Case of Study: Off-Design Performance Simulation of a Combined Cycle Taking Account Environmental Conditions in Peru

Héctor J. Bravo, MSc. in Energy[1,2], Pascual H. Adriazola, MSc. in Production Management[1], and Gregorio E. Robles, PhD in Energy[1]
[1] Universidad Nacional de San Agustín de Arequipa, Peru, hbravop@unsa.edu.pe padrizola@unsa.edu.pe, erobles@unsa.edu.pe
[2]Universidad Tecnológica del Perú, Peru.

Abstract

The demand for electricity in the country and the world has increased consistently in recent years. Electricity from thermoelectric plants is a viable option that allows diversifying the electricity market and using fossil fuels such as natural gas.
The influence of weather conditions on the performance of any machine can mean its viability or infeasibility from a technical and economic point of view. Combined cycle power generation plants are no exception to this fact, but rather their great physical magnitude as machines make them more vulnerable to climatic variables.
Environmental conditions such as air temperature and atmospheric pressure significantly influence the performance of a thermoelectric plant, this thesis studies the effect that these climatic variables have on combined cycle comprising a Brayton cycle fed with natural gas and a Rankine cycle of three levels of pressure. Considering the local atmospheric pressure and ambient temperature in La Joya and Majes (located in the department of Arequipa) 8760 off design simulations for each hour of the year in both locations were performed and subsequently the results were compared.
Having the same thermoelectric plant installed, the results show that the energy generated in the life of the plant is greater in La Joya than in Majes by 6.64% and that translates into an average annual electricity cost of 6.59% more in Majes than in La Joya, while the efficiency of the plant is 0.2414% higher in La Joya than in Majes.
The model also determines the behavior of the plant in certain periods of the year, being the winter and autumn when the performance of the plant increases while in summer and spring periods performance is reduced. While early morning and evening hours are more productive for the plant than daylight hours.

Keywords—NGCC, Natural Gas Combined Cycle, Gas Cycle, Brayton Cycle, Rankine Cycle, Steam Cycle, Off-design.

I. Introduction

Energy is a resource that is totally indispensable and necessary for the progress of a modern society. New statistics show that the world's energy demand will grow in the next three decades, mainly in countries outside the Organization for Economic Cooperation and Development (OECD) such as Peru. [1]

The growth of the electric energy market is one of the most dynamic of all energy sectors and proof of this is that in recent decades it has been the form of final energy with the highest growth worldwide. (International Energy Agency, 2018). [2] The growth of electricity in non-OECD countries averages 2.5% per year between 2012 and 2040, mainly due to the increase in people's quality of life. [1]
The Peruvian energy sector has registered significant growth due to the internal increase in demand, which is associated with the economic development of products and services. Energy demand will continue to grow in the coming years due to the development of mining and industrial projects, as well as the facilitation of investments and the development of the main metropolises of the country. [3]
At present, the use of natural gas for power generation is justified due to its cost, in most cases high efficiency facilities are typically combined cycle thermal plant. [4]
The energy sector in the country has had an important growth due to the increase in demand that has occurred especially in the last decade. This demand is due to the country's economic growth and competitive prices in oil activities, and rates resulting from auctions in the market for natural gas production and electricity generation [5]. Because electrical energy is the easiest to transport, it is a key part of the country's development, in fact many human development factors are measured in proportion to the energy a country consumes.
Energy demand will continue to grow due to mining and industrial projects, as well as investments and development in the main cities of the country [5].

II. Modeling

The study involves a thermodynamic model at the design point, a thermodynamic model in off design conditions and an economic model, the models were coded in a commercial programming software that had thermodynamic properties included.

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Initially a search for combined cycle configurations was performed, and the configuration of Manente [6] was chosen. The study of Manente [6] is about a combined cycle under design conditions in the Thermoflow software (without having access to a programming code) and it does not take into account the environmental conditions in the performance of the plant.

The combined cycle chosen consists of a natural gas turbine, a three-pressure steam cycle, 15 heat exchangers, three steam turbines and three pressure levels as can be seen in Fig. 1.

The combination of a gas turbine fed by natural gas and steam turbine working with the exhaust gases of the Brayton cycle is known as a Natural Gas Combined Cycle (NGCC).

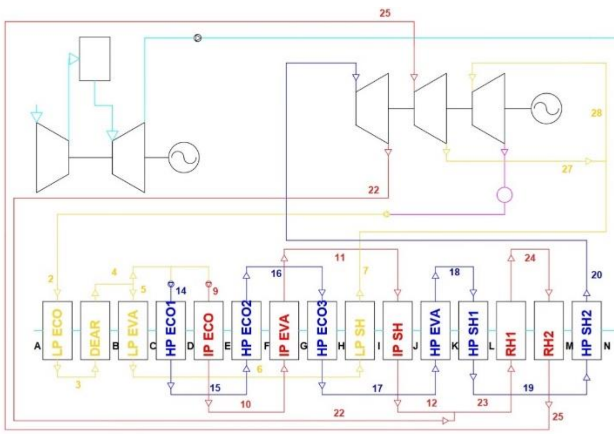

Fig. 1 NGCC Configuration

A. *Modeling in design point considerations*

The analysis at the design point is defined as an energy and mass analysis where the weather conditions are in ISO conditions (1 atmosphere of atmospheric pressure and 15 C °).
Design point model is into two parts: (1) natural gas cycle and (2) rankine cycle.

Natural Gas Cycle

In this section only technical data of a natural gas turbine was used and there is not any calculation anaylsis. Then to validate the calculation of the thermodynamic analysis, the input data of the Manente model was used, it indicates that the gas cycle has the Siemens turbine model SGT5-4000F whose technical characteristics are specified in Table 1.

TABLE 1
GAS TURBINE DESIGN PARAMETERS SIEMENS SGT5-4000F

Parameters	Siemens SGT5-4000F
Pressure ratio	17.00
Air flow (kg/s)	643.10
Exhaust temperature (°C)	600.40
Generator power output (kW)	254250.00

The mentioned turbine will only be used to validate the model in ISO conditions but not in off design conditions this because this turbine has a frequency of 50 Hz that is different from the one that has to be delivered to the network (this being 60 Hz).

Rankine Cycle

For the Manente Rankine cycle model, the exhaust gas compounds of the Siemens gas turbine model SGT5-4000F have been considered. The table 2 shows the percentage composition of the gases considered at the exit of the gas turbine that will be used in the HRSG for the Rankine cycle.

TABLE 2
COMPONENT CONCENTRATION OF EXHAUST GASES

Gas	Cocentration (%)
Nitrogen (N2)	72
Oxigeno (O2)	18
Carbon dioxide (CO2)	5
Water steam (H2O)	5

Once the exit gas were defined the rankine configuration must be explained, as it can be seen in Fig. 1, heat exchangers of yellow color are LP ECO, DEAR, LP EVA and LP SH being an economizer, an integrated de-aerator, an evaporator and a superheater respectively. All yellow section of the Fig. 1 is in low pressure of approximately 4 bar but due to the length of the pipes this varies throughout the course of the configuration, for the simplification of the calculation a constant pressure of 4.04 bar was taken.

Red exchangers as they are shown in Fig. 1 are the medium pressure section and they are IP ECO, IP EVA, IP SH, RH1 and RH2 that are an economizer, an evaporator, a superheater and two reheaters correspondingly. The constant pressure for this section was considered 14.93 bar.
The blue exchangers in Fig. 1 are considered the high pressure section of the cycle with a pressure of 99.5 bar. They are HP ECO, HP ECO2, HP ECO 3, HP EVA, HP SH1 and HP SH2 meaning t economizers the three first ones, an evaporator the following one and the two last ones are superheaters.

Equation (1) is used in all heat exchangers without into consideration evaporators.

$$E = \frac{(Tout_{steam} - Tin_{steam})}{(Tin_{gas} - Tin_{steam})} \quad (1)$$

E is the effectiveness of the exchanger, out is the suffix for output point of the fluid flow and in is the entrance point of it. T is the temperature and gas and $steam$ are suffixes for exhaust gas fluid of the gas turbine and the working fluid of the steam cycle respectively.

Also energy balance is defined by equation *(2)* and it is used in all kind of evaporators.

$$Cp_{gases}(Tent_{gases} - Tsal_{gases})M_{gas} = (Hsal_{vapor} - Hent_{vapor})M_{vapor} \quad (2)$$

where, H is the enthalpy and M is the mass flow and Cp_{gases} is average specific heat at constant pressure of the exhaust gases of the turbine, it can be calculated with equation (*3*).

$$Cp_{gases} = \frac{Cp_{gas-in} + Cp_{gas-out}}{2} \quad (3)$$

Where Cp_{gas-in} and $Cp_{gas-out}$ are the specifics heat at constant pressure in the inlet and outlet of the heat transfer respectively.

The approach point and the pitch point are being taking in consideration in equations *(4)* and *(5)* respectively.

$$T_{in\ evap} = T_{evaporation} - AP, \quad (4)$$

$$T_{gas} = T_{in\ evap} + PP. \quad (5)$$

AP is approach point, PP is pinch point and $T_{evaporation}$ is the saturation temperature.

Power generated in the steam turbine is calculated with equation *(6)*.

$$P_{SteamTurbine} = M(H_{in} - H_{out-real})(\varepsilon_G), \quad (6)$$

being M the mass flow, H_{in} the input enthalpy of the work fluid in the turbine, $H_{out-real}$ is the real output enthalpy in the turbine and ε_G the electrical efficiency of the generator.

To calculate $H_{out-real}$ the equation *(7)* is used,

$$E_{SteamTurbine} = \frac{H_{in} - H_{out-real}}{H_{in} - H_{out-iso}}, \quad (7)$$

where $H_{out-iso}$ is the output enthalpy of the turbine in isoentropic conditions.

The power consumed by the pumps is calculated using equation (*8*)

$$P_{pump} = M(H_{in} - H_{out-real}). \quad (8)$$

To calculate $H_{out-real}$, equation *(9)* is used.

$$E_{PUMP} = \frac{H_{out-iso} - H_{in}}{H_{out-real} - H_{in}}, \quad (9)$$

being E_{PUMP} the isentropic efficiency of the pump. To calculate the heat transferred in the condenser equation *(10)* is used.

$$Q_{CON} = M(H_{in} - H_{out}) \quad (10)$$

B. Modeling in off design considerations

Natural Gas Cycle

For the analysis under design conditions, the Siemens ST5-4000F 50 Hz frequency turbine was considered, but because the places where the plant is to be located is 60 Hz, a turbine with similar characteristics was chosen techniques but with different frequency.

The turbine chosen was the SGT6-5000F mainly due to its mass exhaust flow, exhaust temperature similar to the ST5-4000F and its frequency of 60 Hz, a summary of its technical specifications can be seen in Table 3.

TABLE 3
GAS TURBINE DESIGN PARAMETERS SIEMENS SGT6-5000F

Parameter	Siemens SGT6-5000F
Pressure ratio	18.90
Air flow (kg/s)	588.00
Exhaust temperature (°C)	598.00
Generator power output (kW)	250000.00

In the off design conditions the physical parameters of the 60 Hz turbine will be taken into account, in other words the model at the design point is with the characteristics of the 50 Hz turbine but in simulation conditions the 60 Hz has been taken.

The performance curves of the SGT6PAC-5000F gas turbine are those that will allow us to model the behavior in conditions outside of the design, for which the curves of [7] were modeled in second degree polynomial functions.

As already mentioned, the ambient temperature and atmospheric pressure are the two physical parameters considered for the evaluation of the plant in off design conditions. All the equations in this section will be a function of these two variables.

The performance curves considered were those of temperature and pressure, the curves in [7] were extracted and converted into functions.

The temperature at the outlet of the gas turbine as a function of the ambient temperature is defined by equation (*11*) and mass flow of exhaust gases at the outlet of the gas turbine as a function of local temperature is represented by the equation (*12*).

$$\begin{aligned}(T_{OUT-OFF\ DESIGN})_T &= 578 + 0.0049\left(\frac{T_{IN-OFF\ DESIGN} - 15}{1.8}\right)^2 \\ &+ 0.9599(T_{IN-OFF\ DESIGN} - 15),\end{aligned} \tag{11}$$

$$\begin{aligned}(M_{G-OFF\ DESIGN})_T &= 1 - 8(10^{-6})(T_{IN-OFF\ DESIGN} - 15)^2 \\ &- 0.0024(T_{IN-OFF\ DESIGN} - 15)\end{aligned} \tag{12}$$

Being $T_{OUT-OFF\ DESIGN}$ the ambient temperature in local conditions.

To take into account the influence of atmospheric pressure in off design conditions on the temperature of the exhaust gases the equation (*13*) is taken.

$$(T_{G-OFF\ DESIGN})_P = (T_{G-OFF\ DESIGN})_T + FC_T \tag{13}$$

where FC_T is the correction factor that is added to take into consideration atmospheric local pressure into the exhaust temperature. FC_T can be calculated with equation (*14*).

$$\begin{aligned}FC_T = \frac{0.0454}{1.8}(P_{OFF\ DESIGN} - 14.696)^2 \\ + \frac{0.4232}{1.8}(P_{OFF\ DESIGN} - 14.696)\end{aligned} \tag{14}$$

where $P_{OFF\ DESIGN}$ is the barometric pressure in local conditions.

The mass flow takes into account the influence of local atmospheric pressure with equation (19),

$$(M_{G-OFF\ DESIGN})_P = FC_P(M_{G-OFF\ DESIGN})_T, \tag{15}$$

where FC_P is defined as the correction factor for the mass flow that takes into account local atmospheric pressure conditions.

$M_{G-OFF\ DESIGN}$ is calculated with equation *(16)*,

$$M_{G-OFF\ DESIGN} = 1 + 0.0688(P_{OFF\ DESIGN} - 14.696). \quad (16)$$

The power generated by the gas turbine in off design conditions is determined by the equation *(17)*,

$$P_{GasTurbine-OFF} = C_{Pre} \times C_{Tem} \times 250000, \quad (17)$$

C_{Tem} and C_{Pre} are the correctors that takes into consideration the ambient temperature and the local atmospheric pressure respectively. C_{Tem} is defined by the equation *(18)*,

$$C_{Tem} = 2.414285(10^{-6})\left(Variation_T{}^2\right) - 1.945088(10^{-3})(Variation_T) + 1, \quad (18)$$

being $Variation_T$ corresponds to the difference between the ambient and nominal temperatures as it can be seen in equation (*19*).

$$Variation_T = T_{ambient} - 15°\mathrm{C} \quad (19)$$

15 ° C is the nominal design temperature of the gas turbine.

C_{Pre} is defined by the equation (*20*),

$$C_{Pre} = 7.083344(10^{-2})(Variation_P) + 1 \quad (20)$$

Where $Variation_P$ is the difference between ambient and nominal atmospheric pressure as it can be seen in equation (*21*).

$$Variation_P = P_{ambient} - 1\ atm \quad (21)$$

Where 1 atm corresponds to the nominal pressure of the gas turbine.

Rankine Cycle

For off design model of the steam cycle, the Ganapathy off model [8] has been used because it does not require the details of the mechanical design details such as pipe size, length, density of the fins, etc.

For this method, initially calculations of three parameters are made: (i) the logarithmic temperature differential (equation (*22*)), (ii) the gas property accounting factor F_g (equation (*23*))), and (iii) the thermal conductivity of gases called K (equation (*24*)).

$$\Delta T = \frac{\left[\left(Tin_{gas} - Tin_{Steam}\right) - \left(Tout_{gas} - Tout_{Steam}\right)\right]}{\ln\left[\left(Tin_{gas} - Tin_{Steam}\right) - \left(Tout_{gas} - Tout_{Steam}\right)\right]} \quad (22)$$

$$F_g = \frac{C_p{}^{0.33} k^{0.67}}{\mu^{0.32}} \quad (23)$$

$$K = \frac{Q}{\Delta T\left(W_{gd}{}^{0.65}\right)F_g} \quad (24)$$

Being T the temperature, the suffixes in and out refer to the input and output respectively, while gas and $steam$ are refering to the exhaust gases and the fluid of work respectively. C_p is the specific heat, k the thermal conductivity of the exhaust gases and μ is the viscosity of the exhaust gases.

Finally, Q is the heat that is transferred through the heat exchanger, ΔT the logarithmic temperature differential, W_{gd} the gas mass flow and F_g the gas properties accounting factor.

Once the values of ΔT, F_g and K are calculated at the design point, they are used in the iterations under off design conditions.

Fig. 2 shows the scheme that is used to obtain the off design condition in configurations and has been running for each hour of the day for both localizations considered.

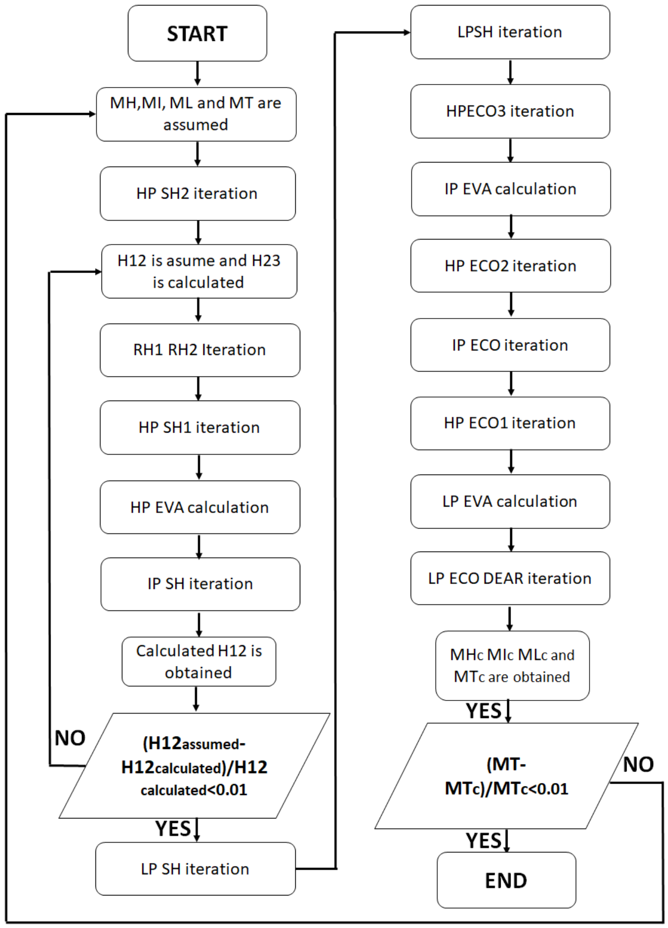

Fig. 2 Iteration Scheme Of NGCC

III. RESULTS

A. Design point

In this section a comparison between the results of this study and in Manente's [6] and Bravo's [10]. Table 4 shows the considered parameters in [6] that are evaluated in this paper.

TABLE 4
INPUT PARAMETERS TO THE MODEL

Parameter	Value
Exhaust air flow (kg/s)	643.10
Exhaust temperature (°C)	600.00
High pressure section (bar)	99.95
Intermedial pressure section (bar)	14.93
Low pressure section (bar)	4.04
Pressure in the condenser (bar)	0.034
Pinch point (°C)	20.00
Aproach point (°C)flow	5.00
Pump efficiency (%)	0.80
Inlet temperature in the high pressure steam turbine (°C)	540.00
Inlet temperature in the intermediate pressure steam turbine (°C)	530.00
Inlet temperature in the low pressure steam turbine (°C)	340.00
Efficiency of the high pressure turbine (%)	86.50
Efficiency of the intermediate pressure turbine (%)	90.89
Efficiency of the low pressure turbine (%)	93.29
Evaporator efficiency (%)	97.00

Table 5 shows a comparison between the results obtained with the thermodynamic model developed and Manente's model and model developed before in other paper that instead considered exhaust gases of the gas turbine as air.

TABLE 5
COMPARISON OF THE MODEL

	Model	Bravo [10]	Manente [6]
Power of the high pressure steam turbine	31881	31481	31680
Power of the intermediate pressure steam turbine	32062	31492	29330
Power of the low pressure steam turbine	72037	70960	76390
Total power	135980	133933	137400

It can be seen in table 6 that the actual model is more accurate compared with the realized before in the other paper that is because considering compounds of exhaust gases instead of just air.

TABLE 6
DEVIATION OF THE MODEL

Reference	Power generated (kW)	Deviation from the original model
Manente [6]	137400	-
Bravo [6]	133933	2.52%
Model	135980	1.03%

As observed in this table, the relative discrepancies between the results of the model and the reference [6] considered, in terms of power is almost just 1%, compared with the model before made [10] that gives us 2.5% approximately.

The values found in Table 3 correspond to the design point with the conditions of the parameters in Table 2 that belongs to Siemens SGT5-4000F that has a frequency of 50 Hz. The real conditions take into consideration Siemens SGT6 5000F whose frequency is 60 Hz. The turbine SGT6 5000F (Table 4) from Siemens was chosen because its mass flow of exhaust and the temperature of this flow is similar to that of SGT5-4000F.

B. Economic modeling

The economic parameters calculated were IRR, the NPV, the PayBack and the LCOE whose formulas are respectively defined in equations

(25),(26),(27) and (28).

$$NPV = -I_0 + \sum_{t=1}^{n} \frac{F_t}{(1+k)^t} \tag{25}$$

$$0 = -I_0 + \sum_{t=1}^{n} \frac{F_t}{(1+IRR)^t} \tag{26}$$

$$PayBack = \frac{I_0}{F} \tag{27}$$

$$LCOE = \frac{CC + O\&M + NG}{Et} \tag{28}$$

F_t is the money flows of each period t, I_0 is the investment made at the initial moment (t = 0), n is the number of time periods and k is the interest rate. CC is the capital cost of the plant, $O\&M$ is operation and maintenance cost NG is the natural gas cost and Et is the total energy generated in the life time of the plant.
The investment cost of the power plant is defined per unit of energy defined by equation (*29*) extracted from [9].

$$Investment\ per\ unit\ of\ energy = 370 + 10.3(10^4)(P)^{-0.438} \tag{29}$$

Where P is the nominal power of the plant. Once the value of Investment per unit of energy is obtained, it is multiplied by P to obtain the total investment cost of the cycle as shown in equation *(30)*.

$$Power\ Plant = P \times\ Investment\ per\ unit\ of\ energy \tag{30}$$

Operation and maintenance costs are defined as a percentage of the investment cost.

$$Operation\ and\ maintenance\ costs\ = \% \times\ I_0 \tag{31}$$

The cost of maintenance with respect to the previous year increases by u, as can be seen in the formula *(32)*

$$O\&M_{year\ n+1} = (1+u) \times O\&M_{year\ n} \tag{32}$$

C. *Off design*

Off-design conditions were made taking into consideration 8760 simulations for both places: La Joya and Majes. Taking into account ambient temperature and atmospheric pressure.

Meteorological real data was considered for both places that was registered in the years 2003, 2004 and 2005 for La Joya and in years 2017, 2018 and 2019 for Majes. Using the equations

Ambient temperature for La Joya and Majes considered for each hour of the year can be seen in Fig. 3 and 4 respectively.

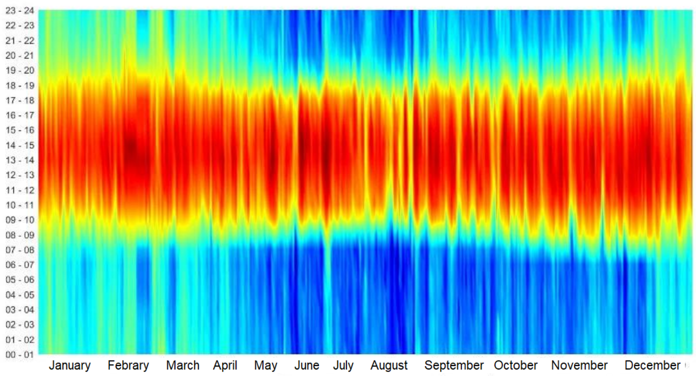

Fig. 3 Heatmap of air temperature of La Joya

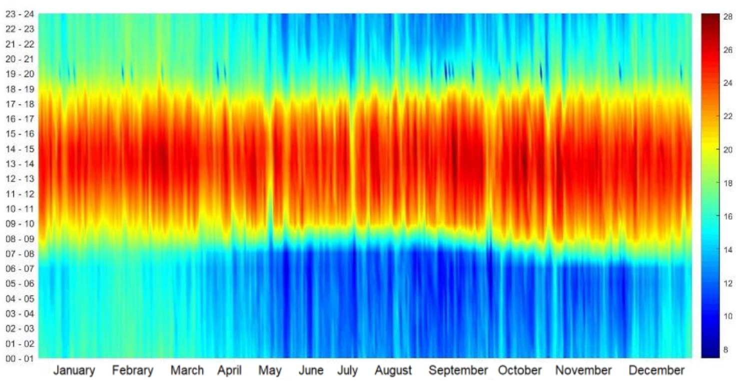

Fig. 4 Heatmap of air temperature of Majes.

For local atmospheric pressure equation *(33)* was taken into consideration

$$p = p_0 \left(1 - 0.0065 \frac{h}{288.15}\right)^{5.2561} \tag{33}$$

where, p_0 is nominal pressure (1atm) and h is the height of the place that is in meters above sea level. Table shows the results of pressure using equation *(33)*.

TABLE 7
HEIGHT CONSIDERED

Location	Height (m.a.s.l)	Reference	Calculated pressure (kPa)
La Joya	1187	[11]	87854.09
Majes	1680	[12]	82704.22

Results of power generated for each hour of the year can be seen in Fig. 5 and 6 for La Joya y Majes correspondingly. Fig. 7 shows the energy generated for each month of the year for La Joya and Majes respectively.

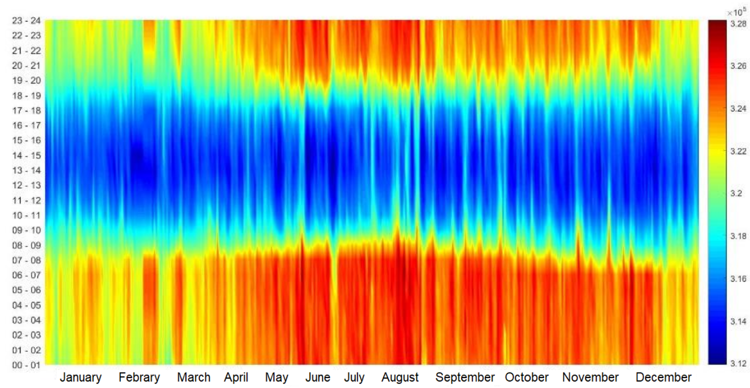

Fig. 5 Heatmap of the energy generated in La Joya.

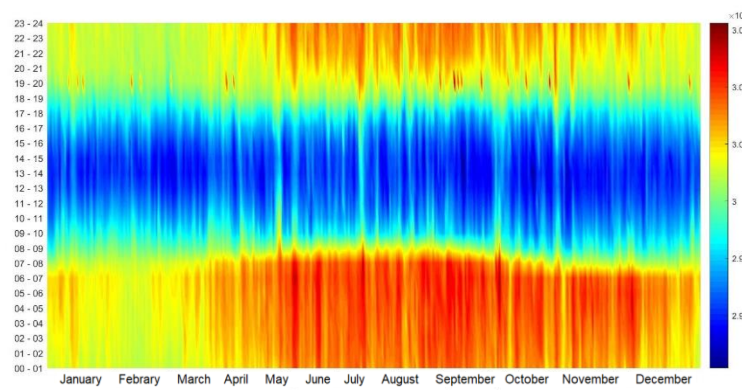

Fig. 6 Heatmap of the energy generated in Majes.

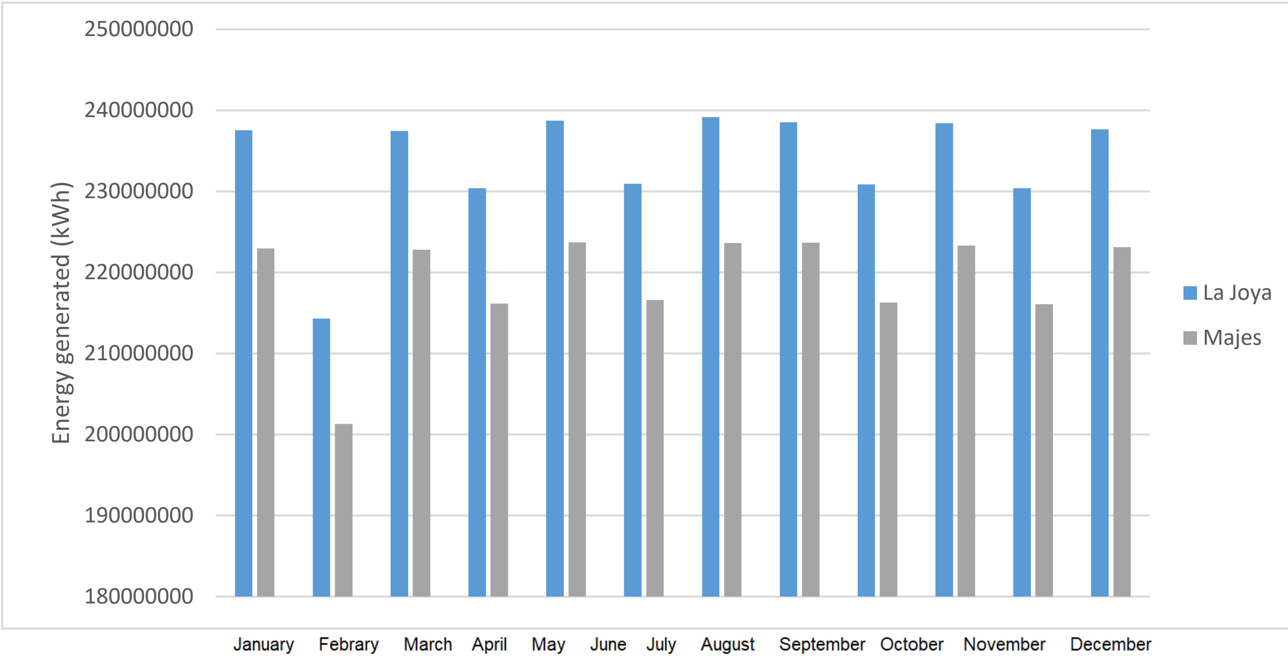

Fig. 7 Total energy generated

It can be noted in Fig. 7 as it is logical that the months when more energy is generated are the months of the seasons with lower temperatures that are autumn and winter being the largest months of July and May generation for La Joya and Majes corresponding. The months with the highest temperatures that are in the spring and summer seasons is the stage of the year that generates less energy being the month of February the month of lower generation, its lower generation is also due to the fact that this is the month with less days of the year.

Table 8 shows the total energy generated in many periods of time for La Joya y Majes.

TABLE 8
ENERGY GENERATED

Location	Average energy generated in a hour (kWh)	Average energy generated in a day (kWh)	Energy generated in a year (kWh)
La Joya	320124.64	7682991.36	2804291846.40
Majes	300187.23	7204493.52	2629640134.80

Table shows that the total energy generated in Majes is 2629665932 kWh and in La Joya 2804357790 kWh in a year being 6.64% more energy in La Joya compared to Majes.

D. *Economic evaluation*

This section describes the main economic indicators (VAN, IRR, PayBack and LCOE) for both locations.
Table 9 shows the economic consideration for both places.

TABLE 9
ECONOMIC VARIABLES CONSIDERED

Economic Variable	Value	Reference
Plant factor	0.80	Estimated
Availability factor	0.85	Estimated
Annualized factor	0.1175	Calculated
Plant life cycle (years)	20	Estimated
Natural gas cost (USD/MMBTU)	1.5572	[13]
Investment rate	0.10	Estimated
Electricity sale price (USD/MWh)	40	[14]

Considering table gives the results in table 10.

TABLE 10
ECONOMIC RESULTS

Location	VAN (USD)	IRR	PayBack (years)
La Joya	174,636,475.14	18.55%	5.12

Majes	127,247,104.32	16.35%	6.70

It can be seen in Table 10 all economic variables are advantageous in La Joya compared to Majes, having a NPV of 174636475.14 USD and 127247104.32 USD in La Joya and Majes correspondingly, which means 37.24% more in La Joya than Majes.

The IRR is higher in La Joya by 13.46% compared to Majes, being in 18.55% and 16.35% in La Joya and Majes respectively. The PayBack is 5 years 1 month and 14 days for La Joya, while for Majes it is 6 years 8 months and 15 days approximately which corresponds to a smaller PayBack in La Joya with respect to Majes of 23.58%.

Table11 shows the LCOE calculated for both locations

TABLE 11
LCOE CALCULATION

Location	LCOE ($/MWh)
La Joya	31.40
Majes	33.47

With Table 11 we can conclude that the cost per energy unit is higher in Majes compared to La Joya by 6.59%

IV. CONCLUSIONS

The performance was simulated considering the local meteorological conditions for the proposed combined cycle plant which indicates that, due to the ambient temperature and the local atmospheric pressure in the combined cycle, its performance has a significant impact, making it generate 6.64% more electricity in La Joya than in Majes.

It was possible to predict the performance of the gas turbine and the Rankine cycle based on the climatic variables having a greater plant efficiency in La Joya at 0.2414% with respect to Majes.

The economic variables were calculated considering the performance of the plant and thus calculating the total energy produced, obtaining a NPV, IRR and PayBack of USD 174636475.14, 18.55% and 5.12 years for La Joya and USD 127247104.32, 16.35% and 6.70 years for Majes. While the cost per unit of energy produced is 6.59% more expensive in Majes than in Joya, being 33.47 and $ 31.40 / MWh respectively.

In addition, a performance pattern of the thermoelectric plant was obtained, which is reduced in summer and spring months due to the increase in air temperature, while in winter and autumn months the opposite effect due to the decrease in this It also concludes that the hours of the morning and night are more productive than those of the day due to the change in temperature.

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