Energy Technology

Advanced Exergy and Exergoeconomic Analysis of a Gas Power System with Steam Injection and Air Cooling with a Compression Refrigeration Machine

--Manuscript Draft--

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Keywords:	Exergy, Exergoeconomic; Sting Cycle, Brayton Cycle; Steam Injection; Compression Cooling System
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Author Comments:	Dear Editor: Please, find attached here with a paper entitle “Advanced exergetic and exergoeconomic analysis of a gas power system with steam injection and air cooling with a compression refrigeration machine”. This manuscript contains the results of the conventional, advanced and exergoeconomic exergyic analysis performed at a Thermoelectric Power Plant located in the city of Cartagena Colombia. The analyzes carried out have aimed to determining the influence of water vapor injection in the combustion chamber of a Brayton Cycle and the cooling of the air entering the compressor on the performance of the plant. In addition, this analyze allow to determining the main sources of irreversibilities, the percentages of destroyed exergy avoidable, not avoidable, the factors or criteria that affect these parameters and the most representative costs associated with the process and equipment of the analyzed thermoelectric power plant. With this study, the foundations are laid for the implementation of this type of analysis in thermoelectric plants with the configuration mentioned above. Once, in the review of the scientific literature this type of analysis was not found for the configuration of the thermoelectric power plant analyzed in this work. In this version of the manuscript, the improved version is being presented, taking into account suggestions and comments made by the group of evaluators assigned by such a distinguished journal, in the review process already carried out by its working group. For this review of our work in your journal, we present the reference number (ente.202000993R1), as well as the following documents: word document with the updated version of the work, document with the detailed response of the suggestions

	made by the evaluators assigned to the work. Best regards, Best regards, Ph.D. Gaylord Enrique Carrillo Caballero Universidad Tecnológica de Bolívar (UTB) Cartagena/Bolívar, Colombia.
Additional Information:	
Question	**Response**
Please submit a plain text version of your cover letter here.	Dear Editor: Please, find attached here with a paper entitle “Advanced exergetic and exergoeconomic analysis of a gas power system with steam injection and air cooling with a compression refrigeration machine”. This manuscript contains the results of the conventional, advanced and exergoeconomic exergyic analysis performed at a Thermoelectric Power Plant located in the city of Cartagena Colombia. The analyzes carried out have aimed to determining the influence of water vapor injection in the combustion chamber of a Brayton Cycle and the cooling of the air entering the compressor on the performance of the plant. In addition, this analyze allow to determining the main sources of irreversibilities, the percentages of destroyed exergy avoidable, not avoidable, the factors or criteria that affect these parameters and the most representative costs associated with the process and equipment of the analyzed thermoelectric power plant. With this study, the foundations are laid for the implementation of this type of analysis in thermoelectric plants with the configuration mentioned above. Once, in the review of the scientific literature this type of analysis was not found for the configuration of the thermoelectric power plant analyzed in this work. In this version of the manuscript, the improved version is being presented, taking into account suggestions and comments made by the group of evaluators assigned by such a distinguished journal, in the review process already carried out by its working group. For this review of our work in your journal, we present the reference number (ente.202000993R1), as well as the following documents: word document with the updated version of the work, document with the detailed response of the suggestions made by the evaluators assigned to the work. Best regards,
Do you or any of your co-authors have a conflict of interest to declare?	No. The authors declare no conflict of interest.
This journal's Expects Data Policy requires a Data Availability Statement (even if no data are shared), which will be published alongside your manuscript if it is accepted for publication. Do you choose to share the research data described in this manuscript?	No. Research data are not shared.

Response to Reviewers:	Dear Editor: Journal of Energy Technology We would like to thank you for the opportunity provided to contribute our work to such a prestigious journal. Additionally, we would like to inform you that in this document, we provide answers to some of the questions and doubts raised by the evaluators of our work. We would like to inform you that we are sending a document with the corrections recently sent by the editor of such a prestigious journal Energy Technology. For this reason, some questions raised in this second review will be answered below. Full Paper, Noente.202000993R1 TitleAdvanced exergy and Exergoeconomic Analysis of a Gas Power System with Steam Injection and Air Cooling with a Compression Refrigeration Machine AuthorsDeibys Barreto Author 1, Juan Fajardo Author 2, Gaylord Carrillo Caballero Author 3, Yulineth Cardenas Escorcia Author 4 Submission date (Original Manuscript)2020-07-04 Revision date (Round 1)2020-10-15 -- ente.202000618 (First Review) Revision date (Round 2)2021-01-25 -- ente.202000993 Revision date (Round 3)2021-02-11 -- ente.202000993R1 Correctionsplease verify text highlighted in blue in the manuscript. Reviewer Comments – Reviewer 2 POINT (1) Equations considered to be erroneous in Table 3: 1-CH1 cond fuel and product exergy should be adjusted as follows: EF = E39-E40 EP = E34-E33 2-CH1 evap fuel and product exergy should be adjusted as follows: EF = E31-E42 EP = E26-E25 3-CH2 cond fuel and product exergy must be corrected as follows: EF = E40-E41 EP = E38-E37 4-CH2 evap fuel and product exergy should be corrected as follows: EF = E35-E36 EP = E28-E27 5-It is also thought that the fuel and product exergy in the combustion chamber are spelled incorrectly. E9 should also be written in fuel exergy. Authors Response to POINT (1) We greatly appreciate your suggestions and efforts to improve the quality of the work, for us it was a great help. Next find the answer to the first point consulted: The errors on the equations in Table 3 were caused by an error in the name of the equipment: the condensers and evaporators of chillers 1 and 2, which were inverted in figure 1. The correction has been made on figure 1(please check figure 1 in the manuscript) and all the definitions of fuel and product of the equipment have been revised in table 3, and later in table 5. Resulting in 1)-CH1 cond fuel and product exergy: EF = E31-E32

EP = E26-E25
2)-CH1 evap fuel and product exergy:
EF = E39-E40
EP = E34-E33
3)-CH2 cond fuel and product exergy:
EF = E35-E36
EP = E28-E27
4)-CH2 evap fuel and product exergy should be corrected as follows:
EF = E40-E41
EP = E38-E37

5)- The respective correction was made. Obtaining the following:

EF = E9
EP = E6-E5-E11

Reviewer Comments – Reviewer 2
POINT (2)

Equations considered to be erroneous in Table 5:
1- CH1 cond, CH1 Evap
2-CH2 cond, CH2 Evap
written cost equations for. I mentioned this fix in my previous review. The places of these equations in the table are confused. Hopefully the authors will make the necessary corrections. If they think the evaluation is wrong, the necessary explanation should be given.

Authors Response to POINT (2)

Revised the equations in table 3 and in table 5, the equations considered errors were corrected and they are as follows:

CH1 Condc_26 Ė _26-c_25 Ė _25=c_31 Ė _31-c_32 Ė _32+Ż _CondCh1
CH1 Evapc_34 Ė _34-c_33 Ė _33=c_39 Ė _39-c_40 Ė _40+Ż _EvapCh1
CH2 Condc_28 Ė _28-c_27 Ė _27=c_35 Ė _35-c_36 Ė _36+Ż _CondCh2
CH2 Evapc_38 Ė _38-c_37 Ė _37= 〖 c〗 _40 Ė _40-c_41 Ė _41+Ż _EvapCh2

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Abstract

Gas turbine power plants are increasingly used worldwide, as one of the most interesting options in electricity generation. Therefore, these types of plants have been widely studied, and as a result the negative effects on their output power and thermal efficiency have been known when operating in atmospheric conditions exceeding ISO conditions (15 °C, 60% RH). For this reason, different technologies and methodologies have been implemented that modify the cycle, aiming to increase the output power and improve the thermal efficiency. Unfortunately, the lack of operational parameters of this kind of system limited its characterization, sizing and implementation of strategies to improve its performance. Advanced exergetic and exergoeconomic analyses have been applied in the study. In order to effectively locate the sources of irreversibilities, their costs and the true potentials of

optimal energy use and capital investments. In this work was used an exergy and exergoeconomic analysis in a Gas Turbine Power Plant, to identify that the primary sources of irreversibilities and more significant costs.

Results obtained in this research shows that the main sources of irreversibilities and higher costs are in Combustion Chamber (CC), Heat Recovery Steam Generator (HRSG) and Gas Turbine (GT). From these components, the components the HRSG and GT have greatest potential for improvement, and this can be achieved by improving the overall configuration of the system, due to the fact that the destruction of exogenous exergy is in more significant measure avoidable. While higher costs of investment can be reduced in Combustion Chamber and Gas Turbine. Methodology implemented in this work can be used to improve energy and economic performance in Stig cycle power plants with air cooling with a compression refrigeration machine, combined-cycle systems, or hybrid plants.

Nomenclature

$\dot{C}$	Total cost (USD/s)	*Superscript*	
c	Specific cost per unit of exergy (USD/kJ)	AV	Avoidable
C_P	Specific heat (kJ/kg°C)	CH	Chemistry
$\dot{E}$	Exergy board (kJ/s)	CI	Investment capital
e	Exergía specific (kJ/kg)	EN	Endogenous
h	Specific Enthalpy (kJ/kg)	EX	Exogenous
$\dot{m}$	Mass flow (kg/s)	PH	Physics
P	Pressure (kPa)	SU	Supply
$\dot{Q}$	Heat transfer (kW)	UN	Unavoidable
R	Constant of ideal gases (kJ/mol-K)	*Abbreviations*	
R_P	Pressure Ratio	C	Compressor
T	Temperature (°C)	CC	Combustion chamber
$\dot{W}$	Work (kW)	CDP	Fraction of steam injected into the pre-combustion chamber
s	Entropy (kJ/kg°C)	CH1	Refrigeration machine 1
		CH2	Refrigeration machine 2
$\dot{Z}$	Investment cost (USD/s)	CP	Pump water to condensers
Greek Letters		CT	Cooling tower
α	Stoichiometric air	AC	Air Cooler
ε	Exergy Efficiency	Cond	Condenser
η	Energy Efficiency	EP	Pump water to evaporators
λ	Excess air	EV	Expansion valve
φ	Operation and maintenance factor	Evap	Evaporator
ω	Humidity	FAR	Fuel air ratio
Sub-index		FWP	Feed water pump
0	Reference state conditions	GT	Gas turbine
D	Destruction	HPC	High-pressure compressor
f	Fuel	HRSG	Heat recovery boiler
i	input	IAC	Air cooling at the compressor inlet
k	*k*-th component	K	Specific heat ratio
o	output	LPC	Low-pressure compressor
P	Product	MWP	Make-up water pump
s	isentropic	RH	Relative humidity
		PEC	Equipment purchase price
		SAR	Air steam ratio
		TIT	Turbine inlet temperature

1. Introduction

Design of gas turbines is based on the recommendations and parameters given by the International Standards Organization -ISO- which specifies as environmental design conditions 15 °C, 60% relative humidity, and pressure at sea level. Therefore, if environmental conditions of the installation site differ from ISO, the output electrical power, thermal efficiency, and cost of the kilowatt/hour generated in the gas turbine plants are affected [1]. For each degree Celsius that the ambient temperature increases above ISO conditions, gas turbine plants lose in output power and efficiency 1.47 MW and 0.1% respectively [2]. There are technologies to compensate the decrease in output power and thermal efficiency in power plants with gas turbines that operate at conditions above ISO, some of these technologies are air cooling at the compressor inlet, steam injection and combined cycles.

Air cooling technologies have been studied in different investigations, some of these are: Comodi et al. [3] evaluated the effects of air cooling at the inlet on a test bench of a 100 kW micro gas turbine. The air cooling technology selected was the electric chiller. Electric power and thermal efficiency gain depends on ambient conditions and it reached up to 8.5 and 1.6 %, respectively, concerning the nominal conditions of the micro gas turbine. Mohapatra et al. [4] focused on comparing the impact of two air cooling methods (steam compression and absorption) on a single gas turbine plant and combined cycle. It was observed that, the system performance with vapor compression inlet air cooling was superior to vapor absorption inlet cooling. The optimum compressor inlet temperature for both scheme was found to be 20°C. Baakeem et al. [5] simulated three air cooling technologies (mechanical steam compression, evaporative cooling, and absorption cooling) in an 85 MW gas turbine under Riyadh desert environmental conditions. The annual power gained in the gas turbine when integrated with evaporative cooling, mechanical vapor compression, and absorption cooling was 9%, 14.9%, and 14.9%, respectively. Air cooling at the compressor inlet has proven to be an effective technology for increasing the output power and efficiency of gas turbine systems, and it has also been shown that applying this technology is profitable since the increase in power offsets the increased investment as shown Zare et at [6], but the irreversibilities introduced by the application of this technology are not known, nor the origin of these.

Also, the effects of steam injection into the combustion chamber systems as gas turbine technology for power increase were studied by Xue et al. [7], obtaining that this reduces the temperature in the burner and slightly increases the loss of total pressure when the fraction of the weight of steam increases. For a 15% steam mass fraction, the pressure loss reaches 4.87%. Zhang et al [8], evaluated the thermodynamic performance of a gas turbine system with steam injection, where it is highlighted that the outlet temperature of the combustion chamber has a great impact on the efficiency of the cycle. For a combustion temperature of 1300 K and pressure ratio of 20, thermal efficiency was 51.13%, and exergetic efficiency was 49.31%. In recent times, integrating air cooling at the compressor inlet (IAC) and steam injection to the gas turbine power generation system (Stig cycle) is one of the most widely used strategies to improve the performance of gas turbines, both alternatives can be implemented without a significant modification of the integrity of the existing basic cycle. Shukla and Singh [9] compared the specific output power of different combinations of power increasing technology for gas turbines: Simple gas turbine cycle (GT) with steam injection (SI), air cooling –(IAC) and steam injection (SI). The output power is increased by 7.2% for GT with SI, 9.5% for GT with IAC and SI. In a later study, the authors analyzed the effect of the integration of air cooling technologies (evaporative cooling and inlet fogging) in a gas turbine cycle with steam injection. For parameters, TIT=1700 K, Rp=24, and environmental conditions Tamb=318 K and RH=35%, the specific output power and thermal efficiency increase 13.2% and 15.92%, respectively. Results showed that the inlet fogging is better than evaporative cooling to achieve lowest temperature at the compressor inlet [10]. Other studies integrating steam injection and combined cycle air cooling technologies: Shukla and Singh [11] investigated the performance of a combined cycle plant with two pressure levels with steam injection and air cooling by absorption refrigeration machine. Specific work and thermal efficiency increase by 17.34% and 6.78% respectively when the steam air ratio increases from 3% to 7% for a given inlet temperature, the pressure ratio of 24 and compressor inlet temperature of 278 K. Athari et al. [12] evaluated energetically and exergetically the gas turbine systems with stig cycle -BIFSG- and combined cycle -BIFCC, in both cases air cooling by fogging and biogás. Evaluations show that the combined cycle is more efficient at low pressure ratio values, unlike the stig cycle plant, which is advantageous at high pressure ratio values. In the case of plants with a stig cycle, it has a higher net power and less exergy lost than the combined cycle for the same conditions.

In a later study, the same authors thermo-economically analyzed the two previous systems. The results showed that electric power production and component costs are higher in the combined cycle than in the steam injection plant [13]. In these studies, energetic, exergy and exergoeconomic analyzes have been carried out in complex systems of gas turbine plants with air cooling and steam injection, the complexity of these systems increases irreversibilities and costs, which can be optimized if necessary with more information about its origin. With advanced exergy analysis it is possible to know the origin of the destruction (endogenous / exogenous destroyed exergy) and the maximum potential for improving the systems (unavoidable / avoidable destroyed exergy). However, the advanced exergetic and exergoeconomic analysis is not considered, as it is done in the present work.

Different thermal systems have been studied with advanced exergetic analysis, Kecebas et al. [14] analyzed a geothermal plant. From the exergy destruction values, they identified that the components that deserve priority to receive modifications are the heat exchangers and the turbines. It was shown that applying the improvements to the system increased the modified efficiency to 18.26%, while the efficiency in real conditions was 9.60%. Acikkalp et al. [15] studied the performance of a 37 MW power generation facility. The relations between the components are weak because of the ratio of the endogenous exergy rates of 70%. The improvement potential of the system is 38%. It may be concluded that one should focus on the gas turbine and combustion chamber for improving the system. Boyaghchi et al. [16] performed an advanced exergetic analysis in a combined cycle power plant; where it is reported that TIT and R_P of the compressor are the variables chosen to study the behavior of the parts of the exergy destruction. Increasing the TIT and R_P of the compressor increases the potential for improvement in most components and decreases the unavoidable part in some components. These authors have obtained the improvement potentials for their study plants, but they do not evaluate the economic implications of implementing them.

Advanced exergoeconomic analysis has also been used for the evaluation of thermal systems, to know the origin of the investment costs and energy destruction, in addition to the possible reductions of these, from the application of improvements to the systems. Acikkalp et al. [17], analyzed a gas turbine system with regeneration, where was determined that the relationships between the components are healthy. The potential for system improvement and reduce investment costs is low. The results of the analysis indicate that the combustion chamber, the high pressure steam turbine and the condenser present a potential for economic improvement due to their high costs of exergy destruction. Similarly, the HRSG and the condenser present significant potential to lower your investment costs. Anvari et al. [18] considered advanced exergetic analysis in a trigeneration plant for the production of heat, cold and electrical power. It concludes that 29% of the total destroyed exergy and its associated costs are avoidable endogenous, that most of the investment costs (58%) are preventable. Author reports that the highest avoidable exergy destruction costs are presented in the combustion chamber, followed by the air heater, HRSG, and gas turbine. Unlike these studies, the present work investigated the effect on advanced energy, exergetic and exergoeconomic indicators of air cooling at the compression input and steam injection in a gas turbine.

Based on the review of the scientific literature presented above, the main contribution of this work is the advanced exergetic and exergoeconomic analysis of Gas Turbine Plant with the Stig cycle and air cooling at the compressor inlet and the study of the performance of the components of this thermal power plants. In this study, the components with the highest destruction of exergy, the real potentials for improving equipment performance, the effect on the exergetic efficiency of a specific equipment due to the operation of the remaining equipment, avoidable capital investments and the unavoidable capital investments are shown for the power plant configuration defined in this research. To develop the analysis proposed in this work, data obtained from the actual operation of a company in the industrial sector were used.

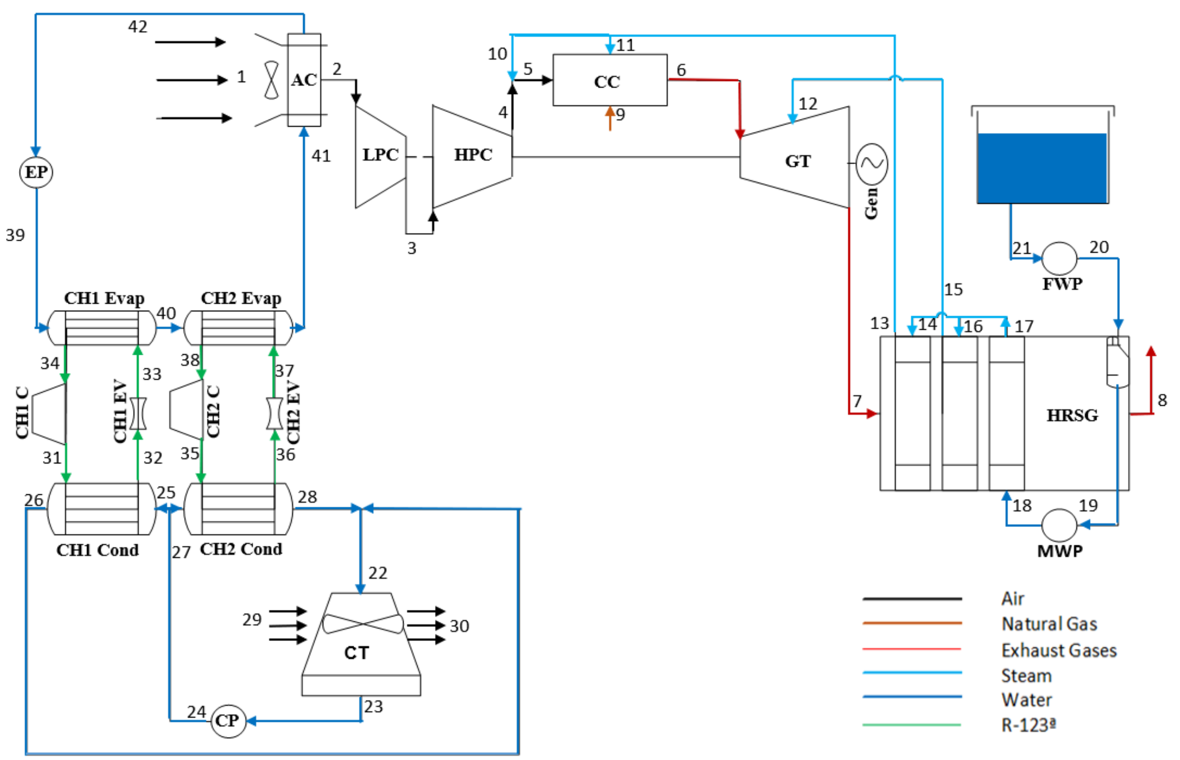
2. Methodology

2.1. Description of the system

The components of the power system with Stig cycle and air cooling (Figure 1), consists of a General Electric LM5000 gas turbogenerator, a recovery boiler (HRSG), and the air cooling system, which are arranged so that the evaporators are in series and the condensers in parallel, this arrangement allows the air cooling temperature to reach 8.8°C, operating condition of the study plant (Figure 1). Atmospheric conditions of the geographic location of the generation plant are, on average, 32 °C of temperature and 80% relative humidity [19]. Some characteristics of the generation plant are show in Table 1.

Table 1 Characteristics of the generation plant

Item	**Value**	**Reference**
The mechanical efficiency (%)	98.5	[11]
Pressure drops (CC, HRSG) (%)	5	[20]
Overall Steam injection (kg/s)	10.306	[21]
Ratio steam injection in pre-combustion chamber (%)	37.2	[21]
Ratio steam injection control generation NOx (%)	20.3	[21]
Ratio steam injection control the turbine temperature (%)	42.5	[21]
Chemical composition of natural gas [21].		
Compound	**% Vol.**	
$\mathbf{CH_4}$	97.9458	
$\mathbf{N_2}$	1.4832	
$\mathbf{CO_2}$	0.2062	
$\mathbf{C_2H_6}$	0.0541	
$\mathbf{C_4H_{10}}$	0.0302	
$\mathbf{C_5H_{12}}$	0.0094	
$\mathbf{C_6H_{14}}$	0.0189	



AC: Air Cooler. LPC: Low-Pressure Compressor. HPC: High-Pressure Compressor. CC: Combustion Chamber. GT: Gas Turbine. Gen: Generator. HRSG: Heat Recovery Steam Generator. CT: Cooling Tower. CP: Condenser pump. FWP: Feed Water Pump. MWP: Make up Water Pump. CH1 C: Chiller1 Compressor. CH1Cond: Chiller1 Condenser. CH1 EV: Chiller1 Expansion Valve. CH1 Evap: Chiller 1 Evaporator. CH2 C: Chiller2 Compressor. CH2 Cond: Chiller2 Condenser. CH2 EV: Chiller2 Expansion Valve. CH2 Evap: Chiller 2 Evaporator

Figure 1 Diagram of the power plant with a gas turbine and steam injection with air cooling by compression cooling system

The flowchart of the algorithm for calculating an advanced exergoeconomic of the system with steam injection is showed in Figure 2. In this algorithm, sequence of the studies elaborated in the present work is presented, it is essential to point out that for each of these analyses parameters determined from the different balances of the system and its components are used.

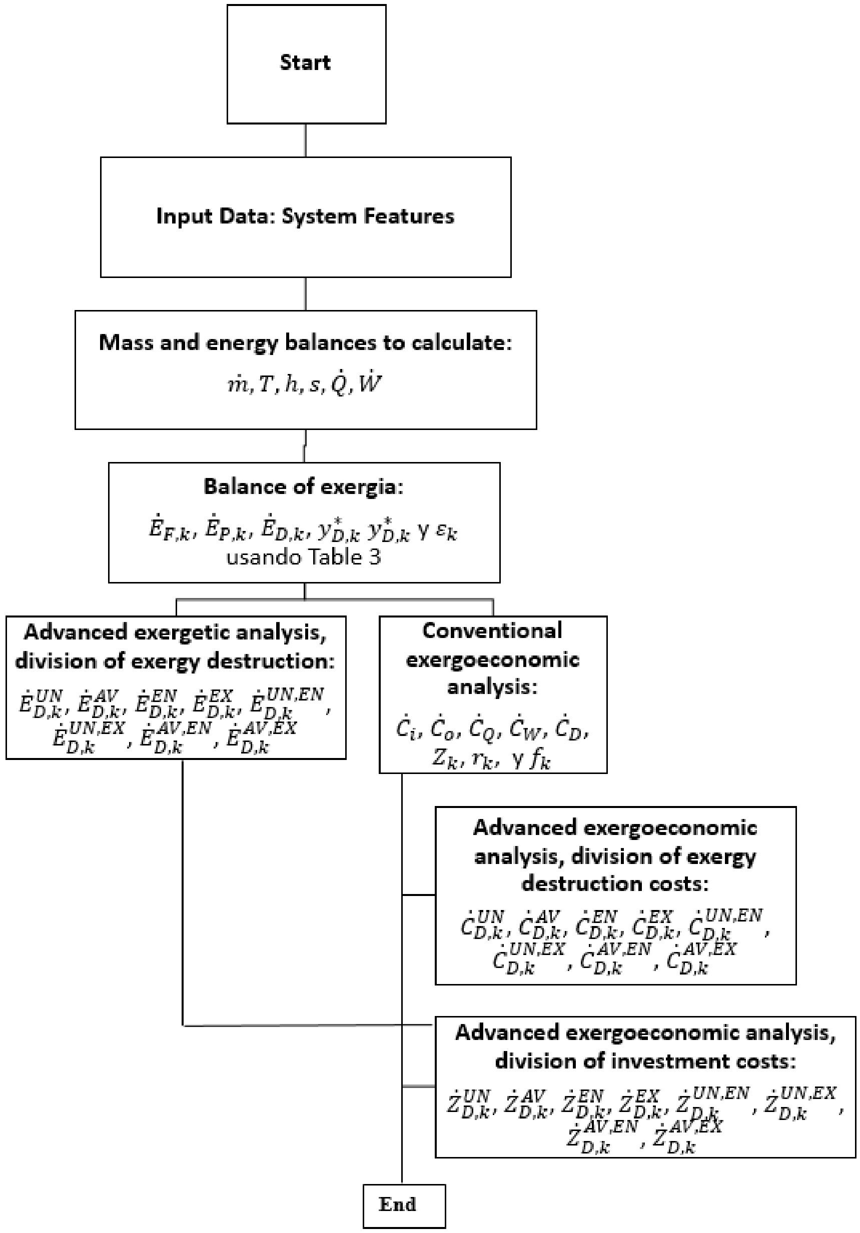


Figure 2 Calculation algorithm of an advanced exergoeconomic analysis for a gas power cycle with the stig cycle and air cooling.

2.2. Mass balance, energy, and thermodynamic model

Equations of the mass (Equation 1) and energy (Equation 2) balances, shown below, are applied, assuming that the components of the analyzed system are in a stable state. Next, the thermodynamic model is developed that allows the evaluation of each of the elements that make up the power system with the Stig cycle and air cooling under study [22].

Mass balance

$$\sum \dot{m}_{i,k} = \sum \dot{m}_{o,k} \ \ (Kg/s) \qquad 1$$

Energy balance

$$\dot{Q}_k - \dot{W}_k + \sum \dot{m}_{i,k} h_{i,k} - \sum \dot{m}_{o,k} h_{o,k} = 0 \ \ (KW) \qquad 2$$

Exergy balance

$$\dot{E}_{Q,k} - \dot{E}_{W,k} + \sum \dot{E}_{i,k} - \sum \dot{E}_{o,k} - \dot{E}_{D,k} = 0 \ \ (KW) \qquad 3$$

The air outlet temperature of the compression units can be obtained as follows [23]:

$$T_o = \frac{T_i}{n_C}\left[RP_C^{\frac{K_C-1}{K_C}} - 1\right] + T_i \ (°C) \qquad 4$$

Air leaving the HPC is mixed with a fraction of steam injected into the pre-combustion chamber. The properties of air at the CC inlet are obtained from the mass balance of dry air and water vapor, and the energy balance of the humidification process in equations 5, 6, and 7, respectively, assuming there is no change in temperature and pressure in the air stream.

$$\dot{m}_{o,Air} = \dot{m}_{i,Air} \ \ (Kg/s) \qquad 5$$

$$\dot{m}_{o,Air}\omega_o = \dot{m}_{i,Air}\omega_i + \dot{m}_{steam}(CDP) \ \ (Kg/s) \qquad 6$$

$$\dot{m}_{o,Air}h_{o,Air} = \dot{m}_{i,Air}h_{i,Air} + \dot{m}_{steam}(CDP)h_{steam} \ \ (kW) \qquad 7$$

The combustion process follows the analysis of reactive systems based on the first law for stationary flows, applying Equation 8 and Equation *9* on a molar basis to obtain the exit temperature of the combustion gases. This model foresees an energy loss in the combustion of 2% of the energy supplied by the reagents (Q_Loss=2% HR), as suggested by Tsatsaronis in *[24]*. To obtain the composition of the exhaust gases, we use the moles balance of the elements present in the combustion C, H, O, N, (Equations 12 to 15) and two complementary equations of simultaneous reaction of chemical equilibrium for the formation of CO and NO (Equations 16 and 17) are used *[22]*.

$$CH_4 + \lambda\alpha(O_2 + 3{,}76N_2) + \left(4{,}76\lambda\alpha\overline{\omega} + n_{NO_X}\right)H_2O \rightarrow aCO_2 + bCO + cH_2O + dO_2 + eN_2 + fNO \qquad 8$$

$$H_{CH_4} + \lambda\alpha\left(H_{O_2} + 3{,}76H_{N_2}\right) + \left(4{,}76\lambda\alpha\overline{\omega} + n_{NO_X}\right)H_{H_2O} \rightarrow aH_{CO_2} + bH_{CO} + cH_{H_2O} + dH_{O_2} + eH_{N_2} + fH_{NO} - Q_{Loss} \qquad 9$$

a,b,c,d,e,f represent the moles of each of the species present in the exhaust gases. Equations 16 and 17 represent the chemical equilibrium equations for the dissociation of $CO_2 \rightarrow CO + \frac{1}{2}O_2$ y $\frac{1}{2}N_2 + \frac{1}{2}O_2 \rightarrow NO$ respectively.

$$\overline{\omega} = 1.608\omega \quad \left(\frac{Mol_{H_2O}}{Mol_{Air}}\right) \qquad 10$$

$$n_{NO_X} = \frac{\dot{m}_{NO_X} MW_{CH_4}}{\dot{m}_{CH_4} MW_{H_2O}} \quad \left(\frac{Mol_{H_2O}}{Mol_{CH_4}}\right) \qquad 11$$

$Balance\ of\ C$

$$1 \rightarrow a + b \qquad 12$$

$Balance\ of\ H$

$$4 + 2\left(4{,}76\lambda\alpha\overline{\omega} + n_{NO_X}\right) \rightarrow 2c \qquad 13$$

$Balance\ of\ O$

$$2\lambda\alpha + \left(4{,}76\lambda\alpha\overline{\omega} + n_{NO_X}\right) \rightarrow 2a + b + c + 2d + f \qquad 14$$

$Balance\ of\ N$

$$2\lambda\alpha(3{,}76) \rightarrow 2e + f \qquad 15$$

$$K_{PCO_2} = \frac{b^{V_{CO}} d^{V_{O_2}}}{a^{CO_2}} \qquad 16$$

$$K_{PNO} = \frac{f^{V_{NO}}}{e^{V_{N_2}} d^{V_{O_2}}} \qquad 17$$

Chemical equilibrium constants for the ideal gas mixture, for each of the simultaneous formation equations, is calculated as [22]:

$$ln(K_{PCO}) = \frac{-\Delta Giss^*_{CO}(T_{Prod})}{R_u T_{Prod}} \qquad 18$$

$$ln(K_{PNO}) = \frac{-\Delta Giss^*_{NO}(T_{Prod})}{R_u T_{Prod}} \qquad 19$$

Mass balance in the CC is presented below based on a fuel mass flow, the fuel air ratio (FAR) and steam air ratio [25].

$$FAR = \frac{\dot{m}_{fuel}}{\dot{m}_{dry\ air}} = \frac{M_{fuel}}{4.76 \propto \lambda M_{Air}} \quad 20$$

$$SAR = \frac{\dot{m}_{Steam}}{\dot{m}_{dry\ air}} \quad 21$$

$$\dot{m}_{gases} = \dot{m}_{dry\ air}\left(1 + \omega + FAR + SAR(CDP + NO_X)\right)\ (kg/s) \quad 22$$

To know the exhaust gas outlet temperature of the TG is evaluated as [23]:

$$T_o = T_i - n_{GT}T_i\left[1 - \left(1 - \left(\frac{1}{RP_{GT}}\right)^{\frac{K_{GT}-1}{K_{GT}}}\right)\right]\ (°C) \quad 23$$

In the HRSG, high-quality steam is produced at the necessary outlet conditions for high pressure and low pressure flows from the efficiency of the first-grade heat recovery boiler, obtaining the exhaust gas outlet temperature for the requirements of steam generation (Equation 24) [12].

$$\dot{m}_{gases}C_P(T_i - T_o) = \dot{m}_{19}h_{19} - \dot{m}_{20}h_{20} + \dot{m}_{17}h_{17} - \dot{m}_{18}h_{18} + \dot{m}_{15}h_{15} - \dot{m}_{16}h_{16} + \dot{m}_{13}h_{13} - \dot{m}_{14}h_{14}\ \ (kW) \quad 24$$

MWP and MWP water pumps for the steam injection system, and P_Evap, P_Cond for the air cooling system are modeled from their pressure ratios and the isentropic efficiency of the pumps to obtain the output conditions, as shown in equation 25 [26].

$$\eta = \frac{h_{o,s} - h_i}{h_o - h_i} \times 100\ \ (\%) \quad 25$$

Cooling tower is modeled to obtain the amount of air needed to reduce the water temperature between the required limits, for which the mass and energy balances of air and water are used, as shown in equations 26, 27, and 28 [22].

$$\dot{m}_{29}h_{29} - \dot{m}_{30}h_{30} + \dot{m}_{22}h_{22} - \dot{m}_{23}h_{23} = 0\ \ (kW) \quad 26$$

$$\dot{m}_{30} = \dot{m}_{29}\ \ (kg/s)\ Air\ mass\ balance \quad 27$$

$$\dot{m}_{29}(\omega_{30} - \omega_{29}) = \dot{m}_{23} - \dot{m}_{22}\ (kg/s)\ Water\ mass\ balance \quad 28$$

For the compression refrigeration machines, which work between the operating pressure limits of the condenser and evaporator is assumed (Equations 29, 30, 31 and 32):

- Isentropic efficiency of the compressor [27] [28].
- Refrigerant in the output of the condenser as saturated liquid [27] [28].
- Iso-enthalpy process in the throttle valve [27] [29].
- Refrigerant as saturated steam in the output of the evaporator [27] [29].
- No heat loss between water and refrigerant in the condenser and evaporator [27].

$$\eta = \frac{h_{o,s} - h_i}{h_o - h_i} \times 100 \ \ (\%) \quad 29$$

$$\dot{m}_{i,water} h_{i,water} - \dot{m}_{o,water} h_{o,water} = \dot{m}_{o,Ref} h_{o,Ref} - \dot{m}_{i,Ref} h_{i,Ref} \ \ (kW) \quad 30$$

$$h_o = h_i \ (kJ/kg) \quad 31$$

$$\dot{m}_{i,water} h_{i,water} - \dot{m}_{o,water} h_{o,water} = \dot{m}_{o,Ref} h_{o,Ref} - \dot{m}_{i,Ref} h_{i,Ref} \ (kW) \quad 32$$

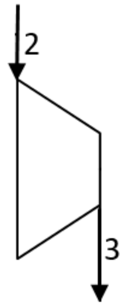
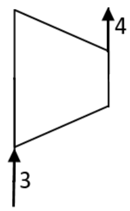
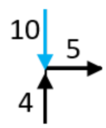
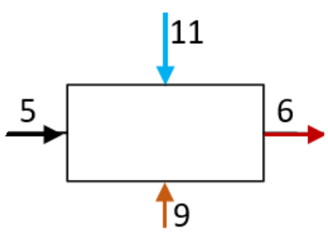
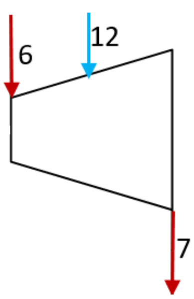
AC was modeled using heat transfer efficiency between air and water from equa11tion 33 [30]. For the conditions of inlet and outlet constant air and water of the model, is obtained the water flow necessary to obtain the desired air outlet temperature.

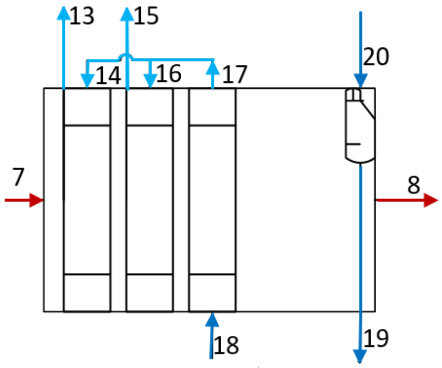
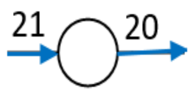
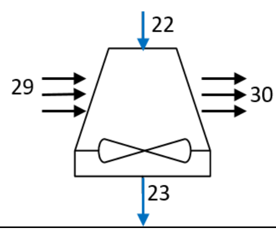
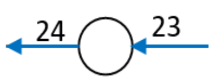
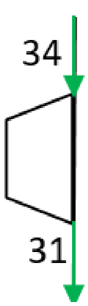
$$\eta_{CCoil} = \frac{\dot{m}_{i,Air} h_{i,Air} - \dot{m}_{o,Air} h_{o,Air}}{\dot{m}_{o,water} h_{o,water} - \dot{m}_{i,water} h_{i,water}} \times 100 \ \ (\%) \quad 33$$

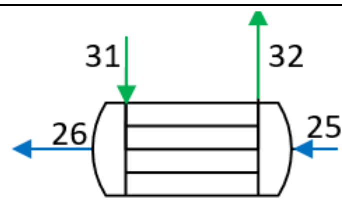
Table 2 shows the energy balances for each of the components of the gas turbine cycle with the Stig cycle and air cooling at the compressor inlet with a compression cooling system.

Table 2 Energy and mass balances in the gas turbine cycle with steam injection and air cooling.

Component	**Mass and energy balance**
LPC	$\dot{W}_{LPC} = (\dot{m}_3 h_3 - \dot{m}_2 h_2)/\eta_{mec}$ $\dot{m}_3 = \dot{m}_2$

	
(HPC 	$\dot{W}_{HPC} = (\dot{m}_4 h_4 - \dot{m}_3 h_3)/\eta_{mec}$ $\dot{m}_4 = \dot{m}_3$
Air humidifier 	$\dot{m}_5 h_5 = \dot{m}_4 h_4 + \dot{m}_{10} h_{10}$ $\dot{m}_5 \omega_5 = \dot{m}_4 \omega_4 + \dot{m}_{10} \quad Water\ mass\ balance$ $\dot{m}_5 = \dot{m}_4 \quad Air\ Mass\ Balance$
CC 	$CH_4 + \lambda\alpha(O_2 + 3{,}76N_2) + (4{,}76\lambda\alpha\overline{\omega} + n_{11})H_2O \rightarrow aCO_2 + bCO + cH_2O + dO_2 + eN_2 + fNO$ $\dot{m}_6 = \dot{m}_5\left(1 + \omega_5 + FAR + SAR(NO_X)\right)$ $\dot{m}_{11} = \dot{m}_5 SAR(NO_X)$ $\dot{m}_9 = \dot{m}_5 FAR$ $n_{11} = \dfrac{\dot{m}_{11} MW_{CH_4}}{\dot{m}_9 MW_{H_2O}}$
GT 	$\dot{W}_{GT} = \left(\dot{m}_6 C_{P,7}(T_6 - T_7) + \dot{m}_{12}(h_{12} - h_7)\right)$ $\dot{m}_7 = \dot{m}_6 + \dot{m}_{12}$ $\dot{m}_7 = \dot{m}_5\left(1 + \omega_5 + \dfrac{1}{AFR} + SAR(NO_X + LP)\right)$
Gen 	$\dot{P}_{Elect} = \eta_{Gen}\dot{W}_{net}$

HRSG 	$-\dot{Q}_{HRSG,out} + \dot{m}_7 C_{P,7}(T_7 - T_8) + \dot{m}_{20}h_{20} - \dot{m}_{19}h_{19} + \dot{m}_{18}h_{18} - \dot{m}_{17}h_{17} + \dot{m}_{16}h_{16} - \dot{m}_{15}h_{15} + \dot{m}_{14}h_{14} - \dot{m}_{13}h_{13} = 0$ $\dot{m}_8 = \dot{m}_7$ $\dot{m}_{20} = \dot{m}_{19} = \dot{m}_{18} = \dot{m}_{17} = \dot{m}_5 SAR$ $\dot{m}_{12} = \dot{m}_{15} = \dot{m}_{16} = \dot{m}_5 SAR(LP)$ $\dot{m}_{13} = \dot{m}_{14} = \dot{m}_{10} + \dot{m}_{11} = \dot{m}_5 SAR(CDP + NO_X)$
MWP 	$\dot{W}_{MUP} = (\dot{m}_{18}h_{18} - \dot{m}_{19}h_{19})/\eta_{mec}$ $\dot{m}_{18} = \dot{m}_{19}$
FWP 	$\dot{W}_{FDP} = (\dot{m}_{20}h_{20} - \dot{m}_{21}h_{21})/\eta_{mec}$ $\dot{m}_{21} = \dot{m}_{20}$
CT 	$\dot{m}_{29}h_{29} - \dot{m}_{30}h_{30} + \dot{m}_{22}h_{22} - \dot{m}_{23}h_{23} = 0$ $\dot{m}_{30} = \dot{m}_{29}$ *Balance de aire* $\dot{m}_{29}(\omega_{30} - \omega_{29}) = \dot{m}_{Makeup}$ Balance of humidity in the air $\dot{m}_{22} = \dot{m}_{23} + \dot{m}_{Makeup}$ *Water balance*
CP 	$\dot{W}_{Pcond} = (\dot{m}_{24}h_{24} - \dot{m}_{23}h_{23})/\eta_{mec}$ $\dot{m}_{24} = \dot{m}_{23}$
CH1 C 	$\dot{W}_{C,Ch1} = (\dot{m}_{31}h_{31} - \dot{m}_{34}h_{34})/\eta_{mec}$ $\dot{m}_{31} = \dot{m}_{34}$
CH1 Cond:	$-\dot{Q}_{cond,Ch1} + \dot{m}_{31}h_{31} - \dot{m}_{32}h_{32} + \dot{m}_{25}h_{25} - \dot{m}_{26}h_{26} = 0$ $\dot{m}_{32} = \dot{m}_{31}$ $\dot{m}_{26} = \dot{m}_{25}$

<table>
<tr><td>
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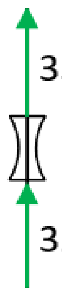
</td><td>$h_{33} = h_{32}$
$\dot{m}_{33} = \dot{m}_{32}$</td></tr>
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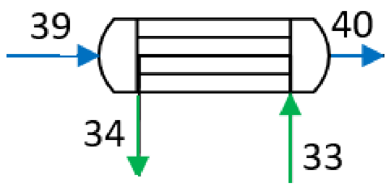
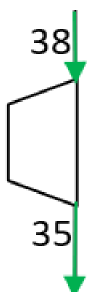

</td><td>$\dot{Q}_{evap,Ch1} + \dot{m}_{33}h_{33} - \dot{m}_{34}h_{34} + \dot{m}_{39}h_{39}$
$- \dot{m}_{40}h_{40} = 0$
$\dot{m}_{40} = \dot{m}_{29}$
$\dot{m}_{34} = \dot{m}_{33}$</td></tr>
<tr><td>CH2 C

</td><td>$\dot{W}_{C,Ch2} = (\dot{m}_{35}h_{35} - \dot{m}_{38}h_{38})/\eta_{mec}$
$\dot{m}_{35} = \dot{m}_{38}$</td></tr>
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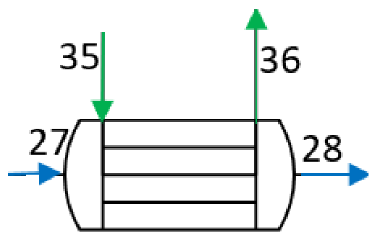
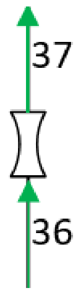

</td><td>$-\dot{Q}_{cond,Ch2} + \dot{m}_{35}h_{35} - \dot{m}_{36}h_{36} + \dot{m}_{27}h_{27}$
$- \dot{m}_{28}h_{28} = 0$
$\dot{m}_{28} = \dot{m}_{27}$
$\dot{m}_{36} = \dot{m}_{35}$</td></tr>
<tr><td>CH2 EV</td><td>$h_{37} = h_{36}$
$\dot{m}_{37} = \dot{m}_{36}$</td></tr>
</table>

<table>
<tr><td>
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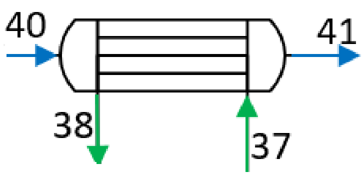
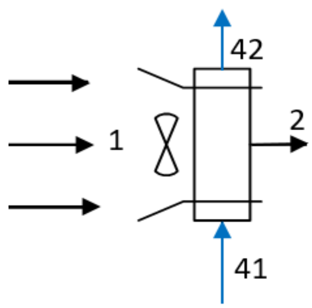
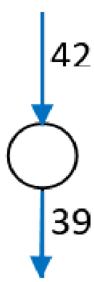

</td><td>$$\dot{Q}_{evap,Ch2} + \dot{m}_{37}e_{37} - \dot{m}_{38}e_{38} + \dot{m}_{40}h_{40} - \dot{m}_{41}h_{41} = 0$$
$$\dot{m}_{38} = \dot{m}_{37}$$
$$\dot{m}_{41} = \dot{m}_{40}$$</td></tr>
<tr><td>AC

</td><td>$$\dot{Q}_{AC} + \dot{m}_1 h_1 - \dot{m}_2 h_2 + \dot{m}_{41}h_{41} - \dot{m}_{42}h_{42} = 0$$
$$\dot{m}_{42} = \dot{m}_{41}$$
$$\dot{m}_2 = \dot{m}_1$$</td></tr>
<tr><td>EP

</td><td>$$\dot{W}_{Pevap} = (\dot{m}_{39}h_{39} - \dot{m}_{42}h_{42})/\eta_{mec}$$
$$\dot{m}_{39} = \dot{m}_{42}$$</td></tr>
</table>

It is considered that the Q in the previous equations represents the loss of energy to the surroundings and cannot be recovered. From the equations and energy analysis presented above, the bases and parameters required to initiate the conventional exergetic and exergoeconomic of the analyzed system are established.

2.3. Conventional exergetic and exergoeconomic analysis

Exergy is a property that measures the maximum working potential of an amount of energy in a specific state [30]. Exergy is composed of two parts, physical exergy and chemical exergy (Equation 34):

$$e_k = e_k^{PH} + e_k^{CH}\ (kJ/kg) \quad 34$$

Physical exergy for a substance is obtained with Equation 35 [30] and 36 [20]:

$$e_k^{PH} = h - h_0 - T_0(s - s_0)\ (kJ/kg) \quad 35$$

$$e_k^{PH} = \frac{1}{M_k}\left(\bar{C}_{P,k}(T - T_0) - T_0\left[\bar{C}_{P,k}\ln\left(\frac{T}{T_0}\right) - \bar{R}\ln\left(\frac{P}{P_0}\right)\right]\right)\ (kJ/kg) \quad 36$$

Where 0 is the reference state, which we assume as the state of least energy $T_{37} = 3.424°C$ y $P_{37} = 38.15\ kPa$, as suggested Kotas [31], Yumrutas [32] y D'Acaddia [33] thus ensuring that all temperatures are higher than those of the dead state.

Specific chemical exergy is calculated in equation 37, and $\bar{e}^{CH}$ is obtained from [31]:

$$e_k^{CH} = \frac{1}{M_k}\left(\bar{x}_i\bar{e}_i^{CH} + \bar{R}T_0\sum\bar{x}_i\ln(\bar{x}_i)\right)\ (kJ/kg) \quad 37$$

Exergy balance is obtained from Equation 38 [30]:

$$\dot{E}_{D,k} = \dot{E}_{F,k} - \dot{E}_{P,k}\ (kW) \quad 38$$

Exergy efficiency is obtained as (Equation 39) [30]:

$$\varepsilon_k = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} \times 100\ \ (\%) \quad 39$$

Exergy destruction ratios are calculated as (Equations 40 and 41) [30]:

$$y_{D,k} = \frac{\dot{E}_{D,k}}{\dot{E}_{F,k}} \times 100\ (\%) \quad 40$$

$$y_{D,k}^{*} = \frac{\dot{E}_{D,k}}{\dot{E}_{D,Total}} \times 100\ (\%) \quad 41$$

The energy balances and the definition of fuel and product exergies for each plant component are shown in Table 3.

Table 3 Definition of Fuel and Product exergies.

Component	Exergy Fuel and Product
LPC	$\dot{E}_F = \dot{W}_{LPC}$ $\dot{E}_P = \dot{m}_3 e_3 - \dot{m}_2 e_2$
HPC	$\dot{E}_F = \dot{W}_{HPC}$ $\dot{E}_P = \dot{m}_4 e_4 - \dot{m}_3 e_3$
CC	$\dot{E}_F = \dot{m}_9 e_9$ $\dot{E}_P = \dot{m}_6 e_6 - \dot{m}_5 e_5 - \dot{m}_{11} e_{11}$
GT	$\dot{E}_F = \dot{m}_6 e_6 + \dot{m}_{12} e_{12} - \dot{m}_7 e_7$ $\dot{E}_P = \dot{W}_{GT}$
HRSG	$\dot{E}_F = \dot{m}_7 e_7 - \dot{m}_8 e_8$ $\dot{E}_P = \dot{m}_{13} e_{13} - \dot{m}_{14} e_{14} + \dot{m}_{15} e_{15} - \dot{m}_{16} e_{16} + \dot{m}_{17} e_{17} - \dot{m}_{18} e_{18} + \dot{m}_{19} e_{19} - \dot{m}_{20} e_{20}$
MWP	$\dot{E}_F = \dot{W}_{MUP}$ $\dot{E}_P = \dot{m}_{18} e_{18} - \dot{m}_{19} e_{19}$
FWP	$\dot{E}_F = \dot{W}_{FDP}$ $\dot{E}_P = \dot{m}_{20} e_{20} - \dot{m}_{21} e_{21}$
CT	$\dot{E}_F = \dot{m}_{22} h_{22} - \dot{m}_{23} h_{23} + \dot{W}_{fan}$ $\dot{E}_P = \dot{m}_{30} h_{30} - \dot{m}_{29} h_{29}$
CP	$\dot{E}_F = \dot{W}_{Pcond}$ $\dot{E}_P = \dot{m}_{24} e_{24} - \dot{m}_{23} e_{23}$
CH1 C	$\dot{E}_F = \dot{W}_{CCh1}$ $\dot{E}_P = \dot{m}_{31} e_{31} - \dot{m}_{34} e_{34}$
CH1 Cond	$\dot{E}_F = \dot{m}_{31} e_{31} - \dot{m}_{32} e_{32}$ $\dot{E}_P = \dot{m}_{26} e_{26} - \dot{m}_{25} e_{25}$
CH1 EV	$h_{33} = h_{32}$ $\dot{E}_F = \dot{m}_{32} e_{32}$ $\dot{E}_P = \dot{m}_{33} e_{33}$
CH1 Evap	$\dot{E}_F = \dot{m}_{39} e_{39} - \dot{m}_{40} e_{40}$ $\dot{E}_P = \dot{m}_{34} e_{34} - \dot{m}_{33} e_{33}$
CH2 C	$\dot{E}_F = \dot{W}_{CCh2}$ $\dot{E}_P = \dot{m}_{35} e_{35} - \dot{m}_{38} e_{38}$
CH2 Cond	$\dot{E}_F = \dot{m}_{35} e_{35} - \dot{m}_{36} e_{36}$ $\dot{E}_P = \dot{m}_{28} e_{28} - \dot{m}_{27} e_{27}$
CH2 EV	$\dot{E}_F = \dot{m}_{36} e_{36}$ $\dot{E}_P = \dot{m}_{37} e_{37}$
CH2 Evap	$\dot{E}_F = \dot{m}_{40} e_{40} - \dot{m}_{41} e_{41}$ $\dot{E}_P = \dot{m}_{38} e_{38} - \dot{m}_{37} e_{37}$

AC	$\dot{E}_F = \dot{m}_{42}e_{42} - \dot{m}_{41}e_{41}$ $\dot{E}_P = \dot{m}_1 e_1 - \dot{m}_2 e_2$
EP	$\dot{E}_F = \dot{W}_{Pevap}$ $\dot{E}_P = \dot{m}_{39}e_{39} - \dot{m}_{42}e_{42}$

The cost balance is applied to each component of the study system and is expressed in Equation 42 [24].

$$\dot{C}_{P,k} = \dot{C}_{F,k} + \dot{Z}_k^{Tot} \ \ (\$/s) \qquad 42$$

$\dot{Z}_k^{Tot}$ the investment cost rate, which includes equipment purchase costs (PEC_k), supply costs (C_{tot}^{su};Refrigerant or raw water) and the operation and maintenance cost rate (φ).

The components of the cost balance are calculated, as shown in Equations 43, 44, 45, 46, and 47 [24]. We use the cost levelization approach [24], in Table 4 shows some parameters for the exergoecnomic analysis.

$$\dot{C}_{P,k} = c_{P,k}\dot{E}_P \ \ (\$/s) \qquad 43$$

$$\dot{C}_{F,k} = c_{F,k}\dot{E}_F \ \ (\$/s) \qquad 44$$

$$\dot{Z}_k^{Tot} = \dot{Z}_k + Z_k^{SU} \ \ (\$/hr) \qquad 45$$

$$\dot{Z}_k = \frac{PEC_k\left[\frac{i_r(1+i_r)^{n_y}}{(1+i_r)^{n_y}-1}\right]\varphi}{3600(RTY)} \ \ (\$/hr) \qquad 46$$

$$Z_k^{SU} = \frac{C_{tot}^{su}PEC_k}{3600(RTY)\sum PEC_k}(\$/hr) \qquad 47$$

Table 4 parameter to exergoeconomic analysis.

Item	**Value**
***RTY* (hours)**	2688
φ (-)	1.06
n_y (years)	20
i_r (%)	6.5

Costs of destroyed exergy were obtained from Equation 48, considering that prices of the product were fixed [24]:

$$\dot{C}_D = c_{F,k}\dot{E}_{D,k}\ (hr) \qquad 48$$

Total costs represent the sum of the investment costs plus the costs of destroying exergy (equation 49) [32]:

$$\dot{Z}_k^{Total} + \dot{C}_D\ (\$/hr) \qquad 49$$

Relative cost difference expresses the relative increase in the average cost per unit of exergy between the inputs and outputs of a component (Equation 50) [33]:

$$r_k = \frac{c_{P,k} - c_{F,k}}{c_{F,k}} \times 100 \quad (\%) \qquad 50$$

Exergoeconomic factor is the reason for the contribution of non-exergetic costs to the increase in total cost (Equation 51) [34]:

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + c_{F,k}\left(\dot{E}_{D,k}\right)} \times 100 \quad (\%) \qquad 51$$

Table 5 shows the cost balances and auxiliary equations for each component of the power system with Stig cycle and air cooling by compression refrigeration machine. The auxiliary equations are derived from the configuration of the system components and are obtained by applying the equations F and P [24].

F Equation: The total cost associated with removing exergy from an exergy stream in one component must equal the cost of the stream at which the exergy supplied to the same flow in the upstream component was removed. The exergy difference of this current between input and output is considered in the definition of fuel for the component.

P Equation: Each unit of exergy that is supplied to any current associated with the product of a component is of equal average cost $c_{(P,\,k)}$. This cost is calculated from the equilibrium cost and equations F.

Table 5 Cost balance and auxiliary equations for each power system component with Stig and IAC cycle

Component	Balance of costs and auxiliary equations
AC	$c_2\dot{E}_2 - c_1\dot{E}_1 = c_{41}\dot{E}_{41} - c_{42}\dot{E}_{42} + \dot{Z}_{AC}$ $c_1 = 0$
LPC	$c_3\dot{E}_3 - c_2\dot{E}_2 = c_{elect}\dot{W}_{LPC} + \dot{Z}_{LPC}$ $C_{elect} = 0.00001999\ USD/kJ$
HPC	$c_4\dot{E}_4 - c_3\dot{E}_3 = C_{elect}\dot{W}_{HPC} + \dot{Z}_{HPC}$
Air humidifier	$c_5\dot{E}_5 = c_4\dot{E}_4 + c_{10}\dot{E}_{10}$
CC	$c_6\dot{E}_6 - c_5\dot{E}_5 - c_{11}\dot{E}_{11} = c_9\dot{E}_9 + \dot{Z}_{CC}$ $c_9 = 0.00004919\frac{USD}{kJ}\ (natural\ gas\ cost)$
GT	$c_{elect}\dot{W}_{GT} = c_6\dot{E}_6 + c_{12}\dot{E}_{12} - c_7\dot{E}_7 + \dot{Z}_{GT}$ $\frac{c_7\dot{E}_7}{\dot{E}_7} = \frac{c_6\dot{E}_6 + c_{12}\dot{E}_{12}}{\dot{E}_6 + \dot{E}_{12}}\ (F)$
Gen	$c_{Pelect}\dot{P}_{Elect} = c_{elect}\dot{W}_{net} + \dot{Z}_{Gen}$
HRSG	$c_{13}\dot{E}_{13} - c_{14}\dot{E}_{14} + c_{15}\dot{E}_{15} - c_{16}\dot{E}_{16} + c_{17}\dot{E}_{17} - c_{18}\dot{E}_{18} + c_{19}\dot{E}_{19} - c_{20}\dot{E}_{20} = c_7\dot{E}_7 - c_8\dot{E}_8 + \dot{Z}_{HRSG}$ $c_7 = c_8\ (F)$ $c_{14} = c_{16} = c_{17}\ (P)$ $\frac{c_{13}\dot{E}_{13} - c_{14}\dot{E}_{14}}{\dot{E}_{13} - \dot{E}_{14}} = \frac{c_{15}\dot{E}_{15} - c_{16}\dot{E}_{16}}{\dot{E}_{15} - \dot{E}_{16}} = \frac{c_{17}\dot{E}_{17} - c_{18}\dot{E}_{18}}{\dot{E}_{17} - \dot{E}_{18}} = \frac{c_{19}\dot{E}_{19} - c_{20}\dot{E}_{20}}{\dot{E}_{19} - \dot{E}_{20}}\ (P)$
MWP	$c_{18}\dot{E}_{18} - c_{19}\dot{E}_{19} = c_{elect}\dot{W}_{MUP} + \dot{Z}_{MUP}$
FDP	$c_{22}\dot{E}_{22} - c_{21}\dot{E}_{21} = c_{elect}\dot{W}_{FDP} + \dot{Z}_{FDP}$
CT	$c_{30}\dot{E}_{30} - c_{29}\dot{E}_{29} = c_{23}\dot{E}_{23} - c_{22}\dot{E}_{32} + c_{elect}\dot{W}_{FAN} + \dot{Z}_{CT}$ $c_{22} = 0$ $c_{29} = 0$
CP	$c_{24}\dot{E}_{24} - c_{23}\dot{E}_{23} = c_{elect}\dot{W}_{Pcond} + \dot{Z}_{Pcond}$ $c_{23} = 0\ (\text{Recirculating fluid, not consumable})$
CH1 C	$c_{31}\dot{E}_{31} - c_{34}\dot{E}_{34} = c_{elect}\dot{W}_{CCh1} + \dot{Z}_{CCh1}$ $c_{34} = 0\ (\text{Recirculating fluid, not consumable})$
CH1 Cond	$c_{26}\dot{E}_{26} - c_{25}\dot{E}_{25} = c_{31}\dot{E}_{31} - c_{32}\dot{E}_{32} + \dot{Z}_{CondCh1}$ $c_{31} = c_{32}\ (\text{F})$
CH1 EV	$c_{33}\dot{E}_{33} = c_{32}\dot{E}_{32} + \dot{Z}_{EVCh1}$
CH1 Evap	$c_{34}\dot{E}_{34} - c_{33}\dot{E}_{33} = c_{39}\dot{E}_{39} - c_{40}\dot{E}_{40} + \dot{Z}_{EvapCh1}$ $c_{34} = c_{33}\ (P)$
CH2 C	$c_{35}\dot{E}_{35} - c_{38}\dot{E}_{38} = c_{elect}\dot{W}_{CCH2} + \dot{Z}_{CCH2}$ $c_{38} = 0\ (\text{Recirculating fluid, not consumable})$

CH2 Cond	$c_{28}\dot{E}_{28} - c_{27}\dot{E}_{27} = c_{35}\dot{E}_{35} - c_{36}\dot{E}_{36} + \dot{Z}_{CondCh2}$ $c_{36} = c_{35}$
CH2 EV	$c_{37}\dot{E}_{37} = c_{36}\dot{E}_{36} + \dot{Z}_{EVCh2}$
CH2 Evap	$c_{38}\dot{E}_{38} - c_{37}\dot{E}_{37} = c_{40}\dot{E}_{40} - c_{41}\dot{E}_{41} + \dot{Z}_{EvapCh2}$ $c_{38} = c_{37}\ (P)$
EP	$c_{39}\dot{E}_{39} - c_{42}\dot{E}_{42} = c_{elect}\dot{W}_{Pevap} + \dot{Z}_{Pevap}$ $c_{42} = 0$ (Recirculating fluid, not consumable)

The next stage of this work consists of elaborating the advanced exergetic and exergoeconomic analysis, as an option to determine the improvement potentials and the exergetic acosts.

2.4. Advanced exergetic and exergoeconomic analysis

2.4.1. Destruction of unavoidable and avoidable exergy

Unavoidable destroyed exergy ($\dot{E}_{D,K}^{UN}$), is part of destruction of exergy within a component that cannot be reduced due to technological or economic limitations *k*th component [35]. Avoidable destroyed exergy ($\dot{E}_{D,K}^{AV}$), is the part of part of destruction of exergy that can be reduced with optimization processes [36].

2.4.1.1. Destruction of endogenous and exogenous exergy

Endogenous destroyed exergy $\left(\dot{E}_{D,K}^{EN}\right)$ is the part of destruction of exergy due to the internal functioning of the *k*th component [37]. $\dot{E}_{D,K}^{EN}$ is calculated when the remaining parts operate in ideal processes except for the *k*th component that works under its real efficiency [20]. Exogenous destroyed Exergy $\left(\dot{E}_{D,K}^{EX}\right)$ is the part of destruction of exergy caused by the interaction of the remaining parts with the study component) [38].

Combination of exergy destruction in its avoidable / unavoidable and exogenous / endogenous parts

Combining the divisions of avoidable/evitable and exogenous/endogenous exergy destruction gives more information on system performance [39]. The destruction of endogenous unavoidable $\left(\dot{E}_{D,k}^{UN,EN}\right)$, is the part of destruction of exergy, which cannot be reduced by technological limitations *k*th component [40]. The destruction of endogenous avoidable exergy $\left(\dot{E}_{D,k}^{AV,EN}\right)$, is part of the destruction of exergy that can be reduced by improving the efficiency of the study component [41]. The destruction of exogenous unavoidable exergy $\left(\dot{E}_{D,k}^{UN,EX}\right)$, is the part of destruction of exergy, which cannot be reduced by the global technological limitations originating from the remaining components [15]. Destruction of exogenous avoidable exergy $\left(\dot{E}_{D,k}^{AV,EX}\right)$, is the part of destruction of exergy that can be reduced by improving the overall efficiency of the components [16]. The equations to splitting the exergy destruction is show in Table 6.

Division of exergy destruction costs and unavoidable/evitable and endogenous/exogenous investment costs

The equations of the division of exergy destruction, the costs of exergy destruction and the costs of investment are shown in the Table 6 [42] [34] [43] [44].

Table 6 Equations used for advanced exergoecnomic analysis

Term	**Splitting the exergy destruction**	**Splitting the exergy destruction cost**	**Splitting the investment costs**
Unavoidable	$\dot{E}_{D,K}^{UN} = \dot{E}_{P,k}\left(\frac{\dot{E}_{D,k}}{\dot{E}_{P,k}}\right)^{UN}$	$\dot{C}_{D,K}^{UN} = c_{f,k}\dot{E}_{D,K}^{UN}$	$\dot{Z}_{D,k}^{UN} = \dot{E}_{P,k}\left(\frac{\dot{Z}_{k}}{\dot{E}_{P,k}}\right)^{UN}$
Avoidable	$\dot{E}_{D,K}^{AV} = \dot{E}_{D,k} - \dot{E}_{D,K}^{UN}$	$\dot{C}_{D,K}^{AV} = c_{f,k}\dot{E}_{D,K}^{AV}$	$\dot{Z}_{D,k}^{AV} = \dot{Z}_{k} - \dot{Z}_{D,k}^{UN}$
Endogenous	$\dot{E}_{D,K}^{EN}$ *is calculated as suggested in* [20].	$\dot{C}_{D,K}^{EN} = c_{f,k}\dot{E}_{D,K}^{EN}$	$\dot{Z}_{D,k}^{EN} = \dot{E}_{P,k}^{EN}\left(\frac{\dot{Z}_{,k}}{\dot{E}_{P,k}}\right)^{Real}$
Exogenous	$\dot{E}_{D,K}^{EX} = \dot{E}_{D,k} - \dot{E}_{D,K}^{EN}$	$\dot{C}_{D,K}^{EX} = c_{f,k}\dot{E}_{D,K}^{EX}$	$\dot{Z}_{D,k}^{EX} = \dot{Z}_{k} - \dot{Z}_{D,k}^{EN}$

Unavoidable Endogenous	$\dot{E}_{D,k}^{UN,EN} = \dot{E}_{P,k}^{EN}\left(\frac{\dot{E}_{D,k}}{\dot{E}_{P,k}}\right)^{UN}$	$\dot{C}_{D,K}^{UN,EN} = c_{f,k}\dot{E}_{D,K}^{UN;EN}$	$\dot{Z}_{D,k}^{UN,EN} = \dot{E}_{P,k}^{EN}\left(\frac{\dot{Z}_{,k}}{\dot{E}_{P,k}}\right)^{UN}$
Avoidable Endogenous	$\dot{E}_{D,k}^{AV,EN} = \dot{E}_{D,k}^{EN} - \dot{E}_{D,k}^{UN,EN}$	$\dot{C}_{D,K}^{AV,EN} = c_{f,k}\dot{E}_{D,K}^{AV,EN}$	$\dot{Z}_{D,k}^{AV,EN} = \dot{Z}_{D,k}^{EN} - \dot{Z}_{D,k}^{UN,EN}$
Unavoidable Exogenous	$\dot{E}_{D,k}^{UN,EX} = \dot{E}_{D,k}^{UN} - \dot{E}_{D,k}^{UN,EN}$	$\dot{C}_{D,K}^{UN,EX} = c_{f,k}\dot{E}_{D,K}^{UN,EX}$	$\dot{Z}_{D,k}^{UN,EX} = \dot{Z}_{D,k}^{UN} - \dot{Z}_{D,k}^{UN,EN}$
Avoidable Exogenous	$\dot{E}_{D,k}^{AVEX} = \dot{E}_{D,k}^{EX} - \dot{E}_{D,k}^{UN,EX}$	$\dot{C}_{D,K}^{AV,EX} = c_{f,k}\dot{E}_{D,K}^{AV,EX}$	$\dot{Z}_{D,k}^{AV,EX} = \dot{Z}_{D,k}^{EX} - \dot{Z}_{D,k}^{UN,EX}$

At the end of the methodology and system characterization stage, the proposed model was validated to guarantee the usefulness of the analysis carried out, and the results obtained.

3. Validation

In this study, a exergetic, and exergoeconomic analysis is carried out with the conventional and advanced methodology of a gas power system with air cooling by compression refrigeration machine and steam injection, the parameters of the system used for the validation process are defined in table 11. For this purpose, a thermodynamic model was developed in EES, where the properties of each one of the system currents and the parameters used in this study are obtained [45]. To guarantee the usefulness of the proposed methodology as a tool to develop the analysis offered in this research, the mathematical validation of the thermodynamic model was elaborated, which constitutes the basis of the analysis of all the methodologies integrated into this work. Thermodynamic model is validated by comparing the operating parameters of the study system (power system with Stig cycle and air cooling) with those obtained from the thermodynamic model. The validation of the thermodynamic model is summarized in Table 7.

Table 7 Validation of the thermodynamic model of power system with Stig cycle and air cooling

Stig cycle and air cooling	**Actual Data**	**Thermodynamic model**	**Difference(%)**
LPC inlet temperature (°C)	12.44	12	**3,6%**
LPC outlet temperature (°C)	113	111.8	**1,1%**
HPC output temperature (°C)	534	551	**3,1%**
Gas turbine outlet temperature (°C)	440	456.8	**3,7%**
HRSG outlet temperature (°C)	240	250.6	**4,2%**
Output power (kW)	45000	45742	**1,6%**
Thermal efficiency (%)	38	36.69	**3,6%**

Table 7 shows results obtained from the comparison of the operating condition of the power generation system installed in Cartagena-Colombia with the Stig cycle and air cooling at 12°C. This system was used as a reference for the analysis developed in this research. From these results is possible to observe a maximum difference of of 4.2% obtained in the temperature of the exit of the recovery boiler (HRSG). In the case of minimum difference obtained from the validation, this was of 1.1% obtained for the exit temperature of the low-pressure compressor (LPC). In general, differences obtained between the compared parameters was less than 5%. These differences can be caused because, unfortunately, there are some operating parameters that do not have a measurement system to determine them during the operation of the study plant. In this situation, it was necessary to make adjustments to the different unknown values in the mathematical model elaborated in this work to obtain results close to those obtained in the study plant.

4. Results

Thermodynamic properties of power plant state shown in figure 1 are shown in Table 8.

Table 8 mass flow rate, temperature, pressure, y exergy to power plant with gas turbine and steam injection with air cooling at compressor inlet to 12°C

State	Substance	$\dot{m}$ (Kg/s)	T $(°C)$	P (KPa)	e^{PH} (KJ/Kg)	e^{CH} $(KJ/Kg$
0	Water	0	3,42	38,15	0	0
0	R-123	0	3,42	38,15	0	0
1	Air	122,2	32	101,3	80,77	7,097
2	Air	122,2	12	100	77,28	6,307
3	Air	122,2	111,8	250	167,9	6,307
4	Air	122,2	551	3000	625,3	6,307
5	Air	122,2	551	3000	651,8	10,01
6	Combustion gases	131,8	1150	2850	1299	157,4
7	Combustion gases	136,2	456,8	98,28	306,5	152,9
8	Combustion gases	136,2	250,6	100	82,12	152,9
9	Natural Gas	2,573	85,6	3404	655,6	50170
10	Steam	3,834	299,3	3249	1190	527,3
11	Steam	2,092	299,3	3249	1190	527,3
12	Steam	4,38	247	1083	1033	527,3
13	Steam	5,926	315	3420	86,93	527,3
14	Water	5,926	180	3600	86,93	527,3
15	Steam	4,38	260	1140	86,93	527,3
16	Water	4,38	180	1200	86,93	527,3
17	Water	10,31	180	5838	86,93	527,3
18	Water	10,31	120	6145	86,93	527,3
19	Water	10,31	119,3	1463	81,61	527,3
20	Water	10,31	30,68	1540	6,772	527,3
21	Water	10,31	30,56	101,3	5,294	527,3
22	Water	324,9	38,89	317,9	9,05	10,05
23	Water	320,9	30,56	113,2	5,306	527,3
24	Water	324,9	30,58	334,6	5,533	527,3
25	Water	173	30,58	334,6	5,533	527,3
26	Water	173	38,89	317,9	9,047	527,3
27	Water	151,9	30,56	334,6	5,527	527,3
28	Water	151,9	38,89	317,9	9,047	527,3
29	Air	280,2	28	101,3	80,02	5,787
30	Air	280,2	33,5	101,3	81,58	7,384
31	R-123	35,62	46,15	159,8	22,84	0
32	R-123	35,62	40,99	159,8	2,632	0

33	R-123	35,62	7,526	45,61	0,5629	0
34	R-123	35,62	7,526	45,61	2,712	0
35	R-123	31,19	47,26	159,8	22,96	0
36	R-123	31,19	40,99	159,8	2,632	0
37	R-123	31,19	3,424	38,15	0.0127	0
38	R-123	31,19	3,424	38,15	0,05908	0
39	Water	224,2	14,59	493,6	1,376	527,3
40	Water	224,2	9,13	468,9	0,6745	527,3
41	Water	224,2	4,44	445,5	0,4151	527,3
42	Water	224,2	14,57	168	1,047	527,3

Results of the exergetic and exergoeconomic analysis are summarized in Table 10, from which it is observed that the exergetic efficiency of the power system is 34.98% and the destruction of total exergy is 65256.05 kW, where 68.51% corresponds to the gas turbine and electric generator, 28.90% is due to components necessary for the steam injection process, and only 2.59% compares to the air cooling system. The elements with the most significant exergy destruction are CC (23288 kW), HRSG (18845 kW), and GT (16384 kW). The combustion chamber, despite being the component that destroys the most exergy due to the irreversibilities present in the chemical combustion reaction, has a high exergetic efficiency as obtained by Fallah et al. [40]. Most of the inefficiencies in the HRSG are presented by radiation, conduction, and in the exhaust gases, which by reducing a few hundred of the components that have the highest ratio of exergy transformation of the fuel into exergy destroyed and consequently those that have the lowest exergetic efficiency are CH2 EV, CH2 Evap, CH1 EV, AC, HRSG, and CH1 EVAP. For the exergoeconomic analysis, the cost functions for each power system component listed in Table 9 were used. The total exergy destruction costs of the power system are \$3066.73/hr, the elements with the highest total costs are GT (\$1639.44/GJ), HRSG (\$1063.33/GJ), CC (\$603.30/GJ), HPC (\$453.91/GJ), and LPC (\$213.21/GJ). A low exergoeconomic component means that total costs are high compared to investment costs, which merits a review of whether higher investment reduces overall costs by lowering exergy destruction costs. The components with the lowest exergoeconomic factors are HRSG, CH2 Cond and CH1 Cond, AC, CH2 Evap, and CH1 Evap. The improvement of this equipment should be sought by increasing the investment in these components; such investment should be directed at improving heat transfer between fluids and decreasing heat exchange with the environment. With the relative cost difference, it is known which are the components that most increase the average cost of the product. The elements with the most significant relative cost difference are CH2 Evap, CH2 EV, CH1 EV, AC, HRSG, MWP, and CH1 Evap. Of all this, it can be indicated that the HRSG and the evaporators of the refrigeration machine are the components that worst exergéticos and exergoeconómicos indicators presents, therefore, must be intervened increasing its investment to reduce the destruction of exergía. CH1 Evap and CH2 Evap, despite being the most inefficient components, do not represent a significant impact on global indicators.

Table 9 Cost Equations of Power System Components with Stig Cycles and Air Cooling

Component	Equipment acquisition cost function	Source
Air cooler	$PEC_{AC} = 130\left(\frac{A}{0.093}\right)^{0.78}$	[46]
Compressor	$PEC_{C} = 7900\left(\frac{\dot{W}}{0.746}\right)^{0.62}$	[42]
Combustion chamber	$PEC_{CC} = \left(\frac{46.08 m_{gases}}{0.995 - \frac{P_i}{P_e}}\right) \times \left(1 + e^{(0.018 T_e - 26.4)}\right)$	[47]
Gas turbine	$PEC_{GT} = \dot{W}_{GT}\left(1318.5 - 98.328 \ln\left(\dot{W}_{GT}\right)\right)$	[47]
Generator	$PEC_{Gen} = 60 \times \dot{W}_{net}$	[48]
HRSG	$PEC_{HRSG} = 8500 - 406(A_{HRSG})^{0.85}$	[47]
Cooling tower	$PEC_{CT} = 253226.835\left(\frac{\dot{Q}_C}{3600}\right)\left(-0.6936 \ln\left(\frac{T_{cw,1} - T_{cw,o}}{2} - T_{wb,amb}\right) + 2.1898\right)$	[49]
Pump	$PEC_{Pump} = 3540 \dot{W}_{bomba}{}^{0.71}$	[50]

Evaporator	$PEC_{Evap} = 1.3(190 + 310A)$	[51]
Condenser	$PEC_{Cond} = 1.3(190 + 310A)$	[51]

Table 10 Exergetic and exergoeconomic analysis of the power system with Stig cycle and air cooling at 12°C.

Compo-nent	E_F (kW)	E_P (kW)	E_D (kW)	ε (%)	y_D (%)	y_D^* (%)	c_f ($/GJ)	c_P ($/GJ)	$\dot{C}_D$ ($/hr)	$\dot{Z}$ ($/hr)	$\dot{Z} + \dot{C}_D$ ($/hr)	r (%)	f (%)
AC	522	141.70	380.50	27.14	72.86	0.58	85.31	337.33	116.86	3.32	120.18	295.41	2.76
LPC	12676	11077	1599	87.39	12.61	2.45	16.43	21.77	94.59	118.62	213.21	32.51	55.63
HPC	58409	55906	2503.	95.71	4.29	3.84	16.43	18.69	148.05	305.86	453.91	13.76	67.38
CC	130783	107495	23288.	82.19	17.81	35.69	6.42	7.82	537.96	65.34	603.30	21.93	10.83
GT	136259	119877	16382	87.98	12.02	25.10	15.53	19.33	915.84	723.60	1,639.4	24.46	44.14
Gen	46675	45742	933.	98.00	2.00	1.43	19.33	20.08	64.93	57.46	122.39	3.85	46.95
HRSG	30551.	11706	18845	38.32	61.68	28.88	15.53	40.76	1053.7	9.64	1,063.	162.42	0.91
MWP	64.78	54.83	9.95	84.64	15.36	0.02	16.42	31.81	0.59	2.45	3.04	93.65	80.63
MWP	18.90	15.22	3.68	80.53	19.47	0.01	16.43	39.03	0.22	1.02	1.24	137.60	82.43
CT	1328.	884.70	443.30	66.62	33.38	0.68	15.22	23.00	24.29	18.77	43.06	51.15	43.60
CP	91.79	73.95	17.84	80.56	19.44	0.03	16.43	32.17	1.06	3.14	4.19	95.82	74.82
CH1 C	881.50	717	164.50	81.34	18.66	0.25	16.43	29.07	9.73	22.89	32.62	76.94	70.18
CH1 Cond	719.90	607.90	112.00	84.44	15.56	0.17	29.06	34.53	11.72	0.22	11.94	18.82	1.83

CH1 EV	93.73	20.05	73.68	21.39	78.61	0.11	29.06	135.86	7.71	0.00	7.71	367.48	0.00
CH1 Evap	157.30	76.55	80.75	48.66	51.34	0.12	66.12	136.64	19.22	0.19	19.41	106.67	0.98
CH2 C	877.70	714.30	163.40	81.38	18.62	0.25	16.43	29.06	9.66	22.81	32.47	76.90	70.24
CH2 Cond	634.00	534.90	99.10	84.37	15.63	0.15	29.05	34.55	10.37	0.19	10.56	18.91	1.83
CH2 EV	82.08	0.40	81.68	0.48	99.52	0.13	29.06	6022.7	8.54	0.00	8.54	20627.3	0.00
CH2 Evap	58.16	1.45	56.71	2.49	97.51	0.09	149.85	6054.6	30.59	0.16	30.76	3940.6	0.54
EP	92.70	73.75	18.95	79.56	20.44	0.03	16.43	32.54	1.12	3.16	4.28	98.07	73.81
Total	130783	45742	65256	34.98	-	-	-	-	3066.7	1358.8	4425.6	-	-

4.1. Advanced Exergetic Analysis Results

Advanced exergetic analysis of the power system with the Stig cycle and 12 °C air cooling is shown in Table 11. Considerations taken into account for the study of the proposed method for the operating conditions, ideal conditions, and certain conditions are listed in Table 11.

Table 11 Assumptions used for advanced exergoeconomic analysis

Component	Theoretical Conditions.	Operating Conditions	Unavoidable Conditions
AC	$\Delta T = 0°C$	$\Delta T = 4.36°C$	$\Delta T = 0.5°C$
LPC	$\eta = 1$ $\eta_{Mec} = 1$	$\eta = 0.85$ $\eta_{Mec} = 0.985$	$\eta = 0.9$ $\eta_{Mec} = 1$
HPC	$\eta = 1$ $\eta_{Mec} = 1$	$\eta = 0.84$ $\eta_{Mec} = 0.985$	$\eta = 0.9$ $\eta_{Mec} = 1$
CC	$\Delta P = 0\%$ $Q_{Loss} = 0\%$ $\lambda = 2.847$	$\Delta P = 5\%$ $Q_{Loss} = 2\%$ $\lambda = 2.847$	$\Delta P = 2\%$ $Q_{Loss} = 0.5\%$ $\lambda = 2$
GT	$\eta = 1$ $\eta_{Mec} = 0.985$	$\eta = 0.888$ $\eta_{Mec} = 0.985$	$\eta = 0.9$ $\eta_{Mec} = 1$
Gen	$\eta = 1$	$\eta = 0.98$	$\eta = 0.995$
HRSG	$\Delta T = 0°C$ $\eta = 1$ $\Delta P = 0\%$	$\Delta T = 140°C$ $\eta = 0.88$ $\Delta P = 5\%$	$\Delta T = 0.5°C$ $\eta = 0.88$ $\Delta P = 3\%$

MWP	$\eta = 1$ $\eta_{Mec} = 1$	$\eta = 0.8$ $\eta_{Mec} = 0.985$	$\eta = 0.9$ $\eta_{Mec} = 1$
MWP	$\eta = 1$ $\eta_{Mec} = 1$	$\eta = 0.8$ $\eta_{Mec} = 0.985$	$\eta = 0.9$ $\eta_{Mec} = 1$
CT	$\Delta T = 0°C$	$\Delta T = 2.56°C$	$\Delta T = 0.5°C$
CP	$\eta = 1$ $\eta_{Mec} = 1$	$\eta = 0.8$ $\eta_{Mec} = 0.985$	$\eta = 0.9$ $\eta_{Mec} = 1$
CH1C	$\eta = 1$ $\eta_{Mec} = 1$	$\eta = 0.8$ $\eta_{Mec} = 0.985$	$\eta = 0.9$ $\eta_{Mec} = 1$
CH1 Cond	$\Delta T = 0°C$ $\Delta P = 0\%$	$\Delta T = 2.1°C$ $\Delta P = 5\%$	$\Delta T = 0.5°C$ $\Delta P = 3\%$
CH1 EV	$s_{27} = s_{28}$	$h_{27} = h_{28}$	$h_{27} = h_{28}$
CH1 Evap	$\Delta T = 0°C$ $\Delta P = 0\%$	$\Delta T = 1.604°C$ $\Delta P = 5\%$	$\Delta T = 0.5°C$ $\Delta P = 2\%$
CH2 C	$\eta = 1$ $\eta_{Mec} = 1$	$\eta = 0.8$ $\eta_{Mec} = 0.985$	$\eta = 0.9$ $\eta_{Mec} = 1$
CH2 Cond	$\Delta T = 0°C$ $\Delta P = 0\%$	$\Delta T = 2.1°C$ $\Delta P = 5\%$	$\Delta T = 0.5°C$ $\Delta P = 3\%$
CH2 EV	$s_{37} = s_{34}$	$h_{37} = h_{34}$	$h_{37} = h_{34}$
CH2 Evap	$\Delta T = 0°C$ $\Delta P = 0\%$	$\Delta T = 1.016°C$ $\Delta P = 5\%$	$\Delta T = 0.5°C$ $\Delta P = 3\%$
EP	$\eta = 1$ $\eta_{Mec} = 1$	$\eta = 0.8$ $\eta_{Mec} = 0.985$	$\eta = 0.9$ $\eta_{Mec} = 1$

The results of the advanced exergetic analysis *of a Gas Power and Steam Injection System with Air Cooling at 12°C are shown below*, data shown in *Figure 3* indicated that from total exergy destruction (65256 kW), the 81% of it, cannot be reduced by the technological and economic limitations of the system components, and only 12395.68 kW of total exergy destruction is avoidable exergy destruction. The destruction of exergy mostly originates in the proper functioning of the components, i.e. 58.9% of the total exergy destruction is endogenous exergy destruction and 41.1% remaining is due to the interaction between the components of the power system under study (*Figure 4*). The components with the greatest avoidable exergy are GT (3002.72 kW), HRSG (2917.54 kW), CC (2605.88 kW) and HPC (1940.31 kW), while the entire air cooling system would only achieve a reduction of exergy destruction of 542.80 kW. From the data shown in figure 5, the components in which the interaction with the remaining components produces greater exergy destruction than the irreversibilities inherent in the operation of themselves are HRSG, AC, MWP and EP. 47.1% of the exergy destruction is unavoidable endogenous, followed by the unavoidable exogenous with 33.9%, the endogenous avoidable exergy destruction and exogenous avoidable exergy are only 11.8 and 7.2% respectively. Much of the components of the power system to reduce their exergy destruction is necessary to improve their own performance while some components such as HRSG and GT need to improve the overall configuration of the system.

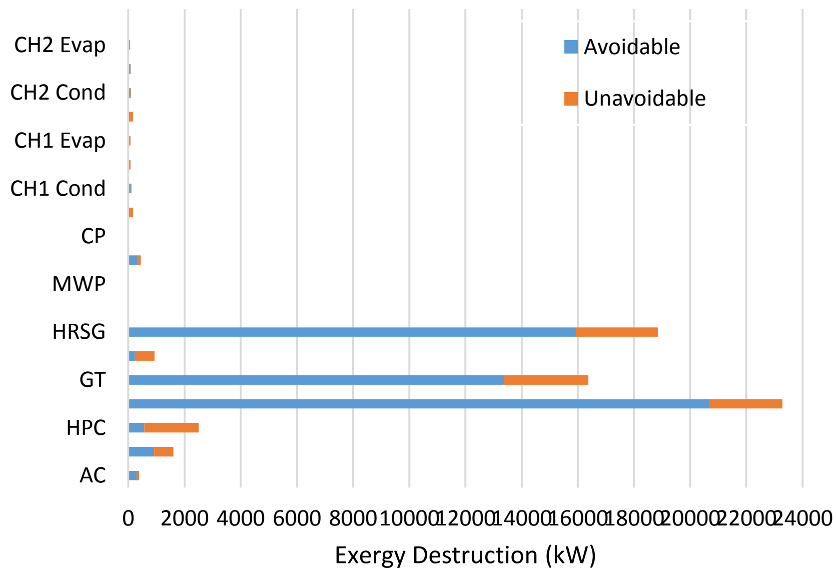


Figure 3 Unavoidable and avoidable exergy destruction of a Gas Power and Steam Injection System with Air Cooling at 12°C.

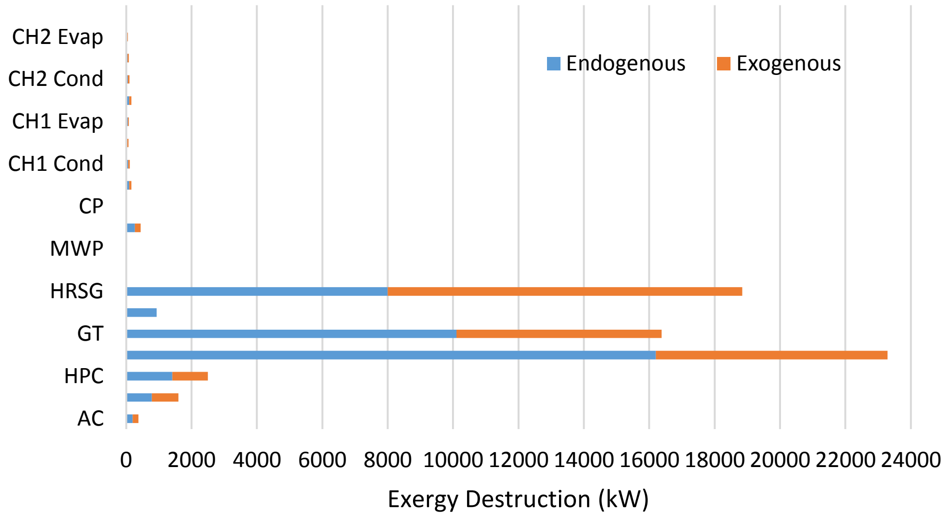


Figure 4 Endogenous and exogenous exergy destruction of a Gas Power and Steam Injection System with Air Cooling at 12°C.

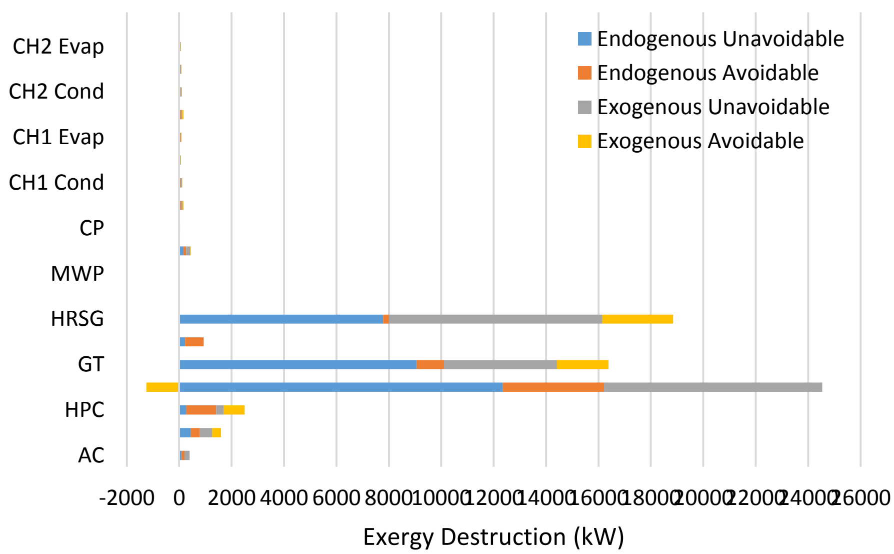
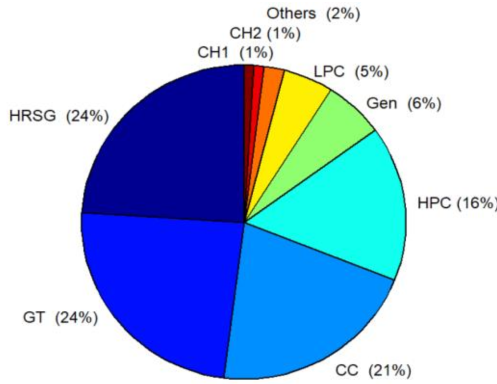
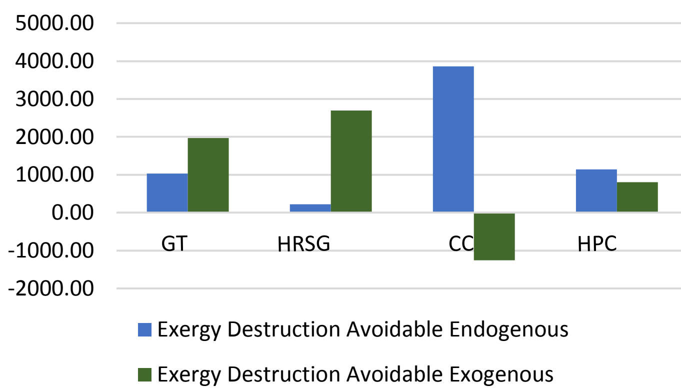


Figure 5 The value of each of the exergy destruction of a Gas Power and Steam Injection System with Air Cooling at 12°C.

Figure 6 shows percentages of avoidable exergy destruction of the gas power system components with steam injection and air cooling by refrigeration machine at 12°C and endogenous/exogenous avoidable exergy destruction of the elements with highest potential for improvement.



(a)



(b)

Figure 6 Avoidable exergy destruction of a Gas Power and Steam Injection System with Air Cooling at 12°C.

Figure 6 highlights the destruction of avoidable exergy, endogenous avoidable, and exogenous preventable (GT, HRSG, CC, and HPC). To improve HPC performance, more significant air cooling can be explored to reduce exogenous exergy destruction (41.2 %). The value of destruction of avoidable exogenous exergy is negative in the CC, due to the fact that when working in the unavoidable condition, greater efficiency of combustion is reached and therefore greater heat transfer is presented, causing a greater generation of entropy than in the real condition of operation. For the HRSG is recommended to increase investment to increase efficiency and reduce radiation and convention losses, considering that from the avoidable exergy destruction greater extent is exogenous.

4.2. Advanced exergoeconomic analysis results

Figure 7, Figure 8 and figure 9 shows costs of advanced exergy destruction of the gas power system with steam injection and air cooling. Total exergy destruction costs are $3066.73/hr, these costs (56.16%) originate from the proper functioning of the components, and the remaining 44.61% is due to the interaction between the components. From the $3066.73/hr of the total exergy destruction costs of the power system, only $662.98/hr can be avoided, of the avoidable exergy destruction, only 46.58% improving the own performance of each component and the remaining 53.42% can be avoided by performing a global optimization of the system. The 78.4% of the total exergy destruction costs cannot be reduced by technological and economic limitations of the system components and comprehensive operation, where 354.16 $/hr of the exergy destruction costs can be reduced by improving the own performance of the elements, and 308.82 $/hr of the expenses must be reduced by starting from a global optimization of the system. Power system components that present avoidable exergy destruction cost more significant than the unavoidable exergy destruction cost are the high-pressure compressor, the generator, the pumps of the air cooling and steam injection systems, and the compressors of the two refrigeration machines. The HRSG, CC, HPC, and LPC have a higher cost of exogenous than endogenous exergy destruction. The exogenous avoidable exergy destruction costs are higher than the endogenous avoidable exergy destruction costs in HRSG ($150.63/hr), GT ($109.96/hr), CH1 EV ($1.89/hr) and EP ($0.34/hr).

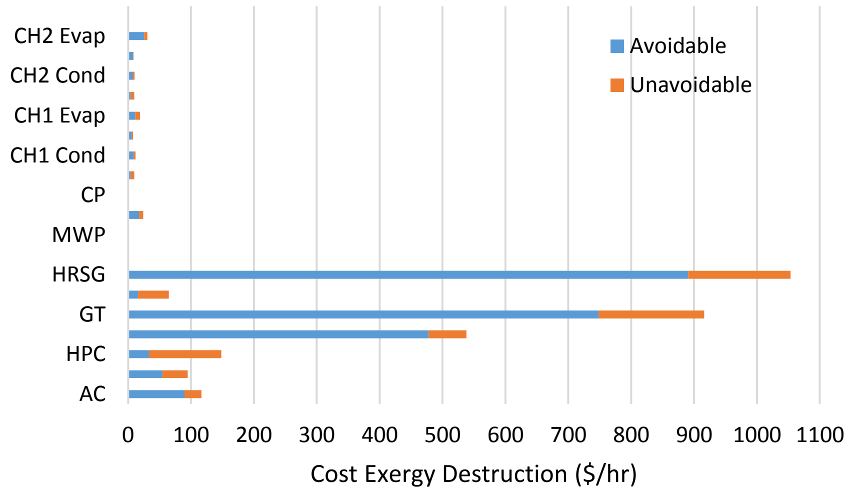


Figure 7 Unavoidable and avoidable cost exergy destruction of a Gas Power and Steam Injection System with Air Cooling at 12°C.

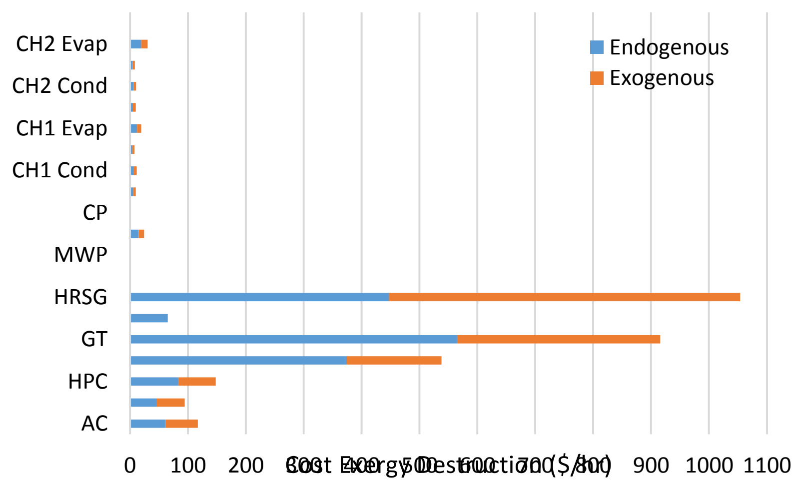


Figure 8 Endogenous and exogenous cost exergy destruction of a Gas Power and Steam Injection System with Air Cooling at 12°C.

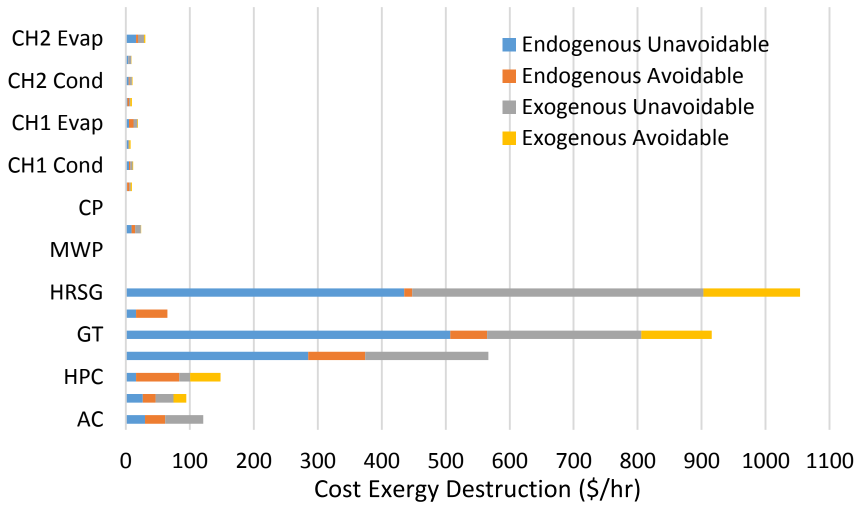


Figure 9 The value of each of the cost exergy destruction of a Gas Power and Steam Injection System with Air Cooling at 12°C.

Investment costs divided into their unavoidable/evitable, endogenous/exogenous, and combination of both for the gas power system and steam injection with air cooling are shown in Figure 10, Figure 11 and figure 12. Overall power system investment costs are $1358.84/hr, of which 97.75% is unavoidable. Investment costs of endogenous origin are $839.33/hr, while those of exogenous source reach $519.51/hr, for HPC, LPC, HRSG, AC, EP, and MWP exogenous investment costs are higher than endogenous. Avoidable investment costs are $30.53/hr, 58.43% of this is endogenous preventable, and 41.57% is unavoidable exogenous, the components that present higher endogenous avoidable investment costs are CC ($4.15/hr), GT ($8.60/hr) and LPC ($1.54/hr).

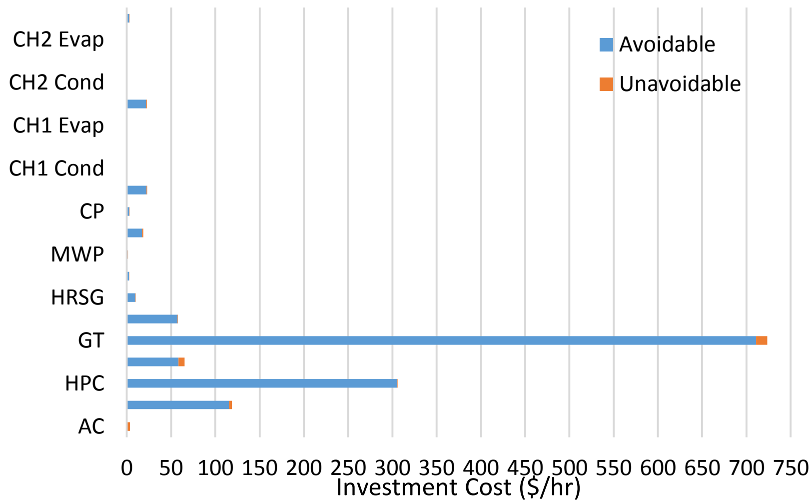


Figure 10 Unavoidable and avoidable investment cost of a Gas Power and Steam Injection System with Air Cooling at 12°C.

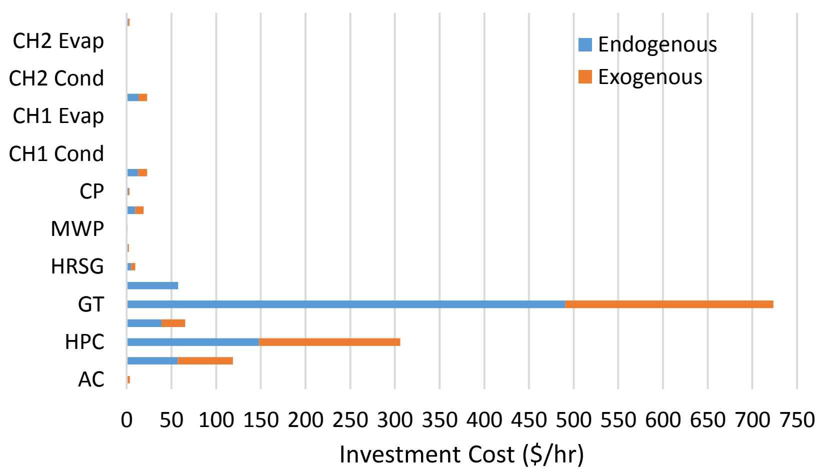


Figure 11 Endogenous and exogenous investment cost of a Gas Power and Steam Injection System with Air Cooling at 12°C.

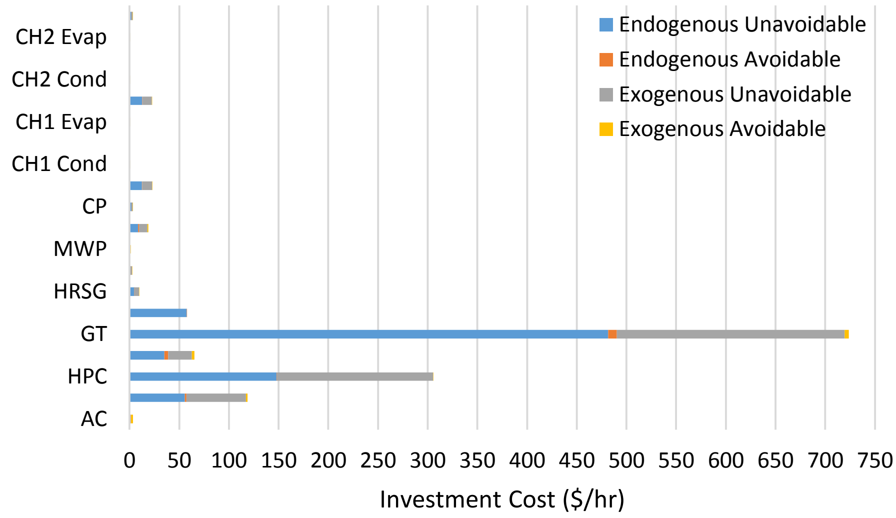


Figure 12 The value of each of the cost exergy destruction of a Gas Power and Steam Injection System with Air Cooling at 12°C.

From results of the conventional exergetic and exergoeconomic analysis, it can be highlighted that components with worst indicators and that would require an intervention to reduce the destruction of exergy are CC, HRSG, GT, AC, Evaporators and expansion valves of refrigeration machines. From information obtained in the advanced exergetic and exergoeconomic analysis the components mentioned above present to a greater extent exergy destruction and unavoidable exergy destruction costs, it is also appreciated that the intervention efforts of the parts should focus on the work consuming equipment and the GT, since their exergy destructions, exergy destruction costs, and investment costs are mainly avoidable and endogenous.

5. Conclusions

Results obtained from the validation show that the methodology (Thermodynamic Model) implemented in this work can be used as a tool to perform conventional and advanced exergetic and exergoeconomic analysis. In the validation, a maximum difference of 4.2%

was obtained for the output temperature of the recovery boiler. This methodology is applicable for any range of operation of a power plant with Stig cycle and air cooling at the compressor inlet, combined-cycle systems, or hybrid plants that have the system configuration described above.

Highest total costs ($\dot{Z}+\dot{C}_D$) were presented in GT, HRSG, CC, and HPC, which represent 37.04%, 24.03%, 13.63%, and 10.26% of total costs respectively. The HRSG presented high total price, low exergoeconomic factor, relative cost ratio, and low exergetic efficiency. It is recommended to explore the possibility of capital investments to improve its performance.

Components with highest exergy destructions were CC, HRSG, and GT, which resulted in 35.69%, 28.88%, and 25.10% of the plant's total exergy destruction, respectively. For the mentioned components, the destruction of exergy is more unavoidable; the removal of avoidable exergy is 11.2% in the CC, 15.4% in the HRSG, and 18.3% in GT of the total exergy destruction of each of the components. The highest potential for improvement is presented in the GT and the HRSG, which would be obtained by improving the overall configuration of the system, while the CC must develop its performance, i.e., combustion efficiency.

Most significant inefficiencies in the gas power generation system with heat recovery boilers are found in the CC, HRSG, and GT, also, the destruction of total exergy of the system is mostly unavoidable and endogenous as reported by Boyaghchi & Molaie in [21], Acikkalp & others in [19].

Highest costs of avoidable exergy destruction are presented in GT (167.87 $/hr), HRSG (163.13 $/hr) and HPC (114.77 $/hr), respectively; for GT and HRSG exergy destruction is mostly exogenous with 65.5% and 92.3% respectively, while in HPC 58.5% is endogenous. Avoidable investment costs represent only 2.25% of the total investment costs. The avoidable investment costs are low, and the exergy destruction costs are considerable. From this, to reduce the total costs of exergy destruction, efforts should be focused on making capital investments to improve the overall performance of the system.

Acknowledgements

Authors wish to thank the Technological University of Bolivar (UTB) for their cooperation and financing the development of this work; as well as, to professor Bienvenido Sarria Lopez for his valuable collaboration and management.

Bibliography

[1] G. T. Udeh y P. O. Udeh, «Comparative thermo-economic analysis of multi-fuel fired gas turbine power plant,» *Renewable Energy,* vol. 133, pp. 295-306, 2019.

[2] A. De Sa y S. Al zubaidy, «Gas turbine performance at varying ambient temperature,» *Applied Thermal Engineering,* vol. 31, pp. 2735-2739, 2011.

[3] G. Comodi, M. Renzi, F. Caresana y L. Pelagalli, «Enhancing Micro Gas Turbine Performance In Hot Climates Through Inlet Air Cooling Vapour Compression Technique,» *Applied Energy,* vol. 147, pp. 40-48, 2015.

[4] A. K. Mohapatra y Sanjay, «Thermodynamic assessment of impact of inlet air cooling techniques on gas turbine and combined cycle performance,» *Energy,* vol. 68, pp. 191-203, 2014.

[5] S. S. Baakeem, J. Orfi y H. Al-Ansary, «Performance improvement of gas turbine power plants by utilizing turbine inlet air-cooling (TIAC) technologies in Riyadh, Saudi Arabia,» *Applied Thermal Engineering,* 2018.

[6] V. Zare, «Performance improvement of biomass-fueled closed cycle gas turbine via compressor inlet cooling using absorpion refrigetration, thermoeconomic analysis and multi-objetic optimization,» *Energy conversion and Management,* vol. 215, p. 112946, 2020.

[7] R. Xue, C. Hu, V. Sethi, T. Nikolaidis y P. Pilidis, «Effect of steam addition on gas turbine combustor design and performance,» *Applied Thermal Engineering,* 2016.

[8] S.-J. Zhang, J.-L. Chi y Y.-H. Xiao, «Performance analysis of a partial oxidation steam injected gas turbine cycle,» *Applied Thermal Engineering,* vol. 91, pp. 622-629, 2015.

[9] A. Shukla y O. Singh, «Performance Evaluation of Steam Injected Gas Turbine Based Power Plant With Inlet Evaporative Cooling,» *Applied Thermal Engineering,* vol. 102, pp. 454-464, 2016.

[10] A. K. Shukla y O. Singh, «Thermodynamic analysis of stem-injected gas turbine cycle power plant with inlet air cooling,» *International journal of ambient energy*, pp. 1-26, 2016.

[11] A. K. Shukla y O. Singh, «Thermodynamic Investigation of Parameter Affecting the Execution of Stem Injected Cooled Gas Turbine Based Combined Cycle Power Plant with Vapor Absorption Inlet Air Cooling,» *Applied Thermal Engineeing,* vol. 122, pp. 380-388, 2017.

[12] H. Athari, S. Soltani, M. Rosen, S. Mohmoudi y T. Morosuk, «Gas turbine stema injection and combined power cycles using fog inlet cooling and biomass fuel: A Thermodynamic assessment,» *Renewable Energy,* pp. 95 - 103, 2016.

[13] H. Athari, S. Soltani, M. Rosen, M. Kordoghli y T. Morosuk, «Exergoeconomic study of gas turbine steam injection and combined power cycles using fog inlet cooling and biomass fuel,» *Renewable Energy,* pp. 715 - 726, 2016.

[14] A. Keçebas¸ y H. Gökgedik, «Thermodynamic evaluation of a geothermal power plant for advanced exergy analysis,» *Energy,* vol. 88, pp. 746-755, 2015.

[15] E. Acikkalp, h. Aras y A. Hepbasil, «Advanced exergy analysis of an electricity-generating facility using natural gas,» *Energy Conversion and Management,* vol. 82, pp. 146-153, 2014.

[16] F. A. Boyahchi y H. Molaine, «Sensitivity analysis of exergy destruction in a real combined cycle power plant based on advanced exergy method,» *Energy Conversion and Management,* vol. 99, pp. 374 - 386, 2015.

[17] E. Acikkalp, H. Aras y A. Hepbasli, «Advanced exergoeconomic analysis of a trigeneration system using a diesel-gas engine,» *Applied Thermal Engineering,* vol. 67, pp. 388-395, 2014.

[18] S. Anvari, R. Saray y K. Bahlouli, «Conventional and advanced exergetic and exergoeconomic analyses applied to a tri-generation cycle for heat, cold and power production,» *Energy,* vol. 91, pp. 925-939, 2015.

[19] CIOH Oceanografia Operacional, «Climatologia Del Caribe,» [En línea]. Available: https://www.cioh.org.co/meteorologia/Climatologia/ResumenCartagena4.php. [Último acceso: 10 10 2017].

[20] Y. Sohret, E. Acikkalp, A. Hepbasli y T. H. Karakoc, «Advanced exergy analysis of an aircraft gas turbine engine: SPlitting exergy destructios into parts,» *Energy,* pp. 1219-1228, 2015.

[21] D. Barreto, J. Fajardo y J. Campillo, «Determination of the optimal range of the compressor inlet air temperature in a power plant with Stig cycle through of advanced exergetic analysis,» *IMECE,* nº IMECE2019-10410, 2020.

[22] Y. A. Cengel y M. A. Boles, Thermodinamic, Mc Graw Hill, 2014.

[23] S. Sanaye, M. Amani y P. Amani, «4E modeling and multi-criteria optimization of CCHPW gas turbine plant with inlet air cooling and steam injection,» *Sustainable Energy Technologies and Assessments,* vol. 29, pp. 70-81, 2018.

[24] A. Bejan, G. Tsatsaronis y M. Moran, Thermal Desing and Optimazation, New York: John Wiley & Sons, 1996.

[25] H. Athari, S. Soltani, M. Rosen, S. M. Seyed Mahmoudi y T. Morosuk, «Comparative Exergoeconomic Analyses od Gas Turbine Steam Injection Cycles with and without Fogging Inlet Cooling,» *Sustainability*, vol. 7, pp. 12236-12257, 2015.

[26] M. Aminyavari, A. H. Mamaghani, A. Shirazi, B. Najafi y F. Rinaldi, «Exergetic, Economic, and Enviromental Evaluations and Multi-objetive Optimization of an Internal-Reforming SOFC-Gas Turbine Cycle Coupled with a Rankine Cycle,» *Applied Thermal Enginering*, 2016.

[27] T. Morosuk y G. Tsatsaronis, «Advanced exergetic evaluation of refrigeration machines using different wirking fluids,» *Energy*, vol. 34, pp. 2248-2258, 2009.

[28] R. Huang, J. Ling y V. Aute, «Comparison of approximation-assisted heat exchanger models for steady-state simulation of vapor compression system,» *Applied Thermal Engineering*, vol. 166, p. 114691, 2020.

[29] Z. Li, E. Chen, Y. Jing y S. Lv, «Thermodynamic relationship of subcooling power and increase of cooling output in vapor compression chiller,» *Energy Conversion Management*, vol. 149, pp. 254-262, 2017.

[30] I. Dincer y M. Rosen, Exergy: energy, environment, and sustainable development, segunda ed., Oxford: ELSEVIER, 2013.

[31] T. Kotas, The exergy method of Thermal Power Plants, Londres: Anchon Brendon, 1985.

[32] R. Yumrutas, M. Kunduz y M. Kano, «Exergy analysis of vapor compression refrigeration systems,» *Exergy an International Journal*, vol. 2, pp. 266-272, 2002.

[33] M. D'Acaddia y L. Vanoli, «Thermoeconomic optimization of the condenser in a vapour compression heat pump,» *International Journal of Refrigeration*, vol. 25, pp. 433-441, 2004.

[34] Szargut, «Egzergia. Poradnik obliczania I stosowania, Widawnictwo Politechniki Shlaskej,» Gliwice, 2007.

[35] M. Mansouri, P. Ahmadi, A. Kaviri y M. Jaafar, «Exergetic and economic evaluation of the effect of HRSG configurations on the performance of combined cycle power plants,» *Energy Conversion and Management*, vol. 58, p. 47–58, (2012).

[36] A. Abusoglu y M. Kanoglu, «Exergetic and thermoeconomic analyses of diesel engine powered,» *Applied Thermal Engineering*, vol. 29, pp. 234 - 241, 2008.

[37] M. Mehrpooya, M. M. Moftakhari Sharifzadeh y H. Ansarinasab, «Investigation of a novel integrated process configuration for natural gas liquefaction and nitrogen removal by advanced exergoeconomic analysis,» *Applied Thermal Engineering*, 2017.

[38] Ligang Wang , Yongping Yang, T. Morosuk y G. Tsatsaronis, «Advanced Thermodynamic Analysis and Evaluation of a Supercritical Power Plant,» *Energies*, pp. 1850-1863, 2012.

[39] S. Anvavi, R. Khoshbakhti Saray y K. Bahlouli, «Employing a new optimization strategy based on advanced exergy concept for improvement of a tri-generation system,» *Applied Thermal Engineering*, 2016.

[40] P. Kelly, G. Tsatsaronis y T. Morosuk, «Advanced exergetic analysis:Approaches for splitting the exergy destruction into endogenous and exogenous parts,» *Energy*, pp. 384 - 391, 2009.

[41] S. Soltani, M. Yari, S. Mahmoudi, T. Morosuk y M. Rosen, «Advanced exergy applied to an exernally-fired combined-cycle power plant integrated with biomass gasification unit,» *Energy*, vol. 59, pp. 775-780, 2013.

[42] J. Chen, H. Havtun y B. Palm, «Conventional and advanced exergy analysis of an ejector refrigeration system,» *Applied Energy*, vol. 144, pp. 139-151, 2015.

[43] M. Fallah, H. Siyahi, R. Akbarpour Ghiasi, S. Mahmoudi, M. Yari y M. Rosen, «Comparison of different gas turbine cycles and advanced exergy analysis of the most effective,» *Energy*, pp. 701-715, 2016.

[44] S. Anvari, R. Saray y K. Bahlouli, «Conventional and advanced exergetic and exergoeconomic analyses applied to a tri-generation cycle for heat, cold and power production,» *Energy*, vol. 91, pp. 925-939, 2015.

[45] A. Palizdar y S. Sadrameli, «Conventional and advanced exergoecnomic analyses applied to ethylene refrigeration system of an existing olefin plant,» *Energy Conversion and Management*, vol. 138, pp. 474-485, 2017.

[46] T. Morosuk y G. Tsatsaronis, «Advanced exergy-based methods used to understand and improve energy-conversion systems,» *Energy*, 2018.

[47] H. Ansarinasab y M. Mehrpooya, «Advanced exergoeconomic analysis of a novel process for production of LGN by using a single effect absorption refrigeration cycle,» *Applied Thermal Engneering*, vol. 114, pp. 719-732, 2017.

[48] F-Chart Software, *EES "Engineering Equation Solver"*, Wisconsin, 2018.

[49] M. Mehrpooya, H. Ansarinasab, M. M. Moftakhari Sharifzadeh y M. A. Rosen, «Conventional and advanced exergoeconomic assessments of a new air separation unit integrated with a carbon dioxide electrical power cycle and a liquefied natural gas regasification unit,» *Energy Conversion and Management*, vol. 163, pp. 151-168, 2018.

[50] A. Shirazi, M. Aminyavari, B. Najafi, F. Rinaldi y M. Razaghi, «Thermal-economic-enviromental analysis and multi-objetive optimization of an internal-reforming solid oxide fuel cell-gas trubine hybrid system,» *Invernational Journal of Hydrogen Energy*, vol. 37, pp. 19111-19124, 2012.

[51] P. Ahmadi y I. Dincer, «Exergoeconomics,» de *Comprehensive Energy System*, Elservier, 2018, pp. 340-375.

[52] A. Alashkar y M. Gadalla, «Thermo-economic analysis of an integrated solar power generation system using nanofluids,» *Applied Energy*, vol. 191, pp. 469-491, 2017.

[53] F. Mohammadkhani, N. Shokati, S. Mahmoudi, M. Yari y M. Rosen, «Exergoeconomic assessment and parametric study of a Gas Turbine-Modular Helium Reactor combined with two Organic Rankine Cycles,» *Energy*, vol. 65, pp. 533-543, 2014.

[54] N. Shokati y S. Khanahmadzadeh, «The effect of different combinatios of ammonia-water Rankine and absortion refrigeration cycles on the exergoeconomic performance of the cogeneration cycle,» *Applied Thermal Engineering*, vol. 141, pp. 1141-1160, 2018.

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Dear Editor:

Journal of Energy Technology

We would like to thank you for the opportunity provided to contribute our work to such a prestigious journal. Additionally, we would like to inform you that in this document, we provide answers to some of the questions and doubts raised by the evaluators of our work.

We would like to inform you that we are sending a document with the corrections recently sent by the editor of such a prestigious journal Energy Technology. For this reason, some questions raised in this second review will be answered below.

Full Paper, No	**ente.202000993R1**
Title	Advanced exergy and Exergoeconomic Analysis of a Gas Power System with Steam Injection and Air Cooling with a Compression Refrigeration Machine
Authors	Deibys Barreto Author 1, Juan Fajardo Author 2, Gaylord Carrillo Caballero Author 3, Yulineth Cardenas Escorcia Author 4
Submission date (Original Manuscript)	2020-07-04
Revision date (**Round 1**)	2020-10-15 -- ente.202000618 (First Review)
Revision date (**Round 2**)	2021-01-25 -- ente.202000993
Revision date (**Round 3**)	2021-02-11 -- ente.202000993R1
Corrections	please verify text highlighted in blue in the manuscript.

Reviewer Comments – Reviewer 2

POINT (1)

Equations considered to be erroneous in Table 3:

1-CH1 cond fuel and product exergy should be adjusted as follows:

EF = E39-E40

EP = E34-E33

2-CH1 evap fuel and product exergy should be adjusted as follows:

EF = E31-E42

EP = E26-E25

3-CH2 cond fuel and product exergy must be corrected as follows:

EF = E40-E41

EP = E38-E37

4-CH2 evap fuel and product exergy should be corrected as follows:

EF = E35-E36

EP = E28-E27

5-It is also thought that the fuel and product exergy in the combustion chamber are spelled incorrectly. E9 should also be written in fuel exergy.

Authors Response to POINT (1)

We greatly appreciate your suggestions and efforts to improve the quality of the work, for us it was a great help. Next find the answer to the first point consulted:

The errors on the equations in Table 3 were caused by an error in the name of the equipment: the condensers and evaporators of chillers 1 and 2, which were inverted in figure 1.

The correction has been made on figure 1(please check figure 1 in the manuscript) and all the definitions of fuel and product of the equipment have been revised in table 3, and later in table 5.

Resulting in

1)-CH1 cond fuel and product exergy:

EF = E31-E32

EP = E26-E25

2)-CH1 evap fuel and product exergy:

EF = E39-E40

EP = E34-E33

3)-CH2 cond fuel and product exergy:

EF = E35-E36

EP = E28-E27

4)-CH2 evap fuel and product exergy should be corrected as follows:

EF = E40-E41

EP = E38-E37

5)- The respective correction was made. Obtaining the following:

EF = E9

EP = E6-E5-E11

Reviewer Comments – Reviewer 2

POINT (2)

Equations considered to be erroneous in Table 5:

1- CH1 cond, CH1 Evap

2-CH2 cond, CH2 Evap

written cost equations for. I mentioned this fix in my previous review. The places of these equations in the table are confused. Hopefully the authors will make the necessary corrections. If they think the evaluation is wrong, the necessary explanation should be given.

Authors Response to POINT (2)

Revised the equations in table 3 and in table 5, the equations considered errors were corrected and they are as follows:

CH1 Cond	$\boldsymbol{c_{26}\dot{E}_{26} - c_{25}\dot{E}_{25} = c_{31}\dot{E}_{31} - c_{32}\dot{E}_{32} + \dot{Z}_{CondCh1}}$
CH1 Evap	$\boldsymbol{c_{34}\dot{E}_{34} - c_{33}\dot{E}_{33} = c_{39}\dot{E}_{39} - c_{40}\dot{E}_{40} + \dot{Z}_{EvapCh1}}$
CH2 Cond	$\boldsymbol{c_{28}\dot{E}_{28} - c_{27}\dot{E}_{27} = c_{35}\dot{E}_{35} - c_{36}\dot{E}_{36} + \dot{Z}_{CondCh2}}$
CH2 Evap	$\boldsymbol{c_{38}\dot{E}_{38} - c_{37}\dot{E}_{37} = c_{40}\dot{E}_{40} - c_{41}\dot{E}_{41} + \dot{Z}_{EvapCh2}}$