2.	Air Compressor 2	AC2	17341.57	1904.39	89.00

3.	Combustion Chamber	CC	52845.56	20884.24	82.00
4.	Reheater	RH	40488.47	17856.10	82.00
5.	Gas Turbine 1	GT1	39336.19	1379.48	97.00
6.	Gas Turbine 2	GT2	39614.80	1389.25	97.00
7.	Intercooler	INC	16938.54	369.42	88.00
8.	Regenerator	REG	56057.75	3254.35	90.00

Figure-3 shows the exergy destruction rates among the components in both cycles. The lowest exergy destruction rate occurred in the air compressor for the SGTC at 4% and for the MGTC it occurred at the intercooler at 1%. This occurrence is dictated by the operating temperatures in the components.



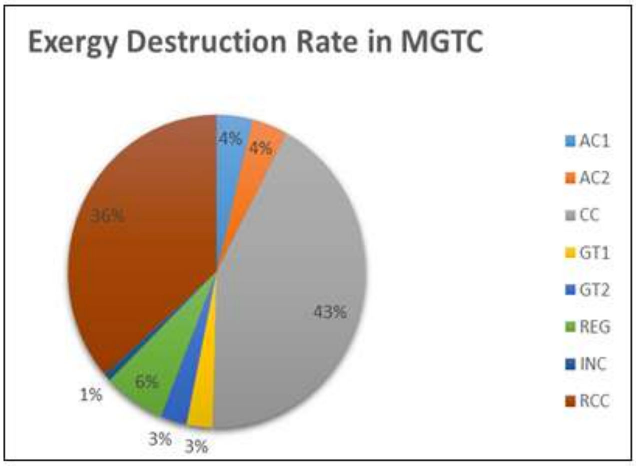


Fig-3: Distribution of exergy destruction rates among components of SGTC and MGTC

The highest exergy destruction rate occurs in the CC in both cycles but it is as high as 90% in the CC as there is no other high temperature component in the SGTC (in the MGTC, there is a reheater which takes 36% of the total exergy destruction rate in the cycle).

The PECs of the SGTC are shown in Table-5 while those of the MGTC are presented in Table-6. The most expensive component in the simple cycle gas turbine is the compressor which cost $4317629.06 while for the modified cycle, the gas turbines are the most expensive; $1671953.98 and $1683796.12. The total purchase equipment cost for the simple gas turbine cycle was $7898754.10 which is higher than that of MGTC with PEC of $5044774.45. This occurrence is because the PEC depends on the pressure ratio and the compressors and the turbines operate at much lower pressure ratios in the MGTC. But, it will take longer and hence cost more to install a MGTC plant as against a SGTC plant.

Table-5: Initial investment, monetary flow rates and levelized capital cost rates for SGTC components

S/No	Component (k)	PEC_k (\$)	$\dot{C}_k(\$/year)$	$\dot{Z}_k(\$/h)$
1.	Air Compressor	4317629.06	803251.22	106.43
2.	Combustion chamber	130127.97	24208.99	3.21
3.	Gas Turbine	3450997.07	642023.10	85.07
	Total	7898754.10	1469483.31	194.71

Table-6: Initial investment, monetary flow rates and levelized capital cost rates for the MGTC

S/No	Component (k)	$\boldsymbol{PEC_k}$ (\$)	$\boldsymbol{\dot{C}_k(\$/year)}$	$\boldsymbol{\dot{Z}_k(\$/h)}$
1.	Air Compressor 1	666224.63	123944.4	16.42
2.	Air Compressor 2	666224.63	123944.4	16.42
3.	Combustion chamber	130127.97	24208.99	3.21
4.	Reheater	131269.14	24421.30	3.23
5.	Gas Turbine 1	1671953.98	311050.10	41.21
6.	Gas Turbine 2	1683796.12	313253.20	41.51
7.	Regenerator	35033.58	6517.64	0.86
8.	Intercooler	60144.40	11189.26	3.24
	Total	5044774.45	938529.29	126.1

The levelized cost rates obtained from the solutions of the simultaneous equations for the cycles are shown in Tables 7 and 8 respectively. The results revealed that the exergy costing method gave 10.303$/GJ for the product (electricity) in the simple cycle gas turbine while that of the MGTC ranged from 7.3339$/GJ and 7.7969$/GJ. This is because less fuel is burnt to produce to produce unit amount of power in the MGTC as against the.

The exergo-economic parameters obtained for the SGTC and MGTC are presented in Tables 8 and 9 respectively. The results are arranged in descending order of $\dot{Z}_k + \dot{C}_{D,k}$. In the simple cycle, the combustion chamber has the highest value for $\dot{Z}_k + \dot{C}_{D,k}$ ($/h) and also it has a low rate of exergo-economic factor f_k. The exergo-economic factors for the compressor and gas turbine fall between 35% to 75% which is the typical value for compressors and turbines [13]. In the modified cycle, the combustion chamber and the reheater have the highest value of $\dot{Z}_k + \dot{C}_{D,k}$ and the lowest values of f_k occurred in the combustion chamber. The exergo-economic factors for the gas turbines are 53.37% and 54.87% respectively. The values for the air compressors are the same; 23.50% for both compressors which falls below the typical value for compressors. Thus, increasing the capital investment of the component in order to improve the compressor efficiency should be considered [13].

Table-6: Levelized cost rates and average cost per unit of exergy at different state points in the SGTC

S/No	$\dot{C}\left(\frac{\$}{h}\right)$	$c\left(\frac{\$}{kWh}\right)$	$c\left(\frac{\$}{GJ}\right)$
1	0.00	0.00	0.00
2	1516.63	0.0408	11.340
3	3118.62	0.0325	9.039
4	794.68	0.0325	9.039
5	1598.78	0.0134	3.731
6	1410.20	0.0371	10.303
7	998.81	0.0371	10.303

Table-7: Levelized cost rates and average cost per unit of exergy for MGTC

S/No	$\dot{C}\left(\frac{\$}{h}\right)$	$c\left(\frac{\$}{kWh}\right)$	$c\left(\frac{\$}{GJ}\right)$
1	0.0000	0.0000	0.0000
2	491.8678	0.0327	9.0880
3	391.3171	0.0327	9.0880
4	865.592	0.0316	8.7759
5	1699.488	0.0293	8.1339
6	2479.202	0.0261	7.2516
7	1416.295	0.0261	7.2516
8	2052.18	0.0246	6.8270
9	1044.421	0.0246	6.8270
10	211.3884	0.0246	6.8270
11	776.5062	0.0134	3.7309
12	632.649	0.0134	3.7309
13	0.0000	0.0000	0.0000
14	102.0333	0.0377	10.4819
15	475.4452	0.0281	7.7969
16	457.8523	0.0264	7.3339
17	628.6758	0.0281	7.7969
18	591.4128	0.0266	7.3757

Table-8: Exergo-economic parameters for the SGTC

S/No	K	c_p ($/GJ)	c_f($/GJ)	$\dot{E}_{D,k}$ (MJ/s)	$\dot{C}_{D,k}$ ($/h)	$\dot{Z}_k$ ($/h)	$\dot{Z}_k + \dot{C}_{D,k}$ ($/h)	f_k(%)
1	CC	9.04	5.54	60.34	1203.65	3.21	1206.86	0.27
2	GT	10.30	9.04	4.38	142.57	85.07	227.64	37.37
3	AC	11.34	10.30	2.96	109.75	106.43	216.18	49.23

Table-9: Exergo-economic parameters for MGTC

S/No	k	c_p (\$/GJ)	c_f(\$/GJ)	$\dot{E}_{D,k}$ (MJ/s)	$\dot{C}_{D,k}$ (\$/h)	$\dot{Z}_k$ (\$/h)	$\dot{Z}_k + \dot{C}_{D,k}$ (\$/h)	f_k(%)
1.	CC	7.2516	5.9367	20.8842	446.34	3.2077	449.5449	0.7135
2.	RCC	6.8270	5.6154	17.8561	360.97	3.2358	364.2031	0.8885
3.	REG	7.5599	6.8270	3.2543	79.98	0.8636	80.8459	1.0682
4.	INT	9.0880	9.0880	0.3694	12.0861	1.4826	13.5686	10.9265
5.	AC1	9.0880	7.7969	1.9044	53.4541	16.4226	69.8767	23.5023
6.	AC2	8.5341	7.7969	1.9044	53.4541	16.4226	69.8767	23.5023
7.	GT1	7.7963	7.2516	1.3795	36.01	41.2141	77.2262	53.3681
8.	GT2	7.3339	6.8270	1.3892	34.1436	41.5061	75.6497	54.8700

CONCLUSION

In this research, comparative exergo-economic analysis was carried out on simple cycle and modified cycle gas turbine power plants. The analysis carried out on both cycles revealed that both the first law efficiency and the second law efficiency of the modified cycle plant are higher than those of the simple plant. This means there is better fuel utilization and lower rate of exergy destruction in the modified cycle plant. The modified cycle gas turbine model produced a higher net-power than the simple cycle gas turbine model. The exergo-economic parameters revealed that the modified gas turbine cycle components showed lower exergy destruction rates. Also, the cost of producing unit amount of electricity is lower in the modified cycle. The largest contributors to the total exergy destruction rates in the modified cycle are the combustion chamber followed by the reheater while that of the simple cycle is the combustion chamber followed by the gas turbine. The highest exergy unit costs in both cycles were achieved at the exits of the air compressors. The comparison between both cycles showed that the modified cycle gas turbine is a more efficient cycle to operate both from the energy, exergy and exergo=economic perspectives.

REFERENCES

1. Breeze, P. (2014). Power Generation Technologies, second edition, Oxford, UK. ed should be deleted.: Elsevier Limited.
2. Kostas, B. (2016). Revisiting the Energy - Development Link Evidence from the 20th Century of Knowledge Based and Developing Economies, no. 1.
3. Sambo, A. S. (2008). Matching Electricity Supply with Demand in Nigeria, International Association of Energy Economics, 32-36.
4. Dincer, I., & Rosen, M. A. (2013). Exergy: Energy, Environment and Sustainability, Oxford: Elsevier.
5. CBN. (2013). CBN 2013 Annual Report, Central Bank of Nigeria.
6. NRC. (2010), Hidden Costs of Energy: Unpriced Consequences of Energy Production and Use, The National Academies Press, Washington DC.
7. Khorasani Nejad, E., Hajabdollahi, F., Hajabdollahi, Z., & Hajabdollahi, H. (2013). Thermo- economic optimization of gas turbine power plant with details in intercooler. *Heat Transfer—Asian Research*, *42*(8), 704-723.
8. McMillan, R., Martin, P., Noden, R., & Welch, M. (2004). Gas Fuel Flexibility in a Dry Low Emissions Combustion System. *Demag Delaval Industrial Turbomachinery Ltd*.
9. Dincer, I., & Zamfirescu, C. (2011). Sustainable Energy Systems and Applications, Newyork: Springer.
10. Rosen, M. A., & Dincer, I. (2003). Exergoeconomic analysis of power plants operating on various fuels. *Applied Thermal Engineering*, *23*(6), 643-658.
11. Oliveira, S. D. (2012). Exergy: Production, Cost and Renewability, Springer Science and Business Media.
12. Ahmadi, A., & Dincer, I. (2018). Exergo-economics, Comprehensive Energy Systems, 1, 340-376.
13. Bejan, A., Tsatsaronis, G., & Moran, M. (1996). Thermal Design and Optimization, John Wiley and Sons Incorporated, NY.
14. Khani, L., Mahmoudi, S. M. S., Chitsaz, A., & Rosen, M. A. (2016). Energy and exergoeconomic evaluation of a new power/cooling cogeneration system based on a solid oxide fuel cell. *Energy*, *94*, 64-77.
15. Cengel, Y. A., & Boles, M. A. (2014). Thermodynamics: An Engineering Approach, 8th ed., McGraw-Hill Publishing Company, NY.
16. Igbong, D., & Fakorede, D. O. (2014). Exergoeconomic analysis of a 100 MW unit GE Frame 9 gas turbine plant in Ughelli, Nigeria. *International Journal of Engineering and Technology*, *4*(8), 463-468.
17. Ahmadi, P., Dincer, I., & Rosen, M. A. (2011). Exergy, exergoeconomic and environmental analyses and evolutionary algorithm based multi-objective optimization of combined cycle power plants. *Energy*, *36*(10), 5886-5898.
18. Oko, C. O. C. (2011). Engineering Thermodynamics: An Algorithmic Approach, Worldwide Communications Limited, Lagos.
19. Moran, M. J. (1982). Availability Analysis: A Guide to Efficient Energy Use, NJ: Prentice - Hall: Englewood Cliffs, 199-215.
20. Shokati, N., Mohammadkhani, F., Yari, M., Mahmoudi, S., & Rosen, M. (2014). A comparative exergoeconomic analysis of waste heat recovery

from a gas turbine-modular helium reactor via organic Rankine cycles. *Sustainability*, *6*(5), 2474-2489.

21. Bagherneiad, A., & Yaghoubi, M. (2011). Exergo-economic Analysis and Optimization of and Intergrated Solar Combined Cycle System (ISCCS) Using Genetic Algorithm. *Journal of Energy Conversion and Management*, 52(3), 2193-2203.
22. Valero, A., Lozano, M. A., Serra, L., Tsatsaronis, G., Pisa, J., Frangopoulos, C., & von Spakovsky, M. R. (1994). CGAM problem: definition and conventional solution. *Energy*, *19*(3), 279-286.
23. Lazzaretto, C., & Tsatsaronis, G. (2006). SPECO: A Systematic and General Methodology for Calculating Efficiencies and Cost in Thermal Systems. Journal of Renewable Energy, 1(3), 65-73.
24. Valero, A., Lozano, M. A., Serra, L., Tsatsaronis, G., Pisa, J., Frangopoulos, C., & von Spakovsky, M. R. (1994). CGAM problem: definition and conventional solution. *Energy*, *19*(3), 279-286.
25. Oyedepo, S. O., Fagbenle, R. O., Adefila, S. S., & Alam, M. M. (2015). Exergoeconomic analysis and performance assessment of selected gas turbine power plants. *World Journal of Engineering*, *12*(3), 283-300.
26. Abam, F. I., Briggs, T. A., Ekwe, E. B., & Effiom, S. O. (2017). Investigation of intercooler-effectiveness on exergo-economic and exergo-sustainability parameters of modified Brayton cycles. *Case Studies in Thermal Engineering*, *10*, 9-18.