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Simulation and Analysis of Direct Injection Strategies for Hydrous Ethanol in Internal Combustion Engines

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Abstract*— This research work deals with the application of a hydrous ethanol direct injection spark ignition engine model. Computational Fluid Dynamics (CFD) software is applied in order to create the moving mesh and to solve the governing equations. The research scope includes the airflow characterization during intake and compression stroke and also the charge stratification. The distribution of mixture formation distribution within the combustion chamber is analyzed. The influence of parameters such as fuel injection pressure, start of injection, injected fuel mas, equivalence ratio and wall-wet film are evaluated. The goal is to obtain the fuel concentration within the cylinder during the spray development and the piston movement and also the film formation in the chamber walls. It was analyzed the differences and benefits between different injection strategies considering single and dual pulse injection. Parameters such as injection pressure and fuel mass injected are considered, helping to trace trends for their relation to fuel vaporization. These results are an important reference in order to help on determining the best injection strategies making feasible engine cold starts at low temperatures without auxiliary devices. As expected, the results obtained showed that injection pressures as well as start of injection are parameters, which influence directly the fuel vaporization, and the amount of wall-wet fuel film formed, mainly on piston.

***Index Terms*— internal combustion engines, hydrous ethanol, strategies of injection.**

I. INTRODUCTION

Brazilian experience with engines operating on hydrous E100 is extensively. Their application as a transportation fuel in Brazil began in the late 1970', based on a government incentive for alternative fuels, declining in popularity in late 1980'. Later on, on the 2000' the Flex Fuel Vehicles technology became known as new opportunity for alternative fuel application as engine fuel in large scale. From this date on, the consumers could purchase vehicles motored by either E22 (a blend of approximately 22% ethanol and 78% gasoline) or E100 (~94% ethanol, 6% water), both of them produced for the Brazilian market.

In the early 2000, the first flex fuels engines were running in the streets. At this time the research and development work were mainly concentrated on the consolidation of the engine structures to withstand the more aggressive alcohol behavior compared to crude oil based gasoline. Furthermore, it was necessary to develop both dedicated fuel discharge devices (injectors) as well as the complex related software for identification of the fuel blend and an optimized control of the mixture preparation.

Despite the difficulties, ethanol presents a considerable number of operational advantages that justifies the effort done by automotive industry to keep on trying in its application. The high octane number of ethanol and the resulting excellent knock performance gives significant benefits, for normal aspirated engines and even better results when applied on turbocharged engines.

Other advantages is the green characteristics of ethanol which has prompted many governments around the world to blend ethanol in varying percentage with gasoline to reduce the emissions attending the new regulation levels, moreover the dependence on fossil fuels decreases due to the diminution of the crude oil importation.

Other intrinsic characteristics of ethanol are challenging for industry, such as, its evaporation characteristics that result in challenges regarding cold start and oil dilution mainly with direct injection application. An important challenge is to enable cold start under 12^{o}C – flash point for pure ethanol – as long as at this temperatures the alcohol vaporization is very deficient, making difficult the cranking.

The water content on Brazilian ethanol can be faced in two sides: it is positive for cooling the charge, and on the other hand, it is negative as long as increase the amount of fuel delivered in order to compensate the energy needed to generate same power output.

Ethanol is manufactured in either form: anhydrous and hydrous, the second one can be found in Brazil and in Sweden. The production of anhydrous ethanol requires a costly second step, enhancing fuel price.

Nowadays the majority of Brazilian flex fuel fleet works with a redundant system used previously on the carbureted engines, a secondary tank of gasoline, which actuates in low temperatures making possible the cranking. The inconvenient for this solution is the low frequency of these low temperatures in Brazil, causing the evaporation of the gasoline in the auxiliary tank, taking to poor or even no-starts when the secondary system is activated.

II. OBJECTIVES

The main purpose of this paper is summarizing part of the studies performed and progress obtained so far regarding alternative fuels development by automobilist industry.

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Moreover, a new approach is discussed based on numerical simulation, for the ethanol injection application in internal combustion engines.

Ethanol chemical characteristics are explained and his behavior when applied as the exclusive fuel to feed engines.

A computational model representing a four cylinder direct injection engine (SIDI) is studied. The goal is to establish a guide for spray strategy applied during calibration phase. The airflow pattern and the mixture formation during the full cycle of engine operation are studied. The wall guided combustion system effects on the in-cylinder charge stratification are considered for the flow characterization.

The commercial codes ES-ICE and STAR-CD (Star CD User Guide) are used for moving meshes generation and solution of the conservation equations, respectively. The STAR-CD GUI is employed for post processing the results.

Thus, after the analysis of results, it is possible to predict the effectiveness of a gamma of injection profiles available, and the best start of ignition. Moreover, it could be predicted the amount of wall-wet film formed, guiding the calibrators to operate with injection strategies that presented less film and an ignitable mixture at the same time.

III. Literature Review

The following topics summarize part of the literature review done for this research work.

A. *SIDI Engines*

It is well known that direct injection spark ignition engines represent the combination of the best characteristics of the compression ignition engines (Diesel) and the spark ignition engines (Otto). Mainly when operating in partial load, these engines present the best working conditions because they combine the reduction of the pumping working losses and the lean mixture combustion.

According to [22], the mixture preparation process and the combustion characteristics of this type of engine, when operating with gasoline, present fewer tendencies to knocking when compared to conventional port fuel injection (PFI) spark ignition (SI) engines. This fact is proven also based on positive results on SIDI engines operating in high compression ratios, which is a pre-requisite when considering E100 flex fuels engines best performance. Moreover, as expected, the volumetric efficiency of a GDI engine is higher than that of a PFI engine, which can be explained based on the intake mass composition, for PFI engines it is composed by air plus fuel vaporized, while in DI engines it is exclusively composed by air.

B. *Stratified Charge*

The direct injection systems designed to operate in stratified charge can be classified as: spray-guided, wall-guided e air-guided [31]. The specific nomenclature depends on the spray dynamics; the spray incidence on the piston surface and the mixture flow, respectively. It is important to mention that in an actual operation, all these systems contribute to charge stratification, but normally one of them predominates.

In their studies, [1] presents that the greatest challenge in stratified operation consists in the mixture formation optimization. It can be achieved in different ways: guided by spray, where the injector is positioned in such a way that the fuel jet is guided towards the spark plug; guided by tumble, where the in-cylinder charge motion is used to fuel deflection as well as for mixture preparation or guided by swirl, where the charge motion is applied to the mixture formation. It is common to apply piston bowls in order to deflect the fuel.

SIDI engines, in this majority, apply the wall-guided system to achieve charge stratification. This system consists in assembling the injector laterally in the cylinder and applying a piston with a deep bowl. In this system, during the stratified charge operation, the fuel is injected directly into the bowl and deflected towards the spark plug. The equivalence ratio near the spark plug must be sufficiently rich to guarantee the ignition and to reduce the misfire occurrence [31]. The predictability of the stratification effects is directly connected to the knowledge of the fuel behavior during the intake stroke. Factors such as the intake air portion where this fuel is concentrated, how the air flow affects the fuel displacement and the fuel concentration near the spark plug at the ignition timing are very important.

C. *Emissions*

It has been well documented in literature [10] that the exhaust gas emitted during cold start and in the subsequent warm-up period contributes in a significant amount for the total vehicle emissions index. The major problem regarding the cranking is because the catalytic converter is still cold, so, it is not operating under its full capacity to retain the pollutants, which affects significantly the driving cycle emissions levels.

This fact can explain why when special attention is given to emissions on these periods of engine cycle operation. According to [20], depending on the engine type and the design of the exhaust gas system for after-treatment, the emissions from these two phases can constitute up to 90% of the total HC-emissions emitted during a NECD test.

Fig. 1 presents the effect of air fuel ratio in formation of some exhaust gases: NO, HC and CO.

A huge number of alternatives has been found and implemented in production vehicles, such as the start-stop systems the commercialization of hybrid vehicles. These solutions contribute to decreases the emissions issues during cranking and warming-up periods, but they are still not economically feasible to be implemented in small passenger cars.

The research work from [34] presented experimental results for emissions components relating those to water content, as can be seen in Fig. 2.

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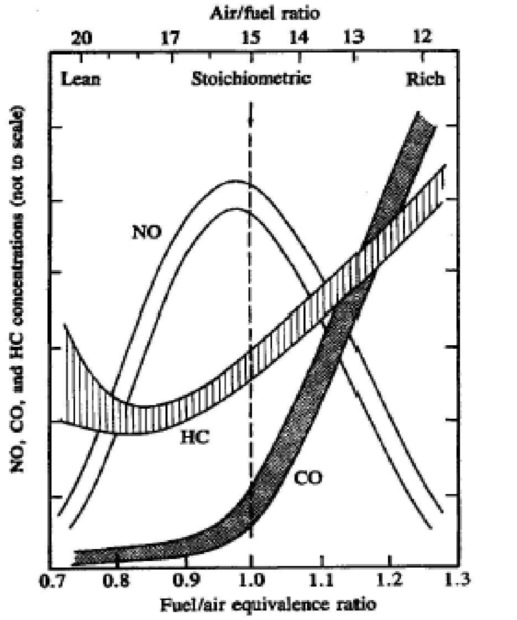

Fig. 1. Air/Fuel mixture effect on NO, HC and CO formation. [10]

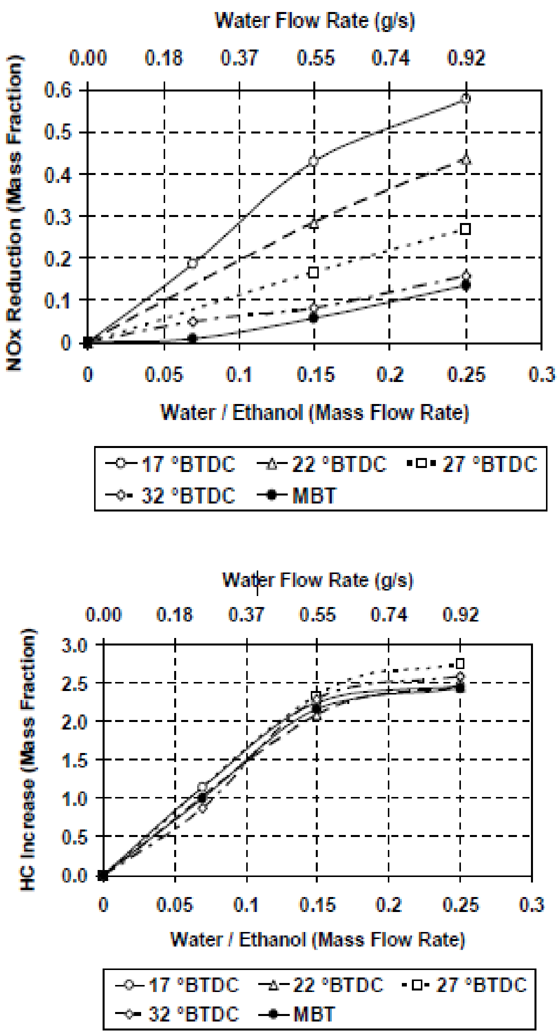

Fig. 2. (a) NOx and (b) HC response at 2000 rpm and 100 kPa MAP [34]

When analyzing same ignition timing, NOx reduction presented lower values as the water content increased. This can be attributed to the decrease in peak combustion temperatures resulting from the specific heat capacity of the added diluent, and to the additional charge cooling results from the transition of phase of water from liquid to vapor. However, the HC emissions presented other trend. As the water content increases, the HC increase as well, a possible explanation for this phenomena could be the increase of quenching, and the fuel water content role on mixture preparation. The CO response did not presented any change related to the differences on water content.

D. Cold Start

Cold start is highly dependent on the fuels ability to vaporize effectively at low temperatures and provide an ignitable mixture at the time of starting. The condition is still worse when considering the conditions of the combustion chamber at this time: the walls are still cold and the intake air velocity is low due to engine low speed. Within a certain temperature range, ethanol blends decreases the temperature at which fuel vaporizes. When operating with ethanol blends, such as E85, E25 or E10, the cold start is dependent of the vaporization or the more volatile fractions of the mixture, in these cases, the gasoline fraction. However, when the operation occurs only with E100, the vapor contains a greater concentration of alcohol than would be expected based on the vapor pressure of the alcohol or its gasoline blends. On this way, more heat is necessary to vaporize the blends containing ethanol.

As consequence, the mixture suffers from enleanment due to the higher alcohol concentration. Studies have shown that it is very difficult to vaporize ethanol at temperatures below 10°C. This difficult to vaporize ethanol is even more pronounced when working with hydrous ethanol, due to water concentration in the fuel.

The requirements applied normally in Europe for minimum fuel temperature is such a manner to guarantee an acceptable cold start performance around -20° C. In Brazil this target is not so severe, being approximately -5 ° C.

In Brazil the lowest temperature from the last decade was recorded at São Joaquim/SC, and can be seen in Fig. 3.

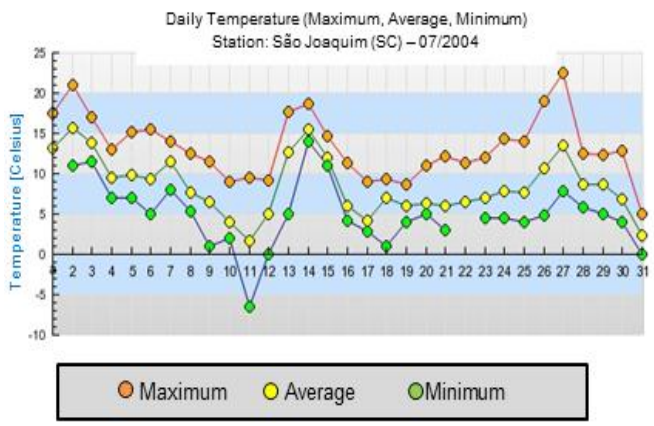

Fig. 3. Lowest temperature recorded in Brazil in last 10 years. São Joaquim, Santa Catarina,

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Since 70´s the application of gasoline sub tanks systems, in carbureted cars was applied and the system have been applied without significant modifications, equipping the flex fuels cars, since late 2000's.

In these vehicles, the solution is applied for cold starts, which occurs typically at 18°C, in Brazil. Fig. 4 shows schematically a gasoline sub-tank system. It consists of a small tank that has an approximate capacity of 1 liter, a small fuel pump, a calibrated orifice and a solenoid valve. The fuel pump is powered by the powertrain control module (PCM), and then the solenoid valve is modulated to deliver the quantity of fuel schedule based on engine coolant temperature. The calibrated orifice can be located in the throttle body beneath the throttle plate or within the intake manifold, and injects the previously amount of fuel calculated either in the plenum or in the intake runners.

After the engine starts, depending on coolant temperature, gasoline may continue to being injected in conjunction with ethanol delivered from the main injectors, in order to optimize warm-up drive-ability and transient throttle response.

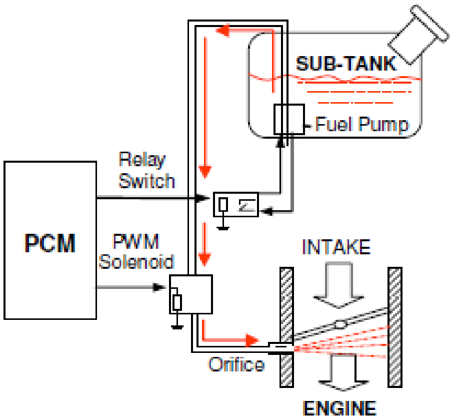

Fig. 4. Gasoline sub-tank system [17]

IV. PROCEDURE

The main purpose of this investigation is to analyze the effect of two different injection strategies considering the stratification effects and mixture motion. Fig. 5 presents the piston geometry applied in this study, which bowl is quite important to achieve the wall guided effect.

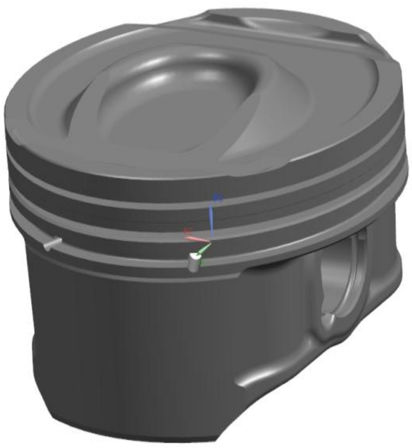

Fig. 5. Bowl geometry

The equivalence ratio around spark plug will be calculated, as well as the amount of liquid film deposited at combustion chamber walls.

The period of simulation considered part of engine cycle, starting at 60 bTDC and finishing at TDC firing (720 cad). The intake stroke is taking into account so the air intake flux characteristics are preserved, representing the real operation of the engine. The combustion was not considered, since the main objective of this work is to analyze the mixture formation conditions at spark plug and film formation before the first combustion cycle. Thus, the results are analyzed considering the mixture flow motion along the piston movement and its equivalence ratio at ignition timing (around 5 cad bTDC).

Fig. 6 presents details regarding mesh used and injector and spark plug positioning.

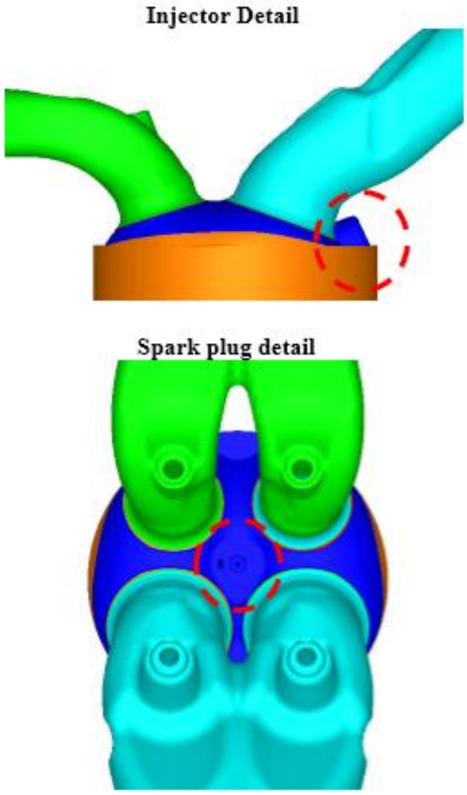

Fig. 6. Mesh applied in the simulations

The engine specifications are given in the Table 1.

Table 1. Cylinder specifications

Diameter	88 mm
Stroke	98 mm
Conrod length	143.8 mm
Compression ratio	11.9:1
Speed	182 rpm

The accuracy of the results obtained from the simulations is directly dependent on the mesh resolution. The mesh resolution used in this work corresponds to 928000 cells at BDC and 516600 cells at TDC, and it guarantees the mesh independence of the results.

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A. Injection Strategies and Initial Conditions

The initial conditions for the intake air are in the Table 2.

Table 2. Initial conditions for the intake air

Intake air pressure	1 bar
Intake air temperature	268 K

For the cylinder head, the cylinder walls and the piston constant temperatures were used, which can be seen in Table 3.

Table 3. Boundary conditions for temperature

Cylinder head	268 K
Cylinder walls	268 K
Piston crown	268 K

The injection timing considered is presented in Table 4. For Case A was applied one single injection pulse, and for Case B a 30%-70% split factor was used and 2 pulses were tested.

Table 4. Injection timing for Cases A and B.

Case	Equivalence Ratio	Start/End of Injection [CAD]	Injection Pressure [MPa]	Split Factor
A	5.5	673.82 – 690.32	20	100%
B	5.5	430.00 – 434.87 678.63 – 690.00	20	30% + 70%

The injection profile for the Cases A and B can be seen in Fig. 7. The injection velocity applied was equivalent to 145 m/s.

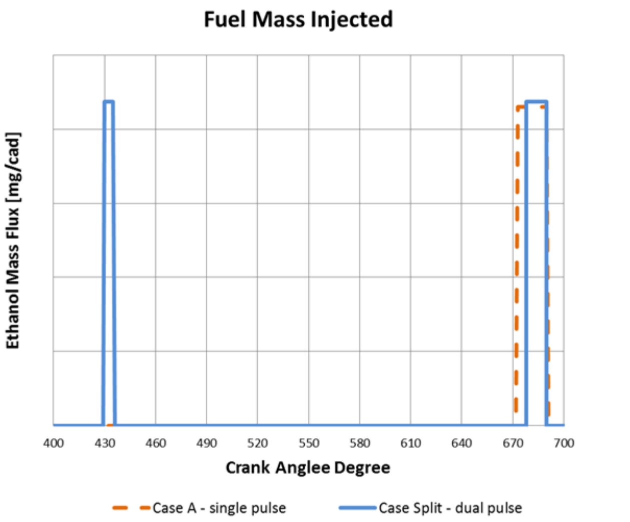

Fig. 7. Injection profile for Cases A, single pulse and Case Split, dual pulse.

V. RESULTS

The focus of this study was on wet-wall fuel film formation and equivalence ratio available around spark plug at ignition timing, considering cold start operation at -5°C degrees. This parameters were analyzed considering 2 different injection strategies, one and dual pulse, as explained in previous item.

The following results will present the conditions achieved in combustion chamber under the operational conditions discussed previously.

The scope of the analysis consists of two cases operating with same injector geometry and same injection pressure, equivalent to 20 MPa. The start of injection for Case A is 673.82 cad and for Case B first and second pulse are respectively 430 and 678.63 cad. Suppliers calculate parameters such as plume angle and Sauter Mean Diameter [32] based on characterization of the spray geometry.

A. Pressure and Temperature

Fig. 8 presents the results achieved for cylinder temperature. As can be observed the case where dual pulse was applied resulted in a slightly lower final temperature, indicating that an advanced pulse partial injection could jeopardize the final combustion chamber temperature as well as deteriorate the fuel vaporization conditions. However around 670 cad, the temperature for dual pulse case is higher than single pulse, indicating that there is a positive situation (higher combustion chamber temperatures), and also that the best injection timing for both pulses must be studied deeply.

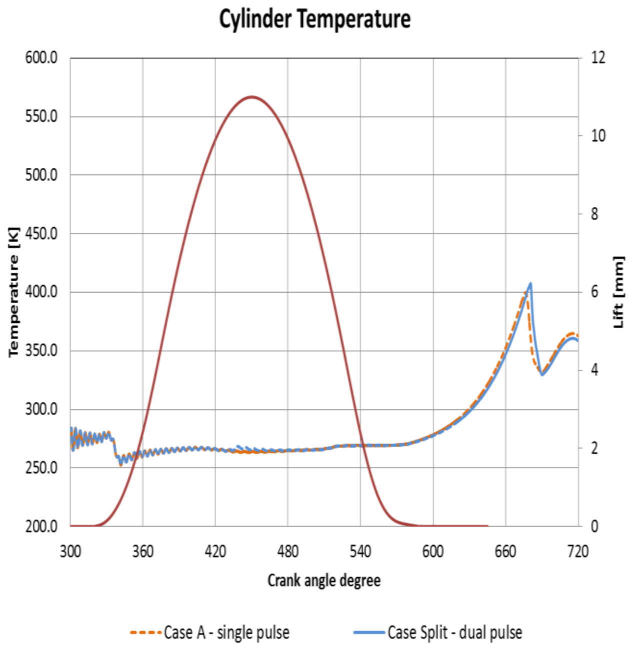

Fig. 8. Cylinder temperature evolution for Cases A and B. Intake valve lift profile, red curve.

Fig. 9 presents the results obtained for cylinder pressure histories.

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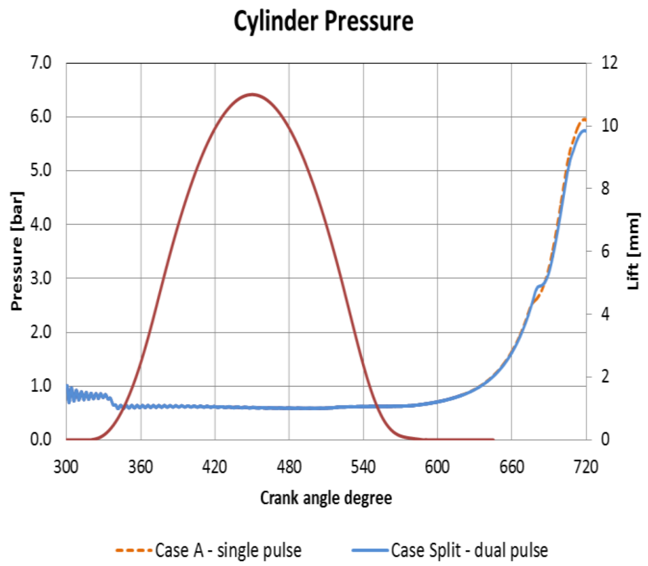

Fig. 9. Cylinder pressure evolution for Cases A and B. Intake valve lift profile, red curve.

The behavior for both cases follows the same trend discussed above for temperature. Both curves present a perturbation during Case A single injection and Case Split second pulse. For the first pulse injection, the perturbation is not captured by the curves probably due the cylinder environmental conditions.

B. Turbulent Kinetic Energy

The evolution of the turbulent kinetic energy is presented in Fig. 10.

The histories for magnitude of turbulent kinetic energy within the cylinder shows a peak before 360 cad, correlated to intake cycle. For dual pulse injection more two pulses can be observed, related to the injection events. For second pulse there magnitude is higher than the achieved for Case A, indicating that a later injection trends to provide more energy, probably due to cylinder environmental conditions be more propitious to this occurrence.

In general, the amount of turbulent kinetic energy observed for both cases are very low until the start of injection, where it achieves the peaks. The very low engine speed in which this simulation was performed is the main reason for that. It is well known that the energy inside the cylinder is directly proportional to air inducted velocity, which is a consequence of engine speed. The increases of the energy in both cases result from the conversion of the swirl and tumble movement into turbulent kinetic energy, in other words, the vortex breakdown into small vortices, dissipating energy. The swirl index is increased by momentum caused by the injection event, and this way the kinetic energy is increased because of its dissipation into kinetic energy. At this moment, the influence of the bowl shape on the flow motion is more significant, indicating that the geometry simulated leads to large vortex dissipation and the consequent increases in turbulent kinetic energy.

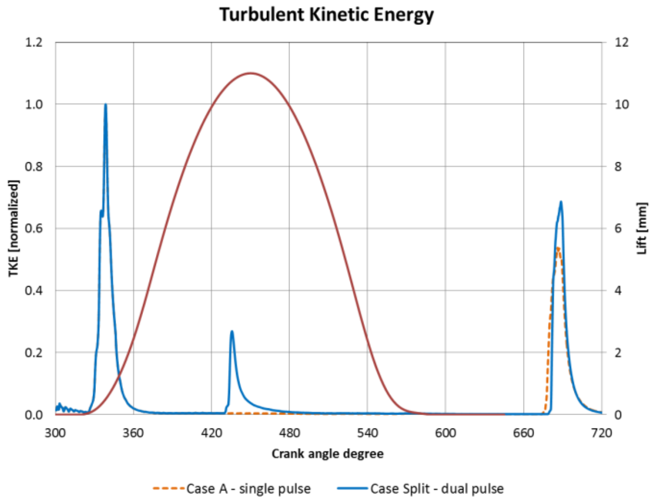

Fig. 10. Average turbulent kinetic energy for Cases A and B. Intake valve lift profile, red curve

C. Equivalence Ratio and Film Mass

A centered spark plug (0,0,Z) was used and a monitored position was set considering a spark plug centered sphere with 5mm of radius.

Fig. 11 shows the equivalence ratio obtained around spark plug in an instant close do spark timing. Case A shows better results for fuel vaporized, indicating that possibly an advanced partial injection could not be positive, at least not the case studied in this paper.

Fig. 12 presents the amount of wall wet film formed summing the film on piston, dome and liner. The Case Split showed a percentage around 50% more film formed than Case A.

This result are coherent with the Fig. 11 results for a lesser EQR for Case Split and also for Fig. 8, where the temperature histories presents lower values for Case Split, interfering on fuel vaporization and favoring liquid fuel deposit on the walls.

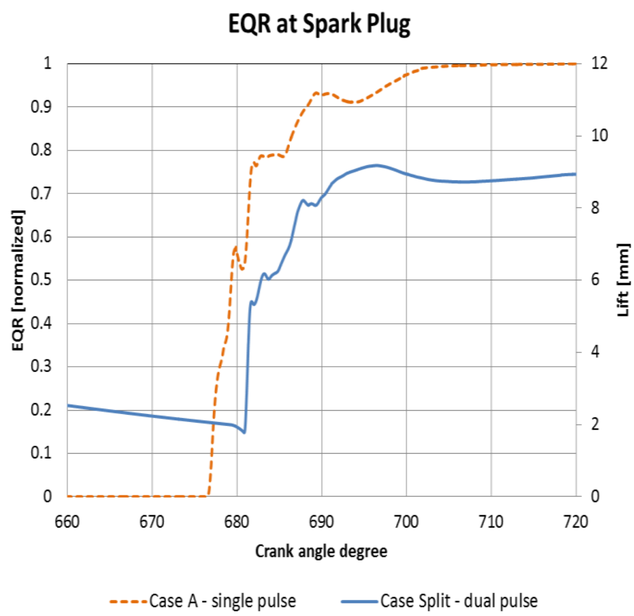

Fig. 11. Equivalence ratio around spark plug for Cases A and B. Intake valve lift profile, red curve.

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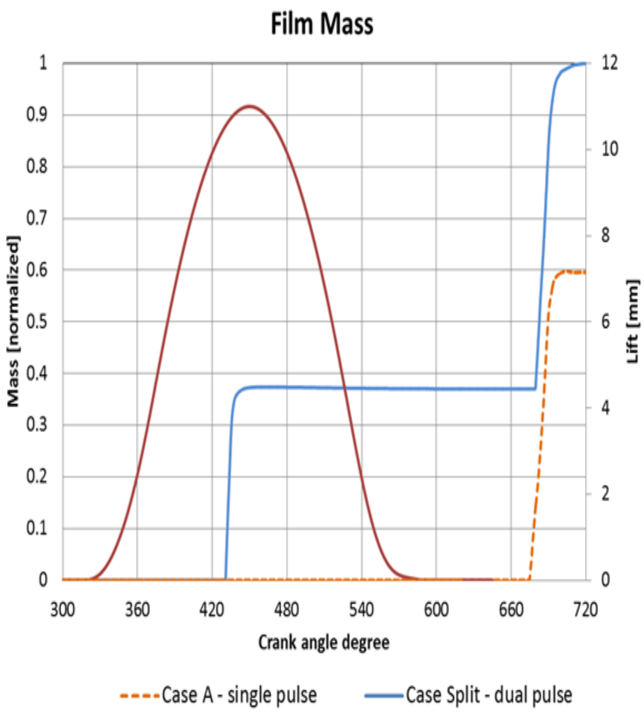

Fig. 12. Film mass on combustion chamber for Cases A and B. Intake valve lift profile, red curve.

VI. Conclusions

In this study, two different injection strategies were analyzed in order to evaluate the best condition for hydrous ethanol direct injection Results for two different injection strategies demonstrate that the injection profile (single or dual) affects the fuel vaporization behavior. The start of injection influences the distribution of the charge in the combustion chamber and affects the vaporization of the fuel. Moreover, the equivalence ratio around spark plug and the amount of wet-wall film was evaluated.

In addition, parameters such as the charge movement, the chamber geometry and the fuel spray development in a DI engine were investigated by means the numerical simulation. Some important characteristics of the mixture formation are identified, such as the stratified charge behavior. The effect of bowl geometry is analyzed regarding the flow field movement within the cylinder. For this specific operational condition, the wall film model applied might be evaluated carefully, as long as, due to the injection close to TDC the piston is completely reached by spray jet, in steady state operation, SOI usually takes place when piston is close to BDC. In forward studies a deeply study regarding the wall film model should be taken into account.

The obtained results can be used as guidelines to estimate the best injection strategy and evaluate the combustion chamber temperatures history through piston motion.

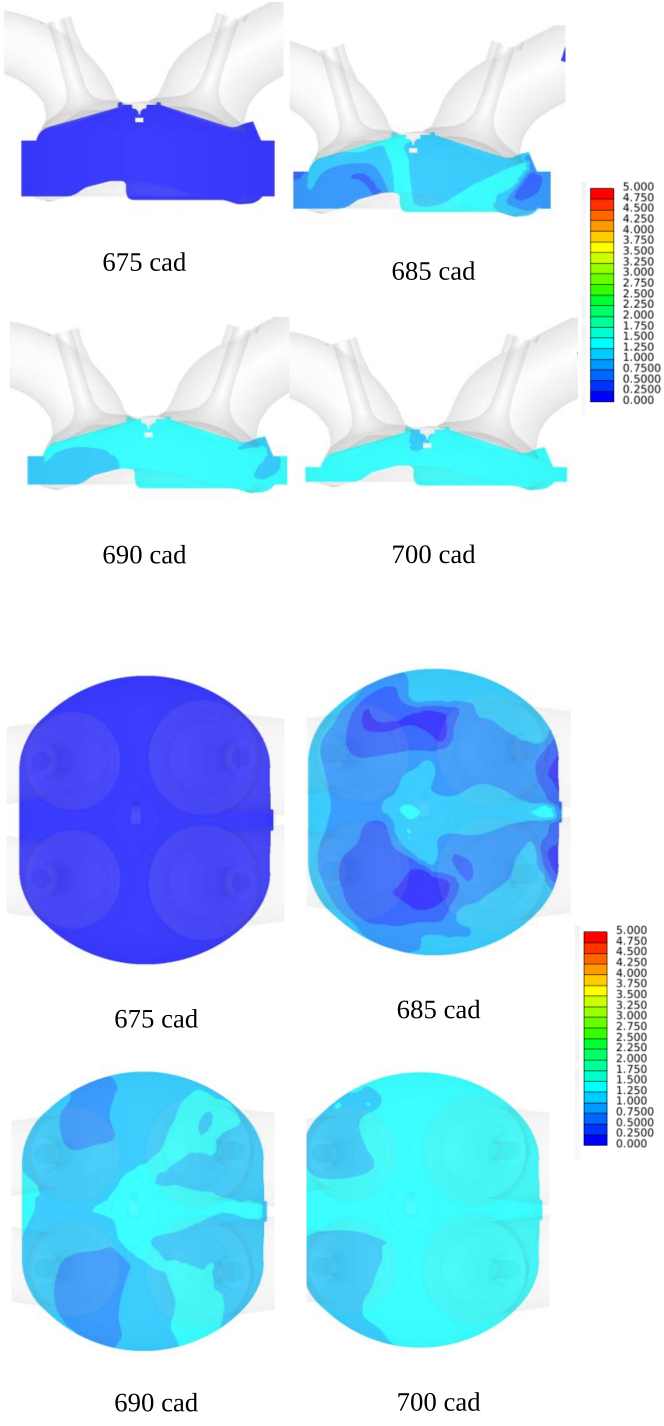

Fig. 13. Contour plots for Equivalence Ratio for Cases A.

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Definitions and Abbreviations

BDC - bottom dead center

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CAD – crank angle degree
CI - compression ignition
DISI - direct injection spark ignition
GDI - gasoline direct injection
PFI - port fuel injection
SI - spark ignition
TDC - top dead center
LIVO - late intake valve opening
LEVO - late exhaust valve opening
IVC - intake valve close
EVC - exhaust valve close

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