ENERGETIC AND EXERGETIC ANALYSIS OF A GAS-TURBINE COMBINED CYCLE POWER PLANT

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Abstract. *A power plant operating in combined cycle utilizes gas and steam turbines associated in a single power plant, both of them generating electrical energy from the burning of the same fuel. Generally, the heat available in the gas turbines exhaustion gases is recovered, producing the steam in recovering kettle with no fuel suplementary burning, but aiming to elevate the power produced, complementary fuel is used in the recovering kettle. In the Brazilian thermoelectrical power plants gas is the most used fuel. However, in order to prevent problems in the gas supply via existing contract with Bolivia and due to the necessity of building new gas ducts in our territory, the study of the possibility of operating these power plants with a second fuel, the diesel, becomes necessary. The power plant generating efficiency operating with diesel fuel compared to the natural gas is thermodynamically analyzed. The analysis of the first Law of Thermodynamics for the power plant operating with both fuels is performed. In final phase, the analysis of the second Law of Thermodynamics is performed and, through the Exergetic Balance comparisons are made for the power plant operating with diesel and natural gas. Finally, a comparison among the levels of thermal, exergetic and irreversibility efficiencies for the plant operating with the fuels considered is made.*

Keywords*:* *Combined Cycle, Thermal efficiency, Exergetic Efficiency, Diesel, Natural Gas.*

Nomenclature

CC	combined cycle	ε	exergetic eficiency
C_p	specific heat constant pressure	η	power plants efficiency
DI	diesel	η_{GT}	gas turbine efficiency
E_{fuel}	energy supplied by fuel (kW)	η_p	pump's efficiency
Ep	power generated (kW)	η_{ST}	steam turbine efficiency
ex	specific exergy (kJ/kg)	η_{total}	total efficiency
Ex	exergy rate (kW)	$\eta_{ISO\ C}$	isoentropic efficiency of compressor
fc	correction factors	$\eta_{ISO\ TG}$	isoentropic efficiency of gas turbine
h	specific enthalpy (kJ/kg)		
i	ith flow stream	*Subscripts*	
I	irreversibility rate (kJ)		
LHV	lower heat value(kJ/kg)	0	reference environment
$\dot{m}$	mass flow rate (kg/s)	C	compressor
NG	natural gas	fuel	combustible
P	pressure (MPa)	g	flue gas
Pr	pressure ratio	gl	global
Qi	fuel lower calorific power ( )	GT	gas turbine
R	universal constant (kJ/kgK)	i	ith flow stream
T	Temperature (° C)	P	pump
I	irreversibility rate (kJ)	S	steam
$\dot{W}$	Work rate or power (kW)	ST	steam turbine
λ	specific heat ratio	g	flue gas
ΔT	temperature variation	gl	global

1. INTRODUCTION

Studies of engineering designs and exergy analysis for power generation systems are of scientific interest and also essential for the efficient utilization of energy resources. For this reason the exergy analysis has drawn much attention by scientists and system designers in recent years. Some devoted their studies to component exergy analysis and efficiency improvement; others concentrate on systems design and analysis (Bilgen, 2000; Sue and Chuang, 2004).

Energetic analysis is based on the conservation of mass and energy and makes it possible to seek an applicable zone of operation for each system. Exergetic analysis identifies the components of the system which are at the origin of the exergy losses and is based on the exergy concept. The exergy concept is related to the energy and the reversible character of a process (Borel, 2001). Indeed, exergy represents the energy supplied in a reversible process. As for the losses due to irreversibilities, there are several kinds: exergy destruction, lost work, irreversible heat. Thus, the exergy analysis includes the First and Second Law of Thermodynamics. This makes is possible to evaluate both quantitatively and qualitatively the degree of degradation of energy, i.e., to calculate accurately the consequences of the various phenomena of thermodynamic irreversibilities and therefore to quantify correcty the thermodynamic losses of a system.

According to Dincer and Rosen (2004), exergy analysis is an effective thermodynamic scheme for using the conservation of mass and energy principles together with the second law of thermodynamic for the design and analysis of thermal systems by reducing the inefficiencies.

Exergy is defined as the maximum work, which can be produced by a system or a flow of matter or energy at it comes to equilibrium with a specified reference environment. Unlike exergy, exergy is conserved only during ideal processes and destroyed due to irreversibilities in real processes (Rosen and Scott, 2003). Although the basic concepts related to exergy analysis were formulated around the mid-to-late 1800s, most developments in the area of practical application occurred during the past few decades (Gagglioli, 1977). At present, exergy analysis is gaining widespread acceptance in many communities. In the last decade, many researchers (Ahern, 1980) have recommended that exergy analysis be used to aid decision making regarding the allocation of resources (capital, research and development effort, optimization, life cycle analysis, materials, etc.) in place of or in addition to energy analysis. The scope of this paper is to examine, from an energetic and exergetic point of view, the possibilities and advantages natural gas – fired combined cycle concept. The performance of a Combined Cycle system along with its each effect was evaluated by using exergy analysis, which aims at better identifying process efficiencies and losses. The second law analysis is used to deal with the exergy analysis of variable mass processes, in which the exergy losses and the corresponding exergy efficiencies for each of the component are calculated.

2. POWER PLANTS DESCRIPTION

To analyze the engineering system design and enhance gas turbine performance, on power plant was selected to perform the experiments in this study. The Figure 1 show the gas turbine system associated to the recovering kettle and steam turbine, from which a corresponding functional diagram can be made. The system proposed will utilize, for comparison purposes, natural gas or diesel as fuel. Under the atmospheric conditions, the air penetrates the compressor and is compressed until it reaches the combustion pressure. Next, it is sent to the combustion chamber, where it is pauselessly burned under constant pressure and the gases from the combustion expand in the gas turbine producing electricity. The gases with high temperature are then sent to the heat recovering, producing more electrical energy. The most adequate type of fuel for this system is the natural gas, which has high energetic density and optimum combustion efficiency (Villela and Silveira, 2006).

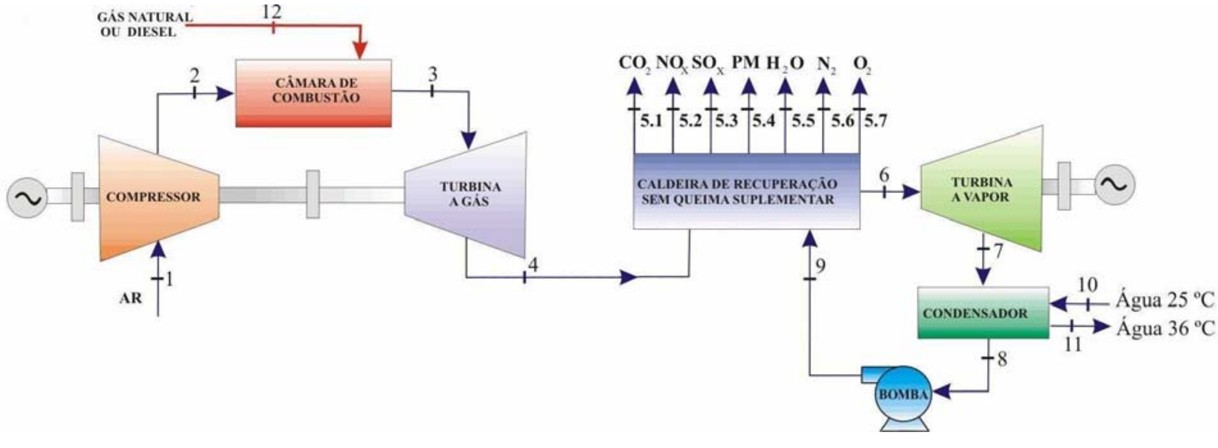

Figure 1. Gas Turbine System Associated to the Heat Recovering and Steam Turbine

2.1. Gas turbine design information

The gas turbine information of plant is presented in Table 2 (GTW Handbook, 2003). Generally, the gas turbine manufacturers quote performance based on ISO and LHV conditions. ISO conditions mean 15 ºC ambient air temperature, 101.325 kPa barometric pressure, and 60% relative humidity. LHV means the lower heating value of the fuel being used (natural gas or diesel)

Table 2. Gas turbine information for power plant

Model gas turbine	Ansaldo Energy V94.3A
Heat rate (Btu/kWh)	8860
Press Ratio	17.0
Exhaust Temperature	583 ºC
Mass Flow (ib/sec)	1146.0
Turbine speed	3000 rpm
ISO Base rating (MW)	265
Weight (kg)	330,000

3. GENERAL RELATIONS

In this system the following considerations are adopted: it works in permanent regime; all components operate with no heat loss, the principles of ideal gas are applied to the air and to the combustion products, a complete combustion is considered. In this case two decision variables are taken in account: temperature (T_5) of the exhaustion gases in the turbine and pressure relation of the selected gas turbine $Pr = P_3/P_4$. These variables were chosen because of the influence on the thermodynamic performance of the system and also according to the criterion of selection utilized in the determination of the gas turbine. Thus, considered P_1 and P_5 are the ambient pressure (101.325 KPa) and represent the pressure loss percentage in the dos gases in heat recovery and in the combustion chamber respectively (Valero, 1994; Silveira and Tuna, 2003).

The temperature (T_3) of the gas turbine inlet and the value de λ_G are defined by the following equations (Cohen and Rogers, 1989; Silveira and Tuna, 2003).

$$T_3 = \frac{T_4}{\left\{1-\eta_{ISO_{GT}}.\left[1-(Pr)^{\frac{(1-\lambda_G)}{\lambda_G}}\right]\right\}} \tag{6}$$

$$\lambda_G(T) = \frac{1}{1-\frac{R_G}{Cp_G(T)}} \tag{7}$$

The temperature T_2 (considering T_1 equal the 25 ºC), and value λ_{air}, are defined by (Silveira and Tuna, 2003):

$$T_2 = T_1.\left\{1+\frac{1}{\eta_{ISO_c}}\left[\left(\frac{P_2}{P_1}\right)^{(\lambda air-1)/\lambda air}-1\right]\right\} \tag{8}$$

$$\lambda_{air}(T) = \frac{1}{1-\frac{R_{air}}{Cp_{air}(T)}} \tag{9}$$

The enthalpys of air and gases in points 2, 3 are defined respectively by (Si-Doek et al. 1996):

$$h_2 = Cp_{air}(T_2).(T_2 - T_0) \tag{10}$$

$$h_3 = Cp_G(T_3).(T_3 - T_0) \tag{11}$$

The air flow rate is calculated from the association of the mass conservation principle with the First Law of Thermodynamics:

$$\dot{m}_{air} = \dot{m}_G - \dot{m}_{fuel} \tag{12}$$

The pump and compressor works are obtained from (Moran and Shapiro, 2004):

$$\dot{W}_p = \frac{\dot{m}_S \,.\, (h_9 - h_8)}{\eta_p} \tag{13}$$

$$\dot{W}_C = \dot{m}_{air} \int_{T_1}^{T_2} Cp_{air(T)} dt \tag{14}$$

Table 3. Fixed parameters (Silveira and Tuna, 2003; Dincer et al. 2004)

Rair	0.287	ΔP_{HR}	0.05
R_G	0.290	η_{ISOC}	0.90
CpD (kJ/kg)	1.193	η_{ISOGT}	0.85
CpGN (kJ/kg)	1.209	η_p	0.90
LHV – NG (kJ/kg)	47966	η_{ST}	0.98
LHV – D (kJ/kg)	44574	η_C	0.86
T_{env} (º C)	25	T_o	25 º C
ΔP_{CC}	0.05	P_o (kPa)	101.325

The gas turbine thermal efficiency is estimated by (Moran and Shapiro, 2004; GTW Handbook, 2003):

$$\eta_{GT} = \frac{1}{\text{Heat rate}} \tag{19}$$

The power ($\dot{W}_{ST}$) and the electricity produced (Ep_{ST}) by the steam turbine are defined by (Sue and Chuang, 2004; Moran and Shapiro, 2004):

$$\dot{W}_{ST} = \dot{m}_S \,.\, (h_8 - h_7) \tag{20}$$

$$Ep_{ST} = \dot{W}_{ST} \,.\, \eta_{ST} \tag{21}$$

The power (Ep_{GT}) of steam turbine and the total power produced by the system are given by (Sue and Chuang, 2004; Moran and Shapiro, 2004):

$$\dot{W}_{GT} = Ep_{GT} \tag{22}$$

$$Ep_{total} = Ep_{GT} + Ep_{ST} \tag{23}$$

Thus the power supplied by the fuel (E_{fuel}) and the plant global efficiency (η_{gl}) are obtained by (Sue and Chuang, 2004; Moran and Shapiro, 2004):

$$E_{fuel} = \dot{m}_g \,.\, LHV \tag{24}$$

$$\eta_{gl} = \frac{Ep_{GT} + Ep_{ST} - \dot{W}_p}{E_{fuel}} \tag{25}$$

Figure 3 shows the global efficiency of the cycle for the two fuels considered by using equation 24. The surrounding temperature is 25 º C with 60% of relative humidity, considering the plant local conditions.

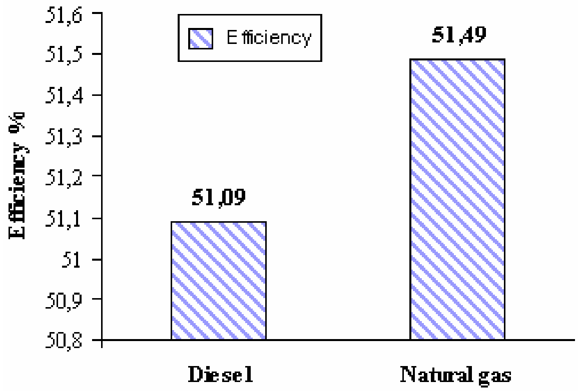

Figure 3. Comparison of the system total efficiency

The turbine exhaust temperature is 176,7 ºC and the gas turbine power are same for both fuels (natural gas, diesel) (Barclay, 1995; Santana, 1999). In this cycle the efficiency of the plant with the use of the natural gas is 1.3% higher than diesel. This difference is mainly attributed to the different fuels and combustion features and is not discussed herein.

4. SECOND LAW ANALYSIS

The second law analysis is an exergy analysis developed rapidly in recent years. The exergy analysis is a very useful tool for understanding the mechanism, performance evaluation, design and optimization of thermoelectrical power plant. The basic procedure for exergy analysis of the cabinet is to determine the exergy values at steady-state points and the reason of exergy variation for the process. The maximum work obtainable from a system using the environmental parameters as reference state called exergy and is expressible in terms of four components: physical exergy, kinetic exergy, potential exergy and chemical exergy. However, the kinetic and potential exergies are usually neglected and because there is no departure of chemical substances from the cycle to the environment, the chemical exergy is zero (Vidal et al., 2006). Therefore, in this analysis the physical exergy (ex) is only considered and is calculated by the general expression:

$$ex_i = (h_i - h_0) - T_0(s_i - s_0) \tag{26}$$

where h and s are the entropy respectively and T_o (298,15 K) is the reference environmental temperature.

In this context, in order to determine the exergy in each point i of the systems proposed, the following expressions are used (Sue and Chuang, 2004; Rosen and Dincer, 2004; Vidal et al, 2006):

The steam exergy is:

$$ex_i = (h_i - h_0) - T_0(s_i - s_0) \tag{27}$$

The exergy of the air and ideal gas are defined by:

$$ex_i = Cp_{air,g}\left[(T - T_0) - T_0.\ln(T/T_0) + R_{air,g}T_0.\ln(P/P_0)\right] \tag{28}$$

For the present calculation it is taken as having a temperature of 298.15K (T_0) and a pressure of 101.325 KPa (P_0) as per recommendations on the standard atmospheric condition (Sirinivasan et al., 2003). The specific heat capacity of air is a function of temperature defined by (Moran and Shapiro, 2004):

$$Cp_{air} = 1{,}04841 - 0{,}000383719\,T + \frac{9{,}45378.T^2}{10^7} - \frac{5{,}49031.T^3}{10^{10}} + \frac{7{,}92981.T^4}{10^{14}} \tag{29}$$

In order to analyze NO_2 in the natural gas combustion products, the following equation for a normalized gas excess is utilized α (Villela and Silveira, 2007; Carvalho and Lacava, 2003):

$$0.893CH_4+0.08C_2H_6+0.008C_3H_8+0.0005C_4H_{10}+0.0005C_5H_{12}+0.005CO_2+0.013N_2+2{,}118\alpha O_2+7.965\alpha N_2 \rightarrow 1.087CO_2+2.064H_2O+7.965\alpha N_2+2.118(\alpha-1)O_2 \tag{30}$$

Considering NO_2 in the diesel combustion products, the equation for a normalized air excess α (Villela and Silveira, 2007; Carvalho and Lacava, 2003) follows:

$$1C_{12}H_{26}+18.5\alpha O_2+69.56\alpha N_2 \rightarrow 12CO_2+13H_2O+69.56\alpha N_2+18.5(\alpha-1)O_2 \tag{31}$$

5. IRREVERSIBILITIES

The thermodynamic inefficiencies of a system consist of exergy destruction associated with the irreversibilities within the system boundaries and of exergy losses associated with the transfer of exergy to the surroundings (Midilli and Kucuk, 2003). Utlu et al. (2006) has also noted that maximum improvement in the exergy efficiency for a process or system is obviously achieved when the exergy loss or irreversibility is minimized.

For the simplicity of presentation, detailed derivation processes are omitted here, and only the final expressions are presented. The exergy loss or irreversibility can be determined for each component of the cycle by the following expression (Midilli and Kucuk, 2003; Utlu et al., 2006; Sozen et al., 2002; Dincer et al., 2004):

$$I = \sum Ex_{input} - \sum Ex_{output} \tag{32}$$

Moreover, for each individual component of the whole systems, the quantify of the exergy losses or irreversibility is calculated by applying Eq. (32) to Eqs. (33) – (40).

Compressor:

$$I_C = Ex_1 + W_c - Ex_2 \tag{33}$$

Combustion Chamber:

$$I_{CC} = Ex_2 + E_{fuel} - Ex_3 \tag{34}$$

Gas Turbine:

$$I_{GT} = Ex_3 - Ex_4 - W_{GT} \tag{35}$$

Heat Recovering:

$$I_{HR} = Ex_4 + Ex_9 - Ex_5 - Ex_6 \tag{36}$$

Steam Turbine:

$$I_{ST} = Ex_6 - Ex_7 - W_{ST} \tag{37}$$

Condenser:

$$I_{CO} = Ex_7 + Ex_{10} - Ex_8 - Ex_{11} \tag{38}$$

Pump:

$$I_p = Ex_8 + W_p - Ex_9 \tag{39}$$

Total Irreversibility: Summing up the above equations, the total irreversibility is obtain as:

$$I_{total} = I_C + I_{CC} + I_{GT} + I_{HR} + I_{ST} + I_{CO} + I_p \tag{40}$$

The values of exergy irreversibilitys (exergy losses) for each component in the system are listed in Table 5. It can be seen that the combustion chamber fot the diesel are the main component yielding the appreciable exergy losses. The gas turbine and the heat recovery boiler are others components that present higher levels of irreversibility.

Table 5. Exergy irreversibility of each component in the system

COMPONENTS	IRREVERSIBILITY	
	Natural Gas	Diesel
Compressor	49287.35	49057.80
Combustion Chamber	246123.55	246546.28
Gas turbine	212891.26	213892.45
Heat Recovery boiler	61493.71	60804.88
Steam turbine	15509.70	15293.68
Condenser	986.91	972.66
total	586842.3	587109.63

6. CONCLUSION

The main conclusion of this paper should be emphasized are following:

(1) The energetic analysis method applied to a combined cycle shows a high thermal efficiency with the use of two different fuels. The efficiency with the use of natural gas is higher compared to the diesel, presenting a 1,3 % difference.

(2) The exergy method, contrary to first law analysis, localizes and quantifies the Thermodynamic losses (irreversibilities). Exergy parameters were good indicators to show the effectiveness of the cycle, as well as to identify the devices where most exergy destruction occurs.

(3) As an example, a 352.2 MW combined cycle power plant and the Thermodynamic feasibility of approaching 51% of efficiency in combine cycle using natural gas and diesel as fuel has been shown.

For complete performance and efficiency evaluation of system, both energy and exergy analysis should be undertaken. Exergy analysis often provides actual and useful informations than energy analysis regarding efficiencies and losses for system, mainly because the losses for the system.

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