THERMODYNAMIC AND ECONOMIC ANALYSIS OF A BIOGAS -FUELED MICRO GAS TURBINE WITH COMPRESSOR PHOTOVOLTAIC DRIVE

G. S. Trindade, and
L. A. Bortolaia

Universidade Federal do Ouro Preto
Morro do Cruzeiro
Ouro Preto, Minas Gerais, Brasil
gustavo.trindade@aluno.ufop.edu.br
luis.bortolaia@ufop.edu.br

Received: Aug 03, 2022
Revised: Aug 10, 2022
Accepted: Aug 13, 2022

ABSTRACT

The increase in global and Brazilian energy demand plus environmental concern due to pollutant emissions are motivating investigation and development of sustainable energy sources and the interaction between them. In this context, electricity generation through solar and solid urban waste energy harnessing has become an effective option for Brazilian energy matrix diversification. Therefore, in order to ratify this trend, a thermodynamic analysis of a 200 kWe micro gas turbine using biogas as fuel and photovoltaic panels to drive compressor shaft is presented in this paper. In terms of methodology, Engineering Equation Solver (EES) Demo was used to run a classic steady state thermodynamic model and a parametric analysis to evaluate MGT performance with a PV system integrated in order to assess specific fuel consumption reduction (SFC) at Belo Horizonte, Minas Gerais, Brazil, operation. Radiasol 2 was the software used to acquire solar insolation data over the year. The fuel economy occurs since power produced in turbine does not need to be discounted to drive compressor shaft during periods when solar insolation is present or when energy generation during the day is sufficient to be credited as Brazilian legislation allows grid integrated systems. In cases that solar energy is not enough or absent, a transmission mechanism similar to a clutch should be considered as well as proposed in other studies to guarantee uninterrupted system operation. Futhermore, an economic analysis was performed to evaluate kWh cost for conventional and hybrid systems. Interest rate, inflation, construction period and amortization period are important parameters taken into account to calculated annuity factor that must be evaluated to predict energy cost. The thermodynamic results showed that SFC=0,718 kg/kWh for conventional operation and SFC=0,266 kg/kWh for hybrid operation for the best operating compressor pressures ratio. The economic results showed that payback time for MGT and MGT with compressor photovoltaic drive are lower than lifetime systems, and hybrid MGT electricity cost is lower when compared to paid ones nowadays.

Keywords: micro gas turbine; biogas; photovoltaic energy; thermodynamic modeling; economic analysis.

INTRODUCTION

Recent work (MosayebNezhad et al., 2019) states that the world demand of energy will undoubtedly increase due to population growth, especially in large urban centers. Moreover, the use of fossil fuels as primary source for power generation over many years has led to a series of worrisome environmental degradations such as ozone layer depletion (Adefarati and Bansal, 2016). According to the International Energy Agency (2019), 80% of the electricity consumed in the world is produced by burning coal, oil, natural gas and nuclear power plant. The comfort coming from non-renewable fuels and minerals exploitation is polluting and dangerous. It is humanity's duty to search for alternatives to minimize such impacts with the use of sustainable resources, those that consider future generation's needs. This is only possible with the alignment between three pillars: economic, social and environmental. Thus, several regulations have recently been created and the development of technologies associated with sources of renewable, clean, non-depleting and sustainable energy economically viable has become essential (Adefarati and Bansal, 2017), as has been ratified in 2015 Paris agreement.

In this context, according to Somehsaraei et al. (2014) micro gas turbines (MGTs), which are usually defined as small gas turbines up to a few hundred kilowatts, are considered as promising power generators since they provide high fuel flexibility, low emissions, small footprint and low maintenance costs, aspects that make it commercially interesting. And according to Gupta et al. (2010) the MGT system might be especially interesting when fueled by green biogas, an abundant, non-polluting and cheap source, obtained from solid urban waste, as well as agriculture waste or even animal excrement.

In Brazil case, a tropical country with high daily insolation rates throughout its territory, solar energy stands out for its enormous available generation capacity. In 2012, there was a regulatory milestone in the country: the National Electric Energy Agency started to allow micro-generation and mini-generation of electricity from renewable and alternative sources with distributed generation systems connected to low voltage electrical grid. Since then, the use of photovoltaic (PV) systems has grown a lot, which is very important to diversify Brazilian energy matrix, highly dependent on hydroelectric plants, sometimes unstable due to water regime. It was also created an electricity credits compensation system for self-producers of energy, which can even export energy to other locations. This possibility is very beneficial, since during the day, when residential consumption is lower, there is an increase in energy supply that sustains the high demand of industries during this period. In this scenario, photovoltaic systems become even more advantageous, as the electricity generated is immune to inflation, different from energy purchased from distribution companies. Once installed, there is a guarantee of PV electricity generation without price increase for at least 25 years - equipment useful lifetime. Generally, shorter time intervals generate enough savings that pay the initial investment.

As solar energy is an intermittent energy source, it is interesting that it complements other sources that are more often available. Therefore, many researches about micro gas turbines and solar energy integration have been developed focusing on increasing system efficiencies, provided by specific fuel consumption reduction, and, as result, decrease in pollutants emissions (Aichmayer *et al.*, 2013; Meriche *et al.*, 2014; Cameretti *et al.*, 2015; Aichmayer *et al.*, 2015; Sami and Marin, 2017; Javidmehr *et al.*, 2018; Chen et al., 2020; Ahmad *et al.*, 2020). Hybrid systems, thus, are originated.

The solarization of gas turbines aligns the strengths of each technology, and the first works were carried out in the 1980s (Spelling, 2013). In this case heat from the Sun basically can be used in two ways: to preheat pressurized air that will participate in combustion process or depressurized air before entering in turbine or to directly generate electricity through photovoltaic panels to drive compressor, previously connected to the same turbine shaft. The focus of this work is on the second option, integration between micro gas turbines and photovoltaic solar energy. Future predictions consider it the most important energy source, due to easy generation, distribution and optimist prognosis favorable to planet preservation (Villalva, 2017).

Thus, the main objective of this study is to develop a performance and quantification model of electricity cost, generated by conventional micro gas turbines and micro gas turbines with compressor photovoltaic drive. A scientific novelty is presented, since it presents the concepts, standardizations and detailed thermodynamic and economic methodologies of a little-reported integration between a sustainable technology on rise with great potential for use and other commercially consolidated.

MICRO GAS TURBINES

A micro gas turbine is a turbomachine based on a Brayton cycle that transfers energy from fluid to a rotor that converts it into kinetic energy and the output shaft work can be used for different purposes. The process of scaling down a gas turbine is not as easy as it looks like, it possesses great difficulties. These gas turbines are used for power generation; also, they can be used for distributed power supply system. These micro gas turbines have also found its application in cogeneration systems, which supply steam and power at the same time (Shukla, 2013).

The integration between micro gas turbines and photovoltaic systems was previously proposed by Jaber (2004) in applications of high electricity demand times. The system contains an electric engine powered by photovoltaic panels responsible for compressor drive when solar radiation is available in order to stock compressed air for moments of need. This engine also acts as a generator when turbine is rotating and there is a clutch-like mechanism that allows coupling and decoupling compressor shaft to electrical engine and turbine shaft to electrical generator, depending on need. Sami and Mari (2017) have also proposed a similar system as represented in Fig. 1.

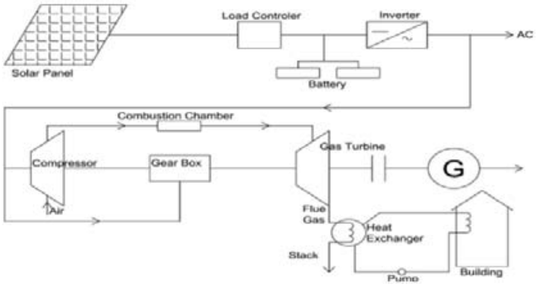

Figure 1. Block diagram of a gas micro gas turbine integrated with PV system (Sami and Marin, 2017).

In this case, the gearbox allows coupling compressor shaft to turbine shaft when solar radiation is not available or not sufficient, as proposed in the present study. The difference between each other consists in connection type. Sami and Marin (2017) presented a stand-alone system that uses batteries to store the generated electricity, and here, a grid-connected photovoltaic system. The last configuration is used in places already served by electricity, where photovoltaic systems are used to generate electricity for local consumption, which can reduce or eliminate consumption from the public grid or even generate surplus power (Villalva, 2017). Lately, the world has moved towards encouraging and rewarding those who adhere this type of energy generation.

THERMODYNAMIC MODELING

The sketch of a conventional regenerative microturbine is represented in Fig. 2 to visualize and number each state.

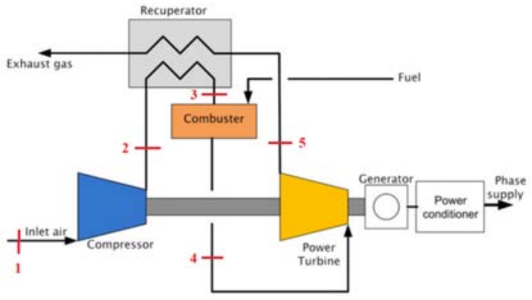

Figure 2. Block diagram of a micro gas turbine (Adapted from Ismail et al., 2013)

Next, the function and the thermodynamic math models of each component will be presented.

Compressor

In the beginning of the cycle, air under ambient conditions is admitted in a compressor where its pressure is increased along with temperature. The pressure ratio (*PR*) is defined as the ratio between the outlet (P_2) and inlet pressures (P_1). The inlet pressure is equal to the ambient pressure minus a pressure drop due to an air filter always installed in intake duct. The ideal compression/expansion process is adiabatic and irreversible, therefore isentropic. For real processes, the compressor isentropic efficiency (η_c) is defined by Eq. (1):

$$\eta_c = \frac{h_{2s}-h_1}{h_2-h_1} \tag{1}$$

h_1, h_2 and h_{2s} are the compressor inlet, outlet and outlet isentropic process enthalpies, respectively, given in kJ/kg and always a function exclusively of temperature for ideal gases, as considered.

The compression power ($\dot{W}_c$) is given in kW by Eq. (2):

$$\dot{W}_c = \dot{m}_a \frac{h_2-h_1}{\eta_m} \tag{2}$$

η_m is the mechanical efficiency and $\dot{m}_a$ the air mass flow rate in kg/s that will be more detailed in combustor analysis.

Recuperator

Recuperators are heat exchangers that preheat compressed air by recovering heat from turbine exhaust gas, thus reducing fuel consumption and improving the system efficiency, typically from 16–20% to 30% . A recuperator with high effectiveness and low pressure loss is mandatory for a good performance (Xiao et. al, 2017). For a wide temperature range the air and combustion gas constant-pressure specifics heat are practically equal. Thus, the recuperator efficiency (η_{rec}) can be approximated by Eq. (3):

$$\eta_{rec} = \frac{T_3-T_2}{T_5-T_2} \tag{3}$$

T represents temperatures, in K. The subscripts *3* and *5* indicate recuperator and turbine outlet states, respectively.

Combustor

The compressed air is then mixed with fuel and the mixture is burnt in a combustion chamber, where its energy gets increased. The combustor treatment is perfomed as suggested by Borglin (1991), whose combustor inlet air is divided into two parts: stoichiometric combustion air and excess air. The air stoichiometric combustion process with fuel produces stoichiometric gas. Thereby, instead of combustion gas leaving combustor, it is treated as a mixture of stoichiometric gas and excess air.

As biogas is mainly composed by methane (CH_4) and carbon dioxide (CO_2), the chemical stoichiometric combustion reaction is represented by Eq. (4):

$$aCH_4 + bCO_2 + Q_{est}(O_2 + 3{,}76N_2) \rightarrow xCO_2 + yH_2O + zN_2 \tag{4}$$

a, *b*, *x*, *y*, *z* and Q_{est} are the number of moles of each substance. For stoichiometric reaction the air fuel ratio (AFR_{est}) is defined by Eq. (5):

$$AFR_{est} = \frac{m_a}{m_f} = GFR_{est} - 1 \tag{5}$$

GFR_{est} is the gas fuel ratio. m_a is the air mass and m_f is the fuel mass.

Other important parameter is the fuel air ratio (*Fa*) given by Eq. (6):

$$Fa = \frac{h_{4a}-h_3}{LHV_f-(GFR_{est}h_{4g}-AFR_{est}h_{4a})} \tag{6}$$

Fa is obtained from an energy balance in combustor. LHV_f is the fuel lower heat value in kJ/kg and the subscripts *4*, *a*, *g* indicate outlet combustor state, air and combustion gas, respectively. h_{4g} is calculated by Eq. (7):

$$h_{4g} = f(T_4, P_4, a) = \frac{Y_{CO2}}{M_{CO2}} h_{CO2} + \frac{Y_{H2O}}{M_{H2O}} h_{H2O} + \frac{Y_{N2}}{M_{N2}} h_{N2} \tag{7}$$

Y and *M,* in kg/kmol, are the molar fraction and molar mass of each substance, respectively.

The species in the combustion gas were calculated by the stoichiometric gas-ratio (SGR) definition given by Eq. (8):

$$SGR = \frac{\dot{m}_{sg}}{\dot{m}_{cg}} = \frac{GFR_{est}Fa}{1+Fa} \tag{8}$$

$\dot{m}_{cg}$ and $\dot{m}_{cg}$ are stoichiometric gas mass flow and combustion gas mass flow, respectively, in kg/s. SGR equals 1 means stoichiometric gas and equals 0 means pure air.

Finally, the combustor outlet enthalpy (h_4), an essential design variable, can be calculated as shown in Eq. (9):

$$h_4 = SGRh_{4g} + (1 - SGR)h_{4a} \tag{9}$$

Ultimately, the combustion heat ($\dot{Q}$), in kW, is calculated by Eq. (10):

$$\dot{Q} = \dot{m}_f LHV_f = a\ \dot{m}_a\ Fa\ LHV_{CH4} \tag{10}$$

The air mass flow rate is a function of the required electrical power ($\dot{W}_{el}$) in kW, the specific turbine/compressor work (w_t and w_c) in kJ/kg, the electrical generator efficiency (η_{el}) and the fuel air ratio as shown in Eq. (11):

$$\dot{m}_a = \frac{\dot{W}_{el}}{(1+Fa)(w_t - w_c)\eta_{el}} \tag{11}$$

The difference between w_t and w_c is the cycle liquid specific work, w_l, since compressor and turbine are both mounted on the same shaft. In hybrid operation mode, when compressor is being powered by photovoltaic system and its shaft is uncoupled from turbine shaft, the air mass flow rate equation must be modified. Then, air mass flow rate for full hybrid operation ($\dot{m}_{ah}$) is calculated by Eq. (12):

$$\dot{m}_{ah} = \frac{\dot{W}_{el}}{(1+Fa)(w_t)\eta_{el}} \tag{12}$$

The power related parameters depend on air mass flow rate. Thus, they all also must be modified in hybrid operation.

Turbine

In turbine the energy transformation takes place. As well as compression process, there are irreversibilities in real expansion processes. The turbine isentropic efficiency (η_t) is defined by Eq. (13):

$$\eta_t = \frac{h_4 - h_5}{h_4 - h_{5s}} \tag{13}$$

The subscript 5 indicates turbine outlet state. As in turbine the working fluid is a mixture and EES library functions are just for single substance fluids, the calculation of T_{5s} and T_5, which depend on s_{5s} (specific entropy on state 5s, in kJ/kgK) and h_5, respectively, must be done with the definition of each property. The increase in specific entropy and enthalpy between states i and j can be calculated for an ideal gas by Eq. (14) and Eq. (15), respectively:

$$\Delta s = \int_{T_i}^{T_j} \frac{C_p(T)}{T} dT - R\ln(\frac{P_j}{P_i}) \tag{14}$$

$$\Delta h = \int_{T_i}^{T_j} C_p(T)\, dT \tag{15}$$

The constant-pressure specific heat, C_p, calculation is discussed in detail by Çengel and Boles (2006).

Is important to note that the integration must be done for each substance. Thereby, the analogous weighting shown in Eq. (7) and Eq. (9) must be done for specific entropy and enthalpy calculation. The same must be done to determine gas constant, *R*, in kJ/kgK.

Thus, turbine power ($\dot{W}_t$) is given in kW by Eq. (16):

$$\dot{W}_t = (\dot{m}_a + \dot{m}_f)(h_4 - h_5) \tag{16}$$

Other important parameter is the specific fuel consumption (SFC), measured in kg/kWh, calculated for conventional operation by Eq. (17):

$$SFC = \frac{3600\ Fa}{w_l} \tag{17}$$

To calculate the specific fuel consumption for both types of operation (SFC_b), Eq. (18), at times conventional, other times hybrid, the operation factor (*Fo*) must be taken into consideration. It quantifies how long micro gas turbine runs with compressor photovoltaic drive. Therefore, *Fo* equals 1 means all-day hybrid operation.

$$SFC_b = Fo.\frac{3600\ Fa}{w_t} + (1 - Fo).SFC \tag{18}$$

Lastly, the electrical efficiency (η_e) is given by Eq. (19):

$$\eta_e = \frac{\dot{W}_{el}}{\dot{Q}} \tag{19}$$

More details about micro gas turbine modeling are presented by Trindade *et al.*, 2020.

PHOTOVOLTAIC PANELS SIZING

To determine the number of photovoltaic panels required to power the compressor, it is necessary to consider the losses associated with conductor wires, electrical inverter and time of PV panels use. According to Villalva (2017), good inverters have efficiency above 90%. Considering a loss of 10% in inverter, 3% in conductor wires (Balfour *et al.*, 2019) and 20% as well as at endly useful life panels, as guided by Pinho and Galdino (2014) and the manufacturers' manuals, the daily energy produced by a photovoltaic panel (*E_d*), in kWh/day, can be calculated by Eq. (20):

$$E_d = 0{,}87.0{,}80.I_d.A_{mod}.\eta_{mod} \quad (20)$$

I_d is the average daily insolation in kWh/m²/day, A_{mod} is the area of each photovoltaic panel in m² and η_{mod} is the efficiency of photovoltaic panels.

By multiplying number of hours per day by compressor power, the energy demanded is determined. Therefore, the number of panels (N_{mod}) can be calculated by Eq. (21):

$$N_{mod} = \frac{24.Fo.\dot{W}_c}{E_d} \quad (21)$$

ECONOMIC ANALYSIS

To calculate electricity generated cost, it must take into consideration all the financial flow involved in equipment acquisition, installation, maintenance and operation. Thus, the electricity generated cost (C), in R$/kWh, can be calculated by Eq. (22):

$$C = \frac{f.C_{inv}+C_m}{H.\dot{W}_{el}} + \frac{C_f.SFC}{\rho} \quad (22)$$

C_{inv}, in R$, C_m, in R$, and C_f, in R$/m³, are the costs of investment, maintenance and fuel, respectively. The investment cost for hybrid system (C_{invh}) must consider the cost of photovoltaic system (C_{PV}) in addition to micro gas turbine cost. Area cost will not be considered, since usually there is space available for PV installation on roofs. H represents the annual period of operation in hours and ρ the fuel specific mass in kg/m³. f represents the annuity factor, dimensionless, that allows to transform a initial investment into a series of annual payments. According to Silveira and Tuna (2013), it can be calculated by Eq. (23):

$$f = \left[\frac{q^{k+CP}-1}{(q-1)q^{k+CP}} - \frac{q^{CP}-1}{(q-1)q^{CP}}\right]^{-1} \quad (23)$$

CP is the construction period, in years, and k the amortization period, in years, generally considered equal to the equipment useful life time. q is an auxiliary parameter, function of inflation (inf) and interest rate (ir), both in %/year, given by Eq. (24):

$$q = \left(1+\frac{inf}{100}\right)\left(1+\frac{ir}{100}\right) \quad (24)$$

C_{PV} can be calculated by Eq. (25):

$$C_{PV} = N_{mod}.P_{STC}.C_{Wp} \quad (25)$$

P_{STC} represents power under STC (Standard Test Conditions), given in watt-peak (Wp) by PV manufacturer, and C_{Wp} represents the watt-peak cost in R$/Wp.

The total financial revenue (FR) obtained - considering PV system useful lifetime equal to 25 years and the cost of electricity purchased from distribution company (C_{edc}) corrected for inflation year over year – with hybrid MGT acquisition can be calculated by Eq. (26):

$$FR = \sum_{i=1}^{25} H.\dot{W}_{el}\left[C_{edc}.\left(1+\frac{inf}{100}\right)^{i} - C_{elmh}\right] \quad (26)$$

The same equation is used to calculate *R* obtained for conventional MGT, just substitute electricity cost generated by hybrid MGT (*C_{elmh}*) for electricity cost generated by conventional MGT. Finally, the payback time can be estimated by dividing total investment cost by average revenue (total revenue divided by 25).

METHODOLOGY

Although solar energy is intermittent, permanent compressor photovoltaic drive, that is, *Fo* equals 1, can be considered, since Brazilian legislation allows generate energy credit to future use. Therefore, in this situation at night, the compressor is driven by grid electricity as credit for extra generation during the day.

According to Villalva (2017), for better sunlight harnessing throughout the day, photovoltaic panels should be oriented facing geographic north (null azimuth) for locations below the equator. In addition, for locations with latitude between 11° and 20°, the tilt angle of the panels should be equal to the latitude itself. Therefore, to find out the average daily insolation at Radiasol 2 for Belo Horizonte (MG), a tilt of 20° and azimuth equals zero were considered. The result found was 5,55 kWh/m²/day.

Since biogas price is usually given considering the methane volume, it is necessary multiply the second term in Eq. (22) by the methane mass fraction and the specific mass considered must be the methane density.

A program was elaborated in EES to calculate the desired parameters and then analyze the influence between them through parametric tables and graphics. The desired output thermodynamic parameters are specific fuel combustion and electrical efficiency for both operation types. The values of thermodynamic input parameters are shown in the Tab. 1.

Table 1. Values of thermodynamic input parameters

Parameters	Value(s) or range considered
Average ambient temperature at Belo Horizonte (K)	295
Atmospheric pressure at Belo Horizonte (kPa)	91.5
MGT Electrical Power (kW)	200
Electrical generator efficiency	0.96
Compressor pressure ratio	3-5
Combustion efficiency	0.98
Compressor isentropic efficiency	0.80 (1)
Turbine isentropic efficiency	0.86 [a]
Recuperator efficiency	0.80
Mechanical efficiency of compressor and turbine	0.98
Turbine inlet temperature (K)	1000; 1100; 1200
Lower heat value of CH_4 (kJ/kg)	50050 [b]
Pressure air drop at compressor inlet (% of atmospheric pressure)	1
Pressure air drop at recuperator (% of compressor outlet pressure)	2
Pressure air drop at combustor (% of recuperator outlet air pressure)	2
Pressure combustion gas drop at recuperator (% of turbine outlet pressure)	2
Pressure combustion gas drop at pipe after recuperator (% of recuperator outlet exhaust gas pressure)	1
Volume composition of biogas (% CH_4)	60

[a] Galanti and Massardo, 2011
[b] Çengel and Boles, 2006

The desired output economic parameters are electricity cost, total revenue and payback time for conventional and hybrid micro gas turbines. The input compressor pressures ratio will be the ones that maximize total financial revenue, as long as it does not exceed the value equal to 5, since this leads to small heat exchanges in recuperator, which is not interesting from the thermodynamic point of view. The values of economic input parameters are shown in the Tab. 2.

Table 2. Values of economic input parameters

Parameters	Value, range considered or consideration
Average daily insolation ($kWh/m^2/day$)	5.55
Dollar (R$/US$)	5.4
Photovoltaic system cost (R$/Wp)	4.24 [a]
Electricity cost purchased from distribution company (R$/kWh)	0.84 [b]
MGT cost (US$/kW)	3150 [c]
Inflation (%)	3.5
Interest rate (%)	3
Construction period (years)	1 [d]
Amortization period (anos)	25
Maintenance cost (US$/kWh)	0.016 [c]
Biogas cost (US$/$m^3$ de metano)	0.35 [e]
Turbine inlet temperature (K)	1100

[a] Falcão and Mighelão, 2018
[b] National Electric Energy Agency (in Portuguese), 2021
[c] Darrow *et al.*, 2015
[d] Galanti and Massardo, 2011
[e] International Renewable Energy Agency, 2017

RESULTS AND DISCUSSIONS

Among 20 models of 4 brands (Canadian Solar, JA Solar, JinkoSolar and Trinasolar), the PV module selected that generates the lowest acquisition cost is: CS3U 360P from Canadian Solar (η_{mod}=0,182, A_{mod}=1,984 and P_{STC}=360 Wp).

The specific fuel consumption behavior, for input data presented in Tab.1 and for different types of operation, are shown in Fig. 3.

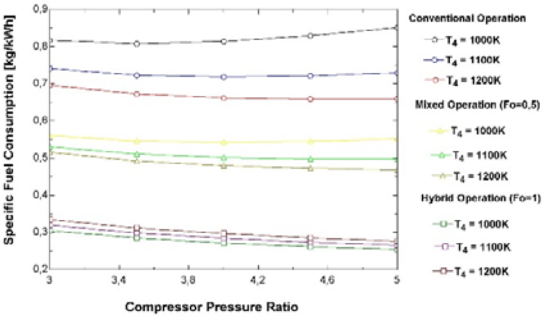

Figure 3. Specific fuel consumptions versus compressor pressure ratio for both operations.

Positive and negative slopes are found along specific fuel consumption curves for conventional operation since there are many variables involved. To justify the slopes it is necessary to analyze the influence degree of each variable point by point. As expected SFC decreases as T_4 increases for conventional operation. For hybrid operation, on the other hand, the specific fuel consumption behavior is decreasing all the time as compressor pressure ratio increases, since in this case it only depends on turbine specific work and no longer on compressor specific work, when *Fo*=1. As turbine expansion increases as air compression increases, the turbine specific work behaves equally. Since specific fuel consumption is inversely proportional to turbine specific work, it decreases as compressor pressure ratio increases is this case. Regarding T_4, the behavior is contrary to that observed for conventional operation, since *Fa* increase is more expressive than w_l increase for this operation type. Futhermore, it can be seen that specific fuel consumptions for hybrid micro turbines (*Fo*=1) are always less than half of the values found for conventional micro turbines. In some cases, less than a third. This reflects on electrical efficiencies as shows in Fig. 4.

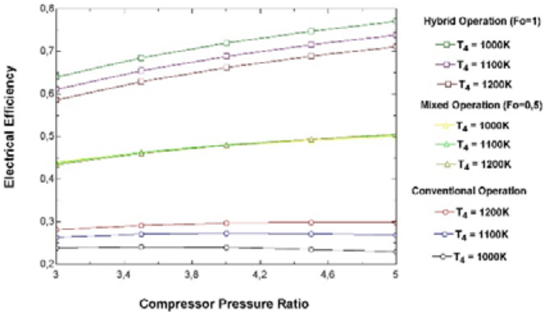

Figure 4. Electrical efficiencies versus compressor pressure ratio for both operations.

For mixed operation, 0<*Fo*<1, intermediate values of specific fuel consumption and electrical efficiency are expected as shown.

The values of output economic parameters are shown in Tab. 3 for the best operating pressure ratio for each configuration:

Table 3. Values of output economic parameters

	Conventional Operation (*PR*=4)	Hybrid Operation (*Fo*=1 and *PR*=5)
Electricity Cost (R$/kWh)	0.988	0.698
Total Financial Revenue (R$)	$1.606.10^7$	$2.875.10^7$
Average Payback Time (years)	5.3	5.9

Good indicators are shown. Positive and expressive revenues are found, as well as lower payback time than system lifetimes. The payback time for hybrid operation is a little higher than payback time for conventional operation, since the initial investment is higher too, but electricity cost found is lower and total financial revenue is significantly higher. Futhermore, electricity cost for hybrid operation is lower than current electricity cost purchased from distribution companies, which is favorable from now on, since positive revenue can already be obtained immediately. Thus, all results prove the compensation obtained with full equipment hybridization.

CONCLUSIONS

In this work a research was done to describe how MGT and MGT with compressor photovoltaic drive work. Its thermodynamic equations were presented together with other mathematical relations to evaluate the economic viability of the proposed alternative systems for electricity generation. An EES program based on methodological data could show how the two main thermodynamic parameters, *SFC* and η_e, behave when others vary and the significant improvement obtained with solarization. Promising economic indicators were also found and the tendency is to get better, since it is expected that acquisition cost of biogas (25% lower in 2040 according to IEA, 2020) and PV system will be reduced. Promising results probably will be obtained for analyses with other fuels, other electrical powers and other national locations, since Brazil stands out for the high insolation rate in its territory. Therefore, investing in research aimed at electricity generation through solar energy and other alternative and clean sources is an important path for society and planet preservation.

ACKNOWLEDGEMENTS

The authors would like to thank Brazilian people, Federal University of Ouro Preto, CAPES, FAPEMIG and Gorceix Foundation.

REFERENCES

Adefarati, T. and Bansal, R.C., 2016. “Integration of renewable distributed generators into the distribution system: a review”. Renewable Power Generation, Vol. 10, No. 7, pp. 873-884.

Adefarati, T. and Bansal, R.C., 2017. “Reliability and economic assessement of a microgrid power system with the integration of renewable energy resources”. Applied Energy, Vol. 206, pp. 911-933.

Ahmad, A.D., Abubaker, A.M., Najjar, Y.S.H. and Manaserh, Y.M.A, 2020. “Power boosting of a combined cycle power plant in Jordan: An integration of hybrid inlet cooling & solar systems”. Energy Conversion and Management, Vol. 214.

Aichmayer, L., Spelling, J. and Laumert, B., 2015. “Preliminary design and analisys of a novel solar receiver for a micro gas-turbine based solar dish system”. Solar Energy, Vol. 114, pp. 378-396.

Aichmayer, L., Spelling, J., Laumert, B. and Fransson, T., 2013. “Micro gas turbine design for small-scale hydrid solar power plants”. Journal of Engineering for Gas Turbine and Power, Vol. 135.

Balfour, J., Shaw, M. and Nash, N.B., 2019. Introduction to photovoltaic system design (In Portuguese). Editora LTC, Rio de Janeiro.

Borglin, S.H., 1991. “A contribuition to the calculation of open gas turbine processes”. ASME COGEN-TURBO, Vol. 6, pp. 115-129.

Cameretti, M.C., Langella, G., Sabino, S. and Tuccillo, R., 2015. “Modeling of a hybrid solar micro-gas turbine power plant". Energy Procedia 82, pp. 833-840.

Çengel, Y.A. and Boles, M.A., 2006. *Thermodynamics: An Enginering Approach (In Portuguese)*. Editora McGraw Hill.

Chen, J., Xiao, G., Ferrari, M.L., Yang, T., Ni, M. and Cen, K., 2020. "Dynamic Simulation of a Solar-hybrid Microturbine System with Experimental Validation of Main Parts". Renewable Energy, Vol. 154, pp. 187-200.

Darrow, K., Tidball, R., Wang, J. and Hampson, A., 2015. "Catalog of CHP Technologies". U.S. Environmental Protection Agency Combined Heat and Power Partnership.

Falcão, S.B., Mighelão, T.R., 2018. *The Brazilian Market of Distributed Solar PV Generation – Annual Report 2018.* Instituto para o Desenvolvimento de Energias Alternativas na América Latina (IDEAL), pp. 34.

Galanti, L., Massardo, A.F., 2011. "Micro gas turbine thermodynamic and economic analysis up to 500 kWe size". Applied Energy, Vol. 88, pp. 4785-4802.

Gupta, K.K., Sidiqui, A.R. and Sarviya, R.M., 2010. "Bio-fuels for the gas turbine: A review". Renewable and Sustainable Energy Reviews, Vol. 14, No. 9, pp. 2946-2955.

International Energy Agency (IEA), 2019. *Power generation in the New Scenario, 2000-2040*, https://www.iea.org/data-and-statistics/charts/power-generation-in-the-new-policies-scenario-2000-2040. Accessed 07 June 2021.

International Energy Agency (IEA), 2020. *Outlook for biogas and biomethane: Prospects for organic growth*, https://www.iea.org/reports/outlook-for-biogas-and-biomethane-prospects-for-organic-growth. Acessed 15 June 2021.

International Renewable Energy Agency, 2017. *Biogas Cost to Boost Sustainable Transport*, https://irena.org/newsroom/articles/2017/Mar/Biogas-Cost-Reductions-to-Boost-Sustainable-Transport. Acessed 10 June 2021.

Ismail, M.S., Moghavvemi, M. and Mahlia, T.M.I., 2013. "Current utilization of microgasturbine as a part of a hybrid system in distributed generation technology". Renewable Power Generation, Vol. 21, pp. 142-152.

Jaber, J.O., 2004. "Photovoltaic and gas turbine system for peak-demand applications". International Journal of Engineering and Technology, Vol. 1, pp. 28-38.

Javidmehr, M., Joda, F. and Mohammadi, A., 2018. "Thermodynamic and economic analyses and optimization of a multi-generation system composed by a compressed air storage, solar dish collector, micro gas turbine, organic Rankine cycle, and desalination system". Energy Conversion and Management, Vol. 168, pp. 467-481.

Meriche, I.E., Baghidja, A. and Boukelia, T.E., 2014. "Design and Performance Evaluation of Solar Gas Turbine Power Plant in South Western Algeria". International Journal of Renewable Energy Research, Vol. 4.

MosayebNezhad, M., Mehr, A.S., Lanzini, A., Misual, D. and Santarelli, M., 2019. "Technology review and thermodynamic perfomance study of a biogas-fed micro humid air turbine". Renewable Energy, Vol. 140, pp. 407- 418.

National Electric Energy Agency (in Portuguese), 2021. *Ranking de Tarifas*, https://www.aneel.gov.br/ranking-das-tarifas. Accessed 10 June 2021.

Pinho, J.T. and Galdino M.A., 2014. *Engineering Handbook for Photovoltaic Systems (In Portuguese)*. Editora CEPEL-CRESESB, Rio de Janeiro.

Sami, S. and Marin, E., 2017. "Simulation of solar photovoltaic, biomass gas turbine and district heating hybrid system". International Journal of Sustainable Energy and Environmental Research, Vol. 6, pp. 9-26.

Shukla, T., 2013. "Micro Gas Turbine – A Review". International Journal on Theoretical and Applied Research in Mechanical Engineering, Vol. 2, Issue 3.

Silveira, J.L. and Tuna, C.E., 2003. "Thermoeconomic analysis method for optimization of combined heat and power systems. Part I". Progress in Energy and Combustion Science, Vol. 29, pp. 479-485.

Somehsaraei, H.N., Majoumerd, M.M., Breuhaus, P. and Assadi, M., 2014. "Performane analysis of a biogas-fueled micro gas turbine using a validated thermodynamic model". Applied Thermal Engineering, Vol. 66, pp. 181-190.

Spelling, J.D., 2013. *Hybrid Solar Gas-Turbine Power Plants: A Thermoeconomic Analysis*. Doctoral Dissertation. KTH Royal Institute of Technology, School of Industrial Engineering and Management, Department of Energy Technology, Stockholm, Sweden.

Trindade, G.S., Bortolaia, L.A., Leal, E.M., 2020. "Biogas-Fueled Micro Gas Turbine Thermodynamic Analysis". In *Proceedings of the 18th Brazilian Congress of Thermal Sciences and Engineering - ENCIT 2020*. Virtual Conference, Brazil.

Villalva, M.G., 2017. *Photovoltaic Solar Energy: Concepts and Applications (In Portuguese)*. Editora Saraiva, São Paulo.

Xiao, G., Yang, T., Liu, H., Ni, D., Ferrari, M.L., Li, M., Luo, Z., Cen, K. and Ni, M., 2017. "Recuperators for micro gas turbines: A review". Applied Energy, Vol. 197, pp. 83-99.