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THERMODYNAMIC ASSESSMENT OF HYBRID SOLAR/ETHANOL MICROTURBINE

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Abstract. *Hybrid solar-biofuel turbines are very interesting from the economic and environmental point of view, thus many researches are being performed in the field, as well as implemented in industries over the last years. The aim of the present work is to analyze the performance of a solar hybrid micro gas turbine in different cycle configurations. The microturbine simulated was the Capstone C30, which is a recuperative gas turbine engine that operates with natural gas in standard condition. To simulate specific operational conditions, it was used the software GateCycle 6.1.2, using ethanol or natural gas as fuel, simulating a solar energy receiver located either in the high pressure or in the low pressure lines. The results showed a significant gain in thermal efficiency as the solar receiver increases the air temperature, particularly when the receiver is located in the high pressure line. Despite the higher net power obtained when using ethanol, the cycle efficiency was higher when using natural gas in all hybrid cases.*

Keywords: *hybrid, gas turbine, ethanol, solar.*

1. INTRODUCTION

Solar energy source is a type of energy highly attractive, because it presents a null level of pollutant emissions. It is also abundant, since the quantity of solar energy available per day could supply the energetic necessities of the mankind for years. The main disadvantage of the system is the high initial investment necessary, once the solar energy is diluted and needs to be concentrated. Another factor is the intermittence of the availability, once there is no energy available during the nighttime and it varies during the day due to clouds and atmospheric phenomenon. The great advantage of this type of source is that it is inexhaustible, clean, free and sustainable. The use of solar energy in Brazil is still few explored, once the share of solar energy is paltry compared to hydropower and wind sources (ONS, 2018).

One of the most recent research areas in gas turbines is its integration to a solar system of energy production. There are already plants operating in hybrid mode, but there are still many limitations regarding the energy supply to the turbine, once the solar light is an intermittent source if energy and the turbine entry temperature must be constant. Besides the reduction of pollutants, the operational cost of a hybrid system is lower, once the cost of the main fuel could be null.

The main concern about using fossil fuels is its inevitable depletion, therefore it is essential to focus resources in order to investigate new sources of renewable energy. The contribution of the solar energy is limited by the intermittency and difficulties in storing it. In a study performed by Cameretti et al. (2017), a hybrid system solar-gas turbine was simulated, the aim was to identify the optimal dimension of the solar sub-system that reduces fuel flow and pollutant emission. The maximum reduction in CO_2 emission obtained was 73 %, but it implied in a high cost due to the large number of heliostats required. Although, a reduction in CO_2 of 51.6 % was obtained when using less than 10 % of heliostats of the previous configuration.

The concept of hybrid turbines is being studied since the 1980 decade. Recently, Rovense et al. (2017) presented a micro gas turbine using solar power only. It was utilized a heat exchanger to cool down the fluid flow, thus maintaining constant the turbine and compressor inlet and outlet temperature. It was obtained the optimal number of solar multiples and found that the modified system could produce 38 % more power.

It was presented by Amelio et al. (2018) a regulation system that varies mass flow to maintain operational conditions during transient loads, in order to control the turbine inlet temperature in 800 °C. The system is able to provide global efficiency of 30 % and produce 1,08 MWh per year.

Investigations in open hybrid cycle was also performed. Barigozzi et al. (2012) modeled an industrial gas turbine with a solar receiver for the four seasons, with a reached temperature in the receiver of 1015 °C in the summer and 800 °C in the winter. The paper showed a significant increase in efficiency and consequent reduction of fuel consumption.

In order to provide sustainable solutions, studies with use of biofuels are also being performed. Cameretti et al. (2015) analyzed a power plant equipped with a hybrid solution of solar tower and a regenerative microturbine. It was simulated the performance for two fuels, natural gas and biogas. It was obtained a reduction of 52 % in fuel consumption during the best condition of combustion, and also a substantial reduction of NO and CO emissions.

Another factor that engages NO_x emissions is the fuel utilized. Emissions when using ethanol in micro gas turbines were verified by Chiong et al. (2018), and compared to diesel and naphtha it was observed a reduction of 50 % in NO_x. Switching fuels requires modification in the injection system but presents a feasible alternative on fueling gas turbines.

In real operation all the equipment present losses, and they were taken into account in a thermodynamic model developed by Merchán et al. (2018). Data from a heliostatic field in Seville, Spain, were utilized combined with real data of irradiance and temperature. It was verified that the economy in fuel could reach 11.5 % for the case analyzed, followed by reduction of CO_2, N_2O and CH_4 emitted.

Due to fluctuations in the solar energy availability, a technology that has been developed recently is storing thermal energy. It was verified by Grange et al. (2016) a raise of system efficiency from 0.295 to 0.307 in periods of the day with no solar light available when using thermal storing. It is known that gas turbines show better efficiencies when operating mid to full load. For this particular turbine, it was pointed by Nascimento (2014) that it achieves the highest efficiencies when operating from 12 kW to full load.

There is still a lot to be investigated for the purpose of optimizing the system, since there are points that can be explored to be achieved a rise in efficiency, in addition to the reduction of pollutant emissions. In this work are presented the effects in net power and efficiency of a Capstone C30 micro gas turbine, when operating with solar power in open cycle with ethanol or natural gas, considering a solar receiver located either in the high pressure line or low pressure line. The aim is to analyze different configurations of the cycle and its performance in power output, efficiency and fuel consumption when operating with different fuels.

2. METHODOLOGY

The performance of a micro gas turbine has been carried out, and it was considered two configuration cycles over the standard one. To simulate these configurations, it was utilized the GateCycle 6.1.2 software. Figure 1 represents the standard configuration. The configuration where the solar receiver is located in the high pressure line (between compressor and combustion chamber) is represented by Fig. 2. Figure 3 represents the configuration when using the receiver in the low pressure line (between turbine and recuperator). These two configurations were considered to verify which one is more efficient and whether it depends on the fuel used.

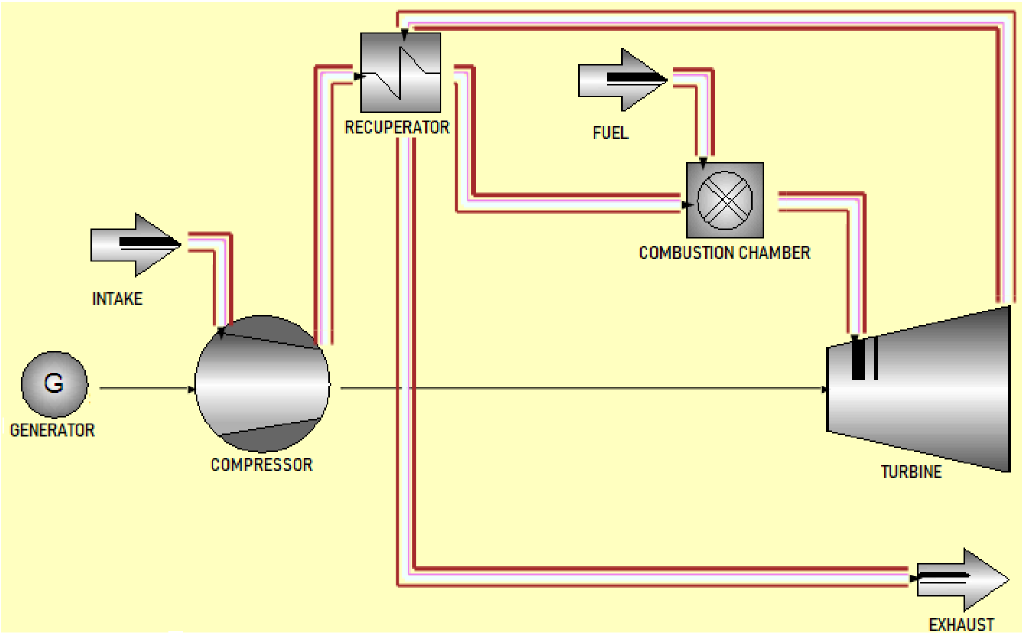

Figure 1. Scheme of the standard microturbine cycle.

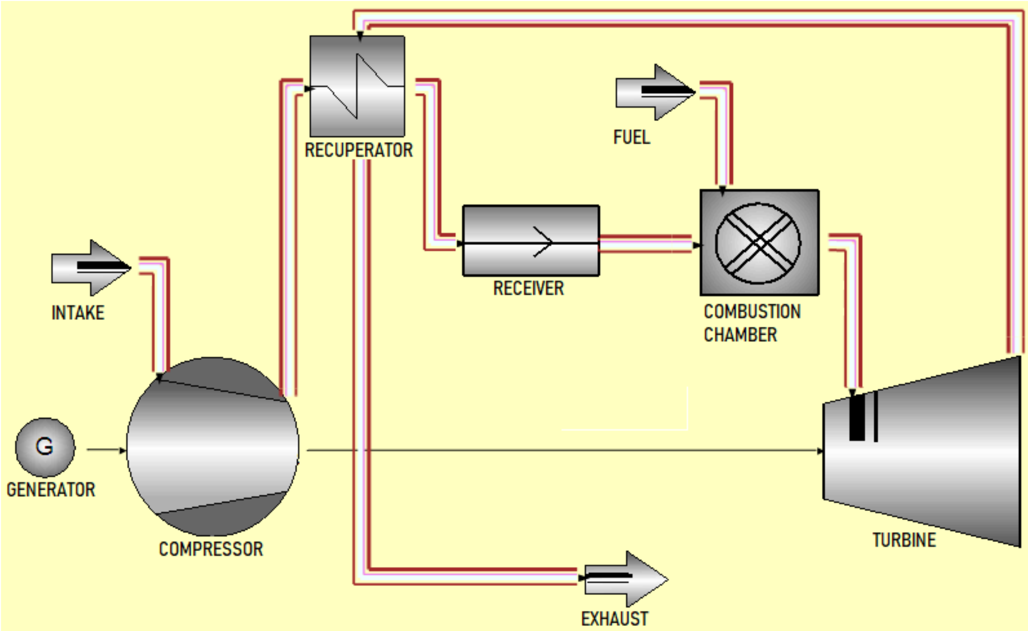

Figure 2. Scheme of the receiver located in the high pressure line.

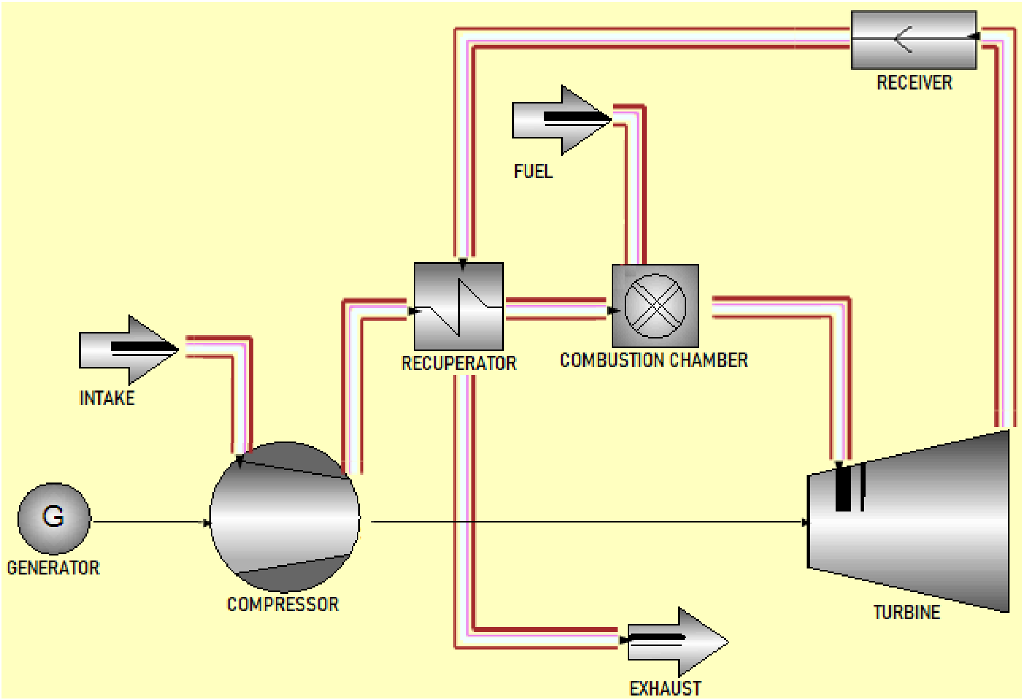

Figure 3. Scheme of the receiver located in the low pressure line.

The receiver considered the solar source of energy in the system as an amount of energy added to a specific part of the cycle. This method was utilized because of depending where the receiver is located, inlet and outlet temperatures of the recuperator will change, and the objective of this work is to find which configuration would be more efficient when a fixed quantity of solar energy is available, therefore the layout and characteristics of the solar system were not considered. The heat transfer coefficient of the recuperator was considered constant and pressure drops in these components were ignored. The main boundary conditions of the simulation are in Tab. 1, where P represents pressure, T temperature, φ relative humidity, ε effectiveness, η efficiency and LHV lower heating value.

Table 1. Boundary conditions for the micro turbine simulation.

Parameter	Value	Parameter	Value	Parameter	Value	Parameter	Value
P_{atm} [kPa]	101.325	$\varepsilon_{receiver}$	1	$\Delta P_{hair\ (recuperator)}$ [kPa]	0.03	$LHV_{natural\ gas}$ [kJ/kg]	47451
T_{amb} [K]	288.15	$\eta_{compressor}$	0.80	$\Delta P_{hgas\ (recuperator)}$ [kPa]	0.04		
φ	0.60	η_{burner}	0.97	$\Delta P/P_{receiver}$	0		
$\varepsilon_{recuperator}$	0.84	$\eta_{turbine}$	0.81	$LHV_{ethanol}$ [kJ/kg]	25737		

2.1 Fuel comparison

The standard configuration of the turbine was built in the software to compare data obtained by simulation with the data provided by the turbine's manufacturer. From this point, it was also simulated the performance in the same design but with ethanol only. The fuel consumption is expected to be higher because of the low LHV of the ethanol compared to natural gas, and can be found through Eq. (1),

$$F_c = \frac{\dot{W}_{liq}}{\eta \cdot LHV} \tag{1}$$

where W_{liq} stands for liquid power output and LHV lower heating value.

It is known that the performance and efficiency depends on the fuel utilized. The molar balance gives ethanol an advantage in terms of power, because of the reaction molar balance and the fuel mass flow, although it causes a drop in temperature due to phase change. Equation (2) can predict the total power output of the turbine based on the pressure ratio and fluid temperatures,

$$W_t = \left(1 + \frac{\dot{m}_f}{\dot{m}_a \cdot \eta_t}\right) \eta_t c_p T_3 \left(1 - \frac{1}{\left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}}}\right) \tag{2}$$

where $\dot{m}_f$ stands for fuel flow, $\dot{m}_a$ air flow, η_t turbine efficiency, c_p fluid specific heat, T_3 turbine entry temperature, P_3 turbine entry pressure, P_4 turbine outlet pressure and k specific heat ratio.

2.2 Configuration comparison

The main advantage of placing the receiver in the high pressure line is that less energy is lost when it does not need to be transferred through the recuperator. In addition, the receiver would have to reach lower temperatures in order to transfer energy to the system, because the initial temperature of the fluid would be lower. Nevertheless, pressure drops in the system would have greater effects on energy loss. According to Saravanamuttoo et al. (2017) it can be estimated through Eq. (3),

$$\Delta P = \frac{PLF\,\dot{m}^2}{2\rho A^2} \tag{3}$$

where PLF represents the power loss factor, $\dot{m}$ mass flow, ρ air specific mass and A maximum sectional area of the combustion chamber.

In this case, the effectiveness of the recuperator would interfere substantially on the global efficiency of the system, and according to Çengel and Boles (2013) it is given by Eq. (4),

$$\varepsilon = \frac{h_5 - h_2}{h_4 - h_2} \tag{4}$$

where h_2 represents the compressor outlet enthalpy, h_4 turbine outlet enthalpy and h_5 recuperator outlet enthalpy.

3. RESULTS

This section shows the results for the different configuration simulated. It was simulated the conditions for the nominal operation (natural gas) and the operation with ethanol. These results are displayed in Tab. 2, where $\dot{m}_g$ stands for exhaust gas mass flow, $\dot{W}_c$ power consumed by the compressor and $\dot{W}_t$ power produced by the turbine. It was observed an increase in power when using ethanol due to the increased mass flow and molar balance, followed by a slightly reduction in efficiency due to a higher pressure loss in the combustion chamber.

Table 2. Results for the standard operation using natural gas and ethanol.

Fuel	**LHV [kJ/kg]**	$\dot{m}_a$ **[kg/s]**	$\dot{m}_f$ **[kg/s]**	$\dot{m}_g$ **[kg/s]**	$\dot{W}_c$ **[kW]**	$\dot{W}_t$ **[kW]**	$\dot{W}_{liq}$ **[kW]**	**η [%]**
Natural Gas	47451	0.3075	0.00247	0.3100	54.24	88.07	30.45	25.91
Ethanol	25737	0.3075	0.00480	0.3123	54.24	89.34	31.6	25.59

It was then simulated the different configurations including the contribution of the solar energy in the high pressure line and low pressure line, each one for both fuels. Figure 4 shows the performance results, power/efficiency versus solar energy, when the receiver is located in the high pressure line, for each fuel utilized. As the energy transferred by the receiver increases, the net power output of the system decreases due to the reduction of fuel consumption and consequently mass flow.

As expected, the opposite is observed in efficiency. The reduction in power was 2.2 kW for ethanol and 1.1 kW for natural gas, between the minimum and maximum solar share considered. The difference in efficiency was 274 % for ethanol and 281 % for natural gas.

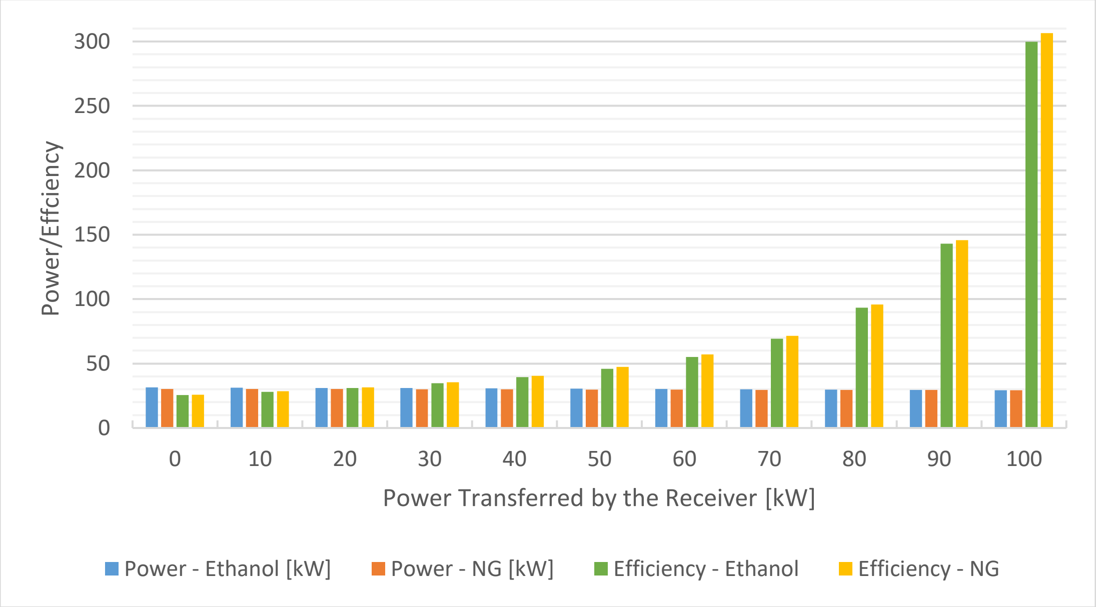

Figure 4. Performance results for high pressure line.

These results are also shown in Fig. 5, but for the receiver located in the low pressure line. It is possible to observe the same pattern regarding the power output and efficiency compared to the high pressure line for both fuels, but it was achieved substantial lower values for efficiency.

The range of solar share simulated was higher, due to losses in the recuperator, which allows the system receiving a larger share of solar energy without reaching the turbine entry temperature. The reduction in power was 2 kW for ethanol and 0.9 kW for natural gas. The efficiency, considering a range of solar share of 120 kW, increased in 257 % for ethanol and 263 % for natural gas. Considering the same solar share of 100 kW, the gain in efficiency was 77.5 % for ethanol and 80.42 % for natural gas.

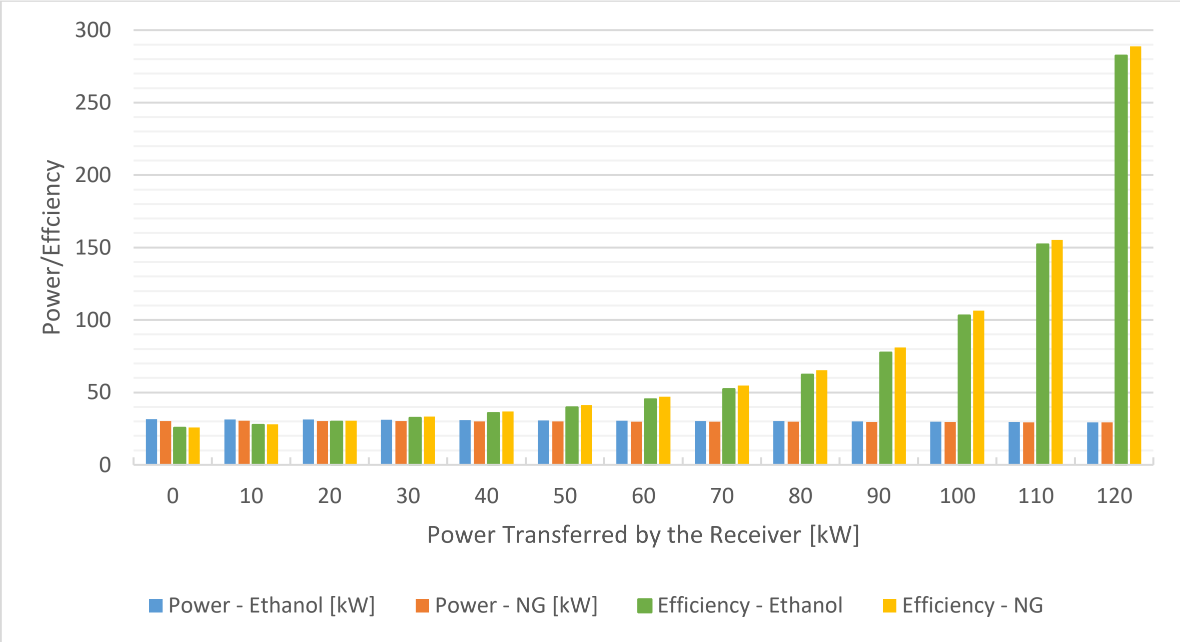

Figure 5. Performance results for low pressure line.

In both cases it is possible to notice a slightly higher power output when using ethanol as fuel (average 2 %), but natural gas provides a slightly higher efficiency in hybrid mode (average 2.7 %). It is also clear that placing the receiver in the high pressure line provides better results in this case, and the system efficiency is elevated abruptly as the solar share increases. It is also noticeable that as the solar share increases, the values of power and efficiency tend to equalize, due to the increase in similarity of the gas composition.

Figure 6 and Fig. 7 display the fuel flow versus solar energy for the high pressure line and low pressure line, respectively. As given by Eq. (1), it is inversely proportional to the LHV of the fuel and the efficiency, therefore expected to be the lowest when using natural gas and the receiver in the high pressure line; and to be the highest when using ethanol and the receiver in the low pressure line.

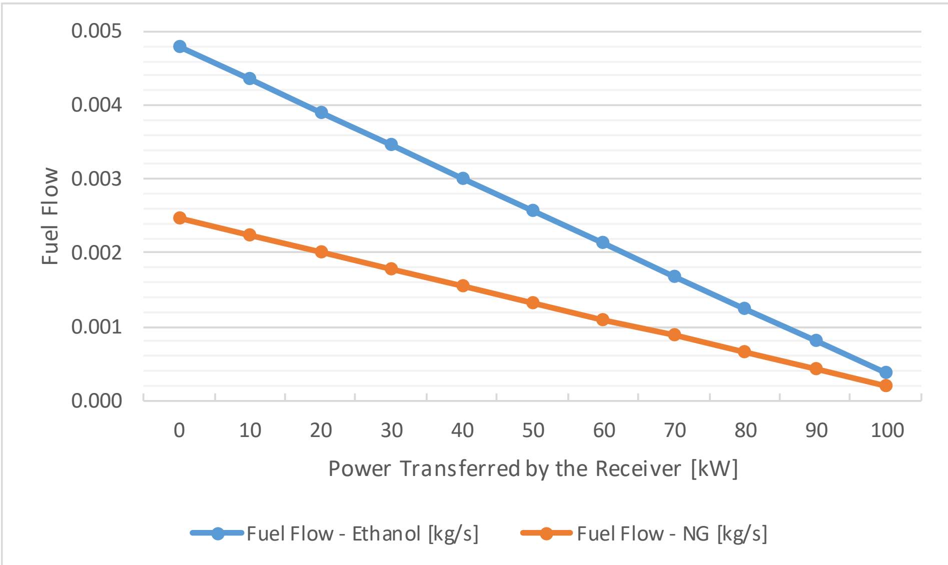

Figure 6. Fuel flow for high pressure line.

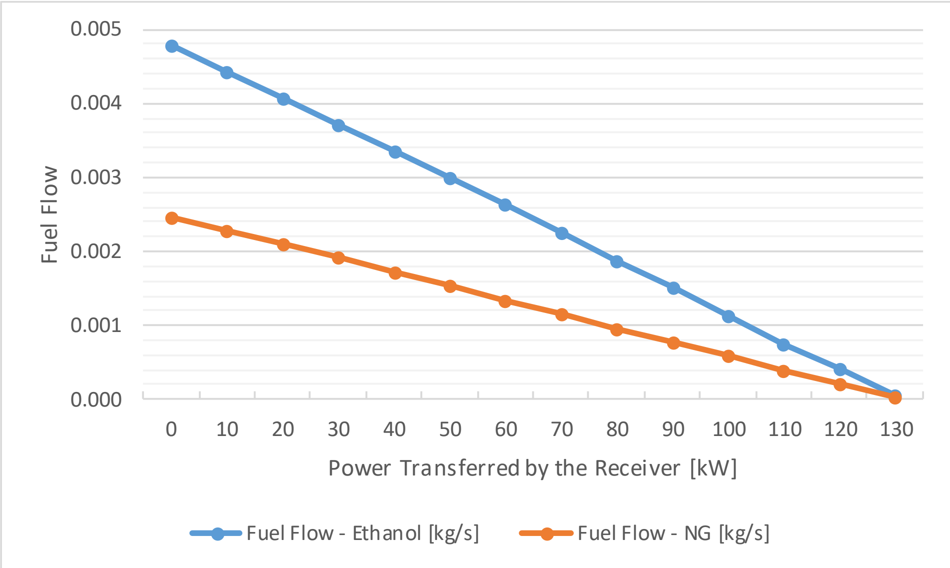

Figure 7. Fuel flow for low pressure line.

When using the receiver in the low pressure line it is achieved similar fuel flow for the solar energy transferred of 120 kW, compared to 100 kW when the receiver is located in the high pressure line, and it varies with the solar power used. The average increase in solar power for the turbine to consume the same amount of fuel is 21.26 %. A small increase in solar share for both cases, beyond the considerated, would make the fuel flow null.

4. CONCLUSIONS

Hybrid solar-biofuels gas turbines have been studied for thirty years, and the challenge is to optimize the integration of the solar system to the turbine, once it requires constant entry temperature. This work suggests there is an optimal configuration to place the solar receiver in the system, in this case when placed in the high pressure line, which enabled the best performance, achieving a rise in efficiency of 274 % with ethanol and 281 % with natural gas using 100 kW of solar power, compared to an increase of 257 % with ethanol and 263 % with natural gas for a 120 kW of solar power in the low pressure line. It means the system requires a 21.26 % higher solar power when adding heat to the low pressure line, for the same fuel flow. The effects of using different fuels on the turbine performance was also considered, achieving 2 % higher net power outputs when using ethanol and 2.7 % higher efficiencies when using natural gas. As expected, this is because the increased mass flow through the turbine and molar balance, in addition to higher pressure loss in the combustion chamber and change in the gas isentropic index, when using ethanol.

Although the results show a better performance for the placement of receiver in the high pressure line, it is necessary to verify the technical feasibility of this configuration for each type of turbine, once the operation conditions vary substantially in terms of temperature and pressure.

5. ACKNOWLEDGEMENTS

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