EFFECT OF COMBUSTION CHAMBER GEOMETRY ON PERFORMANCE, COMBUSTION, AND EMISSION OF DIRECT INJECTION DIESEL ENGINE WITH ETHANOL-DIESEL BLEND

by

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In the present paper, the effect of combustion chamber geometry on performance, combustion and emissions of ethanol-diesel blend operated in direct injection Diesel engine is discussed. The main air motions are generated in the cylinder by the intake – induced swirl, the piston motion, and its geometry. The piston bowl is modified from traditional hemispherical combustion chamber to the toroidal (re-entrant) combustion chamber and operated with Neat diesel and 40% ethanol diesel blend to improve better evaporation and mixing during the compression stroke on a single cylinder Diesel engine. It is found that the toroidal combustion chamber creates better turbulence, squish, and swirl at high compression ratios of 19.5:1 compared to that of traditional one. Further, the combustion is significantly enhanced due to increased swirl. It is concluded that the brake thermal efficiency for toroidal combustion chamber is 33% and the peak pressure in the cylinder as well as peak heat release rate is also increased. Further, it is also concluded that 60% of CO emission, 20% of HC emission, 40% of NO_x emission, and 90% in smoke emissions were reduced for toroidal combustion chamber, compared to that of hemispherical combustion chamber.

Key words: *compression ratio, toroidal, hemispherical, combustion chamber, ethanol, emissions*

Introduction

In recent years, ethanol has proven a promising renewable energy source in promoting the use of domestic renewable resource to the high power Diesel engine, which can be locally produced. Ethanol fuel is comparatively less expensive than diesel and easily available. Usage of ethanol blend in Diesel engine is more convenient because of less modification to the engine, because it is a kind of oxygenated fuel. There are some complications in the use of ethanol in Diesel engine because of its low cetane number, long ignition delay, and high latent heat of vaporization, poor auto ignition capability and stability, corrosion, lubricity and wear protection [1]. This can be overcome by increasing the compression ratio (CR) and by varying the combustion chamber geometry to minimize flame travel distance and by increasing the knock limit.

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The combustion and emission formation processes in Diesel engines have a close relationship with the piston bowl geometry which can strongly affect the air fuel mixing before the combustion starts. It has the potential of achieving a more homogeneous and leaner mixture close to top dead center (TDC) by effective turbulence, squish, and swirl flow. In-cylinder air motion in Diesel engines has a major impact on air-fuel mixing, which results in complete combustion [2]. Increase of turbulence, swirl, squish, and tumble components is highly depend on the design of the intake port, intake valve shape and geometry as well as shape of the combustion chamber and further it is influenced by fuel injection equipment (FIE) variables such as injection pressure, injection timing, injection rate, and number of nozzle holes. Better in-cylinder fluid motion enhances mixing of fuel and air for complete combustion and shortens the ignition delay [3, 4]. Those are the control factors to increase the fuel spray penetration, vaporizes, and mixing of fuel vapor with air entrainment [5].

An engine equipped with swirl supported combustion systems having injection duration of 25° to 30° crank angles has been used in this work. The number of injector holes must be optimized for the specific combustion system. The spray cone angle must be selected with regards to the specific combustion bowl design. The toroidal type bowl design requires a smaller cone angle than the hemispherical type of bowl design [6].

The main function of the bowl geometry would appear to be confining to the fuel during the flame propagation process. It is found that when the engine is operated at light load, the swirl flow improves specific energy consumption. The use of deep bowls is found to degrade homogeneous combustion significantly. Using their higher laminar flame speed will allow them to run with a leaner, or more dilute, air fuel mixture [7, 8]. The better volatility and the presence of more homogeneous mixture are preventing loss of brake thermal efficiency (BTE) at higher loads and fewer chances for production of smoke [9, 10].

The reduced NO_x emission reflects the lower in-cylinder temperature with the oxygenated fuel blend due to the reduced heat of combustion and leaner overall mixture in this combustion [11]. The solubility of ethanol in diesel is affected mainly based on HC composition, water content and temperature of ethanol. By using ethyl acetate and biodiesel as surfactant, the micro emulsion fuels can supplement diesel fuel in a compression ignition engine without any observed problem [12]. In general, using ethanol, CO, HC, and smoke emission increase with the increase of ethanol blends, and leads to lower cetane number and higher ignition delay [13]. Ethanol provides oxygen component and improves local air fuel ratio and thereby increases NO_x formation and on the other hand it suppresses NO formation because of its higher latent heat of vaporization and ignition temperature. Smoke formation is avoided by leaning the mixture or increasing air entrainment prior to ignition [14]. Diesel engines are designed to accommodate cetane number of above 40 and then increases with increase in ethanol percentage in ethanol diesel blend, and leads to engine knocking and incomplete combustion. To overcome this problem, CR and turbulence in the engine are to be increased or cetane improvers such as isoamyl nitrite and diethyl ether are to be added to ethanol-diesel blend [15]. The addition of ethanol causes to lower the viscosity and surface tension compared to that of diesel which improves vaporization and atomization and better mixing with air which leads to complete combustion [16]. All fuels give a rapid premixed combustion phase followed by a phase of diffusion combustion and the premixed combustion phase is more dominated when ethanol blend is used. The diffusion combustion phase becomes more dominant at higher engine load. Ignition is further retarded and most fuel burns during the expansion stroke when the ethanol volume fraction is greater than 79% [17].

Ethanol diesel blends have a much lower flash point than diesel fuel and higher vapor formation potential in confined spaces. It is also found that an increase in the swirl ratio can further reduce the specific energy consumption, which is attributed to improve the fuel transport, more rapid evaporation of fuel in the film on the piston-bowl surface and the reduced fuel dispersion. It is found that only 50% of the injected fuel is vaporized by 20° bTDC on compression. This indicates that the particular geometric configuration is analyzed and it would not be able to operate in the stratified charge mode [18]. It is necessary to set a minimum time interval between the ends of the fuel injection. It is necessary to set a minimum time interval between the ends of the fuel injection. It is found that the spray tip penetration and trajectory is strongly dependent on both swirl ratio and the combustion chamber geometry [19, 20].

The present work aims at to investigate two different combustion chambers of piston bowl geometry. Toroidal shape and hemispherical shape are introduced to increase the turbulence and swirl for effective combustion and to quickly spread throughout the combustion chamber of ethanol blends in Diesel engines. In order to increase the turbulence intensity in the combustion chamber, the piston crown is modified and to achieve a sufficient ignition temperature of ethanol blend, CR are maintained at 19.5:1. Effect of combustion chamber geometry on the performance, combustion and emission of the direct injection (DI) Diesel engine are studied and compared to that of 40% ethanol -diesel blend.

Experimental set-up and procedure

The pictorial view of the experimental set-up is shown in fig. 1. Experiments have been conducted in a single cylinder, water cooled, four 4-stroke DI Diesel engine made by Kirloskar. The engine develops 5.2 kW power at constant speed of 1500 rpm and is directly coupled with AG 250 model water cooled eddy current dynamometer. Experiments were conducted at a CR of 19.5:1. The figs. 2 and 3 show the piston bowl having toroidal (re-entrant) combustion chamber geometry and hemispherical combustion chamber geometry. Further, the engine is operated by ethanol diesel blends (E40).

Figure 1. View of the experimental set-up

Figure 2. Views of the piston bowl combustion chamber geometry

The main objective of this study is to investigate the influences of turbulence, squish, and swirl on toroidal (re-entrant) combustion chamber and hemispherical chamber on a single cylinder Diesel engine. The combustion and performance parameters are measured by computerized systems. The set-up is instrumented for combustion pressure and crank angle measurements. These signals interface with a computer through engine indicator for P-θ and P-V diagrams. The engine performance analysis software package named *Engine soft* is provided for online performance and combustion evaluation. A piezo-electric pressure transducer (PCB piezotronics) is used for recording the cylinder pressure for 20 consecutive cycles for combustion studies. The initial arrangements of the system are properly checked and test fuel is filled in the fuel tank. The engine is operated at different loads such

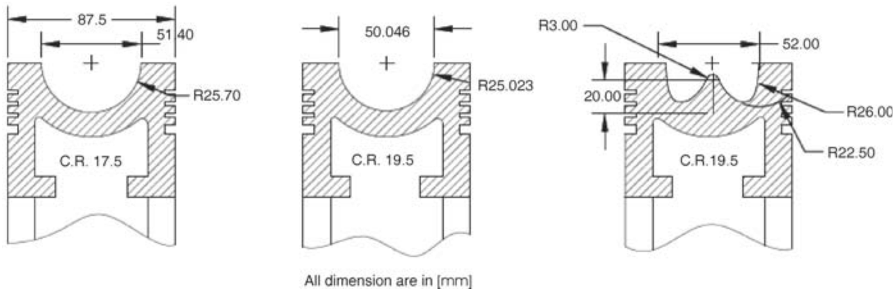

Figure 3. Detailed drawing of the piston bowl combustion chamber geometry

as NO load, 3 kg, 6 kg, 9 kg, 12 kg, 15 kg, and 18 kg. For each load, air flow, fuel flow, temperatures at water inlet and exhaust, Rota meter values are noted. The smoke level of the engine is measured by using an AVL 437C smoke analyzer and values of exhaust gases like CO, HC, CO2, NO_x, and O_2 were noted by using AVL Di Gas 444 gas analyzer. Previous values are noted for different piston bowl geometry and the same procedure is repeated for an E40 blend of ethanol. The performance and emission values are tabulated and compared with each other.

Table 1. Physical and chemical properties of ethanol-diesel

Properties	Diesel	Ethanol
Chemical formula	$C_nH_{1.5n}$	C_2H_5OH
Density [kgm^{-3}] at 20 ºC	830	789
Calorific value, [MJkg^{-1}]	42.700	26.800
Latent heat of vaporization [kJkg^{-1}]	230	840
Auto ignition temperature [°C]	250	395
Stechiometric air/fuel ratio, kg air/kg fuel	14.5	9.06
Flammability limit [%] volume	1.0-6.0	0.30-2.06
Cetane number	45-50	8

Preparation of ethanol-diesel blends

Physical and chemical properties of ethanol and diesel are shown in tab. 1. Since ethanol is azeotropic in nature, it fails to mix with diesel fuel after 15% by volume and offers a phase separation. In order to overcome this problem a surfactant or emulsifier (co-solvent) of the required quantity has been added to the blend. It helps to provide a stable blend avoiding phase separation. The quantity of surfactant or emulsifier required for providing a stable blend has been determined by the preliminary laboratory experiment. The surfactant, ethyl acetate of 2% (by volume) has been added to prepared stable blends. In the modified combustion chamber E40 performs well and shows better performance and lower emission. The E50 is ineffective because the experimental results show that lower performance, combustion and emission characteristics in Diesel engine due to the excessive reduction of required fuel properties. Hence, the maximum percent of ethanol utilization in DI application is limited to 40% by volume.

Results and discussion

Performance characteristics

Brake thermal efficiency

Figure 4 shows the variation of BTE with respect to brake power for ethanol-diesel blend of toroidal and hemispherical combustion chamber at CR 19.5:1. The BTE of ethanol diesel blend fuel is higher than that of diesel in both the combustion chamber geometries. It is also observed that the BTE are higher for a toroidal combustion chamber for E40 at all loads. The BTE strongly depends on the percentage of ethanol blend, CR and combustion chamber geometry. The heating value of ethanol exists about two third than that of diesel fuel. A large quantity of fuel is required to obtain the same power of diesel. Hence, better mixing of ethanol fuel and air can be achieved by producing toroidal combustion chamber bowl. The ethanol shows that combustion in an engine can be controlled by mixing rather than by vaporization and the oxygen atom in ethanol can improve combustion in higher load regions. Further, the droplet collision is higher for the toroidal chamber resulted in lower density of ethanol droplet. The increase in surface area of toroidal chamber causes an increase in vaporization, which plays an important role in the turbulent kinetic energy. Hence, the BTE gets increased during stable operation of the engine. Interestingly, at lower loads the BTE is increased for toroidal combustion chamber compared to the hemispherical combustion chamber for Neat diesel. This may be due to high vaporization and mixing of fuel and air through in cylinder swirl which contribute breaking up more fuel molecules, increase reaction kinetics which in turn reduce physical and chemical delay. This phenomenon can be obtained at lower load for optimum fuel air mixing. Consequently, combustion will occur early even at lower load, high suppression of knock is observed in lower load which could increase efficiency by reducing fuel requirements at lower load.

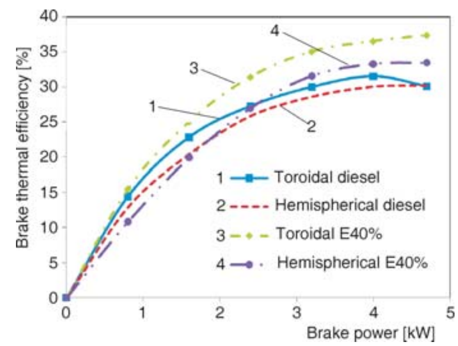

Figure 4. Variations of BTE for ethanol-diesel blend at various combustion chamber geometry

Cylinder pressure

Figure 5 shows the cylinder pressure *vs*. crank angle for toroidal chamber and hemispherical chamber pistons. It can be observed that the slight increase of the premixed combustion and earlier rise in the peak pressure are obtained for a toroidal combustion chamber for both diesel and ethanol blend. The ethanol shows a late start of combustion compared with diesel and this is due to increase in ignition delay resulting from a low cetane number and higher latent heat of vaporization of ethanol. The toroidal combustion chamber has a high swirl velocity, which creates an effective mixing of ethanol-diesel blend. The diesel fuel would initiate the ignition process than ethanol due to low self-ignition temperature of diesel. Ethanol fuel would evaporate more quickly than diesel since ethanol has a lower boiling point. The initiated auto-ignition of diesel would burn the fully evaporated ethanol than that of diesel. The toroidal combustion chamber has more positive effect with ethanol-diesel blend than that of hemispherical chamber. The premixed combustion phase becomes more dominant for higher ethanol blend

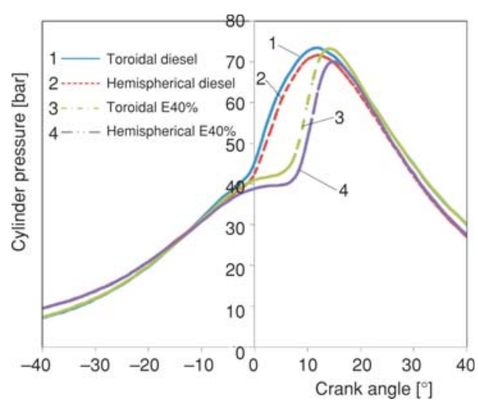

Figure 5. Variations of cylinder pressure for ethanol-diesel blend at various combustion chamber geometry

and at higher engine load than the toroidal combustion chamber. Ignition retards and combustion duration shortens due to rapid combustion of ethanol and more dominated premixed combustion phase. The turbulence level at start of injection is higher in re-entrant combustion chamber than hemispherical combustion chamber. This increased turbulence will improve the air entrainment under ignition condition and reduce the ignition delay, combustion mechanism, and chemical process. Further, the formation of in-cylinder pressure and temperature, enhance the air fuel mixing processes and improve the combustion process. The ethanol having lower density, surface tension and viscosity lead to better fuel atomization and mixing. The increased premixed flame causes a higher maximum rate of heat release, high rate of pressure rise and maximum combustion pressure. Improving mixing phenomena are responsible for accelerated rate controlled diffusion combustion, resulting in a decrease in the total combustion duration. Since the higher CR increases the swirl ratio from the starting of compression itself, this leads to better mixing during the expansion stroke and major part of the combustion of ethanol blend occurs on the premixed stage. Only a minor part of the combustion occurs in diffusive stage. This is due to higher surface area in torodial chamber and turbulent macro and micro mixing of fuel.

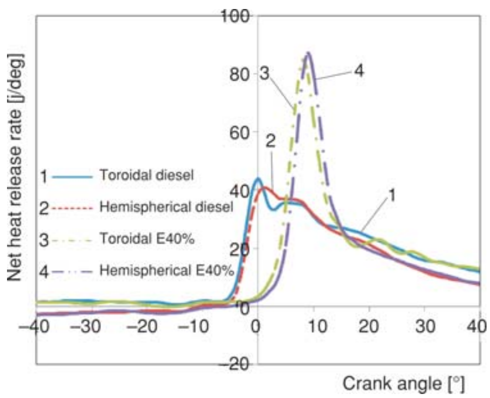

Figure 6. Variations of net heat release rate for ethanol-diesel blend at various combustion chamber geometry

Heat release rate

Figure 6 shows net heat release rate with crank angle. Here, the relative difference in heat release rate between toroidal and hemispherical is experimentally investigated. In general, the heat release rate and cylinder pressure increase with increase in load. In the toroidal combustion chamber, the combustion starts early for both diesel and ethanol diesel blend. It has a higher heat release rate for ethanol blend due to longer delay and it can be seen that the start of combustion is delayed especially for E40 ethanol blend having a lower cetane number. There is a reduction in the ignition delay and significant increase in heat release rate with higher turbulence. The CR breaks up the molecule chain by higher turbulence kinetic energy, thus the bulky amount of fuel is burned in the premixed burning phase. The re-entraining combustion chamber is compact for enhancing fuel air mixing, which shortens the flame travel distance resulting in increasing the burning rate and flame propagation. This result increases the heat release rate in the ethanol, diesel blend engine and so shortens ignition, and combustion duration. However, increase in the turbulence in the toroidal chamber generates high velocity gradient, pressure, and temperature. This influences in re-entry combustion chamber to increase the peak heat release rate of premixed combustion. This depends on the amount of mixture burned at this stage. The ethanol blend provides significant greater output thermal power than diesel. The heat release curve has a slight negative depression during the ignition delay period, which is mainly due to heat loss from the cylinder during the fuel-vaporizing phase. Due to low boiling point of ethanol, it has more cooling effect compared with diesel. Using re-entering piston, buoyancy is also responsible for the increased turbulence kinetic energy. This increases due to the swirl motion. This causes the increase in vaporous spray spreading angle, enhancing greater mass of liquid fuel being vaporized by air entrainment. The improved mixing is due to the increased momentum of the vaporized fuel jet, which can entrain more air and accelerated mixture formation and increased ignition delay are observed, leading to increased premixed combustion.

Emission characteristics

Carbon monoxide emission

Figure 7 shows CO emission in both the toroidal and hemispherical combustion chamber using ethanol diesel blend and Neat diesel. The CO emissions occur generally due to incomplete combustion, insufficient oxygen and lower in-cylinder temperature, which increase the ignition delay. The intermediate temperature regions where the OH radical concentration becomes significantly diminished and controlled by hydrogen peroxide (H_2O_2) results in less conversion of CO to CO_2. Using toroidal combustion chamber spray penetration distance decreases and increases turbulence with enriched combustion chamber temperature, results in short ignition timing, quicker the oxidation and reducing the CO emission. The C/H ratio of ethanol blend compared with diesel is lower and produces low CO and CO_2. The CO emission for ethanol, diesel blends increase remarkably. It can be observed that at lower load, latent heat of ethanol causes lower combustion temperature and results in a lower oxidation rate and higher CO emission. Even though ethanol has more oxygen in the fuel, the oxidation of atom takes place at higher combustion temperature.

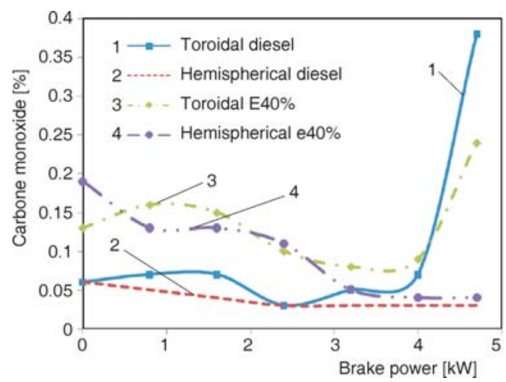

Figure 7. Variations of CO emission for ethanol-diesel blend at various combustion chamber geometry

It facilitates the complete reaction of all oxygen in the fuel at higher load and it drastically decreases CO. The CO emission is higher at lower and medium load for ethanol with toroidal chamber and lower at higher load conditions when compared to diesel fuel. At higher loads, CO emission is drastically reduced due to complete oxidation. However, in higher ethanol blend, combustion time retards and advances turned off to avoid engine damage. The lower heating value of ethanol blend has lower viscosity; high volatility and oxygen in the fuel, lower penetration length with higher cone angle break the molecules. Entrainment of fuel might lead to good decomposition and pre oxidization of the fuel and better mixing and leading to the more homogeneous charge. This results in reduced CO emission in reentry combustion chamber. Hemispherical combustion chamber has deep bowl and this leads to increased CO emission.

Hydrocarbon emission

Figure 8 shows HC emission with various combustion chambers for both Neat diesel and ethanol diesel blend. The main source of HC emission is increased by flame quench layer, the shorter timely arrival of the flame, crevices, and partial burn. The ethanol has a high latent heat of vaporization thus absorbed more heat and produces a high cooling effect in the combustion chamber causing more HC formation at lower load. Combustion chamber design plays an important role in HC emission. The compact combustion chamber reduces quenching thickness and faster motion of fuel air mixture to reduce HC emission. Under heterogeneous mixture, fuel may be too rich or too lean condition. The mixture is too lean to ignite within the flammability

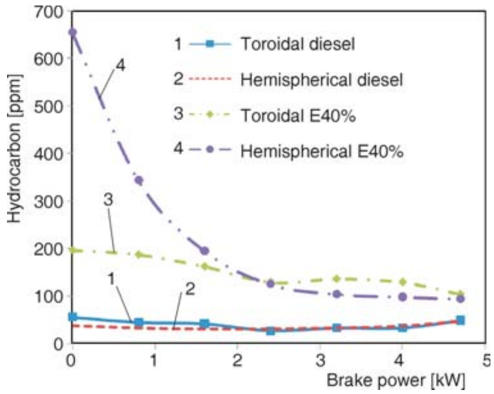

Figure 8. Variations of HC emission for ethanol-diesel blend at various combustion chamber geometry

limits, and hence produces HC emission. Ethanol is easy to be evaporated because of lower boiling point, viscosity and molecular weight than diesel fuel, which may cause better fuel atomization in too leaner mixture and longer ignition delay ensure more HC emission. With the increase of ethanol, unburned HC emissions are enhanced at lower load due to higher heat of vaporization resulting in an increase in ignition delay. Part of the fuel burns at expansion stroke which increases quenching layer thickness and form more HC emission. While using at higher load, toroidal chamber reduces the ignition delay causing most of the combustion to occur at premixed combustion which reduces HC emission. The HC emissions strongly depend upon fuel chemistry, increase of the cylinder charge air temperature and combustion chamber design. The density and angular momentum tend to move the fuel vapor inward towards the center of the swirl, which is beneficial for the reduction of HC emissions. Hemispherical chamber configuration minimizes the air fuel spray disruption, weak flow, additional time for mixing thus decreasing homogeneity at the time of ignition thus forming more HC emission.

Oxides of nitrogen emission

Figure 9 shows that it is possible to achieve high efficiency and low NO_x emissions by using high CR and lean mixture. The NO_x emissions increase rapidly when the load is increased. Increase in load leads to increased time available for combustion duration. This leads to complete combustion and increased temperature. At low and medium engine loads, the lower combustion temperature and more cooling effect of ethanol may result in a significant reduction of thermal NO_x formation. The NO_x production occurs mainly in the hot reaction zone around the periphery of the spray. It is observed that swirl can help to have very lean air-to-fuel ratios of the premixed burning and can improve air utilization and mixing flame propagation combust the mixture progressively and offers a lower flame temperature at all points of toroidal combustion bowl. This is the main reason for the lower NO_x emission. The low NO_x emission is achieved in the lean and diluents air, premixed mode by 12% in toroidal combustion chamber and 16% with respect to hemispherical chamber for diesel combustion where as toroidal is more effective than that of hemispherical for ethanol diesel blend. The hemispherical leads to substantially higher NO_x at the higher load because of the high temperature shift near the injector and more accumulated mixture. The main reason is that the heat absorption by ethanol vaporization

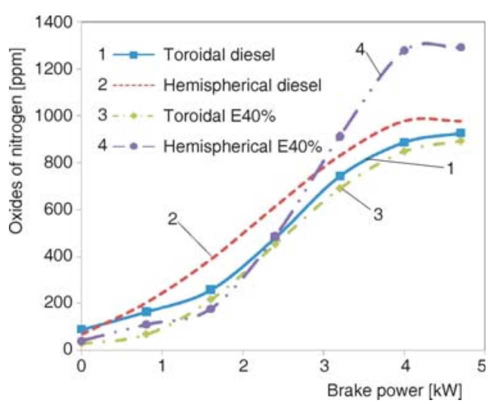

Figure 9. Variations of NO_x emission for ethanol-diesel blend at various combustion chamber geometry

causes a decrease of local adiabatic flame temperature and therefore reduces the chemical reaction in the gas phase to produce thermal NO_x.

Smoke emission

Figure 10 shows smoke emission for various combustion chambers for Neat diesel and ethanol diesel blend. Smoke depends on the air entrainment in the lifted portion of the jet, oxygen in the fuel, and structure of HC in the fuel. Ethanol blend composition and molecular structure play an important role in smoke formation. In general, presence of higher amount of oxygen and hydrogen molecules in ethanol-diesel blend reduces the emission of smoke. It is observed that in re-entry combustion chamber, greater entrainment of the air achieve the maximum degree of homogeneity with fuel. This enhances the turbulence squish and swirl close to TDC and it may result in better fuel air mixing which reduces the combustion duration, and hence pyrolysis of fuel is decreased which in turn increases the oxidation of fuel. Therefore, in toroidal combustion chamber, ethanol-diesel blend is burned in diffusion mode, which leads to reduced smoke formation. The use of toroidal bowl piston increases the hot surface area and produces particle less vapour immediately after the injection. This helps to establish charge homogeneity throughout the cylinder and assists auto-ignition which is an important reason for the improved combustion behavior and lower smoke emission. In hemispherical combustion chamber, the momentum and entrainment of fresh air of the jet is reduced. The imperfect fuel air mixing results in increasing the ignition delay. This causes more smoke in lean mixture. Lower cetane number of ethanol increases both fuel consumption and smoke emission at lower loads. At lower load, the cylinder temperature is also too low, resulting in bad combustion which in turn causes high smoke emission in a hemispherical combustion chamber. At higher loads, the oxygen atom in ethanol is more effective in reducing smoke formation.

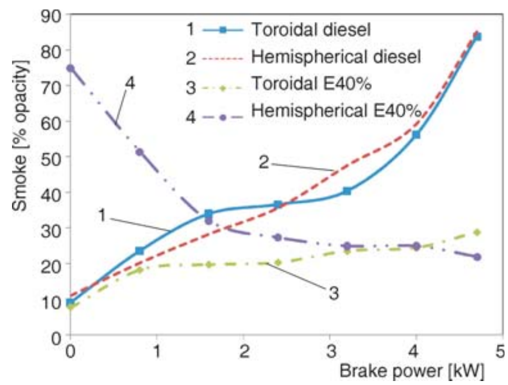

Figure 10. Variations of smoke emission for ethanol-diesel blend at various combustion chamber geometry

Conclusion

In Diesel engine, the BTE is reduced at lower load, but increased at higher load for ethanol blend. The engine operating with ethanol fuel shows the peak BTE of nearly 35%. It is observed that the absolute level of HC emissions from a Diesel engine can be quite low for ethanol diesel blend. It is observed that for ethanol diesel blends, HC increases by 17%, CO decreases by 9%, NO_x decreases by 2%, and smoke reduces by 30% relative to diesel fuel. The peak heat release rate of premixed combustion of ethanol diesel blend is higher than diesel. An increase in the swirl intensity of toroidal chamber reduces the temperature gradient due to increasing degrees of mixing which in turn reduces NO_x emission.

References

[1] Yilmaz, N., *et al.*, Some Perspectives on Alcohol Utilization in a Compression Ignition Engine, SAE 2005-01-3135, 2005

[2] Nandha, K., Abraham, J., Dependence of Fuel Air Mixing Characteristics on Injection Timing in an Early Injection Diesel Engine, SAE paper 2002-01-0944, 2002
[3] Li, J., *et al*., Effect of Piston Bowl Geometry on Combustion and Emission Characteristics of Biodiesel Fueled Diesel Engines, *Fuel, 120* (2014), Mar., pp. 66-73
[4] De Risi, A., *et al.*, Optimization of the Combustion Chamber of Direct Injection Diesel Engine, SAE paper 2003-01-1064, 2003
[5] Gebert, K., *et al*., An Evaluation of Common Rail, Hydraulically Intensified Diesel Fuel Injection System Concepts and Rate Shapes, SAE paper 981930, 1998
[6] Arcoumanis, C., Gavaises, M., Application of a FIE Computer Model to an In-Line Pump-Based Injection System for Diesel Engines, SAE paper 970348, 1997
[7] Matthew, B., *et al*., High Efficiency and Low Emissions from a Port Injected Engine with Neat Alcohol Fuels, SAE paper 2002-01-2743, 2002
[8] Prasad, B.V. V. S. U., *et al*., High Swirl Induction Piston Bowls in Small Diesel Engines for Emission Reduction, *Applied Energy, 88* (2011), 7, pp. 2355-2367
[9] Li., D. T., *et al*., Temperature Measurement in the Swirl Chamber of an IDI Engine Using More Deflectometry, *Applied Thermal Engineering, 19* (1999), 5, pp. 543-554
[10] Singh, B. P., Performance and Emission Characteristics of Conventional Engine Running on Jatropha Oil, *Journal of Mechanical Science and Technology, 27* (2013), 8, pp. 2569-2574
[11] Kannan., M., *et al.*, Feasibility and Performance Study of Turpentine Fueled DI Diesel Engine Operated Under HCCI Combustion Mode, *Journal of Mechanical Science and Technology, 28* (2014), 2, pp. 729-737
[12] Reed, H., *et al*., Piston Bowl Optimization for rcci Combustion in a Light Duty Multi Cylinder Engine, SAE paper 2012-01-0380, 2012
[13] Song, J., *et al*., Effect of Oxygenated Fuel on Combustion and Emissions in a Light Duty Turbo Diesel Engine, *Energy and Fuels, 16* (2002), 2, pp. 294-301
[14] Chandra, R., Kumar, R., Fuel Properties of Some Stable Alcohol-Diesel Micro Emulsions for their Use in Compression Ignition Engines, *Energy and Fuels, 21* (2007), 6, pp. 3410-3414
[15] Lapuerta M., *et al*., Effect of Ethanol on Blending Stability and Diesel Engine Emission, *Energy and Fuels, 23* (2009), 9, pp. 4343-4354
[16] Krisman, A., *et al*., On the Potential of Ethanol Fuel Stratification to Extend the High Load Limit in Stratified-Charge Compression -Ignition Engines, *Fuel, 99* (2012), Sep., pp. 45-54
[17] Liu., S., *et al*., The Effect of a Certain Number (CN) Improver on Combustion and Emission Characteristics of a Compression Ignition (CI) Engine Fueled with an Ethanol -Diesel Blend, *Energy and Fuel, 24* (2010), 4, pp. 2449-2454
[18] He, B.-Q., *et al*., Homogeneous Charge Combustion and Emission of Ethanol Ignited By Pilot Diesel On Diesel Engines, SAE paper 2004-01-0094, 2004
[19] Wickman, D. D., *et al*., Spray Piston Geometry Optimized for HSDI Diesel Engine Combustion, SAE paper 2003-01-0348, 2003
[20] Rakopoulos., C. D., *et al*., Investigation of Piston Bowl Geometry and Speed Effects in a Motored HSDI Diesel Engine Using a CFD Against a Quasi-Dimensional Model, *Energy Conversion Management, 51* (2010), 3, pp. 470-484

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