EXPERIMENTAL INVESTIGATION OF THE EFFECTS OF FUEL INJECTION PARAMETERS ON DIESEL ENGINE PERFORMANCE AND EMISSIONS

Younis Jamal[1], Asad Naeem Shah[1], Wasif Amin Butt[1], Ahmad Naveed[1*], Muhammad Usman[1]

Abstract

The climate changes and increase in global temperature are the key factors that have mainly amplified the exploration studies of changing fuel injection and other constraints of compression ignition (CI) engines for the abatement of exhaust emissions. In current study, a direct injection (DI), CI engine was run on a test bench for the performance and emission analyses using different nozzles and injection timings. During the experiments, two types of nozzles known as sac and valve covered orifice (VCO) were used with hemispherical cavity and toroidal cavity pistons, respectively. Besides an already existing set of sac type nozzles, six distinct combinations of nozzles with varying cone angles and tip penetration (protrusion) lengths (designated as 135° × 3.5 mm, 140° × 3.5 mm, 145° × 3.5 mm, 150° × 3.5 mm, 150° × 2.5 mm, and 150° × 1.5 mm) were used at three different injection timings comprising 16° before top dead center (BTDC), 13°BTDC and 10°BTDC. Experimental results reveal that VCO nozzles in toroidal combustion chamber (CC) are better than sac nozzles with hemispherical CC, and that wider cone angle nozzles at 10° BTDC give the optimum results in terms of emissions and performance, relative to those of narrower cone angles. The 150° nozzles with 1.5 mm tip penetration give abated carbon monoxide (CO), hydrocarbon (HC) and smoke emissions along with better performance characteristics such as brake specific fuel consumption (BSFC) and brake power (BP), while exhibit slightly higher oxides of nitrogen (NO_x) relative to other combinations. Moreover, the same combination also proves to be effective on emission control at 8 mode steady-state cycle.

KEYWORDS: *Compression ignition engine, regulated emissions, injector nozzle, tip penetration, injection timing*

INTRODUCTION

During the last couple of decades, the automotive sector has paid its attention on the improvement of advanced technologies related to emission control due to the escalating dynamic market trends for successfully reducing the emission problems along with better fuel economy. It is important to have an atmosphere within environmental standards for industries. For this purpose, different manufacturers and engineers are making their efforts to advance distinct strategies in operations for reduction in exhaust emission of petrol as well as diesel engine. Moreover, the automotive manufacturers have imposed strict standards related to emissions for making advance engine models to comply with them.

Researchers are working hard for decreasing the harmful gases present in the exhaust of a diesel engine including the alteration in engine. The design of combustion chamber and injection nozzle for fuel spray are the two important parameters to decrease exhaust emission along with improved fuel economy. This is a very serious and challenging target for the automotive engineers to optimize the emissions with fuel economy and higher power.

The geometrical parameters of injector nozzle have been reported to affect the transient behavior of fuel spray jet formation for various conditions of pressures such as needle opening, ambient pressure and injection pressure[1,2]. Also the spray patterns are directly influenced by different parameters for example the spray orifices and cone angles with their shape, length and diameter, the particular position of the fuel injector with respect to piston forming combustion chamber and lift of needle [3]. The formation of spray jet as well as its impingement depends on these designed variables which show their direct effect on exhaust of engine.

The number of spray orifices that forms a jet is very important as for as air fuel mixture is concerned. There exists a direct relation between homogeneity of fuel-air mixture and number of orifices. The combustion process during such condition will produce less exhaust emissions[1,3,4].

1*Department of Mechanical Engineering, University of Engineering and Technology, Lahore 54000, Pakistan.

Sanuri and Ismail[5] have explored the performance of diesel engine by considering the effect of diameter of spray orifices with its numbers on indicated power, indicated torque and indicated specific fuel consumption (ISFC). They depicted that nozzle with five holes was better for ISFC, indicated power and torque for a wide range of engine speed while seven holes nozzle was best optimized at low engine speeds.

In order to decrease the diesel engine exhaust emissions, flow through nozzle has also been studied by few researchers[2,6]. The specific fuel consumption and engine power is greatly affected by the cone angle of nozzle that also determines the pattern of air-fuel mixture[7,8,9]. The penetration length of fuel jet is inversely proportional to the cone angle. Sayin and Canakci[10] improved BSFC with less smoke by increasing fuel injection pressure.

A small residual volume at tip of Sac type injection nozzle produces more fuel droplets to be dribbled in the combustion area at the end of main injection. This surplus amount of droplets increases HC emissions. The VCO (valve covered orifice) type injection nozzles are sac-less and for these covering of spray holes is done directly by needle valves. However, VCO nozzles produce more ignition delays along with higher intensity flames when compared to Sac-type nozzle under same operating conditions[4].

The shape of piston combustion chamber describes flow characteristics such as turbulence and swirling. The re-entrant shaped toroidal combustion chambers are better for air fuel mixture as compared to open combustion chambers.

Emissions and performance characteristics of compression ignition engine was experimentally investigated by Raja et al[11]. For this experiment, oil and different blends of rice bran with diesel (pure fuel) were used. For obtaining a reentrant and toroidal configuration, several new shapes of combustion chamber were developed. Pistons used for the experiment already existed or some modifications were made in existing pistons. After experimentation, it was revealed that for each blend, the amount of hydrocarbons and CO emissions was decreased. Changing geometry of combustion chamber had a significant effect on it.

NOx are have relatively higher potential threat to surrounding among other emissions from exhaust of diesel engine. One of the effective techniques to control these emissions is by retarding the injection timing[12,13]. In order to determine NOx emission stoichiometry and flame temperature are considered most important factors. By retarding the injection timing from 38^o to 34^o before top dead center (BTDC), the observed decease in exhaust emissions and brake specific fuel consumption was 40% and 6% respectively[14]. By retarding injection timing NOx emissions are reduced at sacrifice of brake specific fuel consumption and increase in smoke emissions[15]. Effect of injection timing on NOx emissions in a low heat rejection type turbocharged DI diesel engine is experimentally determined by Buyukkaya and Cerit[16]. They revealed that by retarding injection time from 20^o to 18^o at constant load and speed, NOx emissions and BSFC was decreased by 11% and 2% respectively.

In this work an effort is made to have deep understanding of effect of fuel injection nozzle parameter, combustion chamber design and injection timing on gaseous emissions i.e. CO, NOx and HC, brake power, smoke, torque and BSFC. Without effecting engine performance and meet emissions legislations, optimum combination of modified piston, fuel injection nozzles and combustion chamber is required. According to authors', no investigation of gaseous emissions and performance parameters of CI engine has been made by simultaneously varying the combustion chamber configurations (toroidal and hemispherical shaped cavity) and fuel injection nozzle geometric parameters (injection pressure, flow rate, cone angle). Thus, aim of this work is to study the effects of above mentioned parameters on emissions of DI-CI engine and it performance. It will also provide guidance for future stringent regulations deemed to be imposed on such an efficient prime mover.

MATERIALS AND METHODS

Test Engine, Dynamometer and Fuel

In this work, an experiment was performed on four stroke, three cylinder, 2.5 Littre, CI, naturally aspirated (NA), water cooled, (AD3.1524, made by Perkins) engine. The test engine was bore * stroke = 91.44 mm * 127 mm; compression ratio = 16.5:1; maximum torque = 171 Nm at 1400rpm; maximum power = 34.6kW at

2250 rpm. The experimental setup is shown in the Fig. 1. The dynamometer has inflexible base plate that taking shaft-rotor assemblage inside metallic casing. When crankshaft of engine will rotate, the shaft coupled with rotor will also rotate at same speed that would cut the magnetic lines. As a result of that eddy currents will be produced that would oppose rotation of the shaft in clockwise direction. It is because eddy currents apply an electric load which further leads to heat generation. This heat is then released by heat dissipation units and cooling water circulating through pipes. The speed is measured by an electromagnetic pick up device fixed on toothed wheel.

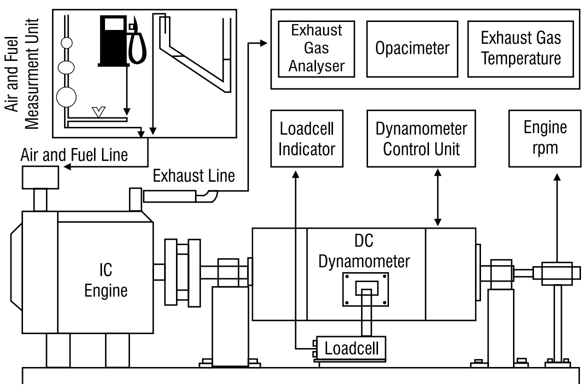


Figure 1: Schematic diagram of experimental setup

In this study, ultra-low sulfur diesel (ULSD) is used as fuel. It has 0.035% sulfur contents by mass. The reason of use of ULSD is to study the behaviour of D3.152 diesel engine and its emissions while varying the piston combustion chamber and nozzle parameters. The properties if ULSD are provided in table 1.

Table 1. Fuel Characteristics

Characteristics	Limits & Units	Test Method ASTM
Cetane Number	Min. 51	D613
Cetane Index	Min. 46	D976
Density @ 15°C	0.820-0.845 Kg/m3	D1298
Distillation	360	D86
Flash Point	Min. 50°C	D93
Kinematic Viscosity @ 40°C	2.0-4.5 cSt	D445
Sulfur content	0.035 %	D1266
Ash content	Max. 0.01	D482
Oxidation stability	25	D2274

Measurements of pollutants and other parameters

To measure nitric oxide (NO), HC, Carbon monoxide (CO) and nitrogen dioxide (NO_2) emissions from exhaust manifold of engine, an exhaust gas analyzer (Model NOVA 7466LK) was used. Data recording was done by inserting long sensitive probe of portable emission analyzer in exhaust manifold.

Using single non-dispersive infra-red detector (NDIR), the HC emissions were detected. To detect NO, NO_2 and CO emissions long life electrochemical sensor was used. These sensors are stimulated by presence of NO, NO_2 or CO and produces small current. This current is proportional to amount of NO, NO_2 or CO in ppm present in the sample. Sample gas is drawn in the analyzer through pump via pre-cooler and sample hose. In analyzer, the sample is passed through final filter. After final filtration, it flows through Teflon liquid block, flowmeter and infrared detector.

Smoke number is measured which ranges from 1 to 10 FSN using a smoke meter. The model of smoke meter is EFAW 68A, by Bosch Ltd. The working principle of smoke meter is absorption of light. Its sampling probe is flexible which is inserted in exhaust pipe to collect gas. Scale of smoke meter is divided into 10 divisions (0 to 10). Reading 0 stands for white disk and 10 stands for disk which absorb all the light.

To find diesel injection timing both at full load and rated speed was measured by diesel injecting timing light. Its model was MT2261 manufactured by Snap-on Ltd.). Correct fuel injection timing relates with position of piston on static timing (intial/base) before the centrifugal advance. With increase in engine speed, the auto advance mechanism of fuel injection pump advances the timing. In the meantime, with decrease in engine speed, injection time is retarded as well. Engine loading also changes the injection timing. When engine loads are high, injection timing gets advances and vice versa.

During the process of combustion, the peak in-cylinder pressure was also recoded. Kistler type pressure sensor (model 6125B) was used for recording the pressure. It was adjusted on the cylinder number 1head (top). It was hypnotized that the pressure inside 2 number cylinder and 3 number cylinder will be same as pressure measured in cylinder number 1.

Types of nozzles, fuel injection angles, and combustion chambers

Three sac-type nozzle with internal radii of 0.5mm were used in baseline engine along with open bowl hemispherical piston cavity and as a result metered amount of diesel fuel in combustion chamber is injected. However in modified engine with toroidal piston cavity , seat-hole type nozzles (with varying spray cone angles, flow and protrusions) or VCO are used. Injecton axis of 25^o from piston top surface (piston crown) is used while mounting and installation of all the nozzles in engine cylinder head i.e (axis of each nozzle is 25^o offset from piston centre). Needle opening pressure of 200 bar was used in case of 4-hole sac type nozzle with 0.28mm diameter , and of 300bar for 5 hole VCO type nozzles with 0.23mm diameter. Fuel cterisflow rate of 0.85L/min was used in sac type nozzle and 1.2L/min for VCO type nozzle. However, the other characteristic of nozzles are identical.

The influence of seat-hole nozzles in baseline engine is determined by procuring four sets of nozzles (each one comprising of three nozzles) varying with respect to cone angles, from a company, Delphi. Tests were performed on the basis of existing sac-hole nozzles geometrical parameters and finally these fuel injection nozzles were selected. Combustion chamber and piston top were designed and developed again so that the VOC nozzles were accommodated for performing experiments with them. The piston shape was changed to toroidal (re-entrant bowl type) from hemispherical shape (open bowl type) based on the design considerations of seat-hole sac and fuel injection nozzle i.e. spray cone angle and nozzle orientation. AutoCAD was employed for early drawing of toroidal and hemispherical shaped pistons and finally 3D modeling was performed using CATIA. Deep discussion related to combustion chamber and piston designing are beyond the scope of this study, so not discussed here.

Washer thickness below the nut seat of injector was changed and as a result protrusion of fuel injection nozzle inside the combustion chamber was changed and their effects on performance and emission were analyzed. Nozzle tip penetration length of 3.5mm. 2.5mm, 1.5mm were achieved by washer thickness of 2mm, 3mm and 4mm.

High pressure streams of fuel in three different pipes connected to fuel injection nozzles were produced by employing mechanical centrifugal governor and rotary distributor (DPA type) fuel injection pump. A timing device is fitted in lower side of the pump that cause the pump to have variable injection timing. According to engine manufacturer standard fuel injection pressure and injection timing are 450 bar and 16° BTDC respectively. Shims made of metal, present in between the spring and piston of fuel pump's timing device are varied in numbers to cause injection timing to retard. Injection timing is retarded or advanced by 3^o BTDC, by shim of size 0.25mm thickness. For required injection retard of 16^o BTDC , seven shims are required. And for retard of 13° and 10° BTDC, six and five shims are required respectively. At full load and rated speed condition, calibration of injection timing was done, after the shims were adjusted, using the diesel injection timing light (Model MT2261)

Scheme of experiments

In this experimental work following variables have been investigated: cone angle of spray nozzle, number of orifices of nozzle spray used, penetration of tip of nozzle, design of piston and combustion chamber and fuel injection timing. Other parameter of engine such as bore of cylinder, length of stroke, volume displaced, volume of air inducted, and intake pressure at the pump were remained same in the whole work. As the objective of research is to determine a combination of design parameters that should suitable for the minimum exhaust emissions and better performance of engine, and the optimum combination of such design parameters was studied through a numbers of experiments, and the same was then used to run the engine consistent with the C1 8-mode (ISO 8178 Type) test cycle in steady-state[17].

In the first part of experiments, we collected data on sac type nozzles with hemispherical piston cavity at 150° × 3.5 mm × 16° BTDC for the baseline measurements. Subsequently, we performed experiments with VCO type nozzles having toroidal piston cavity at 135° × 3.5 mm × 16° – 10° BTDC (cone angle× injection angle × tip penetration), 140° × 3.5 mm × 16° – 10° BTDC, 145° × 3.5 mm × 16° – 10° BTDC, 150° × 3.5 mm ×

16° – 10° BTDC for the test numbers of 1 to 4 respectively to find out the optimum cone angle. 150° was the optimum cone angle and the rest of sets of experiments were carried out at the same angle. During the next sets of experiments, ran the engine on VCO nozzles having 150°cone angle but tip penetration varying from length of 2.5mm to 1.5 mm, from 16° to 10° BTDC. For the 8 mode test during the second part of experiments, the optimum combination of nozzles with 150° × 1.5 mm × 10° BTDC geometric characteristics was utilized finally.

RESULTS AND DISCUSSION

Performance analysis

In the first part of experiments, we ran the engine on full load rated speed at three various injection timings of 16°, 13° and 10° BTDC for the performance and emission analyses. The comparison of performance between present diesel engine (Sac nozzles, 4-hole, with pistons hemispherical) and improved engine (5-hole VCO nozzles having variable geometric parameters with toroidal cavity pistons) is provided in terms of following design parameters:

a) Brake specific fuel consumption

Figure 2 represents the variations of BSFC for nozzles (VCO and Sac) at three various timings of injection (16°, 13° and 10° BTDC). The minimum brake specific fuel consumption was studied with nozzle having 150° cone angle and with the length of tip penetration of 1.5 mm. However, nozzles with 145 and 150 degree with 3.5 and 2.5 mm tip of length produced same results as produced by nozzles at BTDC 13° and 10° of Sac type. Moreover, the higher BSFC was showed by the 135° and 140° nozzles. The BSFC increases slowly by decreasing the timing of injection. When the injection timing retards, engine power decreases. Since the brake specific fuel consumption is the ratio of fuel intake to the brake power, thus BSFC increases.

b) Brake power

Figure 3 represents the variations in brake power at three different injection timings. We know that at higher loads, engine needs more BP or torque to perform the particular work. However, when the fuel injection timing retards, BP reduces because less fuel is required owing to a shorter ignition delay period. Therefore, all the nozzles show the decreasing trends of BP with decrease in injection timings, without any substantial relative difference among them. This shows there is not a significant penalty in BP using VOC nozzles, relative to the baseline sac nozzles.

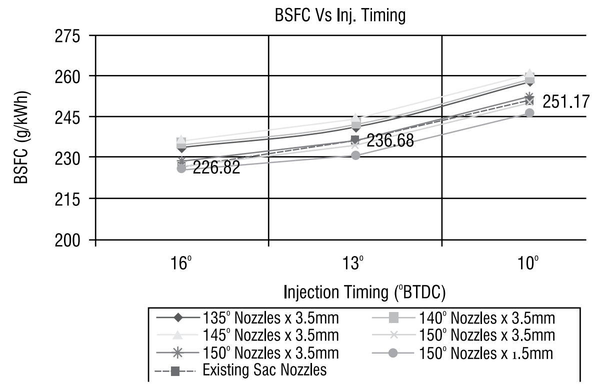


Figure 2: The variation in BSFC at different Injection Timings

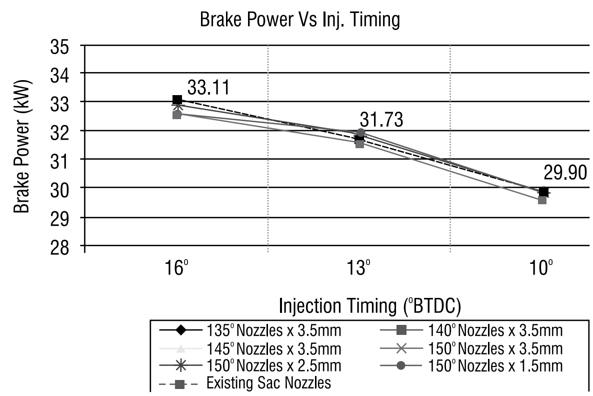


Figure 3: The variation in Brake Power at different injection timings

c) In-cylinder peak pressure

Figure 4 represents the change of in-cylinder highest pressure at three different injection timings. On retardation of timing, top cylinder pressure is decreased. Now it is clear that the peak pressure is also reduced by lessening injection timing. At the lower injection timings, the lesser fuel is burnt. Thus reduced pressure along with burning gases is formed. However, it is observed that various nozzles are behaving different in-cylinder pressures. Engine of 140° and 135° cone angle nozzles shows relatively higher pressure, while engine with 150° cone angle and length of 2.5 mm tip penetration shows the minimum cylinder pressure. The peak cylinder pressure makes the combustion environment more comfortable for producing higher concentration of NO_x because of higher temperature is expected to be developed under this situation[18]. Hence, the higher in-cylinder pressure is one of the significant parameters of the higher concentration of NO_x pollutants and vice versa.

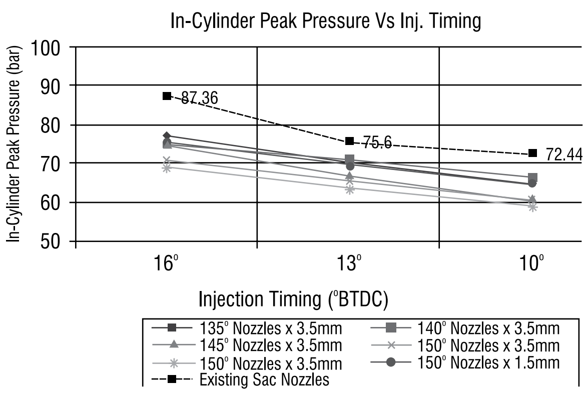


Figure 4: In-cylinder peak pressure at different injection timings

Exhaust emission analysis

a) NO_x emissions

NO_x are composed of nitric oxide (NO) and nitrogen dioxide (NO_2), and they are the most critical emission elements from compression ignition engines. The formation of NO_x is greatly reliant on subsequent three main features: temperature in cylinder, concentration of oxygen and the time for chemical reactions to occur in the combustion chamber[3,5]. At three different injection timings, the variations in the NO_x emissions for both types of nozzles are shown in Fig. 5. With the retardation of injection timings, the NO_x emissions are reduced. This is because by delaying the injection timing, the highest in-cylinder pressure is decreased, more fuel is required when the piston reaches to TDC and the peak temperature results in decrease. Thus, NO_x concentration is reduced at delayed injection timings. 150°Nozzles and 1.5 mm tip penetration showed the relatively higher concentration of NO_x among the VCO category. However, the emissions were reasonably lower as compared to the sac type nozzles. Similar results were given for advanced and retarded injection timing by Sayin and Canakci[10].

b) HC emissions

Figure 6 represents the HC emissions by 4-hole Sac and 5-holes VCO nozzles at three different injection timings. The HC emissions are relatively lower for VCO nozzles at standard injection timing of 16° BTDC. Because of enhanced turbulence, impingement area, the 5-hole VCO nozzles with toroidal formed combustion chamber pistons show improved combustion. On delayed injection timing of 10° and 13°BTDC, an increasing trend is there of HC emissions. This is because of highest cylinder pressure and temperature are less with the reduction of injection timing. Furthermore, time is short for air-fuel mixture formation in comparison to the modified injection timing so it appears in the formation of poor mixture and due to incomplete combustion would produce an unburnt emission of HC. The minimum hydro carbon emissions of 17.51, 12.64 and 12.37 g/h found with nozzles of135° on timings of injection of BTDC 10°, 13° and 16° accordingly amongst VCO nozzles. Weighty improvements in the decrease of HC emissions were shown by nozzles of 150^0 with tip protrusion of 1.5 and 2.5 mm tip. In literature similar concerns have been described of greater HC emissions at delayed injection timings[14,19].

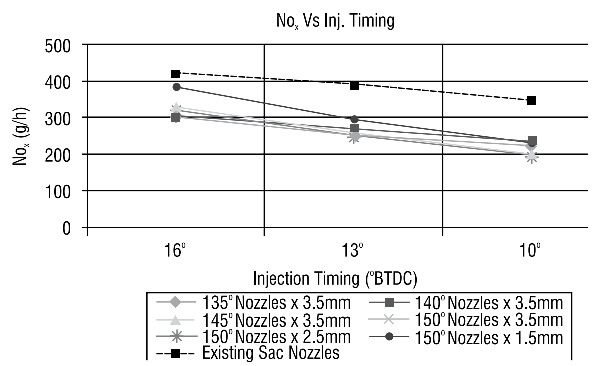


Figure 5: NOx emissions at different injection timings

c) CO emissions

Figure 7 represents the variations in CO emissions with existing Sac and VCO nozzles. At each of the injection timings, CO emissions are being contributed by the 4-hole Sac nozzles with hemispherical CC. It may cause in current combustion chamber partial air-fuel mixing. Comparatively improved outcomes due to enhanced efficiency of combustion of pistons with toroidal cavity were shown with VCO 5-hole nozzles.

CO emissions were minimum with the nozzles at cone angle of 150° with tip penetration lengths of 1.5, 2.5 and 3.5 at all the three timings of injection, and comparatively greater CO emissions by 140° and 135° nozzles are under the same conditions. The CO level is steadily increased by the delay of the injection timing, as the oxidation process reduces at retarded injection timing. Thus incomplete combustion takes place and it results an increase in the CO emissions. Furthermore, the lesser highest in-cylinder pressure and temperature can also be a cause of unsatisfactory combustion of diesel fuel at delayed injection timing with the consequential increase in CO emissions[20,21].

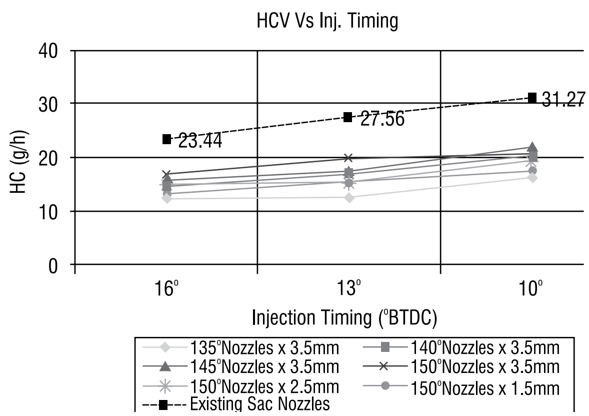


Figure 6: HC emissions at different injection timings

d) Smoke emissions

For VCO and Sac nozzles at three different injection timings (BTDC 10°, 13° and 16°) the variations of opacity of smoke are shown in Fig 8. The smoke rate by current Sac nozzles (i.e. 5.23 FSN) is considerably greater as associated to the VCO nozzles at ordinary injection timing of 16° BTDC. 135° and 140° nozzles show the maximum smoke value of 3.5 FSN among the VCO nozzles. Again, 150° nozzles with varying tip penetrations have been observed to show the minimum smoke emissions during all the three different injection timings. It is clear that the smoke values increase as the injection timing impedes. This is because of the reason that air-fuel blend does not have adequate time to make a complete flammable mixture. Furthermore, temperature within the cylinder is not suitable for the oxidation at late injection timing. Consequently, all these influences are liable for the development of rich mixture resulting in increased smoke emissions.

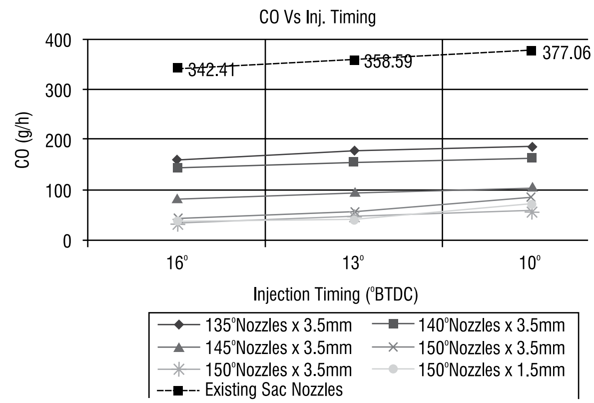


Figure 7: CO emissions at different injection timings

In accordance with the literature[10,19,22], the oxidation gets the sufficient time to promote with the help of advanced fuel injection and there is improved air-fuel droplets collaboration which finally takes to greater cylinder temperatures in the expansion stroke. Thus, ignition period is prolonged causing the smoke and soot emissions to be attenuated.

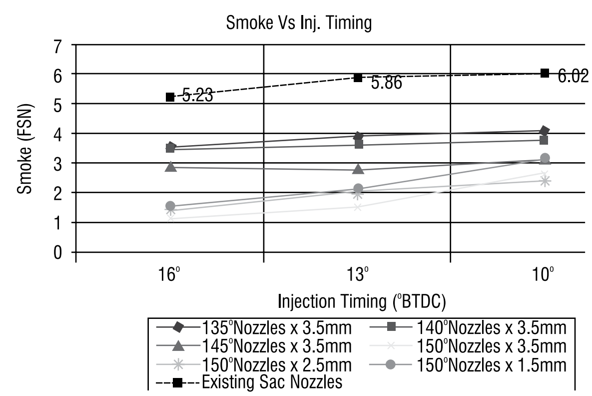


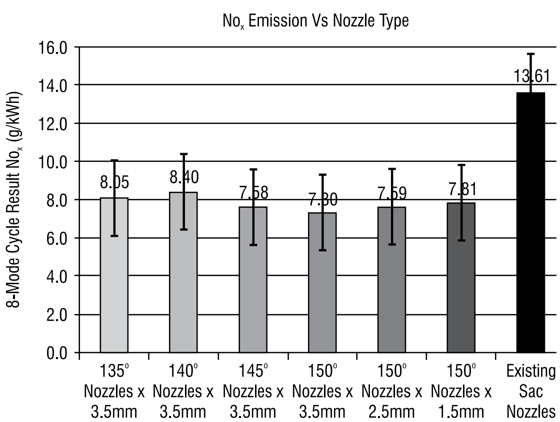
Figure 8: Smoke emissions at different injection timings

Analysis of Emissions at steady-state 8-mode test

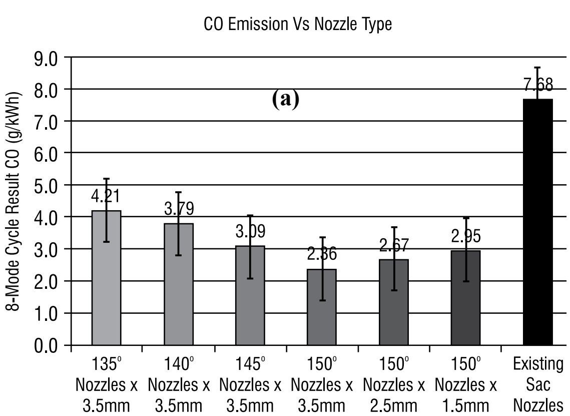
After examining at the condition of fully loaded with the rated speed on VCO nozzles containing 5 holes (chamber of toroidal shape) and 4-hole Sac nozzles (with combustion chamber of hemispherical shape) at three different injection timings, it is found out that only nozzles with VCO 5-hole are capable of meeting the EU Stage II standards of emission due to their best possible performance-emission characteristics. Furthermore, the NO_x contents in exhaust were least at the most retarded injection timing of 10° BTDC amongst the three injection timings. Consequently, 5-hole VCO nozzles (150° × 1.5 mm) were considered for testing at 10° BTDC for the second phase of experimentation, following the ISO 8-mode C1 8178 test cycle with different modes as given: 1 to 4 modes were chosen at rated speed of 2250 rpm considering engine loads in the sequence of 100, 75, 50, and 10%, and then engine was run at a speed of 1400 rpm with 100, 75, and 50% loads for the succeeding 3 modes. The 8th mode was an idle one. For the first three and 8th mode, weighting factor (WF) is 0.15; however WF is 0.10 for other modes of the test[17].

a) NO_x emissions

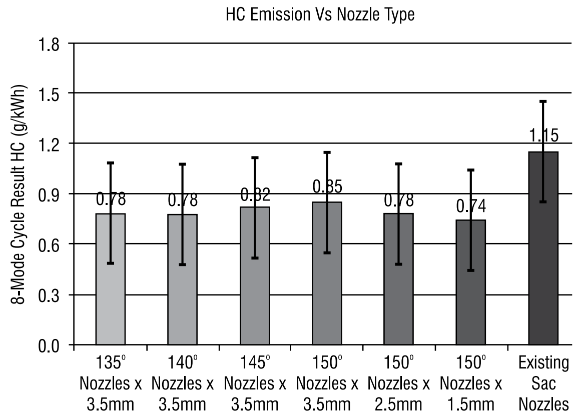
Figure 9(a) shows EU Stage II standards for NO_x along with NO_x 8-mode emission cycle results which is 8.0 g/kWh. The engine was fitted with 150° and 145° VCO nozzles shaping toroidal cavity pistons that meet the toroidal cavity pistons which meet the NO_x emission standards of Stage II.



(a)



(b)



(c)

Figure 9: The 8-Mode test results for (a) NOx, (b) HC, and (c) CO emissions

The existing baseline diesel engine has 13.61 g/kWh emissions of NO_x (70.12% greater than EU Stage II limit for NO_x emission). The shape of combustion chamber is hemispherical with 4-hole Sac type nozzles. NO_x emissions with 4-hole Sac nozzles are much greater as compared to VCO nozzles. The tip lengths of penetration of 1.5, 2.5 and 3.5 mm of 150° VCO nozzles among all show 7.81, 7.59 and 7.3 g/kWh emission levels of NO_x which are 2.38, 5.13 and 8.75% less than standard value respectively. While 8-mode results for nozzles with 145° are 7.58 g/kWh (which are 5.25% less than II-stage limits). However, emissions for 135° and 140° are greater than limits of II-Stage.

Thus the most appropriate nozzles in order to have lower NO_x emission level are having 3.5 mm tip, 150° cone angle with piston of toroidal cavity at optimum value of timing of injection of BTDC 10°. If the timing of injection is advanced to BTDC 16 or 13° BTDC, the engine cannot meet Stage II standards of NO_x emissions. The mitigated NO_x emissions are noted for 150° cone angle nozzles at 10° BTDC along with three tip penetration lengths. NO_x can be reduced further by using after treatment technology either selective catalytic reduction (SCR) or direct water injection on the mentioned combinations.

b) HC emissions

Figure 9 (b) shows result of the weighted brake specific HC emissions that were calculated according to ISO 8178 C18-mode cycle for all set of nozzles. Stage II EU standard of emission for brake specific HC depicts 1.5

g/kWh at test cycle of 8-mode. HC emissions are found well within the limits of standard when the engine was fitted with both VCO and Sac nozzles having toroidal and hemispherical type combustion chambers. But the lowest test results are observed with cone angle nozzles 150° with tip penetration 0f 2.5 and 1.5 mm when compared with 145°, 140° and 135° nozzles for the same speed and load conditions.

HC emission is 23.33% less than the standard value of 1.5 g/kWh for Sac type 4-hole nozzles with existing baseline diesel engine and combustion chamber of hemispherical shaped. However, VCO nozzles have much lower emissions than Sac 4-hole configuration of nozzles. This outcome clearly describes that VCO nozzles are in a healthier match with reentrant combustion chamber of toroidal type.

The nozzles of VCO type 150° cone angle with 2.5 and 1.5 mm length of tip penetration have 0.78 and 0.74 g/kWh HC level of emission which are 48 and 51% less than emission standards respectively. Similarly, the 8-mode cyclic results for 140° and 135° nozzles also reveal 0.78 g/kWh HC emission i.e.48% less than the Stage II limit. As a result, the 150° cone angle nozzles with 1.5 mm tip are the most appropriate with toroidal shaped chamber for lower HC emissions. The air fuel mixture is better now which improves the process of oxidation and comparatively complete combustion and thus lower HC contents in the exhaust. Out of many, one main advantage of VCO nozzles is that the needle valve presents in the body of nozzle, covers the spray holes directly at the end of fuel injection which benefits in a way that no extra fuel droplet gets into combustion space and hence considerable decrease in HC emissions is observed. But in Sac type nozzles working, surplus fuel droplets move inside the chamber after injection when needle dribbles. This is because of the small sac volume usually for fuel reservoir that is just under the needle tip.

c) CO emissions

The Figure 9 (c) illustrates weighted brake specific CO emissions for each set of different cone angle nozzles under the steady-state test. The standard value EU legislation limit for CO is 5.5 g/kWh for the 8-mode test cycle. The CO emissions are found within the range of Stage II standards for AD3.152 engine with toroidal combustion chamber fitted with VCO nozzles. However, 150° cone angle nozzles demonstrate the lowest cycle results when compared to 145°, 140° and 135° ones for the same speed and loading conditions.

The 150° nozzles with 1.5, 2.5 and 3.5 tip penetration lengths have 46.36, 51.45 and 57.09% less emission contents of CO when compared to standard limiting value in Stage II. In the same way, the 8-mode test results for 145°, 140° and 135° nozzles are 3.09, 3.79 and 4.21 g/kWh respectively which are 43.8, 31.1 and 23.5% less than the Stage II limit. It shows that 150° cone angle nozzles with 3.5 mm tip are best suited with toroidal combustion chamber as for as lower CO emission level is concerned. Again, better air fuel mixture giving better oxidation process and hence improved combustion efficiency that yields less CO.

CONCLUSIONS

A four stroke, three cylinders, direct injection, naturally aspirated, water cooled diesel engine was run on an eddy current dynamometer by coupling the engine crankshaft with the dynamometer rotor. All the experiments were performed at full load and rated speed of existing diesel engine (hemispherical cavity pistons, 4-hole Sac nozzles) and improved engine (nozzles with VCO 5-hole with pistons toroidal cavity type) with the retarding injection timings of BTDC10°, 13° and 16° BTDC for the performance and emission analyses. Subsequently, the experiments were carried out with the nozzles revealing the optimum characteristics following an 8 mode steady-state test. The experimental results show that by varying the VCO nozzle parameters (cone angle and tip protrusion) with modified piston and CC, small variations are observed in engine performance characteristics such as BP and BSFC at all the three injection timings of 16°, 13° and 10° BTDC. Though, comparatively greater in-cylinder peak pressure is observed with 150° injector nozzles, compared to other VCO nozzles. The VCO type nozzles contribute to the least regulated gaseous emissions including smoke without much affecting the BP, in comparison to Sac type baseline nozzles with combustion chamber hemispherical shaped. The nozzles with greater cone angle withstand noteworthy impingement with the toroidal shaped combustion chamber walls with the resultant uniform air to fuel mixing, and thus exhibit the optimum performance and emission results in comparison to nozzles with lower cone angles (135° and 140°), particularly, at injection timing of 10° BTDC. Moreover, injector nozzles with same cone angle of 150° but varying tip penetration lengths (3.5, 2.5 and 1.5 mm) show different results in each case. The 150° nozzles with 1.5 mm tip penetration show minimum CO and relatively lower HC and smoke emissions, while 150° with 2.5 mm penetration gives lower NO_x emissions. In addition to this, the former combination at 10°BTDC proves to be a suitable option to meet the criteria of 8 mode test.

ACKNOWLEDGEMENTS

Authors are thankful to the laboratory staff for their support in conducting the tortuous series of experiments. All the experiments were performed in the "engine test laboratory" of Engine Assembly Plant at Millat Tractors Limited Lahore Pakistan. The current work is the part of M.Sc. Thesis submitted by Wasif Amin Butt for the fulfillment of master in Mechanical Engineering under the supervision of Prof. Dr. Younis Jamal.

REFERENCES

1. *Koo, J.Y., (2003), "The effects of injector nozzle geometry and operating pressure conditions on the transient fuel spray behavior", J. Mech. Sci. Tech., Vol. 17, pp. 617–625.*

2. *Bakar, R.A., Ismail, S., Ismail, A. R., (2008) "Fuel injection pressure effect on performance of direct injection diesel engines based on experiment", Am. J. Appl. Sci., Vol. 5, pp. 197–202.*

3. *Soma, S., Ramirez, A.I., Longman, D.E., Agarwal, S.K., (2011) "Effect of nozzle orifice geometry on spray, combustion, and emission characteristics under diesel engine conditions", Fuel, Vol. 90, pp. 1267–1276.*

4. *Singh, I., Zhong, L., Chai, M., Lai, Henein, N., Bryzik, W., (2003) "Effect of nozzle hole geometry on a HSDI diesel engine-out emissions", SAE Tech. pap. Ser. No. 2003–01–0704.*

5. *Sanuri, S., Ismail, A.R., (2010) "Effect of injector nozzle holes on diesel engine performance", Fuel Injection, InTech Publisher ISBN: 978–953–307–116, Vol. 9, pp. 83–94.*

6. *Puhan, S., Jegan, R., Balasubbramanian, K., Nagarajan, G., (2009) "Effect of injection pressure on performance, emissions, and combustion characteristics of high linolenic linseed oil methyl ester in a DI diesel engine", Renew. Energ., Vol. 34, pp. 1227–1233.*

7. *Blessing, M., Braun, H., Koenig, G., Krueger, C., Raab, A., (2006) "Compression-ignition internal combustion engine", United States Patent Publication No.: US 2006/0243242 A1.*

8. *William, Primus, Bhaskar. (2008). "Diesel combustion system with re-entrant piston bowl", United States Patent Publication No.: US 7431012 B1.*

9. *Sono, H., Sasaki, N., Uchimoto, T., Ikeya, K., Tajima, Y., Shibata, M., Yamaya, Y., (2009) "Direct fuel injection diesel engine", United States Patent Publication No.: US 2009/0314253 A1.*

10. *Sayin, C., Canakci, M., (2009) "Effects of injection timing on the engine performance and exhaust emissions of a dual-fuel diesel engine", Energ Convers Manage., Vol. 50, pp. 203–213.*

11. *Raja, A.S., Rao, G.L.N., Nallusamy, N., Kumar, M.S.G., (2004) "Effect of combustion chamber design on the performance and emission characteristics of a direct injection (DI) diesel engine fuelled with refined rice bran oil blends", ASME Conf. Proc. Pap. No. ICEF 2004–0950, pp. 427–435.*

12. *Herzog, P.L., Burgler, L., (1992) "NOx reduction strategies for DI diesel engines", SAE Tech. pap. Ser. No. 920470.*

13. *Muramatsu, G., Abe, A., Furuyama, M., Yosdiha, K., (1993) "Catalytic reduction of NOx in diesel exhaust", SAE Tech. pap. Ser. No. 930135.*

14. *Parlak, A., Yasar, H., Hasimog lu, C., Kolip, A., (2005) "The effects of injection timing on NOx emissions of a low heat rejection indirect diesel*

injection engine", Appl. Therm. Eng., Vol. 25, pp. 3042–3052.

15. *Payri, F., Benajes, J., Arregle, J., Riesco, J.M., (2006) "Combustion and exhaust emissions in a heavy duty diesel engine with increased premixed combustion phase by means of injection retarding", Oil Gas Sci. Technol., Vol. 61, pp. 247–258.*

16. *Buyukkaya, E., Cerit, M., (2008) "Experimental study of NOx emissions and injection timing of a low heat rejection diesel engine", Int. J. Therm. Sci., Vol. 47, pp. 1096–1106.*

17. *ISO 8178-4, (2007) "Reciprocating internal combustion engines - Exhaust emission measurement -Part 4", Steady-state test cycles for different engine applications.*

18. *Shah, A.N., Yunshan, G., Shah, F.H., Mughal, H.U., Naveed, A., (2015) "Performance evaluation of continuously regenerating traps for controlling the carbonyl emissions from a turbocharged compression ignition engine", Iranian J. Sci. & Technol., Trans. Mech. Engg., (Accepted for publication).*

19. *Ganapathy, T., Gakkhar, R.P., Murugesan, K., (2011) "Influence of injection timing on performance, combustion and emission characteristics of Jatropha biodiesel engine", Appl. Energ., Vol. 88, pp. 4376–4386.*

20. *Canakci, M., Usla, K., Sayin, C., (2008) "Influence of injection timing on the exhaust emissions of a dual-fuel CI engine", Renew Energ., Vol 33, pp. 1314–1323.*

21. *Canakci, M., Sayin, C., Ilhan, M., Gumus, M., (2009) "Effect of injection timing on the exhaust emissions of a diesel engine using diesel–methanol blends", Renew Energ., Vol. 34, pp. 1261–1269.*

22. *Suziki, T., Tsujita, M., Mori, Y., Suzuki, T., (1986) "An observation of combustion phenomenon on heat insulated turbo-charged and inter-cooled D.I. diesel engines", SAE Tech. Pap. Ser. No. 861187.*