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GAS TURBINE ENGINE OFF-DESIGN PERFORMANCE SIMULATION USING SYNGAS FROM BIOMASS DERIVED GAS AS FUEL

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ABSTRACT

The use of syngas from gasified biomass as fuel for electric power generation based on gas turbine engines has been seriously studied over the past last two decades. Few experimental power plants have been built around the world. A small review of the use of syngas from gasified biomass and a cleaning system for gas turbine engines are presented. In this paper a computational program was presented and validated to simulate the design and off-design performance analysis of simple cycle gas turbine engines with one and two shafts. The aim was to assess the behavior and performance of the gas turbine engine without accounting for auxiliary syngas fuel compressor when the gasifier is atmospheric. It shows the behavior and performance at the off design condition of these two types of hypothetic gas turbine engines. The two engines were designed to use kerosene as fuel and at off-design conditions, and they were run using syngas from gasified biomass. The results show that the running line in the compressor characteristic moves towards the surge line and that the performance changes when the engine runs with the syngas.

INTRODUCTION

Due to its technological development and constant drop in costs gas turbine engines have been widely used as power generation elements, playing the role of other traditional engines. Furthermore, gas turbine engines have been taking part in more advanced generation plants such as Combined Cycle, Cogeneration Cycle (Combined Heat and Power), Steam Injection Cycle, etc.

Most of these plants are operated using conventional fuels. However, in the past decades another alternative to the conventional fuels was presented: the so called Low Calorific Value (LCV) fuels. Among them is the gas from gasified biomass, which significantly changes the performance and behavior of the engines, operating both at design point and at off-design conditions. Due to its relative low calorific value, the fuel mass flow is much larger than when the fuel is, for instance, natural gas, increasing substantially the mass flow through the turbine. This large mass flow rises the compressor pressure ratio, moving the working line towards the surge.

A small review regarding the use of syngas from gasified biomass and cleaning systems for gas turbine engines are presented, showing that there are problems when this fuel is used.

The use of syngas contributes to the reduction of NOx emissions due to its very low content of nitrogen and low flame temperature in the combustion primary zone. When the biomass comes from sustainable production, the CO_2 emissions can be decreased by 95%, since the CO_2 produced by the power plant will be reabsorbed by the plantation during its growth [1].

In this paper a computational program was presented and validated to simulate the design and off-design performance analysis of simple cycle gas turbine engines with one and two shafts. The aim was to assess the behavior and performance of the gas turbine engine without accounting for auxiliary syngas fuel compressor when the gasifier is atmospheric. It shows the behavior and performance of these two types of hypothetic gas turbine engines at off design condition. The two engines were designed to use kerosene as fuel and at off design conditions, they were run using syngas. The results show the running line in the compressor characteristic moves towards the surge line and the performance changes when the engine runs with syngas.

NOMENCLATURE

LCV – low calorific value
HHV – higher heat value, GJ/ton
LHV – lower heat value, GJ/ton
BIG/GT – biomass integrated gasification/gas turbine
IC – internal combustion engine
CIG/GT – coal integrated gasification/gas turbine
η_c – compressor isentropic efficiency
η_t – turbine isentropic efficiency
η_{pt} – power turbine isentropic efficiency

η_{cycle} – cycle thermal efficiency
η_{pb} – combustor pressure loss, %
m_{ar} – air mass flow, kg/s
m_c – fuel mass flow, kg/s
m_g – combustion gases mass flow, kg/s
w_u – specific useful work, kJ/kg
TET – turbine entry temperature, K
PR – pressure ratio
N – Rotational speed, rpm
SFC – specific fuel consumption, kg/kWh
$\frac{\Delta p_b}{p}$ - Combustion chamber pressure loss
N_p – design point rotational speed, rpm
Syngas – synthetic gas.

GASIFIED BIOMASS IN GAS TURBINE ENGINES

Biomass is an attractive alternative when compared to coal, which is the most commonly used source of energy in solid state and whose reserves are the largest. Biomass, in turn, contains less fixed carbon, ash, sulfur and nitrogen than the coal, and more hydrogen and oxygen (Table 1), contributing to its conversion into fuels and electricity through chemical and thermo-chemical processes. In most of the cases, it is not necessary to remove the sulfur. On the other hand, the wide variety of biomass needs specific designs for each sort of fuel. Other problems are related to its density. Dry biomass is required to avoid substantial reduction in the heating value of syngas.

At a larger scale than Internal Combustion Engines (IC engines), Gas Turbines fuelled by syngas is attractive as far as power generation is concerned. In contrast to steam-cycle technology, the unit capital costs of Brayton-cycle (gas turbine) systems are relatively insensitive to scale, this way, from a capital cost perspective, the gas turbine is a good candidate for achieving higher conversion [2].

In a BIG/GT (Biomass Integrated-Gasifier/Gas Turbine) power plant, shown in Fig. 1, the biomass may be gasified in an atmospheric (the gas fuel leaves the reactor at atmospheric pressure) or pressurized reactor (the gas fuel leaves the reactor above atmospheric pressure) and the products have to be free from particulates and tar before being compressed and burnt in the combustion chamber. The pressurized gasification is preferred to atmospheric gasification because it avoids thermodynamic losses related to the hot fuel gas compression; these losses are typically larger than those related to the compression of the gasification agent. The air-blown gasification becomes feasible due to the high costs related to oxygen gasification plants [3]. However, this paper do not consider the auxiliary gas fuel compression work, although the authors understand that up to 20% of the gas turbine net work can be used for this purpose [4].

Table 1 – Typical composition of some types of biomass and coal

Fuel	Proximate Analysis (Weight %)			Ultimate Analysis (Weight %)						HHV (GJ/ton)
	Volatile	Fixed Carbon	Ash	C	H	O	N	S	Ash	
BIOMASS										
Eucalyptus grandis+	82.55	16.93	0.52	48.33	5.89	45.13	0.15	0.01	0.49	19.35
Wood*	83.6	14.7	1.70	52.3	6.3	40.5	0.1	0.0	0.8	21.0
Bark*	70.6	27.2	2.20	56.2	5.9	36.7	0.0	0.0	1.2	22.0
Sugarcane bagasse+	73.78	14.95	11.27	44.80	5.35	39.55	0.38	0.01	9.91	17.33
COAL										
Ill. No 6 betumin+	37.50	43.40	19.10	65.34	4.20	6.59	1.02	4.55	18.30	26.67
W. Kentucky bet+.	33.12	48.18	18.70	65.78	4.62	4.86	1.26	4.74	18.74	27.81
Texas Lignite*	38.9	44.5	16.6	65.1	4.8	16.9	1.1	1.2	10.8	29.6

+ Source: [2]
* Source: [5]

The quality requirements for gas turbine engines are different from those needed for other applications. The gas has to be much cleaner than the one used, for instance, for heating systems or IC engines. Nevertheless, if the hot cleanup is done, the tar removal is not necessary unless it remains as vapor until it is burnt into the combustion chamber. Thus, one of the most important technical problems that has plagued BIG/GT systems would be eliminated. Another complication, however, is the need to remove trace amounts of alkali vapor from the gas before it enters the gas turbine and corrodes the blades [2].

Biomass integrated gasifier/gas turbine systems will be similar in some aspects to coal integrated gasifier/gas turbine (CIG/GT) systems, but biomass is more reactive than coal and so it can attain very high gasification efficiencies with gasification temperatures lower than those required for coal. This permits a variety of alternative gasifier designs to be considered – all involving dry-ash removal – with the potential for reduced costs. Also, most biomass contains little or no sulfur. Its removal at elevated temperatures is the key to the commercialization of economically viable CIG/GT technology. Against these benefits for biomass are alkali and moisture contents that are higher than for coal, and physical characteristics that make feeding biomass into a gasifier more challenging [3].

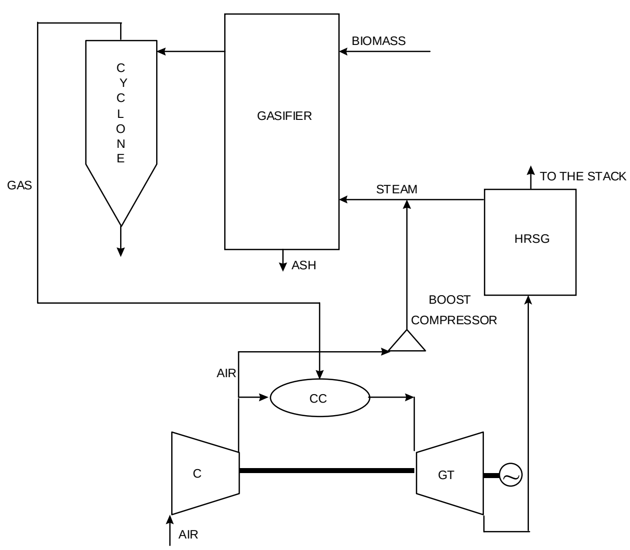

Fig.1 – BIG/GT system

THE GAS CLEAN UP

The clean up of the gas that will be used in gas turbine engines is the most important issue and it must be paid special attention. The gas turbine engines are very sensitive machines and the presence of particulates, tar and/or alkalis can easily affect their behavior and performance. Such compounds cause erosion and corrosion of the turbine blades, reducing its lifetime and increasing the maintenance costs. Due to little practical experience in using gas turbines with syngas, the real limits of contaminants that the engine can deal with are still not very precise [3].

Particulates, even in small quantities, can cause erosion in the gas turbine blades. The aero derivative gas turbine engines are more susceptible because they are not designed to the heavy industrial environment. Some manufacturers specify very small limits to particulates, for example, General Electric (GE) specifications for its turbines (both heavy duty and aero derivative engine varieties) require a total concentration below 1 ppmw at the turbine inlet, with 99 percent of the particle presenting less than 10 µm of diameter. This corresponds to a particulate concentration in uncombusted low heat value gas of about 3 – 5 ppmw [3].

Alkalis corrode the turbine. For these elements the manufacturers specify a maximum concentration of 4 ppbw in combustion products of aero derivative gas turbines and 2 – 3 times more to heavy-duty engines. During biomass gasification, alkali metals are vaporized (sodium and potassium > 600 °C) and leave the gasifier with the gas. To remove those metals, it is necessary to cool the gas (about 350 °C – 400 °C) in the presence of liquids or solids in which they can be condensed and be removed from the gas stream. Or one can use a scrubber, resulting in a more significant cooling and total removal of the alkalis. However, the cycle efficiency is penalized.

Another problem is the formation of tars during the gasification. They account for 0.5 to 1.5% of mass [3] depending on the temperature and the type or reactor used. If the tars are condensed in cooled surfaces, serious operational problems may occur, for example, valves and filters blockage. Tars constitute a very important energetic compound and remove them will reduce the calorific value of the gas.

The gas characteristics will depend on the type of the gasifier. Although oxygen gives better quality, the atmospheric gasifier with air is the most economically feasible, and for this reason this kind of gasifier can became the most commonly used for commercial applications. The composition of the gas, given in Table 2, corresponds to a typical composition of a gas obtained from the gasification of vegetable biomass.

Table 2 – Typical composition of syngas (by volume)

CO	21%
CO2	11%
CH4	3%
C2H4	1%
H2	17%
H2O	3%
N2	44%
LHV	5.52 MJ/kg

Source: [1], p. 512

THE CALCULATION METHOD

A computer program has been developed to simulate design point and off design of simple cycle, one and two shaft engines (Fig. 2 and Fig. 3). For both engines, the design point simulation may use kerosene fuel and the off design may also use syngas fuel, and also the design point and off design simulation may be for kerosene or syngas. The composition of the syngas used in the simulation is displayed in Table 2.

The calculation method and the equations used in the computer program (SIMULATION code) for steady state condition is presented by [4], which is based on thermodynamic cycle calculation, non-dimensional parameters, individual component performance characteristics, rotational speed, mass and energy compatibility among the gas turbine engine components, particularly the compressor and the turbine because they are directly coupled together.

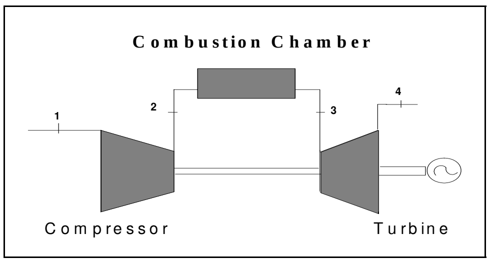

Fig. 2 One-shaft engine scheme

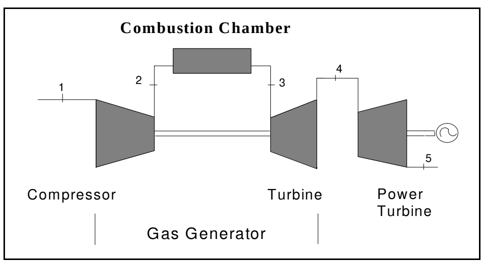

Fig. 3 Two-shaft engine scheme

In order to carry out the calculations some considerations had to be done:

- constant pressure loss in the combustion chamber;
- constant combustion efficiency;
- no bleeds in the compressor;
- losses in the filters, in the intake and exhaust systems are neglected;
- the turbine is considered choked all over the engine operation range;
- the auxiliary gas fuel compressor used to compress the gas fuel is not considered in calculations.

SIMULATION Code Validation

The results from the developed computer program (Simulation code) were validated against the results generated by TURBOMATCH [6] (Fig. 4 and Fig. 5), which has been developed at Cranfield University, UK, for more than 30 years. The same design parameters were assumed for the off design simulation process in both computer programs.

Figures 4 and 5 show the results of the specific fuel consumption – SFC versus percentage of design power for TURBOMACH and SIMULATION code for off design point, considering the design point TET=1300K, PR=20 and ISO condition for one-shaft engine (Figure 4) and two-shaft engine (Figure 5).

The results in Figures 4 and 5 were obtained for electrical load and kerosene fuel, varying power output from 100% to 40% in both computer programs.

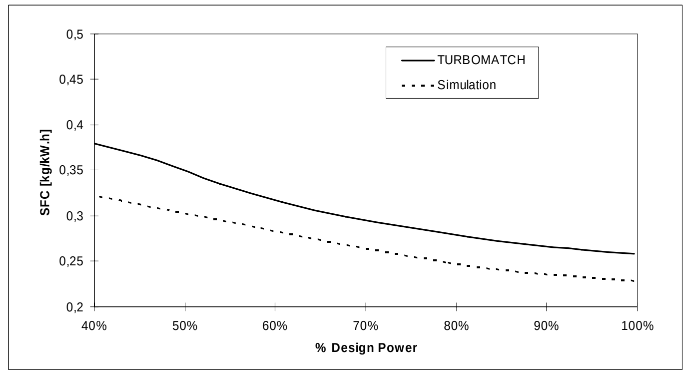

Fig. 4 - Results validated for the one-shaft engine

In Figure 4, the difference between the two results is not more than 12% from 100% down to 60% of design power output and Figure 5 it is not more than 10% from 100% down to 40% of design power output. The difference between the results of TURBOMATCH and SIMULATION code is caused by the characteristics of the components and the calculation methodology in SIMULATION code, which are slight different from TURBOMATCH.

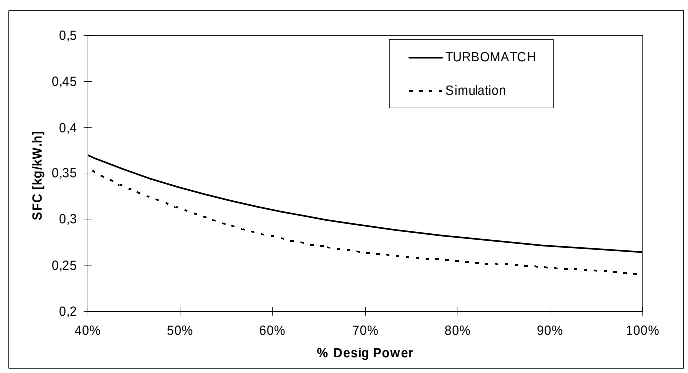

Fig. 5- Results validated for the two-shaft engine

ENGINE PERFORMANCE

After the validation process, the SIMULATION code was used for simulating design and off design performance of one and two shaft engines at the conditions written below.

Design Point

For the design point simulation the following conditions were assumed: the engine performance calculations were carried out according to ISO conditions; the Turbine Enter

Temperature (TET) was assumed at 1300K; and the pressure ratio was 20. These values were chosen by taking the current technology of gas turbine engines into account. Atmospheric gasified was considered without syngas compressor for the simulation of the one and two shaft engines. The aim was to investigate the engine behaviour and performance.

The chosen power output was 41 MW at constant rotational speed (Electrical application). The component efficiencies and the pressure loss in the combustion chamber were taken according to those found in the literature, η_c = 85%, η_t = 87%, η_{pt} = 87% and $\Delta P_b/P$ = 2% (percentage of compressor delivery). Finally, for the off design simulation, the characteristic maps of axial compressor and turbines were taken from [7].

Table 3 shows the main performance parameters results for design point simulation of two one-shaft engines (one for syngas and other one for kerosene as fuel) and two two-shaft engines (one for syngas and other one for kerosene as fuel) at TET=1300K, PR=20, power output of 41 MW and ISO condition. m_{ar} is the compressor inlet air mass flow and m_c is the fuel mass flow, so that the exhaust mass flow is the sum of air mass flow and the fuel mass flow.

Table 3 shows that, regarding the kerosene, the performance does not change much between the one and two shaft gas turbine engines. However, when the syngas is used, the two-shaft configuration shows a much better performance than the one-shaft gas turbine engine.

When the engines operate with syngas, the compressor reduces air mass flow and increases the net specific power, as shown in Table 3.

The syngas fuel flow is higher than he kerosene fuel flow because its heat value is lower than the kerosene one. Hence, more syngas fuel mass flow has to be fed to reach the same power output required.

In addition to this, the gas mass flow and its properties through the turbine are different, when the gas turbine engine is operating with syngas fuel. In the two-shaft engine, the cycle expansion rate occurs in two stages (compressor turbine and power turbine). This improves the engine performance even more, reducing compressor inlet mass flow.

These effects are higher in two-shaft engines. As far as the thermal efficiency is concerned, it is higher for the two-shaft engine working with syngas, about 43%. This thermal efficiency does not account for auxiliary syngas compressor to feed the gas turbine engine.

The design point simulation suggests that the two-shaft engine is better appropriated choice to use syngas.

Table 3 – Simulation results of the main performance parameters for design point of one and two shaft gas turbine engines operating with kerosene and syngas at TET 1300K, PR= 20 and ISO condition.

Design Configuration	Design Fuel	m_{ar} kg/s	m_c kg/s	m_g kg/s	w_u kJ/kg	η_{cycle} (%)
Single shaft	Biomass-derived gas	205.48	19.61	225.09	199.5	37
	Kerosene	241.41	2.98	244.39	169.8	39
Two-shaft	Biomass-derived gas	152.24	17.75	169.99	269.3	43
	Kerosene	247.99	3.07	251.06	165.3	38

Off design Point

The off design point performance simulation were carried out changing TET in order to simulated partial loads. The one and two shaft engines were designed for kerosene and the off design could be for kerosene or for syngas. This is to simulate an existing engine that has been modified to operate with syngas as fuel.

Figure 6 shows the thermal efficiency versus percentage of power of the design point with TET 1300K and pressure ratio 20 for the one-shaft engine. One can see that the one-shaft engine designed to run with kerosene presents a lower efficiency running with syngas. This happens because the large amount of mass, due to the use of syngas, leads to a lower TET, consequently to a lower difference in temperature in the expansion. Another cause is that when running with a fuel of lower LHV, the components are constrained to operate in regions of lower isentropic efficiency and the gas properties after the combustion chamber are different.

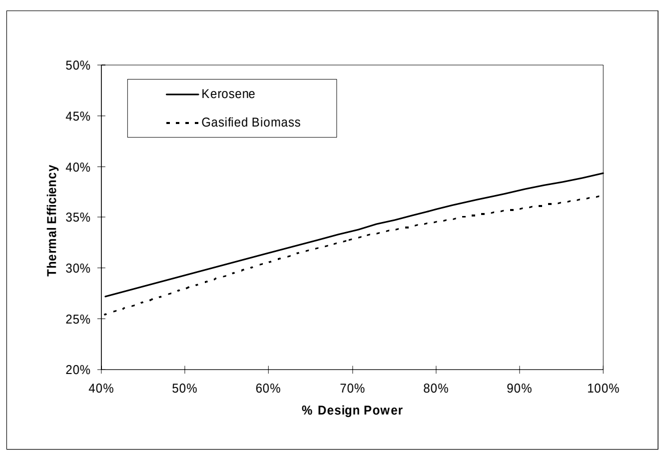

Fig. 6- Thermal efficiency vs. Power for the one-shaft engine

Figure 7 shows the thermal efficiency versus percentage of power of the design point with TET 1300K and pressure ratio 20 for the two-shaft engine. In Figure 7 it is possible notice that, for two-shaft engines, the opposite occurs, i.e., the thermal efficiency of this cycle is better when syngas is used. Due to the larger mass flow the free power turbine delivers more useful work, leading to a higher thermal efficiency.

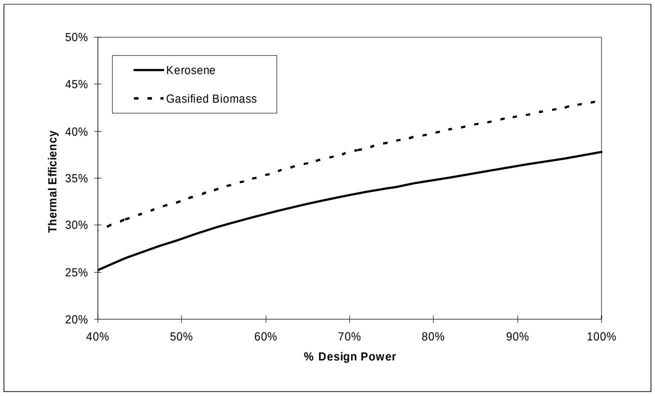

Fig. 7. Thermal efficiency vs. Power for the two-shaft engine

Figure 8 shows the running line in the compressor map for the two-shaft engines operating with the designed fuel, i.e., kerosene, and syngas which is very important to assess the engine stability. One can see that the surge margin is better for the kerosene when the gas turbine runs in high rotational speed. The use of syngas reduces the surge margin, in other words, it moves the running line towards the surge. This happens because the turbine nozzle is chocked and the gas mass flow is limited.

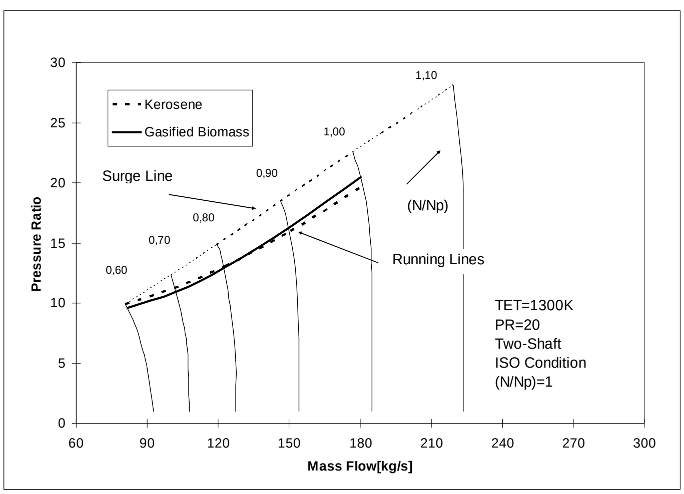

Fig. 8- Running lines for kerosene and syngas for the two-shaft engine

As the running line is approaching the surge line, the gas turbine engine operation gets more aerodynamically instable.

Because the turbine nozzle is chocked, no more mass flow can go through the turbine vanes. However, when the syngas is used more fuel flow is needed to reach the specified TET and the compressor has to reduce air mass flow in order to keep the turbine mass flow constant. If the one-shaft gas turbine engine operates at constant rotational speed the compressor goes to high pressure and low isentropic efficiency in the compressor map, reducing the surge margin. In the two-shaft gas turbine engine the gas generator reduces air mass flow reducing rotational speed, increasing the pressure slightly and keeping the isentropic efficiency higher than the one-shaft engine.

When the engine rotational speed goes to a lower value, the turbine nozzle gets unchocked and more gas mass flow can go through the vanes moving the running line away from the surge line as it can be seen in Figure 8, in the range of no dimensional rotational speed between 0.6 and 0.8.

CONCLUSIONS

The use of syngas fuel in gas turbine engines seems to be technically feasible. Some modifications are required in the combustion chamber to accommodate the syngas increased volume, residence time, starting fuel, flame stability, etc. and the gas cleaning is one of the most important issues concerning to the use of this gas. The engine simulation shows that the type of engine configuration, one or two-shaft, plays an important role in the thermal efficiency. In some cases it is about 16% higher, and the two-shaft engine presented a good performance at this power, which indicates that the use of aero derivative two-shaft cycles is the most promising engine for syngas .

Another important effect that was show in the simulation is the reduction in the surge margin to high rotational speeds, due to the large mass flows. Depending on the case this reduction in the surge margin would not be tolerated because of the engine operation stability. Solutions to this problem have been already developed such as variable inlet guide vanes (VIGVs), bleeding, etc.; all of them associated to suitable control systems.

When syngas fuel auxiliary compressor is used, the thermal efficiency of the system (the gas turbine engine and the auxiliary gas fuel compressor) decreases to a lower value due to the energy balance. However, the engine behaviour for different configurations must be taken in account as shown in this work.

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