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Characterization of Auto-Ignition Phenomena in Spark Ignition Internal Combustion Engine using Gaseous Fuels Obtained from Biomass

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Studies have been carried out on the phenomenon of auto-ignition in liquid fuels and natural gas, but research on the application of gaseous fuels obtained from biomass is limited. Existing investigations about auto-ignition mainly focused on the combustion kinetics to determine the delay time, but not on the prediction of the set of parameters that encourage the presence of the phenomenon. In the present research, a thermodynamic model is developed for the prediction of the auto-ignition in Spark Ignition Internal Combustion Engine operated with gaseous fuels, which are obtained from the process of gasification of biomass. The formulated model can handle variable compositions of gaseous fuels and to optimize the main operational parameters of the engine, to verify its influence on the phenomenon under study. Results show the application of this type of alternative fuels in commercial engines that operated with natural gas, varying engine operational parameters while maximizing the power output of the engine.

1. Introduction

In the global context, due to the shortage of natural gas and its high cost, some countries have been forced to look for new forms of gaseous fuels. These new fuels must be not only economically attractive but also friendly to the environment and easy to implement in internal combustion engines, where the fuel they are currently using is natural gas (Amran, Ahmad, & Othman, 2017). Similarly, in recent years, important research has been carried out on the impact of the use of alternative gaseous fuels in the performance of internal combustion engines. The use of gaseous fuels in combustion internal combustion engines has been developed a lot, but there is a limitation of the increase in efficiency as a function of the increase in the compression ratio (CR). The above is because the CR increases the autoignition phenomenon, which occurs when combustion starts before ignition of the spark, which leads to the high-pressure peaks and a shock wave is generated inside the combustion chamber (Gersen, Darmeveil, & Levinsky, 2012).
The current studies on the phenomenon of autoignition in internal combustion engines operated with gaseous fuels produced by gasification (producer gas or syngas) are limited and are focused on the engines of natural aspiration and low power (< 25 kW) (Bika, Franklin, & Kittelson, 2012; P Boivin, Sánchez, & Williams, 2017; Mittal, Sung, & Yetter, 2006). Several of these studies (Pierre Boivin, Jiménez, Sánchez, & Williams, 2011; Yu, Vanhove, Griffiths, De Ferrières, & Pauwels, 2013) concluded that the phenomenon of autoignition is not accentuated in naturally aspirated engines, and that it is very much harmful to supercharged engines of high CR (Duarte et al., 2014).
Azimov et al. (2011) concerned with engine experiments and spectroscopic analysis of premixed mixture ignition in the end-gas region combustion in a pilot fuel ignited the natural gas dual-fuel engine. The results reveal the characteristics and operating parameters that induce and affect this combustion mode. In combustion, the heat release gradually transforms from the slower first-stage flame rate to the faster second-

stage rate. For the characterized H_2 / CO combustion and reaction chemistry under high-pressure and moderate-temperature operating conditions, Ihme (2012) analyzes several chemical-kinetics, and hydrodynamic processes have been identified as being responsible for the discrepancies between experimental measurements and kinetic predictions of syngas ignition delay times. The above allows characterizing the auto-ignition in motors operated with syngas.

Pal et al. (2015) analyses the reaction front speed and front Damköhler number, were employed to characterize the propagating ignition front. The ignition behaviour shifted from spontaneous propagation (strong) to deflagrative (weak), as the initial mean temperature of the reactant mixture was lowered. The conclusion is that sufficiently low temperatures, the strong ignition regime was recovered due to the faster passive scalar dissipation of the imposed thermal fluctuations relative to the reaction timescale, which was quantified by the mixing Damköhler number. On the other hand, Przybyla et al. (2016) summarize results of experimental trials with the SI and HCCI engine fueling with a producer gas substitute. Engines were fueled with the producer gas substitute that simulated real producer gas composition. The SI engine was charged with a stoichiometric air/fuel mixture, while the HCCI engine with a lean air/fuel mixture with an equivalent ratio of 0.5 and the main variable control in the SI engine operation was the spark timing.

Amador et al. (2017) explored the feasibility of using Syngas with low methane number as fuel for commercial turbocharged internal combustion engines. The effect of methane number (MN), compression ratio (CR), and intake pressure on auto-ignition tendency in spark ignition internal combustion engines was determined. An error function, which combines the Livengood–Wu with ignition delay time correlation, to estimate the knock occurrence crank angle (KOCA) was proposed. The results showed that the KOCA decreases significantly as the MN increases. Results also showed that Syngas obtained from coal gasification is not a suitable fuel for engines because auto-ignition takes place near the beginning of the combustion phase, but it could be used in internal combustion engines with reactivity controlled compression ignition (RCCI) technology.

In the present research, a thermodynamic equilibrium model is proposed, which allows predicting the point where auto-ignition will occur in the compression stroke as a function of the geometric and operational parameters of the engine. This model has been validated in turbocharged engines, which allows verifying the viability of the use of syngas in commercial engines used for the generation of electric power.

2. Methodology

In the present research, the following methodological steps were developed, to achieve significant results on the study of auto-ignition applied to spark ignition engines operated with gaseous fuels. It began with a review of state of the art, focusing on the latest studies on auto-ignition in thermal engines and characterization of biomass product of gasification. Then, a thermodynamic model was implemented to predict the temperature in the combustion chamber before the ignition point, which applies to turbocharged engines. Validation was carried out using natural gas as fuel, to extrapolate the model to other types of gaseous fuels. Based on the state of the art review, the typical auto-ignition temperature of different gaseous fuels obtained by gasification was identified. The application of the developed model leads to a predicted temperature inside the combustion chamber at the ignition point, which is compared with the auto-ignition temperature of each fuel analyzed. As the ignition point is a function of the spark advance, with the proposed methodology, the ignition angle can be optimized so that ignition does not occur. The iterative process, in turn, allows maximizing the output power, since according to de Faria (de Faria, Bueno, Ayad, & Belchior, 2017), the ignition point has a direct relationship with the power developed by the engine. Finally, in the study the influence of the compression ratio (CR) and intake pressure on auto-ignition is verified, to find the set of operational parameters at which a certain syngas can operate in a commercial engine.

3. Proposed Model

Because the engine under study is of stationary application (generation of electrical energy), speed engine does not vary in its nominal operating condition. The above implies that it is possible to assume a quasi-stationary state and describe it as a polytropic process. The relationship between pressure (P), volume (V) and polytropic coefficient (n) is described by:

$$\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = \frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{(n-1)} \qquad (1)$$

It will be assumed that the amount of heat released by the burned gases stored in the volume defined by the top dead center in the combustion chamber, which will be at the temperature of the exhaust gases, is received by the air/fuel mixture, which occurs at high engine speed (Duarte et al., 2014). For the calculation of thermodynamic properties, correlations of gas mixtures as a function of temperature are proposed, so that the property can be adjusted for each temperature increase (with a step of 1 ° C). Figure 1 shows the scheme of the system to be analyzed and its intermediate points:

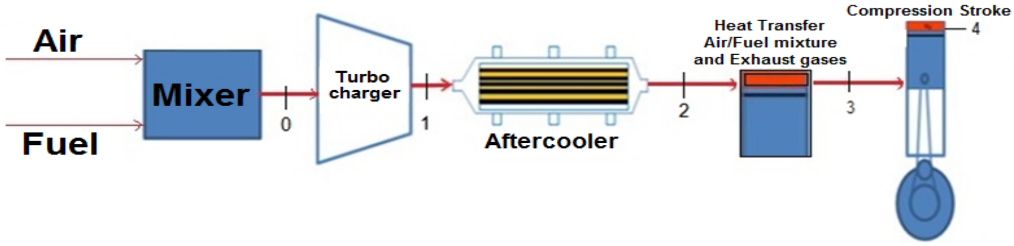

Figure 1: Stages of analysis in the engine under study

As the system to be analyzed is in series from the mixer, the effect of each of the elements involved in the system must be coupled. Starting at 0 and ending at 4. 0 - 1 is the process in the Turbocharger, 1 – 2 aftercooler, 2 – 3 Heat Transfer Air/Fuel mixture and Exhaust gases and 3 – 4 Compression stroke. The equation (2) is obtained for the estimation of the temperature at the ignition point through an analysis of equilibrium thermodynamics applying equation (1) and the volume adjustment model of the combustion chamber in internal combustion engines developed by Duarte (2016).

$$T_4 = \frac{\dfrac{\dfrac{\rho_{EG} \cdot Cp_{EG} \cdot T_{EG}}{\rho_M \cdot Cp_M \cdot (C_R - 1)} + \left[T_E \cdot \left(\dfrac{P_1}{P_0}\right)^{\frac{n_{01}-1}{n_{01}}} - \Delta T_{IC}\right]}{1 + \dfrac{\rho_{EG} \cdot Cp_{EG}}{\rho_M \cdot Cp_M \cdot (C_R - 1)}}}{\left\{V_c \cdot \left[1 + \left(\dfrac{C_R - 1}{2}\right) \cdot \left(R + 1 - cos\theta - (R^2 - sen^2\theta)^{\frac{1}{2}}\right)\right] + \left(\dfrac{\pi \cdot D^2}{4}\right)\left[\dfrac{K_{DEF} \cdot L_{CR} \cdot (P_4 \cdot A_P + m_i \cdot a)}{E_{STEEL} \cdot A_{CR}}\right]\right\}^{n_{34}-1}} \quad (2)$$

The terms of Eq. 4 are specified in Table 1.

Table 1: Definition of terms Ec. (2)

Terms	Description
T_4	Temperature at the ignition point [K]
Cp_{EG}	Heat capacity of exhaust gases[kJ/kg K]
Cp_M	Heat capacity of the air/fuel mixture, which is a function of the composition of the gaseous fuel [kJ/kg K]
ρ_{EG}	Density of exhaust gases [kg/m^3]
ρ_M	Density of the air/fuel mixture, which is a function of the composition of the gaseous fuel [kg/m^3]
T_{EG}	Exhaust gas temperature [K]
T_E	Environmental temperature [K]
P_i	Pressure at point i [Pa]
C_R	Compression ratio
n_i	Polytropic coefficient at point i
ΔT_{IC}	Temperature drop in the intercooler [K]
V_c	Clearance Volume [m^3]
R	Relation between connecting rod length and crank radius
θ	Crankshaft angle [°]
K_{DEF}	Coefficient of deformation, adjusted for each engine and taking into account that not all parts of the mechanism are made of steel
L_{CR}	Length of the connecting rod [m]
A_P	Piston area [m^2]
m_i	mass Connecting rod, piston and pin [kg]
a	Acceleration of connecting rod, piston, and pin [m/s^2]
E_{STEEL}	Young's module of steel [Pa]
A_{CR}	Cross section area of the connecting rod [m^2]

Eq. (4) take into account the mechanical deformations due to pressure and inertia in the connecting rod-piston mechanism. These deformations are manifested through variations in the instantaneous volume, especially in the vicinity of TDC, which affects the prediction of the actual temperature inside the combustion chamber. P_4 is a function of T_4, so Eq. 4 is an implicit expression, so an algorithm was designed to find the solution.

4. Results Analysis

For the validation of the proposed model, the intake temperature and the compression ratio were varied. The characteristics of the engine used for the simulation are the following:

Table 2: Characteristics of the engine used in the study

Characteristic	Description
Manufacturer	Yamaha
Model	EF2400
Displacement [cc]	2400 cc
Test Power [kW]	2.4
Engine speed [rpm]	3600
Compression ratio range	8 - 10
Turbocharger discharge pressure range [bar]	1 – 1.2
Intake Temperature (held fixed for the study) [°C]	35
A/F Ratio (held fixed for the study)	15
Ignition angle (held fixed for the study) [°BTDC]	18
K_{DEF} (adjusted with motored test)	1.2

The adjusted value of the polytropic coefficients of the engine used in the study n_{01} and n_{34} were 1.382 and 1.365, respectively. Initially, the performance of the model was verified using natural gas as fuel, then the behavior of the model with wood gas is validated. According to Malenshek et al. (2009) and Amador et al. (2017), the characteristics of the fuels and their respective methane numbers (MN) to be modeled are the following.

Table 3: Characteristics of gaseous fuel used in the study

Components [%]	Natural gas (MN = 61.3)	Wood gas (MN = 61.3)
CH_4	97.3	8.3
H_2	0.0	39.8
N_2	0.3	2.7
CO	0.0	24.2
CO_2	0.0	24.1
C_2H_6	2.6	0.0
C_3H_8	0.2	0.0
C_4H_{10}	0.1	0.0

The autoignition temperature is a characteristic of the fuel described in Table 2. Figure 2 shows the behavior of the model using natural gas as a fuel and the comparison with the autoignition temperature defined by Amador et al. (2014).
Figure 3 shows the behavior of the model using Wood gas as a fuel and the comparison with the auto-ignition temperature defined by Malenshek et al. (2014).

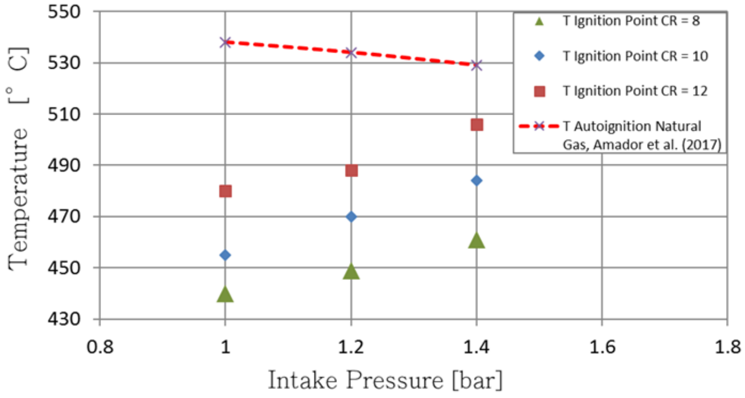

Figure 2: Comparative modeled temperature at the ignition point and Auto-ignition temperature of Natural Gas

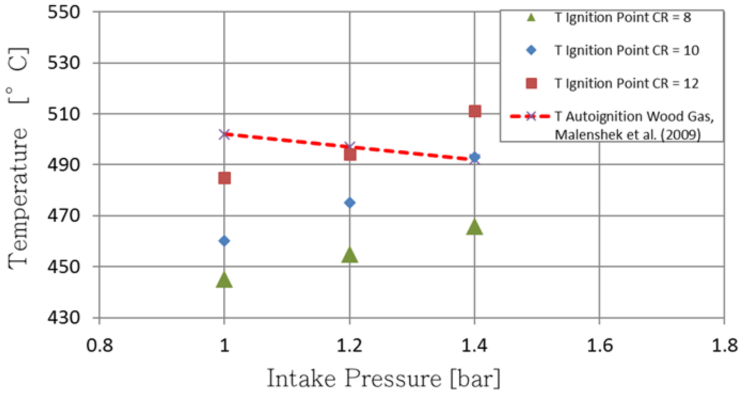

Figure 3: Comparative modeled temperature at the ignition point and Auto-ignition temperature of Wood Gas

For the results of the model illustrated in Figure 2 and 3, the same engine speed was used, taking into account that the engine under study is used for the generation of electrical energy, where the engine speed must remain constant to maintain utility frequency. In both figures it is observed that the temperature at the ignition point increases proportionally with the increase of the intake pressure. Air/Fuel ratio and intake temperature were kept constant in the study, so as not to influence the results of the prediction of the temperature at the ignition point and adjust to the auto-ignition temperature values defined by Malenshek et al. (2009) and Amador et al. (2017).

5. Conclusions

Figure 2 and 3 show the influence of the intake pressure and the compression ratio on the temperature at the ignition point. As the intake pressure increases, the temperature at the ignition point increases. The same behavior is observed with the compression ratio (CR), which is consistent with the results obtained in previous works (Duarte et al., 2014, Amador et al., 2017). For the engine under study, for the range of CR of 8 to 12 and Intake Pressure of 1 to 1.4 bar, it can be seen in Figure 2 the absence of auto-ignition in the compression stroke when operating with natural gas, since the auto-ignition temperature is not reached.
In the case of Wood Gas, Figure 3 shows that the engine under study can be operated up to an intake pressure of 1.2 bar and a compression ratio of 12. In the case of Intake Pressure of 1.4 bar, auto-ignition is presented, except for CR = 8. At the point of operation with CR = 10 and CR = 12 and Intake Pressure of 1.4, it is evident that the temperature at the ignition point is above the auto-ignition. It is necessary, as a strategy for mitigating this harmful phenomenon, delay the angle of ignition (increase) or change A/F Ratio so that the auto-ignition temperature of the Wood Gas increases.

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