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Effect of Nozzle Hole Geometry on Direct Injection Diesel Engine Combustion Process

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Abstract

The aim of the current paper is to link nozzle geometry, and its effects on spray characteristics, with combustion characteristics in the chamber of IC engine. The combustion and formation in a diesel engine is governed mainly by spray formation and mixing. Important parameters governing these are droplet size, distribution concentration and injection velocity. Smaller orifices are believed to give smaller droplet size, even with reduce injection pressure, which leads to better fuel atomization, faster evaporation and better mixing. For this purpose, three 6-hole sac nozzles, with different orifices degree of concavity, have been used. The analysis of all the results allows linking nozzle geometry, spray behavior and combustion development. In particular, CH-radicals have shown to appear together with vapor spray, both temporally and in their location, being directly related to nozzle characteristics.
Keywords: Nozzle Geometry, Diesel Engine Emission

I. Introduction

Fuel injection systems of diesel engines have designed to obtain higher injection pressure. So, it is aimed to decrease the exhaust emissions by increasing efficiency of diesel engines. When fuel injection pressure is low, fuel particle diameters will enlarge and ignition delay period during the combustion will increase. Engine performance will be decrease since combustion process goes to a bad condition. When injection pressure increased of fuel particle diameters will become small. Since formation of mixing of fuel to air becomes better during ignition period, engine performance will be increase. If injection pressure is too higher, ignition delay period becomes shorter. Possibilities of homogeneous mixing decrease and combustion efficiency falls down. The geometry of the nozzle in an injector plays a vital role in controlling diesel spray atomization and combustion. In order to bring fuel droplet size small, the nozzle-hole size is required to be reduced to produce smaller droplets .By decreasing the nozzle hole size, the spray tip penetration is reduced due to the low spray momentum. High injection pressures with small nozzles are common in the modern diesel engine as they reduce injection duration and improve combustion efficiency. Computational simulations were performed to study the effect of reduced nozzle-hole size and nozzle tip hole configuration on the combustion characteristics of a high speed direct injection diesel engine. Nozzles with more and smaller orifices have contributed significantly to the improvement of direct injection (D.I.) diesel engine performance characteristics and to the curtailment of diesel exhaust emissions.

II. How does nozzle geometry affects the combustion process?

The combustion process in diesel engine is mainly depends on fuel spray behavior and the behavior of fuel spray is depends on injection pressure, injector nozzle hole size, nozzle type and geometry. The reduction in orifice diameter affects the fuel spray in different ways.

1) The spray tip penetration will be reduced,
2) The droplet sizes will be reduced,
3) The mass ratio of entrained gas to total fuel injected greatly increases,
4) The mixing rate air and fuel improves.

In addition, the flame lift-off length is relatively longer with decreased orifice diameter .The increases in ambient gas entrainment and the relative flame lift-off length lead to the fuel spray with fewer fuel rich zones that can form soot .By using an injector tip with 50 micron orifice, an isolated fuel jet was non-sooting at an ambient gas temperature and density of 1000°K and 14.8 kg/m^3, with ambient oxygen concentrations between 21% and 10% in a pressurized vessel. In engines, this results in lower engine-out particulate matter (PM) emission with a smaller orifice diameter at light- and medium-loads .However,the reduced spray tip penetration can be a drawback with decreased orifice diameter, resulting in poor spatial distribution of fuel and air utilization concluded that the spray tip penetration, rather than spatial mean diameter (SMD), should be the overriding factor in emissions for heavy-duty diesel engines; nozzles that produce sprays with shorter penetration lengths give higher PM and lower nitrogen oxides (NOx). The spray–spray interactions and the reduced spray penetration, the air–fuel mixing rate actually becomes worse for 12 and 18 orifices injectors, causing a reduction in the soot oxidation and thus increasing the soot emission. Some other researchers also argued that especially under high-speed and high-load conditions a smaller-orifice-diameter tends to increase the exhaust soot, which occurs because the jet momentum is lower; the penetration of the fuel spray is excessively decreased; the adequate fuel–air mixing effect from the spray–wall impinging is obtained with difficulty, and consequently, the air in the piston cavity cannot be fully utilized. .The two micro-orifices are parallel to each other, or with a small included angle (divergent or convergent) between them. Fig.1 shows the concept of a group-hole nozzle compared to a conventional nozzle.

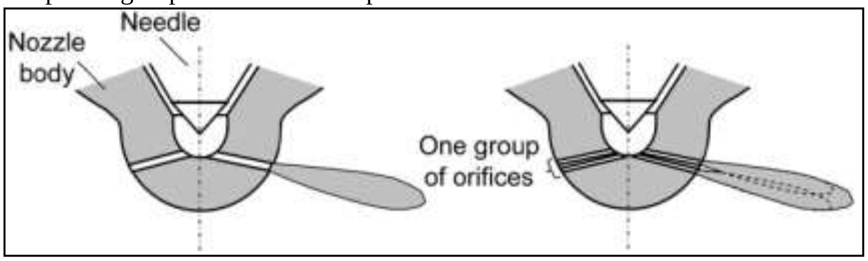

Fig. 1: Concept of a Group-Hole Nozzle (Right) Compared to a Conventional Multi-Hole Nozzle (Left)

In fact minimization of nozzle orifices improves fuel atomization and evaporation, and the very closely-distributed orifices help hold the momentum, maintaining the penetration of the spray. By using a parallel-orifices group-hole nozzle with a 3L (4cylinders) diesel engine, along with the addition of a cooled exhaust gas recirculation (EGR), pre-mixed combustion was achieved and engine-out PM and NOx could be simultaneously reduced. By using group-hole nozzles stated that the overall flame luminosity presents a lower level by adopting an optimal group hole nozzle in a single-cylinder optical diesel engine, indicating the potential for suppressing the soot formation. spray behavior of a group-hole nozzle with close and parallel orifices under various ambient and injection conditions, in comparison with the conventional single hole nozzle with the same total cross section area of orifice(s).

Sprays of a series of group-hole nozzles with included-angles ranging from −10 deg. (convergent) to 10 deg. (divergent) between the two orifices. For free sprays, it is reported that the group-hole nozzle spray penetrates shorter distance and disperses widely as the divergent included-angle increases. While, significant increase in evaporation ratio (ratio of fuel vapor mass to total fuel injected) can be found compared to the single-hole nozzle spray.

Table – 1
Nomenclature

Symbol	*Discription*	*Symbol*	*Discription*
A	*Constant in the ignition delay correlation that represent E_A/R*	C_{mv}	*Fuel mass concentration needed in the spray axis to get complete evaporation*
C_a	*Contraction coefficient*	C_v	*Velocity coefficient*
D_{eff}	*Outlet effective diameter of a nozzle orifice*	D_{eq}	*Equivalent diameter of a nozzle orifice. Defined as* $D_{eq}= D_o \sqrt{\frac{\rho_l}{\rho_a}}$
D_i	*Inlet diameter*	D_o	*Outlet diameter*
EA	*Activation energy*	*ET*	*Energizing time*
K	*Constant used in the ignition delay correlation*	*k-factor*	*Nozzle conicity. Defined as* $k\text{-factor} = 100 \cdot \frac{Di - Do}{L}$
K_P	*Constant in LL analysis, including the effect of cone angle*	*n–m–l*	*Coefficients used in the ignition delay correlation*
L	*Nozzle length*	*LL*	*Liquid length*
P_{back}	*Backpressure*	P_{inj}	*Injection pressure*
SOE	*Start of energizing*	*SOI*	*Start of injection*
t_{mv}	*Time for a fuel parcel in the axis of a stationary spray to reach a concentration equal to C_{mv}*	T	*Temperature in the engine injection chamber*
u_{eff}	*Effective velocity at the outlet orifice, defined as* $u_{eff} = Ca^{\frac{1}{2}} \cdot D_o$	u_{th}	*Theoretical velocity, obtained from Bernoulli equation as* $u_{th} = \sqrt{\frac{2\Delta P}{\rho_l}}$
ΔP	*Pressure drop, $\Delta P = P_{inj} - P_{back}$*	λ	*Wave length*
ρ_a	*ambient density*	*TDC*	*Top Dead Center*
ρ_l	*Fuel density*	T	*Time elapsed from the start of the injection to start ofcombustion (ignition delay)*

III. EXPERIMENTAL METHODOLOGY

All the experimental test were done using standard common rail injection system, including a high pressure pump and a rail, able to regulate pressure inside. Fuel used was Repsol CECRF-06-99.

A. *Nozzles*

In order to make a study in terms of nozzle geometry effects, three 6-hole sac nozzles have been used. Despite Bosch flow number is the same for the three nozzles, they are made with different degrees of conicity, represented by k-factor. Inlet diameter is fixed to approximately 175 µm for all of them. This group of nozzles includes one cylindrical and two convergent geometries. Complete information about their geometry, including inlet and outlet diameters are as follow.

Table – 2
Real Orifice Nozzle Geometry Characterization by Silicone Methodology

Nozzle	*D_i [µm]*	*D_o [µm]*	*K-factor*
1	*175*	*155*	*2*
2	*176*	*160*	*1.6*
3	*175*	*175*	*0*

B. *CH and OH Chemiluminescence*

A single-cylinder two stroke engine is used for combustion visualization studies. The CH- and OH-radicals can be visualized because they emit light intensity in a well-defined wavelength band (k) of the emitted light spectrum. For this reason, using a proper optical filter, emission corresponding to each species can be discerned from the total amount of light coming from combustion process. In this study, CH is acquired using a filter for k between 375 and 405nm, while OH corresponds to the range 305–315 nm. Due to velocity limitations of the camera, only one image is taken at each injection process. In order to get information about CH- and OH-radicals evolution along the engine cycle, the camera trigger is delayed properly depending on injection pressure for statistical analysis purpose, three repetitions are taken at each time step.

Table – 3
Engine Test Matrix Conditions

Inj.P. [MPa]	*E.T. [ms]*	*P at TDC [MPa]*	*T at TDC [K]*	*Density [Kg/m^3]*	*Δt [µm]*	*Repetitipons*
30	*2*	*5*	*950*	*18*	*30*	*3*
30	*2*	*5*	*800*	*22*	*30*	*3*
30	*2*	*7*	*950*	*26*	*30*	*3*
30	*2*	*7*	*800*	*30*	*30*	*3*
80	*1*	*5*	*950*	*18*	*20*	*3*
80	*1*	*5*	*800*	*22*	*20*	*3*
80	*1*	*7*	*950*	*26*	*20*	*3*
80	*1*	*7*	*800*	*30*	*20*	*3*
160	*1*	*5*	*950*	*18*	*20*	*3*

C. *Image Processing*

Images obtained have been analyzed with purpose-made software. Each image is divided in sectors, corresponding to each nozzle orifices. Once each sector is isolated, software analyzes intensity in terms of radial position, calculating a mean value in the whole sector. In Fig. 2, where intensity coming from OH-radicals visualization at a specific time instant and for a single sector has been plotted. Result belongs to nozzle 2, chamber conditions of 7 MPa and 950 ˚ K, and an injection pressure of 80 MPa. As it can be seen. Nevertheless, in order to get a better understanding of combustion process, it is interesting to analyze simultaneously time of appearance and radial position. According to this idea, 2-D contour maps have been chosen as the optimal way of representation. In this figure, the measured intensity (emitted by OH-radicals) is represented by means of a grey scale labeled contour map. This type of graphs allows the simultaneous representation of spatial (in Y-axis) and temporal (X-axis) data obtained from each test point to be performed.

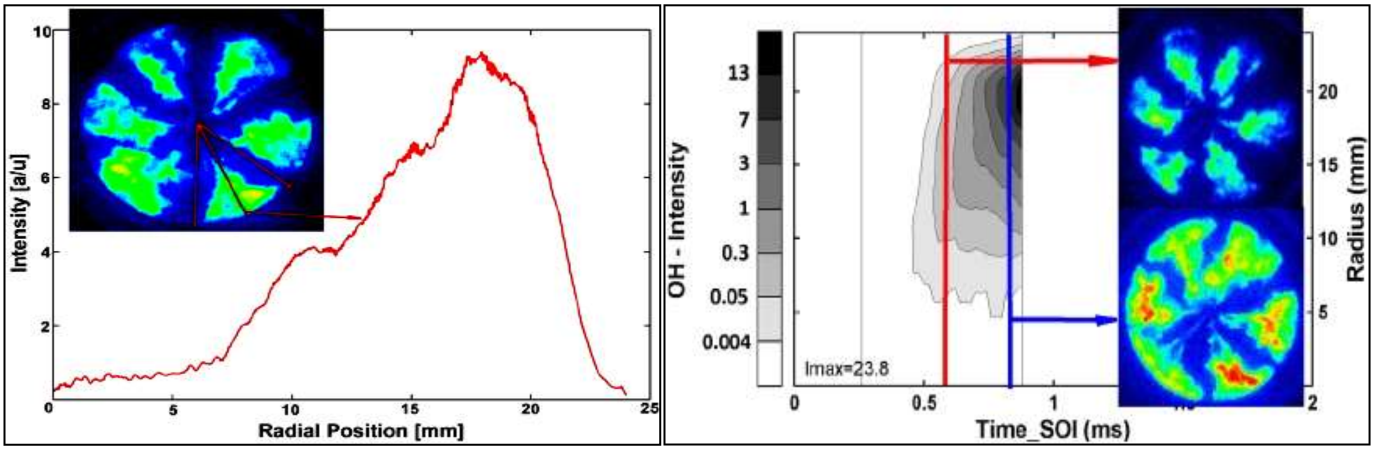

Fig. 2: Image Processing; mean Intensity for Each Radial Position & One Sector

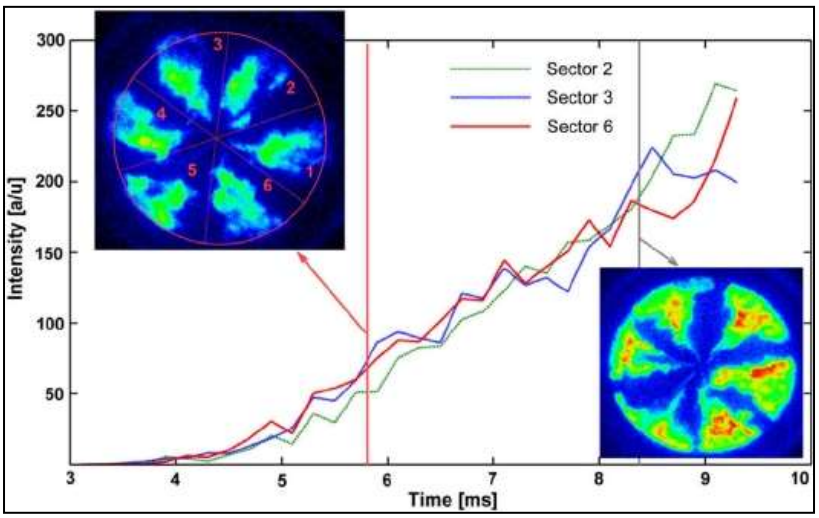

Fig. 3: Temporal Distribution of Intensity and Dispersion between Sectors

IV. EXPERIMENTAL RESULTS

The combustion process is depends upon the spray behavior which is stabilized liquid length which is as follow.

A. Liquid Length Penetration Results

Liquid spray in evaporative conditions was previously studied for the presented nozzles. Assuming that vaporization process is controlled by mixing and using turbulent spray theory for a fuel parcel in quasi-steady spray, the following expression is obtained for the stabilized liquid length:

$$LL = \frac{K_p^2(\frac{1}{4}\cdot\frac{\pi}{4}\cdot C_a)^{\frac{1}{2}}\cdot D_0\ \cdot\rho_l^{1/2}}{C_{mv}\cdot\rho_a^{1/2}} = \frac{K_p^2(\frac{1}{4}\cdot\frac{\pi}{4}\cdot C_a)^{\frac{1}{2}}\cdot D_{eq}}{C_{mv}} \quad (1)$$

Where, K_P is a constant that includes the dependence on spray cone angle,

C_a is a contraction coefficient .D_o the outlet diameter of the nozzle, D_{eq} the equivalent diameter ρ_a and ρ_l densities of air and fuel, respectively, and C_{mv} is the value of fuel mass concentration in the axis at which liquid fuel is totally evaporated, due to the entrainment of warm air. It can be seen that LL shows to be proportional to Deq. For a given engine condition, the parameters C_{mv}, ρ_l, ρ_a are fixed, so that variations in spray penetration are justified in terms of outlet diameter, contraction coefficients (which depends on cavitation regime) or spray cone angle (included in Kp).

Here cylindrical nozzle (N3) provided the highest values of LL at low injection pressure (30 and 80 MPa), due to its higher diameter and the absence of cavitation (C_a=1) .On the other hand when comparing the two conical nozzles, no clear difference are shown.

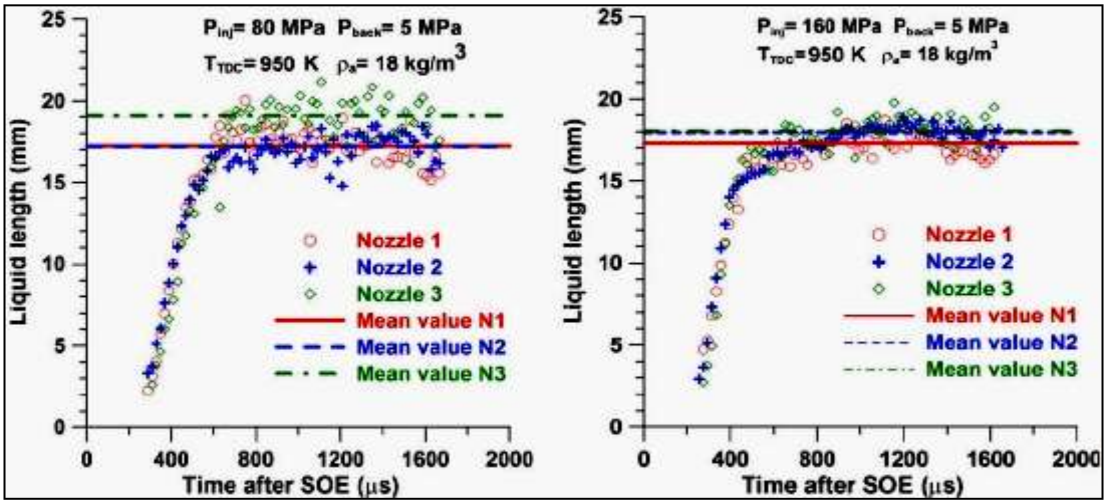

Fig. 4: Examples of Liquid Spray Penetration Results

In this case, there are opposite effects from outlet diameter, higher for N2, and contraction coefficient, for which N3 shows higher values. Nevertheless, when increasing injection pressure, cavitation was more severe in the cylindrical nozzle, and the effect of diameter was compensated by the decrease of Ca. For this reason, the three nozzles have quite similar values of liquid length in the tests performed. Nozzles can also be compared in terms of vaporizing time. Following a similar analysis, this expression can be found for the time to mix and vaporize.

$$t_{mv} \alpha \frac{K_p^2 \cdot \frac{1}{4} \cdot (\frac{\pi}{4} \cdot C_a)^{\frac{1}{2}} \cdot D_{eq}}{C_{mv}{}^2 \cdot C_V \cdot u_{th}} \tag{2}$$

where tmv represents the time that a particle of a quasi-steady diesel spray needs to mix with hot air and vaporize completely, C_v a velocity coefficients and u_{th} is the theoretical velocity (defined using Bernoulli's equation between the inlet and the outlet of the nozzle orifice. Values of t are obtained using values of effective velocity (u_{eff}), already obtained for all the nozzles combining injection rate and spray momentum measurements. Fig. 5 shows t_{mv} values for two different chamber conditions, and for the three nozzles. In general terms, nozzle with higher outlet diameter (N3) shows also the higher time to mix and vaporize. Conical nozzles (N1 and N2) perform with similar behavior, as it happened with stabilized liquid length. As it was seen in liquid length analysis, difference between cylindrical and convergent nozzles becomes almost negligible at high injection pressures, due to the effect of cavitations phenomena in N3. Nevertheless, it is important to underline in this case that only a qualitative comparison between nozzles can be made in terms of tmv because it is not an experimental measurement and values of Cmv are not known. This is the reason why the unit of tmv in Fig. 5 is arbitrary (a/u).

B. Chemiluminescence Technique

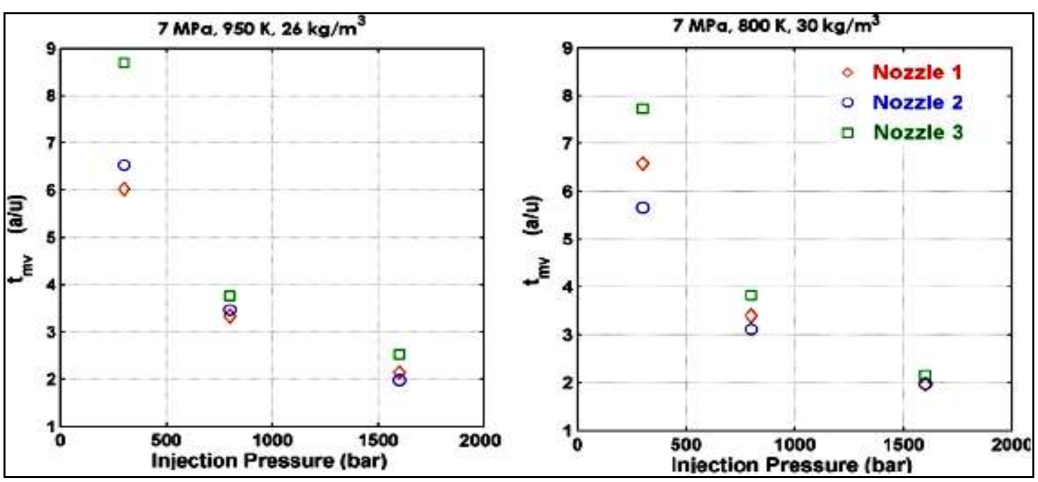

Fig. 5: Time to Mix and Vaporize in Terms of Injection Pressure

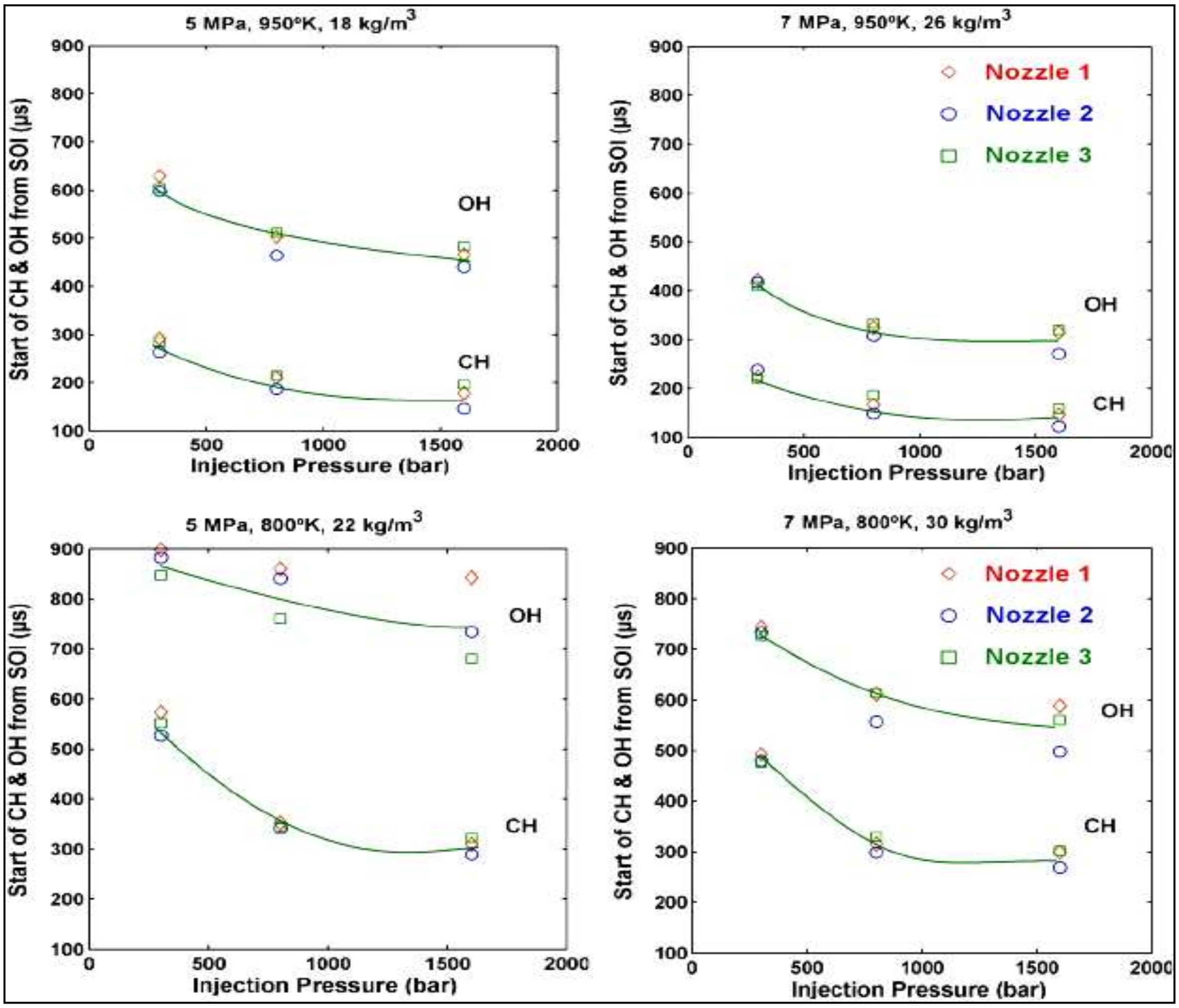

Fig. 6: CH- Appearance (below) and Ignition Delay (Above)

V. ANALYSIS OF COMBUSTION

A. Pre-Reactions Study

As it has been said before, CH-radicals are an indicator of low temperature reactions that only could take place once the fuel is completely mixed with air and vaporized. Frst appearance of CH-radicals seems to be located next to the liquid spray. It is supposed that CH-radicals should appear beyond the position of the stabilized liquid length. Nevertheless, CH intensity is detected before liquid phase penetration reaches LL value. In order to make the deeper analysis is about CH-radicals and spray penetration, theoretical time of mixing and vaporized and time of appearance of pre-reactions are compared in Fig 7.

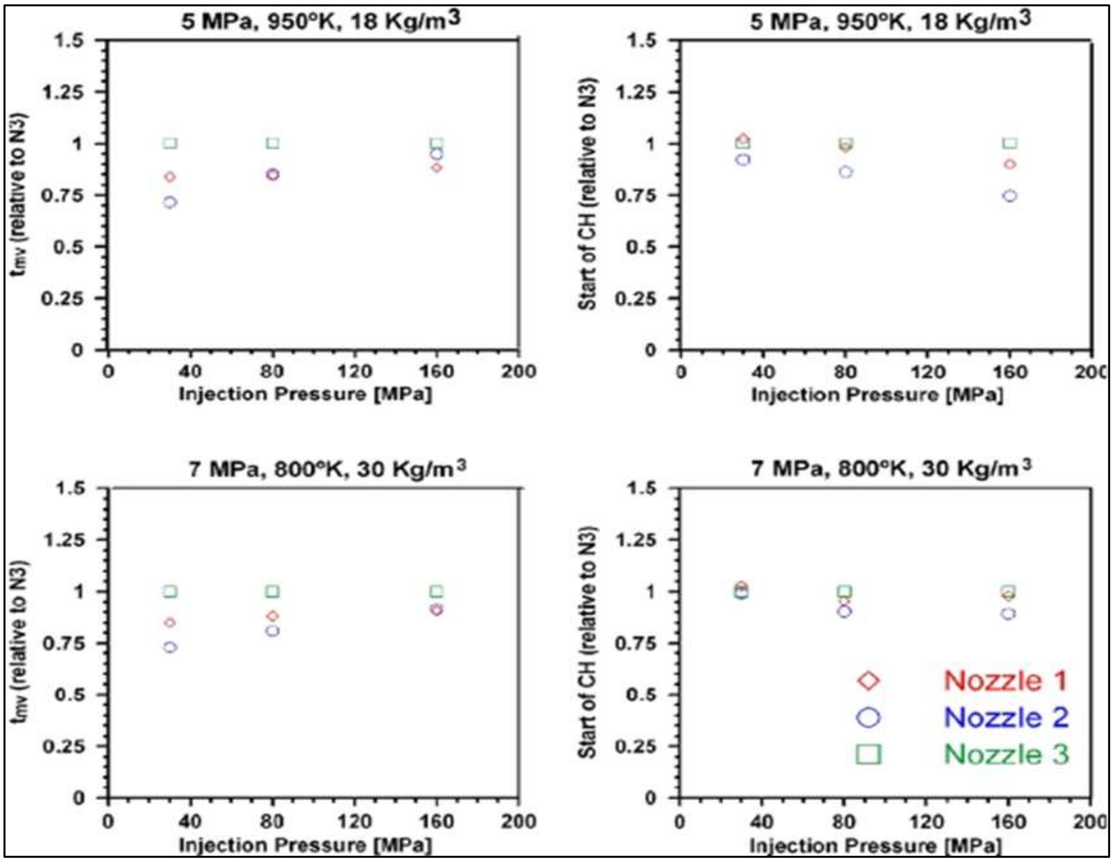

Fig. 7: Comparison between Time to Mix and Vaporize and CH-Appearance

These values are relative to Nozzle 3 behaviour, in order to make non-dimensional analysis. It can be seen that, in almost all the cases, relative values are lower than 1, which indicates that cylindrical nozzle needs, higher times both for vaporizing and appearing CH-radicals. In the same sense, Nozzle 2 shows the lowest relative values for the two parameters presented in the plot .These facts indicate the relationship between vaporization and pre-reactions.

B. Ignition Delay Correlation

As we know , OH chemiluminescence can be used to defined ignition delay, because OH radicals production are a result of the high temperature reactions present in the flame front. Several authors have tried to perform semi-empirical equations for predicting ignition delay as a function of engine conditions.

$$\tau = k \,.\, P_{back}^{n} . exp(\frac{E_A}{R \cdot T}) \tag{3}$$

Where τ, is the delay between start of injection and start of combustion, E_A the activation energy and T and Pback represent temperature and pressure in the combustion chamber.
Considering injection pressure above equation becomes

$$\tau = k \,.\, P_{back}^{n} . exp(\frac{E_A}{R \cdot T}) \,.\, P_{inj}^{m} \tag{4}$$

For this purpose, Pback and T were and temperature at Top Dead Center. All the pressures will be included in MPa, time of combustion in ms, and temperature in K. In order to simplify the statistical process, E_A /R was reduced in a common term, called A. The ignition delay is divided into chemical and physical delay. For chemical delay consider chamber conditions are taken into account and for physical delay controlled by ΔP (Pinj -Pback). But tmv is is linearly depends on effective diameter. For that following relation is used,

$$\tau = k \,.\, P_{back}^{n} . exp(\frac{E_A}{R \cdot T}) \,.\, P_{inj}^{m} \,.\, D_{eff} \tag{5}$$

If effective diameter exponent was fixed to 1, keeping the dependency observed for this variable in the theoretical analysis the following relation is made by considering the effect of K-factor.

$$\tau = k \,.\, P_{back}^{n} . exp(\frac{E_A}{R \cdot T}) \,.\, P_{inj}^{m} \,.\, D_{eff} \cdot (1\text{+K-factor}) \tag{6}$$

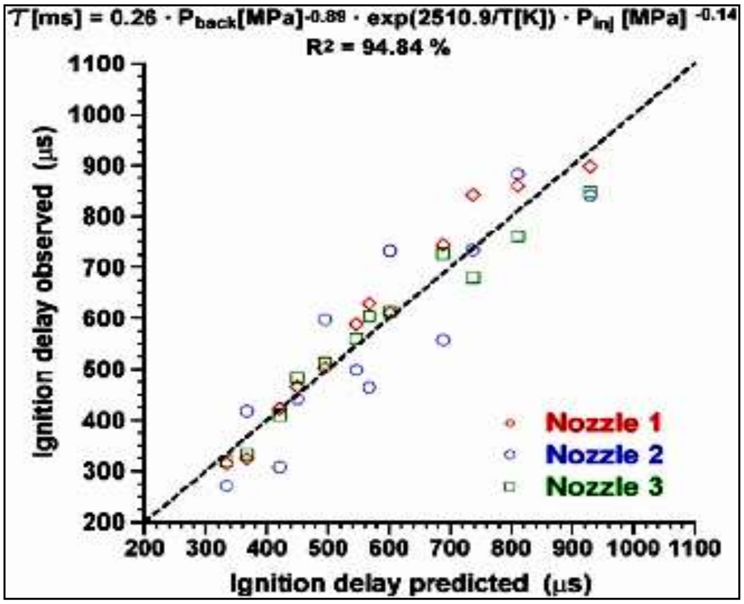

Fig. 8: Observed Vs. Predicted for Ignition Delay Correlation

VI. CONCLUSION

Topics covered in seminar shows that the effect of nozzle geometry on combustion process under different conditions. For this purpose, CH and OH chemiluminescence techniques have been used. Images were taken with an intensified CCD camera in an optically accessible engine, for three nozzles, differing on k-factor. Result are compared with previous geometric and hydraulic characterization, as well as liquid phase spray measurements, previously made in the same engine and for the same conditions.

REFERENCES

[1] Jian Gao, Seoksu Moon, Keiya Nishida, Yuhei, (2009, October). On flame structure of wall-impinging diesel fuel spray injected by group hole nozzles.ELSEVIER Journal of Combustion and Flame 156 (2009) 1263–1277, pp.1-4. Available:http //www.elsevier.com/locate/combustflame.

[2] Xiang Wang, Ke Li, Wanhua Su,.(2011, June). On Experimental an Numerical investigation on internal flow characteristics of diesel nozzle under real fuel injection condition. ELSEVIER Journal of thermal and fluid science 42 (2012) 204–211,pp1-2. Available: http // www.elsevier.com/locate/etfs.

[3] R. Payri, F.J.Salvador,J. Gimeno,J.de morena.(2010, February)On Effect of nozzle geometry on direct injection diesel engine combustion process. ELSEVIER Journal of Applied Thermal Engineering 29 (2009) 2051–2060,pp.1-8. Available: http // www.elsevier.com/locate/apthermeng.

[4] Prashanth K. Karra. (2010, February).On Experimental Study on Effects of Nozzle Hole Geometry on Achieving Low Diesel Engine Emissions", ASME Journal of Engineering for Gas Turbines and Power Vol. 132 / 022802-1,pp1-2. Available: http://www.asme.org/terms/Terms_Use.cfm.

[5] Tong Hao,, Li Yonga, Zhang Long, Li Baoquan,.(2012, July). On Mechanism design and process control of micro EDM for drilling spray holes of diesel injector nozzles", ELSEVIER Journal of Precision Engineering37 (2013) 213– 221,pp1-7. Available: http // www.elsevier.com/locate/precision.