IMPROVING THE PERFORMANCE OF A SPARK IGNITION ENGINE FUELLED WITH SYNGAS BY ADDITION OF HHO GAS

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ABSTRACT

Biomass gasification plants play an important role in rural electrification projects. They can eliminate agricultural waste that may damage the environment while providing a local source of energy, helping to develop areas isolated from the grid. However, the low calorific value of syngas compared to gasoline can reduce the performance of the generators in these plants. Therefore, it is proposed in this article to enrich syngas by adding HHO gas to improve the performance of the spark-ignition engine coupled to the alternator. Obtained by electrolysis of water, this additive fuel is made up of a volume of 2/3 hydrogen and 1/3 oxygen, hence its name HHO. To study the effects of enrichment, a single-cylinder 4-stroke engine powered by the syngas-HHO mixture is modeled on GT-Power, a component of the GT-Suite software. The objective of the study is to determine the optimal syngas-HHO mixture for maximum power and efficiency while reducing pollutant emissions. Simulations show the benefits of HHO enrichment of syngas. The best engine performance is obtained with a mixture of 90% syngas and 10% HHO, and an Air-Fuel ratio of 5. Improvements are seen in engine power output, specific fuel consumption, and lower pollutant emissions compared to gasoline operation. The HHO enrichment of syngas therefore improves the electricity production in biomass power plants. The study presented in this article thus contributes to increase the energy autonomy of rural areas and so can promote their sustainable development.

Keyword: *Spark-ignition engine - Syngas - HHO - Power – Efficiency – Pollutants*

Nomenclature

ρ	fuel density	$HVAP_f$	fuel heat of vaporization, J/kg
η_{comb}	combustion efficiency	I_{ct}	crankshaft inertia, kg.m^2
$[x]$	element x concentration, mol/cm^3	k_{CO}^+	rate constant for CO formation
A	cross-sectional flow area	k_i^+	rate constant for NO_x formation
AA	anchor angle	K_p	pressure loss coefficient
A_s	heat transfer surface area	LHV_f	fuel lower heating value, J/kg
$avgrpm$	average engine speed (per cycle)	$Loss_{comb}$	combustion losses
BE	burned fuel percentage at duration end	m	mass of the volume
BEC	burned end constant	m_a	air mass
b_{eff}	engine efficiency	m_f	fuel mass
bkw	brake power	$mf_{,i}$	injected fuel mass
BM	burned fuel percentage at anchor angle	m_u	unburned zone mass
BMC	burned midpoint constant	$\dot{m}$	boundary mass flux into volume, $\dot{m} = \rho A u$

BS	burned fuel percentage at duration start	$\dot{m}_f$	fuel consumption rate
BSC	burned start constant	$\dot{m}_{f,i,liq}$	instantaneous mass flow rate of fuel entering the cylinder controlled volume i in liquid state, kg/s
bsfc	brake specific fuel consumption	$\dot{m}_{f,i,gas}$	instantaneous mass flow rate of fuel entering the cylinder controlled volume i in gaseous state, kg/s
CE	fraction of fuel burned	*N*	engine speed, rpm
C_f	fanning friction factor	n_r	engine rotation per cycle (1 for 2-stroke engines, 2 for 4-stroke engines)
D	combustion duration	*P*	pressure
Di	equivalent diameter	*p*	cylinder pressure
dp	pressure differential acting across *dx*	*po*	number of pole pairs
dx	length of mass element in the flow direction (discretization length)	Q_b	burned zone heat transfer rate
e	total specific internal energy	Q_u	unburned zone heat transfer rate
E	Wiebe exponent	*SOC*	start of combustion
engfen	fuel power entering the cylinder	*t*	time
e_u	unburned zone energy	$T_b(t)$	brake torque
f	frequency	T_{fluid}	fluid temperature
GT	Gamma Technologies	T_{wall}	wall temperature
H	total specific enthalpy	*u*	velocity at the boundary
h	heat transfer coefficient	*V*	volume
h_a	enthalpy of air mass	V_u	unburned zone volume
h_f	enthalpy of fuel mass	*WC*	Wiebe constant
$h_{f,\,i}$	enthalpy of injected fuel mass	$\dot{\omega}_{ct}$	crankshaft acceleration, rad/s^2

1. INTRODUCTION

Biomass gasification is one of the solutions currently used for the partial or even total substitution of fossil fuels in rural electrification projects. It generates, from agricultural residues, a combustible gas called syngas, which is made up of carbon monoxide, hydrogen, methane, and other non-combustible components. This syngas can be used directly as a fuel for spark-ignition engines or as an additive fuel for dual-fuel diesel engines [1] [2].
However, as the energy density of syngas is relatively low compared to gasoline [1] - [2], it is proposed to enrich it with HHO, highly flammable gas with a high calorific value, to achieve rapid and complete combustion. This enrichment is supposed to increase the power of the engine while reducing the consumption of syngas. In other words, this innovation should allow increasing the amount of electrical energy produced by the power plant with the same amount of syngas.
In the scientific literature, other studies have already been carried out on the enrichment of syngas with other types of fuel. One example is the work of Hagos et al. [3]–[5] who mixed methane with syngas. Szwaja et al. [6], and Nadaleti and Przybyla [7] also used biogas to improve the performance of engines running on syngas. But the originality of this work lies mainly in the choice of HHO to enrich syngas. Produced by the electrolysis of water, HHO has a volume composition of 1/3 oxygen and 2/3 hydrogen.
The objective of this work is to show the improvements in the performance of a spark-ignition engine fuelled with syngas by the addition of HHO. The parameters for measuring this performance are power, efficiency, fuel consumption, and emissions of nitrogen oxides (NOx) and carbon monoxide (CO).
The main issues addressed in this study can be expressed by the following questions:
- What is the stoichiometric air-fuel ratio of syngas?
- How much HHO should be mixed with syngas to maximize engine performance?
- What is the stoichiometric air-fuel ratio of the syngas-HHO mixture?
To answer these questions, the engine powered by HHO-enriched syngas is modeled on the GT-Power tool. The performance of the engine operating under different conditions is then calculated and discussed according to the stated objectives.

2. ENGINE MODELLING

To study the performance of the spark-ignition engine fuelled with HHO enriched syngas, it is proposed in this article to model it on GT-Power to simulate its operation. Simulation allows the study of the operation of an engine under different conditions while overcoming the various obstacles encountered during experimental tests such as the cost and availability of the test bench, the difficulty in measuring certain parameters without compromising the phenomenon to be measured (example: temperature and pressure in the combustion chamber) [8].

2.1 GT-Power tool

The GT-POWER tool is a component of the GT-SUITE software developed by Gamma Technologies especially designed for the simulation of internal combustion engines and vehicles. It is based on the one-dimensional simulation of fluid dynamics, i.e. material flows and heat transfers, in the various manifolds and other system components. The modeling is based on predefined components in the GT-POWER library that will be configured according to the characteristics of the system to be simulated and then linked together to form the model [9].

2.1.1 Fluid dynamics modeling

To model the dynamics of the fluids circulating in the engine, the various pipes and manifolds that make up the engine are discretized into several volumes (see **Fig -1**). These volumes are connected by boundaries. The variables characterizing the fluid circulating in the tubing are classified into two types [10]:
- scalar variables (pressure, temperature, density, internal energy, enthalpy, the concentration of each species, etc.) considered uniform in each discrete volume,
- and vector variables (mass flow, velocity, etc.) calculated at the boundaries.

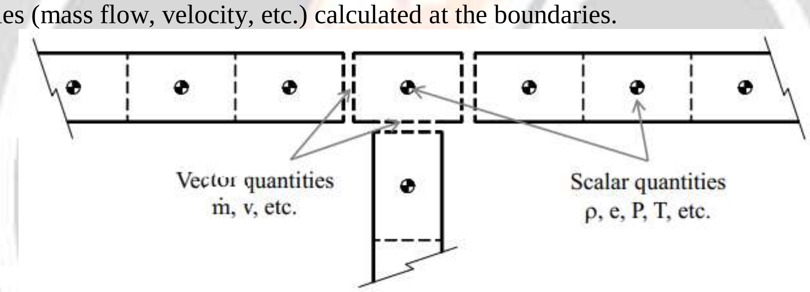

Fig -1: System discretization [10]

The determination of these variables is based on the conservation equations of continuity (1), energy (2), enthalpy (3), and momentum (4). [10]:

Continuity : $$\frac{dm}{dt} = \sum_{boundaries} \dot{m} \qquad (1)$$

Energy : $$\frac{d(me)}{dt} = -p\frac{dV}{dt} + \sum_{boundaries}(\dot{m}H) - hA_S(T_{fluid} - T_{wall}) \qquad (2)$$

Enthalpy : $$\frac{d(\rho HV)}{dt} = \sum_{boundaries} + V\frac{dp}{dt} - hA_S(T_{fluid} - T_{wall}) \qquad (3)$$

Momentum : $$\frac{d\dot{m}}{dt} = \frac{dpA + \sum_{boundaries}(\dot{m}u) - 4C_f\frac{\rho u|u|}{2}\frac{dxA}{Di} - K_p\left(\frac{1}{2}\rho u|u|\right)A}{dx} \qquad (4)$$

2.1.2 Combustion modeling

The combustion chamber is modeled using the two-zone method:
- the "unburned" zone which includes the mass of air and fresh fuel,
- and the "burned" zone (see **Fig -2**).

Combustion, therefore, corresponds to the transfer of a defined quantity of mass of fuel and air with the corresponding enthalpy from the "unburned" zone to the "burned" zone; with the release of chemical energy from the air-fuel mixture. The resulting products and their concentrations are then evaluated according to this transfer [8].

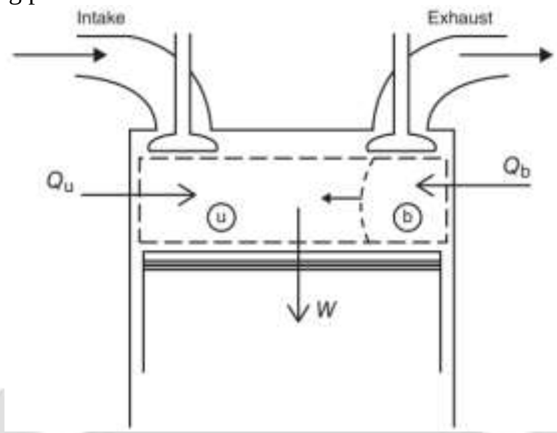

Fig -2: Two-zone combustion method [8]

The variations of the energy contained in each of these two zones are evaluated by Equations (5) and (6) [9].

Unburned Zone:
$$\frac{d(m_u e_u)}{dt} = -p\frac{dV_u}{dt} - Q_u + \left(\frac{dm_f}{dt}h_f + \frac{dm_a}{dt}h_a\right) + \frac{dm_{f,i}}{dt}h_{f,i} \qquad (5)$$

Burned Zone:
$$\frac{d(m_b e_b)}{dt} = -p\frac{dV_b}{dt} - Q_b - \left(\frac{dm_f}{dt}h_f + \frac{dm_a}{dt}h_a\right) \qquad (6)$$

The combustion rate, i.e. the evolution of the transfer from the unburned zone to the burned zone, is given by the Wiebe model (7), (8), (9), (10), (11), (12) which predicts the combustion rate in spark-ignition engines in the absence of experimental measurements of the pressure in the cylinder [11] [12].

Cumulative Burn Rate :
$$Combustion\ (\theta) = (CE)[1 - e^{-(WC)(\theta - SOC)^{E+1}}] \qquad (7)$$

Wiebe Constant
$$\mathrm{WC} = \left[\frac{D}{BEC^{\frac{1}{E+1}} - BSC^{\frac{1}{E+1}}}\right]^{-(E+1)} \qquad (8)$$

Start of Combustion
$$SOC = AA - \frac{D(BMC)^{\frac{1}{E+1}}}{BEC^{\frac{1}{E+1}} - BSC^{\frac{1}{E+1}}} \qquad (9)$$

Burned Midpoint Constant
$$BMC = -\ln(1 - BM) \qquad (10)$$

Burned Start Constant
$$BSC = -\ln(1 - BS) \qquad (11)$$

Burned End Constant
$$BEC = -\ln(1 - BE) \qquad (12)$$

2.1.3 Modeling of NO_x formation

The model used for the calculation of NO_x formation during combustion is based on the extended Zeldovich mechanism, represented by the three chemical reactions (13), (14), and (15) [13].

$$O + N_2 = NO + N \qquad (13)$$

$$N + O_2 = NO + O \qquad (14)$$

$$N + OH = NO + H \qquad (15)$$

The evolution of the concentration of NO and N is then given by Equations (16) and (17), where the values of the rate constants k_1^+, k_2^+, $k3^+$ according to the temperature T in the cylinder are given in **Table -1** [13]

$$\frac{d[NO]}{dt} = k_1^+[O][N_2] + k_2^+[O][O_2] + k_3^+[N][OH] \qquad (16)$$

$$\frac{d[N]}{dt} = k_1^+[O][N_2] - k_2^+[O][O_2] - k_3^+[N][OH] \tag{17}$$

Table -1: Rate constants for NOx formation mechanism [13]

Reaction	Rate constant, cm^3/mol.s	Temperature range, K
(1) $O + N_2 \rightarrow NO + N$	$7.6 \times 10^{13} \exp[-38000/T]$	2000 - 5000
(-1) $N + NO \rightarrow N_2 + O$	1.6×10^{13}	300 - 5000
(2) $N + O_2 \rightarrow NO + O$	$6.4 \times 10^{9}\, T \exp[-3150/T]$	300 – 3000
(-2) $O + NO \rightarrow O_2 + N$	$1.5 \times 10^{9}\, T \exp[-19500/T]$	1000 – 3000
(3) $N + OH \rightarrow NO + H$	4.1×10^{13}	300 - 2500
(-3) $H + NO \rightarrow OH + N$	$2.0 \times 10^{14} \exp[-23650/T]$	2200 - 4500

2.1.4 Modeling of CO formation

For the formation of CO, the calculation is based on the main oxidation reaction of CO represented by Equation (18) [13].

$$CO + OH = CO_2 + H \tag{18}$$

The relationship between the value of the rate constant k_{CO}^+ and the temperature T in the cylinder is given by Equation (19) [13].

$$k_{CO}^+ = 6.76 \times 10^{10} \exp\left(\frac{T}{1102}\right)\ cm^3/gmol \tag{19}$$

2.2 Performance parameters

Besides the NO_x and CO emissions evaluated by the models described in paragraphs **2.1.3** and **2.1.4**, the brake power (*bkw*), the brake specific fuel consumption (*bsfc*), the combustion efficiency (η_{comb}), and the engine efficiency (b_{eff}) are also indicators of the engine performance.

The brake power is the mechanical power received by the alternator shaft which is then converted into electrical power. It corresponds to the power obtained by combustion of the fuel mixture minus the power used for pumping, i.e. the power relative to the intake and exhaust phases, and the power lost in the form of friction (Equations (20), (21)) [9].

$$bkw = \frac{\oint T_b(t)Ndt}{\oint dt}\left[\frac{2\pi}{60000}\right] \tag{20}$$

$$T_b(t) = T_s(t) - I_{ct}\dot{\omega}_{ct}(t) \tag{21}$$

The specific fuel consumption (*bsfc*) is the ratio between the flow rate of fuel consumed ($\dot{m}_f$) to the power supplied by the engine (*bkw*) (Equation (22)) [9].

$$bsfc = \frac{\dot{m}_f}{bkw} \tag{22}$$

The combustion efficiency is given by Equation (23) where the combustion loss ($Loss_{comb}$) is the ratio of the chemical energy contained in the exhaust gas to the total energy of the fuel entering the cylinder [9].

$$\eta comb = 1 - Loss_{comb} \tag{23}$$

Finally, the engine efficiency (*beff*) is the ratio of the brake power (*bkw*) to the power of the fuel entering the cylinder (*engfen*) (Equations (24), (25)) [9].

$$b_{eff} = 400\frac{bkw}{engfen} \tag{24}$$

$$engfen = \left[\frac{avgrpm}{60000 n_r}\right]\sum_{i=1}^{\#Cylinders}[LHV_f \oint \dot{m}_{f,i,gas}dt + (LHV_f - HVAP_f)\oint \dot{m}_{f,i,liq}dt \tag{25}$$

2.3 Modeling on GT-Power of the spark-ignition engine fuelled with HHO-enriched syngas

To study the influences of HHO enrichment of syngas, a 4-stroke, single-cylinder spark-ignition engine (cf. **Fig -3**) was modeled on GT-Power.

Fig -3: The single-cylinder spark-ignition engine studied

The engine characteristics given by the manufacturer or measured are shown in **Table -2**.

Table -2: Engine characteristics

Characteristics	Specifications
Cycle	4
Number of cylinders	1
Bore	67.97 mm
Stroke	53.54 mm
Connecting rod length	84 mm
Compression ratio	11.4
Intake manifold length	138 mm
Intake manifold inner diameter	19 mm
Exhaust manifold length	122mm
Exhaust manifold inner diameter	25 mm
Inlet valve diameter	24.98 mm
Exhaust valve diameter	24.02 mm

As the engine will be used to run a 2-pole (po =1) alternator which must provide an alternating voltage of f = 50 Hz, the shaft speed must be 3000 rpm according to Equation (26).

$$N = \frac{60\,f}{po} \tag{26}$$

Several assumptions were made in the settings of the different elements of the model. For the external environment, an atmosphere formed of air with 76.7% nitrogen and 23.3% oxygen, with a pressure of 1 bar and a temperature of 300 K was assumed. Concerning the engine speed, the throttle regulating the airflow to be mixed with the fuel is always considered wide open.

For the combustion model, the values of the parameters characterizing the Wiebe model are given in **Table -3** [9]

Table -3: Wiebe Model inputs [9]

Inputs	Value
Anchor Angle (AA)	8°
Combustion Duration (D)	25°
Wiebe Exponent (E)	2
Fraction of Fuel Burned (CE)	1
Burned Fuel Percentage at Anchor Angle (BM)	50%
Burned Fuel Percentage at Duration Start (BS)	10%
Burned Fuel Percentage at Duration End (BE)	90%

Syngas used to fuel the engine is a purified gas from forest residue, composed of 15% CO, 17% H_2, 4% CH_4, 15% CO_2, 0.14% O_2, 48.86% N_2, and has a higher heating value of 6 MJ/Nm^3 [14].
Finally, the model of the engine on GT-Power is shown in **Fig -4**.

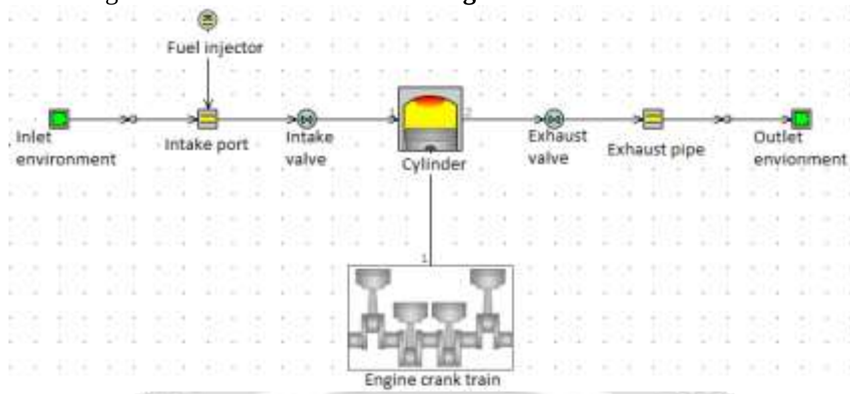

Fig -4: GT-Power model of the engine

3. RESULTS

3.1 Stoichiometric ratio of syngas

Before evaluating the improvements provided by the addition of HHO in syngas, it is first necessary to determine the stoichiometric ratio of pure syngas. This ratio corresponds to the ideal proportion of air and syngas to be mixed to gain maximum energy from the combustion.
To determine this ratio, the power and efficiency of the engine fuelled by syngas are calculated according to the variation of the Air-Fuel ratio. Note that in this article, the term "fuel" refers to the combustible portion of the fuel mixture; it can be gasoline, syngas, HHO, or a mixture of these.
Fig -5 shows the power and efficiency of the engine versus the Air-Fuel ratio. The power is maximum for an Air-Fuel ratio equal to 4, i.e. to recover the maximum energy, 4 volumes of air will be needed for one volume of syngas. Beyond this value, the power decreases because of the lean mixture. It is also noted that the engine efficiency is the highest in the surroundings of this Air-Fuel ratio equal to 4 and this value will therefore be retained for the comparison between the use of gasoline and that of the syngas.

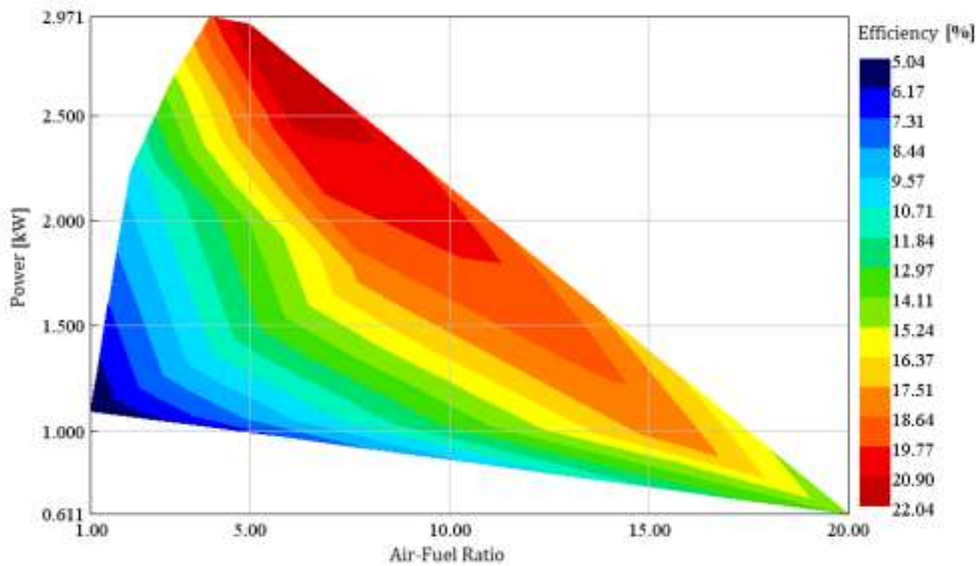

Fig -5: Power and efficiency of the syngas fuelled engine versus Air-Fuel ratio

3.2 Effects of syngas enrichment with HHO gas

To improve the performance of the engine running on syngas, the latter is mixed with hydrogen (HHO) which has a heating value of 33 kWh/kg [15].

Hydrogen is produced by electrolysis of water, which is an electrochemical process consisting of decomposing water with an electric current to obtain hydrogen and oxygen gas. According to Avogadro's law, the resulting gas contains twice as much hydrogen as oxygen, which is why it is called HHO. The addition of HHO in the syngas, to fuel the engine, thus acts both on the fuel by the enrichment of hydrogen and on the oxidizer by the supply of oxygen [16]. In this study, the mixture of syngas and HHO injected into the cylinder of the engine is assumed to be homogeneous.

Fig -6 shows the variation of the power and efficiency of the engine versus the HHO content in the mixture of syngas + HHO. The addition of HHO increases the power delivered by the engine but beyond the 10% content, corresponding to the maximum power (9.56% increase), the power decreases, and from 35% HHO, it even goes below that delivered with pure syngas.

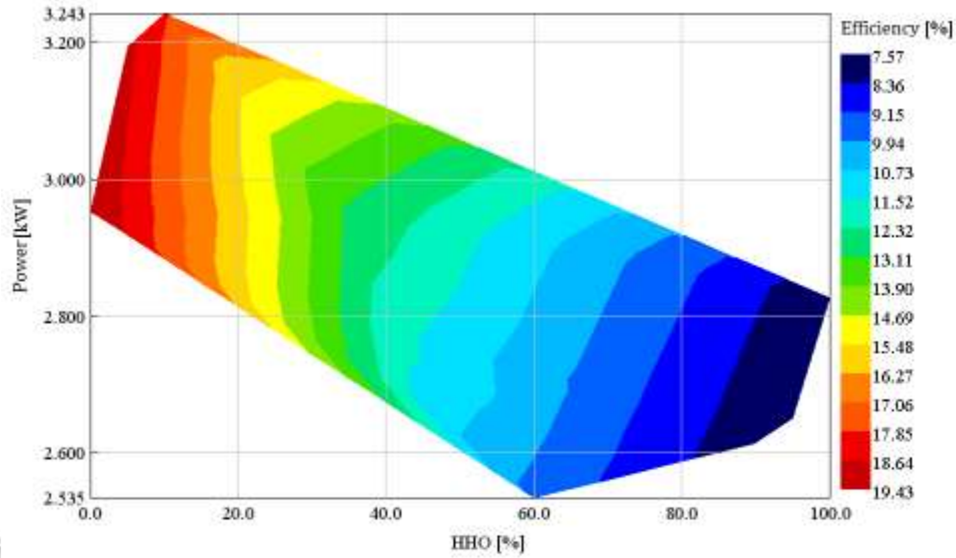

Fig -6: Engine power and efficiency versus HHO content in the syngas + HHO mixture

Since the excess energy provided by HHO is not completely recovered when the mixture is burned, engine efficiency decreases with the addition of HHO. This combustion loss is illustrated by the combustion efficiency curve in **Fig -7** where the fraction remaining unburned increases with the HHO content. The addition of HHO in the syngas, therefore, increases the energy entering the cylinder by up to 11.96% with 10% HHO. But with the decrease of the burnt fraction, the energy exploited decreases. Thanks to its energy input, the enrichment of the syngas with HHO also reduces fuel consumption. For a 10% addition of HHO, a 10.76% reduction in fuel consumption is achieved.

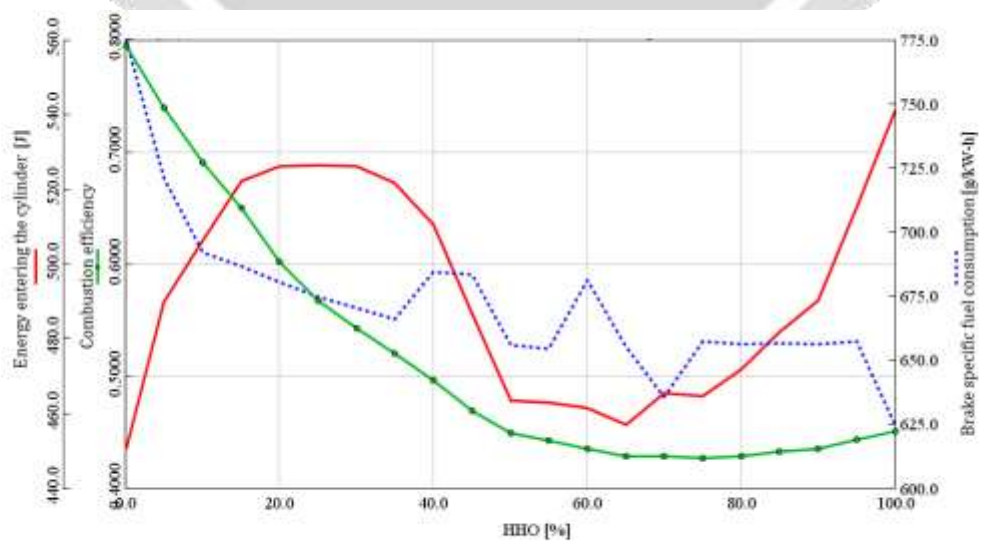

Fig -7: Energy, combustion efficiency, and fuel consumption versus HHO content in the syngas + HHO mixture

Concerning pollutant emissions, **Fig -8** shows that the addition of HHO in syngas reduces the concentration of NO_x in the exhaust gas. Indeed, increasing the HHO content reduces the amount of nitrogen in the syngas and thus inhibits the formation of NO_x. At 10% HHO, the NO_x concentration is reduced by 89.95%. The CO concentration increases slightly to 9.43% at 10% HHO and then decreases as the mixture contains more HHO. Indeed, on the one hand, the addition of HHO increases the temperature in the combustion chamber and thus favors the formation of CO, but on the other hand, the quantity of CO contained in the syngas decreases in favor of HHO.

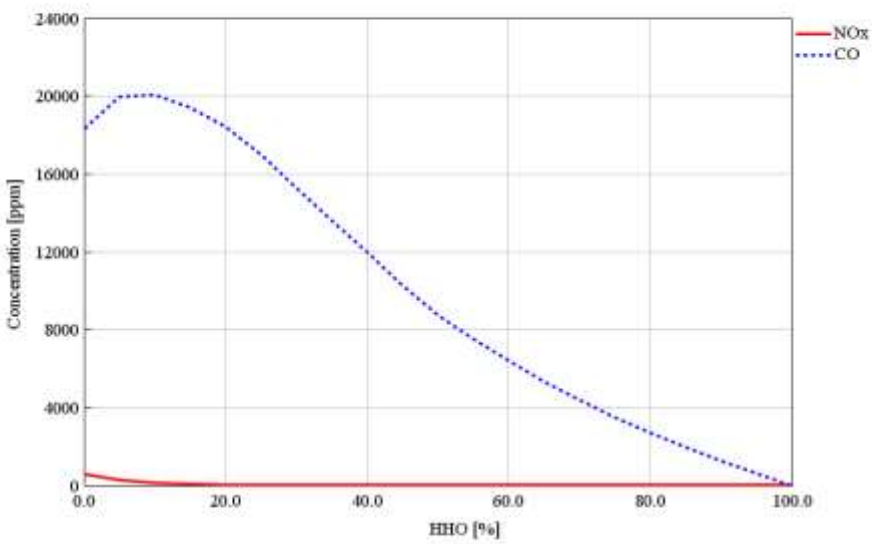

Fig -8: NO_x and CO emissions versus HHO content in the syngas + HHO mixture

3.3 Stoichiometric ratio of the syngas + HHO mixture

Since HHO gas mixed with syngas contains both hydrogen and oxygen, its addition to the syngas may alter the stoichiometric ratio of the mixture. If the hydrogen increases the energy of the fuel, the oxygen increases the amount of oxidant. The stoichiometric ratio of the mixture may therefore be different from that of pure syngas.
In **Fig -9**, it is shown that the power and efficiency of the engine are maximum for an Air-Fuel ratio of 5. For this ratio, the power is increased by 7.64% compared to an A/C of 4, and the efficiency is improved by 21.84%. The A/C ratio of 5 allows better combustion of the fuel consisting of 10% HHO and 90% syngas and represents the stoichiometric ratio of this mixture.

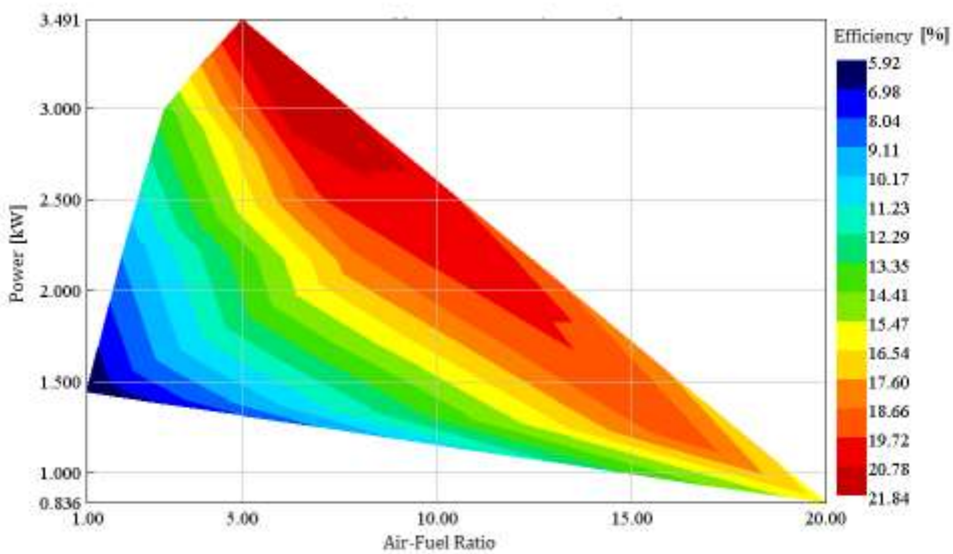

Fig -9: Power and efficiency of the engine fuelled with syngas + HHO versus Air-Fuel ratio

Fig -10 shows the variation in energy, combustion efficiency, and fuel consumption according to the values of the Air-Fuel ratio. The energy entering the cylinder decreases when the fuel mixture contains more air because the proportion of fuel is reduced in favor of the oxidizer. But below the Air-Fuel ratio of 5, the combustion efficiency is well below 1 because there is not enough fuel to burn all the mixture. However, the right proportion of air and fuel allows a reduction in fuel consumption and at stoichiometry (A/C=5) a fuel saving of 17.89% is achieved.

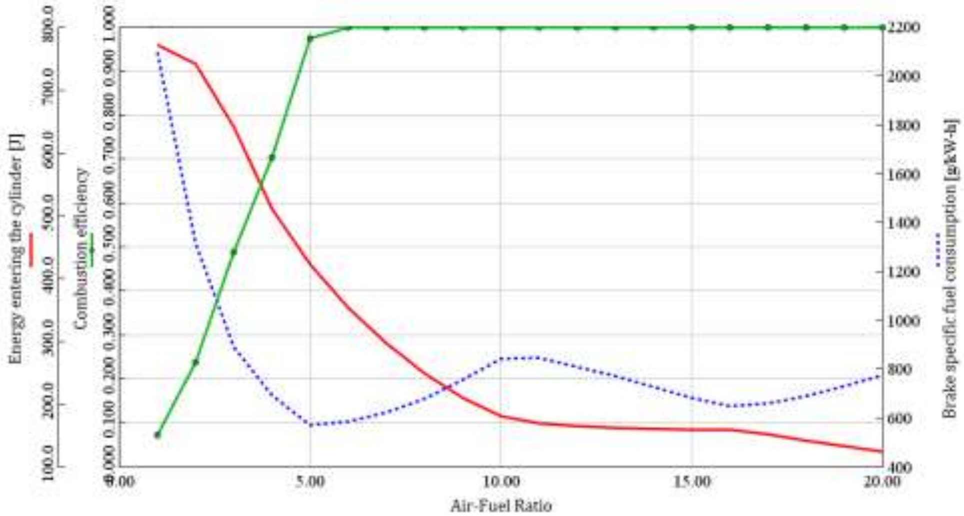

Fig -10: Energy, Combustion Efficiency and Fuel Consumption versus Air-Fuel ratio

Concerning pollutant emissions, finding the stoichiometric ratio of the mixture allows a considerable reduction in CO emissions because, as can be seen in **Fig -11**, mixing 5 volumes of air for one volume of fuel allows a 9-fold reduction in CO concentration compared to that obtained with an Air-Fuel ratio of 4. For NO_x emissions, the concentration is 16.56 times higher, but this value remains lower compared to the pollution caused by the use of gasoline or pure syngas.

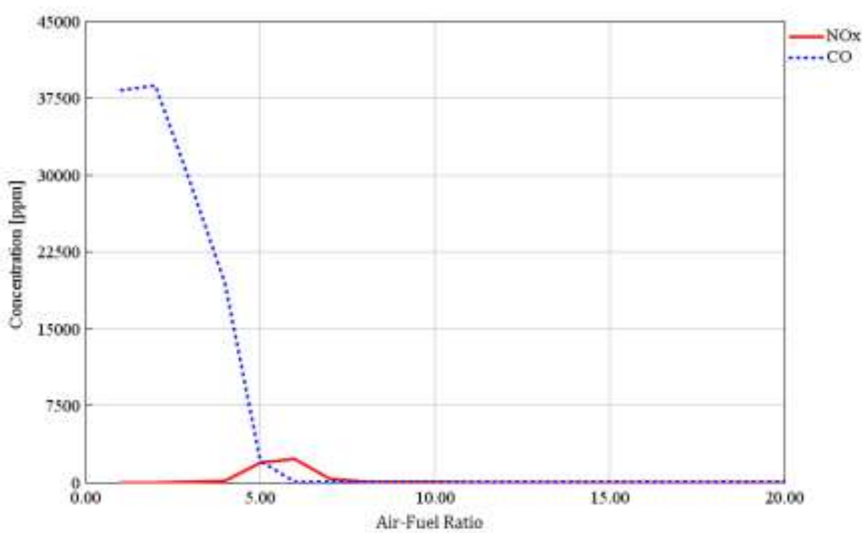

Fig -11: NO_x and CO emissions versus Air-Fuel ratio

4. CONCLUSIONS

The performance of a spark-ignition engine fuelled with HHO-enriched syngas has been studied in this article. The engine was modeled and simulated on the GT-Power tool of the GT-Suite software. The syngas is obtained by gasification of forest residue and the HHO gas by electrolysis of water.

The best performance of the engine is obtained for a mixture of 90% syngas and 10% HHO, with an Air-Fuel ratio equal to 5. This mixture allows the engine to generate 17.94% more power than the starting mixture (100% syngas and an Air-Fuel ratio of 4). Efficiency is also improved by 12.4% and the engine consumes 26.67% less fuel. Compared to gasoline fuelled engines, the use of hydrogen-enriched syngas also reduces pollutant emissions. NOx emissions are thus reduced by 53.11% while CO emissions are cut by 70.69%.

However, the values found in this study may vary according to the fluctuation of the syngas components depending on the quality of the biomass used and the gasification parameters. Similarly, the optimal proportion of HHO to be mixed with the syngas as well as the volume of air required may also change depending on the type and characteristics of the engine used. From the perspective of implementing the HHO enrichment system in biomass power plants, this study shows the importance of simulation in determining the optimal amount of HHO to be produced and thus in properly sizing the hydrogen generator.

The results of the simulations demonstrated the advantages of HHO enrichment of the syngas. This study could therefore help reduce the dependence of rural communities on fossil fuels and thus contribute to their sustainable development.

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