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STUDY OF THE INFLUENCE OF INJECTION TIMING AND DIRECTION IN THE PERFORMANCE OF A SPARK IGNITION ENGINE FUELED WITH GASOLINE-ETHANOL

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Abstract. One of the main energy sources in the transport sector in recent decades due to its high performance and reliability, internal combustion reciprocating engines have been facing international obligations related to pollutant emissions and operations with renewable and more efficient fuels. Ethanol was one of the biofuels that received much attention due to its physicochemical properties and ability to be a partial or complete substitute for gasoline. One of the areas of analysis to obtain better thermodynamic and emission performance is fuel injection. Several configuration possibilities can be made, depending on the system configuration, which can provide great variability in thermodynamic behavior, in the optical behavior of the flame and in pollutant emissions. With this, it is essential to better understand the behavior and spatial propagation of the flame, for the development of more efficient engines, minimizing modifications in the engine design, less emissions and allowing to fully exploit the benefits of biofuels supply. The present work aims to study the effect of the injector position and its possible configurations, providing analysis of the best configuration that allows maximum efficiency and better operating conditions. To carry out the tests, an internal combustion reciprocating engine AVL 5406 with spark ignition with optical access was used. A high-speed PCO.Dimax S1 camera coupled with a VS4-1845HS Scope dual intensifier and 105 f/4.5 UV-Nikon lens was used to obtain the optical data. For the engine and injector used, the configurations allowed analysis of the behavior of the spray with a jet of fuel directed towards the intake valve or the spark plug. In addition, the best angle for the start of injection was also analyzed, observing the influence of the residence time of the fuel inside the cylinder. For the spray directed to the intake valve, condensation of the fuel was observed, which provided inefficient combustion. As for the spray directed to the spark plug, the possibility of greater wear of the spark plug due to direct contact with the fuel and subsequent carbonization, especially at the beginning of the operation, stands out. Also noteworthy is the residence time inside the combustion chamber. The longer residence time for ethanol blends contributes to a better air-fuel mixture.

Keywords: gasoline-ethanol blend, combustion, optical engine, injection timing, direct injection

1. INTRODUCTION

The current environmental scenario added to the moving global policies highlight and demand the shift from fossil fuel consumption to alternative fuels (IPCC, 2020). The transportation sector, which mainly uses fossil fuels such as diesel and gasoline, is a major contributor to greenhouse gases, air pollution. From this, several biofuels emerged with the intention of meeting this market need. One of the biofuels that received much attention due to its physicochemical properties and ability to be a partial or total substitute for gasoline was ethanol (Awad, 2018). This fuel is widely used in countries like Brazil, USA and China (Turns, 2013; Heywood, 2018; Assad, 2011). Ethanol is also an excellent alternative to liquid fuel for hybrid vehicles in the search for a reduction in the use of petroleum-based fuels and also for lower pollutant emissions.

To make the operation with better thermodynamic and emission performances, the injection of fuel inside the combustion chamber is one of the important areas of analysis (S. Gopinath et. al, 2020 and Varde, 2009). The spray development process within the combustion chamber plays a significant role in the quality of combustion in internal

combustion reciprocating engines. The contact of the injected fuel with the surfaces present inside the chamber can provide condensation of the atomized drops which, in addition to generating a less efficient burning, can cause carbonization in the various parts causing a shorter life of the equipment (Heywood, 2018 and S. Gopinath et. al, 2020).

As the penetration of the fuel spray inside the chamber depends on the properties of the fuel mixture, the analysis of tests using biofuels either as a complete or partial substitute is essential (Dunn-Rankin, 2016, Augoye, 2014 and S. Gopinath et. al, 2020).

With the various current models in engines, different configuration possibilities can be made, depending on the system configuration, which, despite being subtle, can provide great variability in thermodynamic behavior, in the optical behavior of the flame and also in the emissions of polluting gases (Merola, 2010, Catapano, 2013 and Desantes, 2015).

With this, it is essential to better understand the behavior of the injected fuel spray, how the engine characteristics influence the distribution of fuel within the combustion chamber and how the injector configuration used can influence the performance of burning and flame propagation. This knowledge becomes fundamental for the development of more efficient engines, minimizing changes in the engine design, lower emission of polluting gases and the possibility of making the most of the benefits of using biofuels.

The use of gasoline-alcohol mixtures, associating the advantages of burning each fuel, has attracted the attention of researchers in several countries, due to the possibility of operating in several internal combustion reciprocating engines with little or no modification. In addition, the reduction of pollutant emissions, reduction of fuel cost and good performance of the mixture in several properties collaborate for its use. Among the properties that are positively affected by the addition of alcohol to gasoline, such as the viscosity of the mixture, calorific value, stability, corrosive characteristics, octane rating, among others.

Thus, the present work aims to study the effect of the injector position and its possible configurations such as fuel spraying direction, injection departure angle and fuel residence time inside the cylinder, thus providing analysis of the best configuration that allows maximum efficiency and better operating conditions using a gasoline-ethanol mixture in the proportion of 73% gasoline and 27% ethanol, commonly used commercially in Brazil.

2. METHODOLOGY AND PROCEDURES

The apparatus used to carry out the tests includes an AVL 5406 internal combustion reciprocating engine with spark ignition with optical access, active dynamometer, acquisition system and control unit. The engine has 4 valves, 2 inlet and 2 exhaust valves, total volume of 530 cm^3, crankshaft 144 mm, engine stroke 90 mm and piston diameter 82 mm.

The combustion chamber allows two-way optical access to the interior. The quartz cylinder wall admits optical access with a side view of the combustion. In addition to this, the 72 mm diameter fused silica window attached to the piston crown allows access and viewing of the chamber from below (bottom view). With the aid of an appropriate UV mirror, inclined at 45° fixed to the bottom of the elongated piston and the radiation produced by the combustion inside the chamber or the aid of LEDs, allow the recording of images by the acquisition system. The described configuration can be seen in Figure 1.

For this study, a mixture of gasoline and ethanol was used as fuel, using the proportion of 27% ethanol and 73% gasoline (G73E27). This fuel has a density (20°C) of 0.7524 kg/m^3 and a higher calorific value of 39.335 kJ/kg, obtained with the aid of the DDM2911 digital density equipment and the IKA C1 calorimeter, respectively.

To obtain the images, a high-speed camera PCO.Dimax S1 was used coupled with a VS4-1845hs Scope dual intensifier and 105 f/4.5 UV-Nikon lens. For the combustion analysis, a region of interest of 864x896 pixels was selected, a frame rate of 5400 fps for a speed of 900 rpm, which corresponds to 1 image for each crank angle (CA). The configuration used allowed a resolution of 91 μm/pixel. For a better investigation of the cycle and combustion, the tests consisted of 62 frames per cycle processing the last 80 cycles. Figure 2 illustrates the view obtained with the high-speed camera of the upper part of the cylinder, where the location of the exhaust and inlet valves, spark plug, pressure transducer and the possible locations for the direct injection injector can be observed. The sail is located 5 mm above the geometric center of the chamber.

Among the thermodynamic data provided by the system are the cylinder pressure, mass fraction burned (MFB), indicated mean effective pressure (IMEP) and the coefficient of variance (COVIMEP). The MFB is the crank angular range required for burning the corresponding part of the fuel. IMEP is the average effective pressure calculated with the indicated power (work). As a parameter that does not depend on the size of the motor, different from power and torque, it is a suitable coefficient for comparison with other works. Due to the cyclical variability and instability of combustion, one method to calculate this variation is the coefficient of variance (COVIMEP).
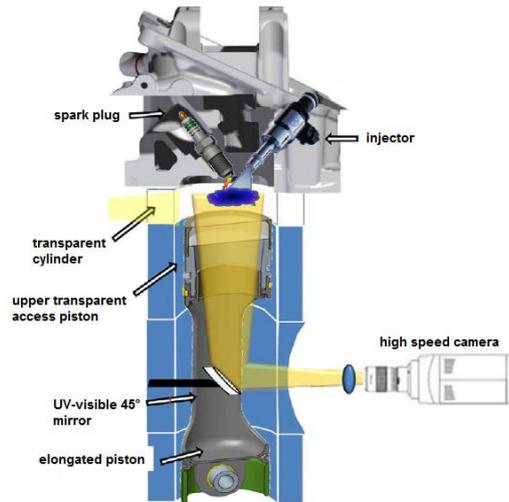
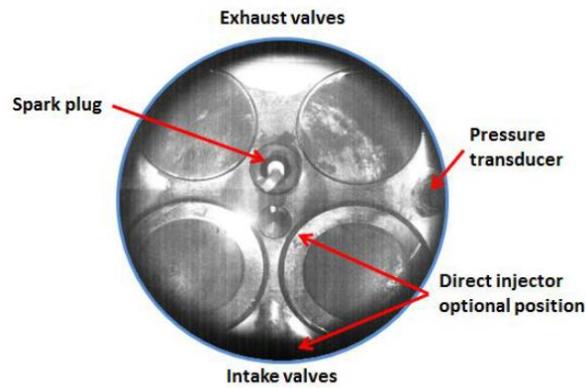Figure 1. Experimental apparatus.

Figure 2. Combustion chamber geometry.

Engine speed was set to 900 rpm and injection timing was adjusted to provide stoichiometric conditions at constant operation. The tests were performed with central direct injection with the start of the injection (SOI) was varied in 290, 220 and 180 CA BTDC (Crank Angle Before Top Dead Center). The different fuel injection start positions tested were selected to analyze how they affect the time of the fuel inside the cylinder and also the contact of the fuel spray with the inlet valves. The fuel injection pressure used for the tests with direct injection (DI) was 100 bar.

Due to the injector orifice distribution configuration, two positions were used, as seen in Figure 3. The settings are relative to the direction of the fuel spray at the time of fuel injection. Thus, the injector directed towards the intake valve (IVP) and with the holes directed towards the spark plug (SPP) was defined by the physical possibilities of the injector and the engine.
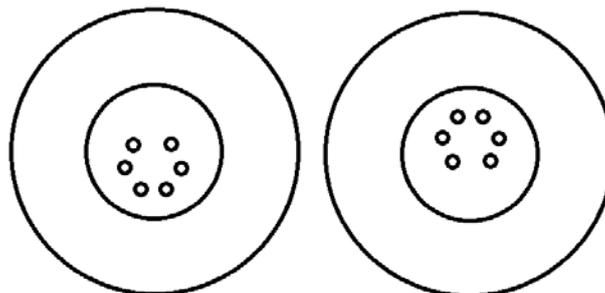

Figure 3. Intake valve (IVP) and Spark plug (SPP) position.

3. RESULTS AND DISCUSSION

As a result of the procedures described above, synchronized records of the propagation profile of the fuel spray to the injector positions were obtained using the high-speed camera. For the tests carried out with SOI 290 and SOI 220, it is observed that due to the configuration of the injector holes and its position, the spray comes into contact with the valve still in the process of opening, mainly for the position of the injector directed towards the inlet valve. This behavior is minimized when injecting to SOI 180.

The inversion of the injector position towards the spark plug, despite providing less contact of the spray directly with the valve, generates another characteristic which is the direct contact of the fuel with the spark plug. This behavior can be harmful to the spark plug, providing carbonization and reducing its useful life. The injection moments in relation to the crank angle for the positions for the inlet valve and for the spark plug are shown in Figure 4 to Figure 7.
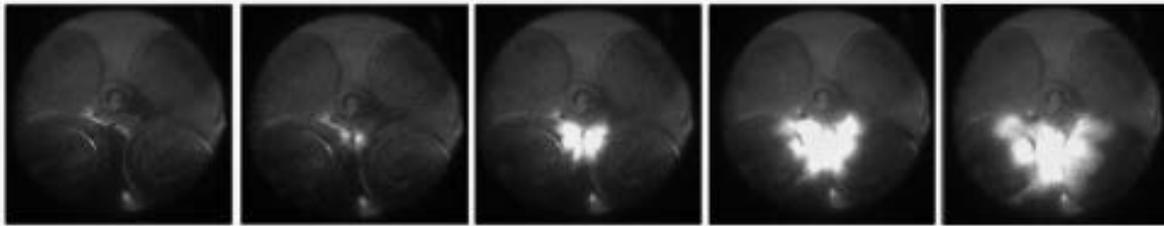

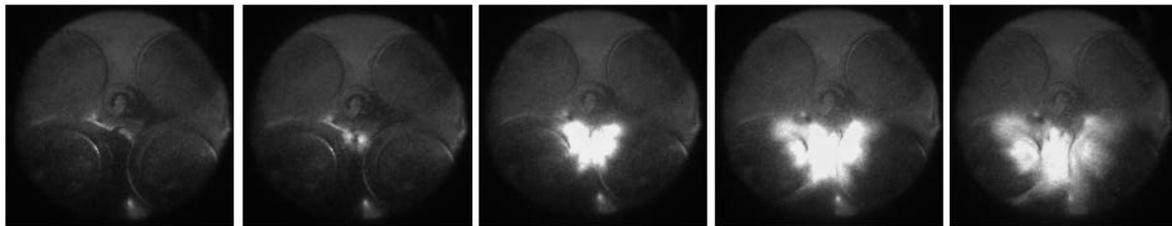Figure 4. Moment of Injection by crank angle in Intake valve position SOI 290 BTDC.

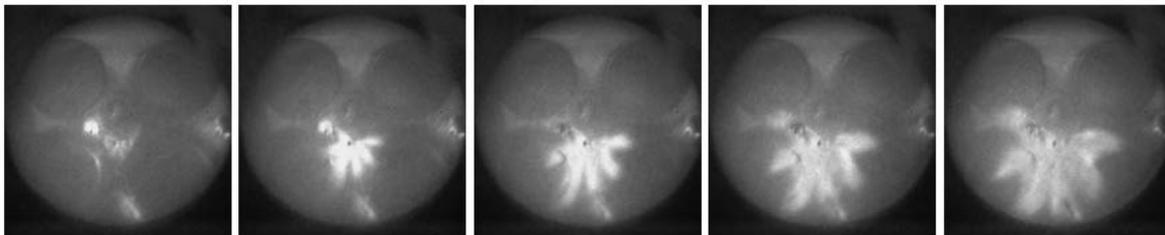Figure 5. Moment of Injection by crank angle in Intake valve position SOI 220 BTDC.

Figure 6. Moment of Injection by crank angle in Intake valve position SOI 180 BTDC.
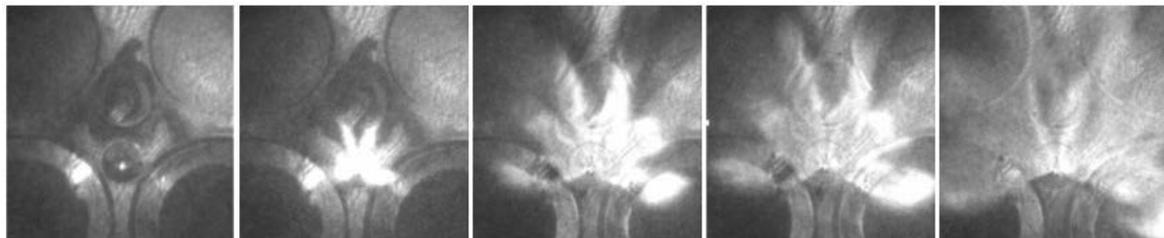

Figure 7. Moment of Injection by crank angle in Spark Plug position SOI 180 BTDC.

In addition to optical data, thermodynamic data were analyzed to understand the effect of injector positioning on engine efficiency. In Figures 8, the cylinder pressure for operations with SOI 180 (Start of injection) for the two injector positions can be observed. It is observed that for this case, where the fuel injection happens with the inlet valve already close to its closing, the highest performance is observed for the injection to the inlet valve. This becomes clearer when looking at Table 1 where the mass fraction of burned (MFB), IMEP (Indicated Mean Effective Pressure), CovIMEP (Covariance of Indicated Mean Effective Pressure) and Maximum pressure (Pmax) are presented. Despite this, it can be seen that the difference is relatively low between the results, with the turbulence and better air-fuel mixture close to the

intake valve provided by the characteristic Tumble movement of the engine which provided this better performance (Abrantes, 2017 and Martinez, 2017).
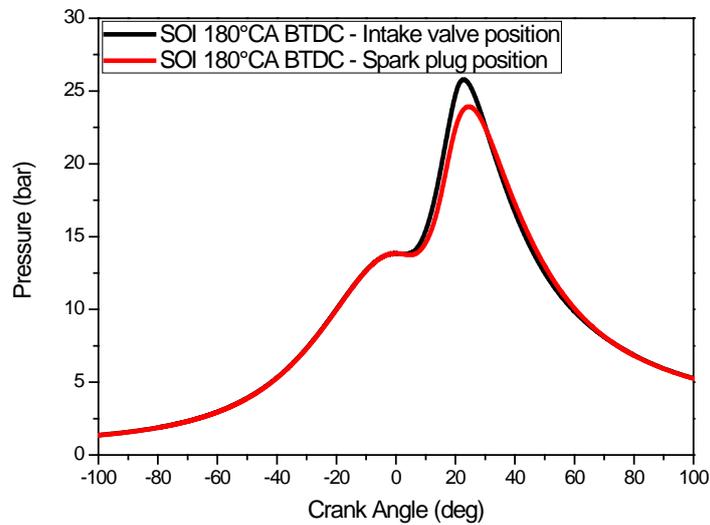

Figure 8. Average cylinder pressure curves for SOI 180 CA BTDC to IVP and SPP.

Table 1. Thermodynamic data of tests performed for SOI 180 CA BTDC to IVP and SPP.

	180 BTDC – IVP	180 BTDC – SPP
MFB* 50% [CAD ASOS]**	19.6	21.4
IMEP [bar]	5.94	5.95
CovIMEP [%]	1.05	3.34
Pmax [bar]	26.6	24.9

* Mass Fraction Burned; ** Crank Angle Degreed After Start of Spark.

Added to the comparison for the different positions of the injector, the thermodynamic behavior for variations in the injection moment was also analyzed. As seen in Figure 9 and Table 2, better performance was obtained by operating with SOI 220, followed by SOI 290 and SOI 180. In addition to less contact with the inlet valve compared to SOI 290, the residence time of the fuel inside the cylinder for better mixing with air is the basis for a more complete burn. This factor may be one of the reasons for the better performance of SOI 220 compared to other modes.
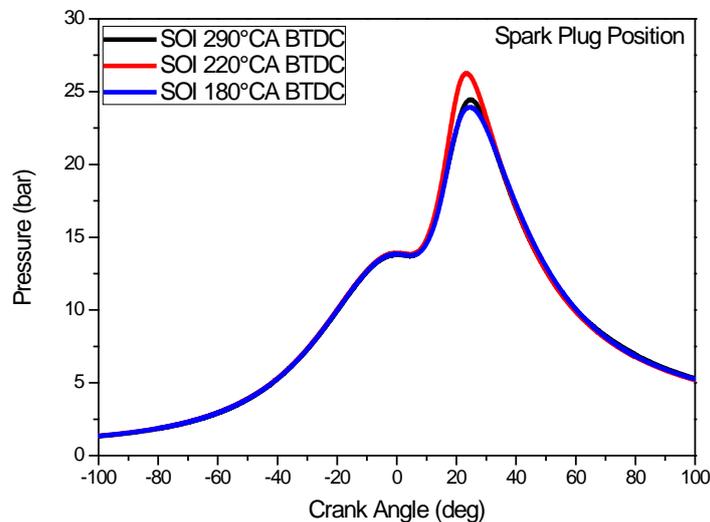

Figure 9. Average cylinder pressure curves for SOI 290, 220 and 180 CA BTDC to SPP.

Table 2. Thermodynamics data from tests performed for SOI 290, 220 and 180.

	290 BTDC	220 BTDC	180 BTDC
MFB* 50% [CAD ASOS]**	21.3	19.6	21.4
IMEP [bar]	5.97	5.96	5.95
CovIMEP [%]	2.12	1.41	3.34
Pmax [bar]	25.0	26.8	24.9

* Mass Fraction Burned; ** Crank Angle Degreed After Start of Spark.

The optical flame propagation data obtained in the tests performed were processed and analyzed. To understand the behavior of the flame and the variability of the process, the 15th crank angle after ignition was selected and fixed for each cycle for the analysis of the behavior of the 80 flame cycles, thus the crank angle 8 CA ATDC was selected, called CA 368. The crank angle selected was chosen because in relation to the flame development, it was in an intermediate position where in the vast majority of tests the flame casing had not reached the cylinder wall and also already had a satisfactory development.

Thus, Figures 10 and 11 show the diameter of the tests performed for SOI 220 for the positions of the injectors. In the results it is possible to observe the great variability of the tests using the direct injection mode. However, tests tend to remain mostly within the calculated standard deviation. It is also observed that after a test in which the flame development in the fixed crank angle is smaller, the next cycle follows with evolution of the larger flame diameter. This is due to the possible accumulation of fuel in the worst development cycle, providing more expressive burning in the subsequent cycle. This possible accumulation of fuel and, consequently, greater variability, may be linked to the characteristic of the search engine used. When comparing the position of the injectors, it is possible to observe an upward behavior of the tests performed with SPP, which generates the possibility of tests with better performance in relation to the IVP position for tests with longer operation.
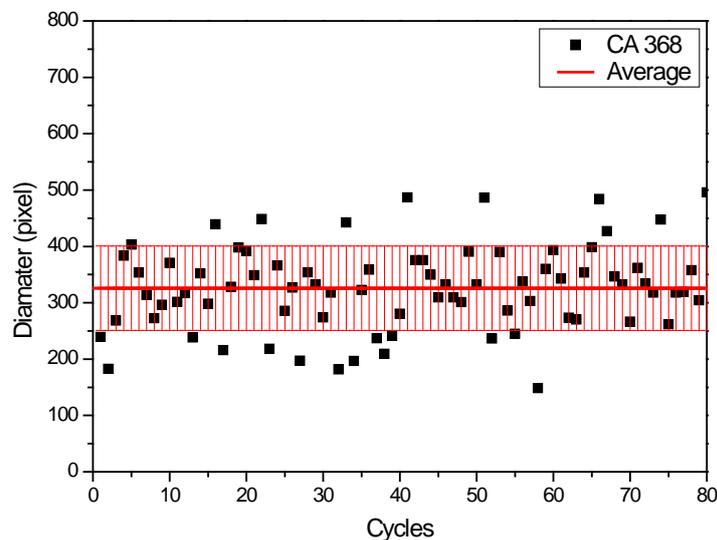

Figure 10. Test performed for SOI 220 for intake valve position (IVP) for diameter of 80 combustion cycles versus cycle for fixed crank angle if 15 CA after ignition.
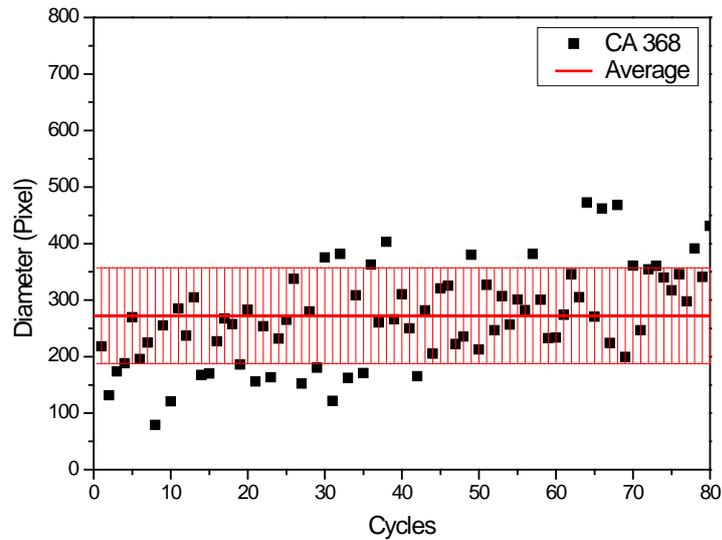
Figure 11. Test performed to SOI 220 for spark plug position (SPP) for diameter of 80 combustion cycles versus cycle for fixed crank angle if 15 CA after ignition..

The dispersion of the flame inside the cylinder was also analyzed and shown in Figures 12 and 13 below. Similar to Figures 10 and 11, the crank angle was fixed after ignition to observe the variability of the flame. When analyzing the images, it is observed that in both cases, the flame distribution centered on the cylinder is slightly shifted towards the exhaust valve. This behavior is mainly caused by tumble motion and high temperature gradient. When comparing the injector positions, a greater displacement of the SPP configuration in relation to the IVP is observed. This is possibly due to the direction in which the fuel was injected and also the greater penetration of fuel to this region, providing greater accumulation of fuel in the area of the exhaust valves.
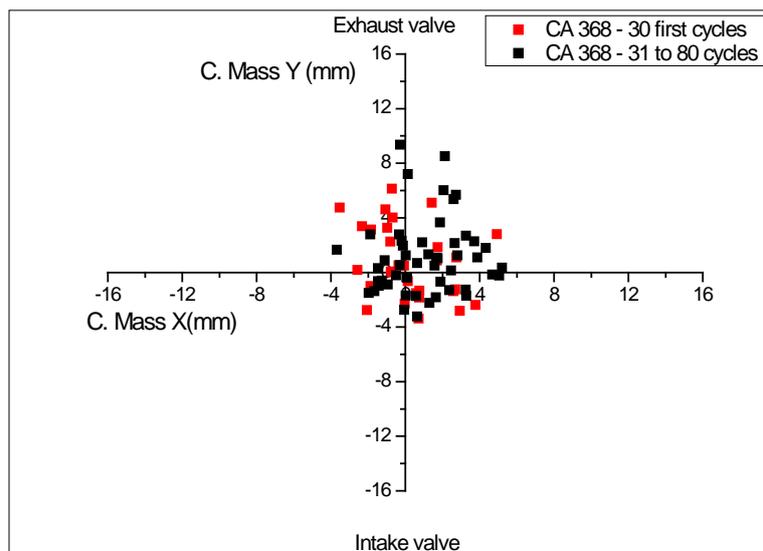

Figure 12. Test performed for SOI 220 for intake valve position (IVP) for image center for 80 combustion cycles at 15 CA after ignition.
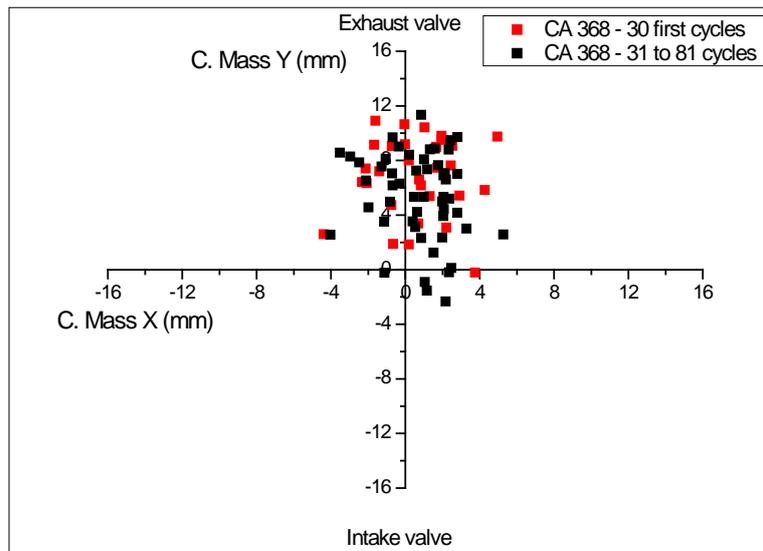
Figure 13. Test performed to SOI 220 for Spark Plug Position (SPP) for image center for 80 combustion cycles at 15 CA after ignition.

4. CONCLUSION

The tests carried out allowed us to understand the influence of the behavior and positions of the engine parts on the operation and efficiency of the engine using the biofuel ethanol as a partial substitute for gasoline in a mixture.

It was observed that the contact of the jet of fuel with the valve, mainly for operations where the valve had its largest openings, can generate, among other resistances, the condensation of the fuel or loss of energy, thus providing an inefficient combustion, which contributes to low thermodynamic and optical development. The contact of this fuel with the intake valve and the condensation of this fuel in the engine parts, causes carbonization in the valve, which can generate a certain increase in pollutant emission and wear on the engine parts.

As for the position of the injector with the holes facing the spark plug, greater possibility of spark plug wear due to direct contact with the fuel and subsequent carbonization can be a result of the operation, mainly for the first combustion cycles due to the low temperature of fuel and parts of the engine. This feature in the long run can lead to lower engine efficiency.

Another important factor that should be highlighted is the residence time inside the combustion chamber. The longer residence time of ethanol blends contributes to a better air-fuel mixture that provides better burning.

It is also highlighted how the tumble and swirl movements, for the engine used, the tumble being the most expressive, influence the dynamics of combustion and can interfere with the strategy adopted.

5. ACKNOWLEDGEMENTS

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