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A Comparative Study of Porous Media Combustion Chamber in Diesel Engine for Different Fuel Blends In Terms Of Performance and Emissions

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Abstract: *The direct injection type of diesel engines are important choice for heavy duty applications as on-road, off- road, marine, and industrial usage particularly, because of their high break thermal efficiency. Based on the regulations of exhaust emissions and fuel economy the new combustion technology development is introducing to improve the efficiency of the diesel engine. we have improved the performance of the diesel engine and reduced the exhaust emissions with reducing of its cost of the fuel by development of porous media (PM) combustion technology for different fuel blends in this paper. The Silicon Carbide porous media (PM) is placed beneath of the fuel injector inside of the combustion chamber for homogeneous combustion. We determined performance and emission characteristics of diesel engine at 80% diesel,15% karanja oil and 5% additives. In this research work ethanol, n-butanal, and diethyl either are considered as additives. The simulation work is also carried out by using the modified KIVA-3V code on performance and emission characteristics of diesel engine combustion with porous media..*
Key Words: *Porous Media, Bio Diesel, Blends, Ethanol, N-Butanol, Diethyl Ether, Kiva-3v*

I. Introduction

The environment norms and concerns about global warming search for cleaner fuel technology is driven by many factors including lack of availability of fossil fuels. For developing cleaner fuel and engines the quest has resulted in a number of technological trials and endeavors. Because of emission constraints we cannot compromise on the performance and efficiency of the engines in spite of the limitations. Being renewable in nature and decreasing effect on HC and CO emissions the alternative fuels, biodiesel has received ample attention due to their attractive characteristics . The use of biodiesel are lower engine power, higher BSFC due to their lower calorific values, higher densities and viscosities on the contrary, major problems. With the use of biodiesel for higher fuel bound oxygen the NOx emission also increases. The use of ethanol, *n*-butanol or diethyl ether in small proportion as additive has come out with great potential recently to overcome some of these difficulties. Similarly for efficient combustion of fuels many researchers have developed new methods. Due to the interaction between two different phases, solid and gas or liquid Porous media combustion, also known as filtration combustion in a packed bed. With exothermic chemical reactions during fluid flow in a porous medium the theory of filtration combustion involves a new type of flame. For combustion of gas flow through porous media the term 'filtration combustion' was introduced by Russian scientists. Still it can be found in special literature as a synonym to combustion within porous media (PM) [1] this term does not correspond to western scientific terminology. With stability in a wide range of reactant fluid velocities, air-fuel ratios, and power density this process facilitates a combustion process PM combustion has some unique characteristics. It gives rise to high flame speed and higher power density, high radiant output, low NOx (Oxides of Nitrogen) and CO (Carbon Monoxide) emissions.

In the literature there are a few studies, to prove its advantages in supporting Mixture formation and improving combustion processes [3–6] which are focused on liquid fuel combustion in PM. By using a collision probability and analyzed one-dimensional, self-sustaining combustion in inert porous media with all modes of heat transfer Martynenko et al. [4] numerically modeled the fuel droplet collision with a high porosity PM. The numerical analysis of combustion of liquid fuel droplets suspended in air inside an inert PM presented by Kayal and Chakravarty [3]. The liquid fuel vaporization–combustion system was developed based on a combined self-sustained. Newburn and Agrawal [4] was designed and experimentally studied for lean prevaporized, premixed combustion to evaluate its heat transfer and combustion performance through counter-flow annular heat recirculating burner fueled with kerosene. The porous media is provided by Mujeebu et al. [5,6] for latest reviews of liquid fuel combustion. Inserting a SiC PM into the cylinder head between the intake and exhaust valves in diesel engine by Durst and Weclas [7, 8] and proposed the concept of the PM engine and performed a systematic experimental study on a test engine. For all combustion events, such as fuel–air mixture formation, fuel vaporization,and homogenization, internal heat recuperation, as well as combustion reactions occurred inside the PM due to injected into the PM volume, and consequently. In comparison with the original one high cycle efficiency, very low emission level, and low combustion noise such results are demonstrated for many attractive characteristics of the PM. The working process of a PM engine fueled with methane and hydrogen respectively, and discussed some important issues concerning practical applications of the PM engine in a numerical study, based on a multizone combustion model by Macek and Pola´sˇsek [9]. Using a two-dimensional numerical model recently, Zhao and Xie [10,11] investigated the interaction between a pressure swirl fuel spray on hot porous medium, as well as the combustion characteristics of compression ignition PM engine, while the heat regenerative cycle in a PM engine is analyzed by Liu et al. [12,13] and based on single zone and two-zone thermodynamic models are evaluated on its thermodynamic performance. The regenerative engine which was proposed by Ferrenberg [14] is another design of PM engine configuration. A porous insert, functioning as a regenerator, is attached to a rod and moves in the cylinder in this regenerative engine, synchronized but out of phase with the piston. The porous medium mounted just beneath the cylinder head during the regenerative heating stroke for most of the period and moves down to the piston while it reaches the top-dead-center (TDC) position, air arrives in intake into the cylinder and is then heated by hot PM. The regenerator moves up during the regenerative in cooling stroke, until the next regenerative heating stroke remains in the original position. Hot exhaust gas flows through the PM after all the fuel has combusted, and delivers part of the reaction heat into the PM.

To prove its advantages in supporting Mixture formation and improving combustion processes [3–6] in the literature there are a few studies, which are focused on liquid fuel combustion in PM. Using a collision probability and analyzed one-dimensional, self-sustaining combustion in inert porous media with all modes of heat transfer by Martynenko et al. [4] numerically the fuel droplet collision with a high porosity PM. Kayal and Chakravarty [3] showed a numerical analysis of combustion of liquid fuel droplets suspended in air inside an inert PM. Combustion system and self-sustained liquid fuel vaporization was developed on model based. It was designed and experimentally studied by Newburn and Agrawal [4] for lean prevaporized, premixed combustion to evaluate its heat transfer and combustion performance through counter-flow annular heat recirculating burner fueled with kerosene. The system was shown to produce low emissions. Mujeebu et al. [5,6] is provided latest reviews of liquid fuel combustion in porous media. Proposed the concept of the PM engine and performed a systematic experimental study on a engine test by Durst and Weclas [7, 8] inserting a SiC PM into the cylinder head between the intake and exhaust valves. All combustion events, i.e., fuel–air mixture formation, fuel vaporization and homogenization, internal heat recuperation, as well as combustion reactions occurred inside the PM due to fuel was injected into the PM volume, and consequently. In comparison with the original one, such as a, high cycle efficiency, very low emission level, and low combustion noise, results demonstrated many attractive characteristics of the PM engine. Discussed some important issues concerning practical applications of the PM engine in a numerical study, based on a multizone combustion model, Macek and Pola´sˇsek [9] modeled the working process of a PM engine fueled with methane and hydrogen, respectively. Investigated the interaction between a pressure swirl fuel spray and a hot porous medium by Zhao and Xie [10,11], recently, using a two-dimensional numerical model, heat regenerative cycle in a PM engine was analyzed as well as the compression ignition and combustion characteristics of a PM engine. Analyzed the heat regenerative cycle in a PM engine by Liu et al. [12,13] and thermodynamic performance evaluated based on single zone and two-zone thermodynamic models.

Based on the PM engine concept proposed by Durst and Weclas [7,8], in which the PM insert is fixed in the combustion chamber as like above mentioned studies. The regenerative engine which was proposed by Ferrenberg [14] which is another design of PM engine configuration. The porous medium remains just beneath the cylinder head for most of the period and moves down to the piston during the regenerative heating stroke, while it reaches the top-dead-center (TDC) position, the intake air arrives in the cylinder and is then heated by passing through the hot PM insert. The regenerator moves up during the regenerative cooling stroke, and remains in the original position until the next regenerative heating stroke. Hot exhaust gas flows through the PM insert and delivers part of the reaction heat into the PM after all the fuel has combusted.

In the presence of a Porous Media (PM) inside the direct injection diesel engine for different fuel blends the primary objective of the proposed work is to model and simulate the combustion and emission process. It consisted 80% diesel, 15% karanja biodiesel and 5% additives are final ratio of improved blends. The idea was to maintain 20% of biofuel into the blends.. According to the Wiebe's combustion model for modeling the combustion, a modified KIVA 3 code is used in this paper.

Emission of NOx and Soot are also suitably modeled similarly. In the form of a modified KIVA code the modeled features are incorporated and the results of the simulation are presented. In regard to fuel consumption, the analysis is specifically carried out of brake power and emissions. At constant speed these analysis were carried out.

Fuel Blends

Most of the researchers define the quality of fuel [15-18] have concentrated their prosperities like density, kinematic viscosity, flash point, and calorific value. During the flow of fluid through various pipelines, nozzles and orifices density and viscosity are the most important parameters. Furthermore, they have great influence on the atomization of fuel which governs the quality of combustion as well as the performance and emission characteristics. The use of ethanol, *n*-butanol and diethyl ether as additives helped to decrease both density and viscosity of the bio diesel because of density and viscosity of biodiesel are higher than diesel. To improve the performance and emission characteristics [19, 20] the number investigations have been carried out on different proportions of ethanol in the biodiesel - diesel blends]. In diesel engine *n*-butanol is a strong alcohol competitor of ethanol as additive to be used which is also a biomass-based renewable fuel. *n*-butanol has higher cetane number, higher heating value, higher miscibility and less hydrophilic tendency than ethanol. Hence, *n*-butanol has got superior characteristics than ethanol to be used as additive. Diethyl ether, produced from ethanol which is another potential additive. Its having very high cetane number, high oxygen content, high miscibility and low auto ignition temperature, in diesel and broad flammability limits. It consisted 80% diesel, 15% karanja biodiesel and 5% additives are improved final ratio blends. The idea was to maintain 20% of biofuel into the blends. They were named D80K15E5, D80K15B5 and D80K15DE5 respectively, blends containing ethanol, *n*-butanol and diethyl ether for the sake of ease. Basically the additives having lower calorific values than biodiesel, so the blends showed the less calorific value than DK20. Flash point also showed a reduced manner. With an exception of D80K15DE5, D80K15B5 and D80K15E5 showed lower values than DK20 regarding to cetane number.
They are listed the fuel characteristics in Table 1.0

Table 1: Characteristics of the Fuels

Type of Fuel	Kinematic Viscosity (@40 0 C – mm^2/ sec)	Density 3 (Kg/m)	Calorific Value (Kj/g)	Cetane Number	Flash Point 0 (C)
Diesel	3.48	833	44.66	47	69.5
D80K20	3.64	834	43.71	49	91.5
D80K15E5	3.26	831	43.08	48	82.5
D80K15B5	3.32	831	43.43	49	83.5
D80K15DE5	3.29	832	43.41	52	80.5

Combustion And Emission Modeling

In internal combustion engines operating with different combustion systems Wiebe function is used to predict the mass fraction and burning rate with fuels [21]. In internal combustion engines Wiebe linked chain chemical reactions with the fuel reaction rate and his approach was based on the premise that a simple one-step rate equation will not be adequate to describe complex reacting systems such as those occurring in an internal combustion engine. For the simultaneous and sequential interdependent chain and chain branching reactions would be time consuming and tedious task moreover, developing and solving rate equations. Based on the concept of chain reactions he argued that for engineering application the details of chemical kinetics of all the reactions could be bypassed and a general macroscopic reaction rate expression could be developed. Prior knowledge of actual overall equivalence ratio is necessary when calculating the heat release. The stoichiometric air-fuel ratio is term equivalence ratio is defined as the ratio of actual air-fuel ratio. This helps in fixing the mass of fuel to be admitted. Fuel and oxidizer react to produce products of different composition in a combustion process. The theory of combustion is a complex process for many years and has been a topic of intensive research. Let us represent the chemical formula of a fuel as $C\alpha H\beta O\gamma N\delta$.

Nitric oxide formation model

To modeling NOx emissions from diesel engines is to use the extended Zeldovich thermal NO mechanism in the current approach and neglects other sources of NOx formation [22]. The mechanisms are extended Zeldovich mechanism it consists of following equations.

$$O + N2 \longleftrightarrow NO + N$$

$$N + O2 \longleftrightarrow NO + O$$

$$N + OH \longleftrightarrow NO + H$$

Net Soot formation model

When the fuel is in rich regions inside the cylinder during combustion the exhaust of CI engines contains solid carbon soot particles that are generated. Traces of other components absorbed on the surface the Soot particles are clusters of solid carbon spheres with HC . When fuel is in rich zones they are generated in the combustion chamber. Where there is not enough oxygen to convert all carbon to CO2. Most of these carbon particles find sufficient oxygen to react and form CO2 subsequently as turbulence motion continue to mix the components. In the combustion chamber the soot particles are formed and consumed simultaneously. By using semi-empirical model proposed by Hiroyasu et. al. (1983) the net soot formation rate was calculated.

Implementation Of Pm Inside An Ic Engine Using Kiva-3v

In this paper, to simulate the working process of the PM engine a modified version of KIVA-3V [23] code was employed. The disk PM regenerator is situated at the top of the cylinder and just beneath the cylinder head in computation mesh of the PM engine. The regenerator has a diameter of 9.4cm and thickness of 10 mm. The computational mesh used consists of 12000 cells at start of computation. With the assumption of zero gradients for temperature of both phase of PM and for species transport through the downstream boundary the boundary condition applied to the momentum and energy equation. Fuel is Diesel and, a solid alumina sphere with a porosity of 0.4 is considered as the PM.

The computational period covers the interval from the intake valve close (IVC) to the exhaust valve open (EVO). Constant temperatures were specified for the main cylinder boundaries; the side boundary of the PM was assumed as adiabatic, while at the top and bottom surfaces of the PM, there was heat exchange with the bulk gas phase for boundary conditions. A cold start of the PM engine was not considered, and the engine cycle was calculated starting at the IVC in a certain compress stroke after several cycles in this paper. During a continuous operation the initial temperature of the PM regenerator was set at a constant value (of 433 K), which should be approximately equal to the average temperature of the PM. For the bulk gas phase in the cylinder volume was specified as 400 K at the initial temperature. Between the PM and the gas, the temperature of the gas phase approaches the temperature of the PM very soon due to the high thermal capacity and a sufficient heat transfer coefficient.

II. Results

The performance of the proposed model to simulate and analyze a modified KIVA 3V code is used. In the Table- 2 the specifications of the engine modeled are illustrated. The necessary modeling parameters are fed and modeled initially using K3PREP solved using the code and processed using K3POST

Table 2 Engine specifications

Bore	13.716 cm^2
Stroke	16.51 cm
Length of Connecting Rod	26.3 cm
Squish	0.4221 cm
Compression Ratio (CR)	15:1
RPM	1500

Similarly in the Table (3.0) other factors that are considered and listed in this model. For further analysis and modeling of combustion and emission these parameters are Considered.

Table 3 Factors Considered in Modeling of Combustion and Emission Process

Cylinder Wall Temperature	438.3K
Head Temperature	534.3K
Piston-Gas Side Surface Temperature	572.0K
Fuel Temperature at Injection	345.0K
Intake Surge Tank Pressure	1.94 e+6 dyn/cm^2
Intake Surge Tank Temperature	327.0K

The performance and emission characteristics has been carried out at a constant speed of 1500 RPM for proper analysis.

Table 4 Engine brake power comparison for different blends without PM insidecombustion chamber at an RPM of 1500

Type of Fuel	Engine Brake Power (KW)
Diesel	31.091
DK20	27.876
D80K15E5	28.546
D80K15B5	29.384
D80K15DE5	31.259

Table 5 Engine Brake Power comparison for different blends with PM inside combustion chamber at an RPM of 1500

Type of Fuel	Engine Brake Power (KW)
Diesel	32.85
DK20	29.22852
D80K15E5	30.45431
D80K15B5	31.63175
D80K15DE5	33.20658

The lowest power for DK20 due to higher density and viscosity can be attributed to its lower calorific value and lower combustion efficiency. At 1500 rpm the blends of D80K15E5, D80K15B5 and D80K15DE5 gave higher increment of brake power than DK20 respectively. These blends showed higher brake power than DK20 which ensures higher combustion efficiency however, in spite of lower calorific value. The result of their decreased densityand viscosity which improved atomization at the higher combustion efficiency.

Table 6 Brake Specific Fuel Consumption for different blends without PM inside combustion chamber at an RPM of 1500

Type of Fuel	BSFC (g/(kW-hr))
Diesel	243.47
DK20	247.62
D80K15E5	249.43
D80K15B5	236.25
D80K15DE5	231.78

Table 7 Brake Specific Fuel Consumption for different blends with PM inside combustion chamber at an RPM of 1500

Type of Fuel	BSFC (g/(kW-hr))
Diesel	228.6407
DK20	231.4851
D80K15E5	233.6426
D80K15B5	222.8639
D80K15DE5	220.9023

The comparison of BSFC is effected at a constant speed and constant full load which means at a certain engine power, then the values of BSFC directly proportional to the fuel mass flow rate obviously. It can be seen that lower BSFC in D80K15B5 and D80K15DE5 has shown corresponding to DK20 as well as diesel. This kind of result can be attributed to good atomization and combustion quality though they have got lower calorific values. Combining at the facts lower BSFC with lower calorific value

Table 8 Brake Thermal Efficiency for different blends without PM inside combustion chamber at an RPM of 1500

Type of Fuel	Efficiency %
Diesel	17.2
DK20	17.8
D80K15E5	18.1
D80K15B5	18.3
D80K15DE5	18.5

Table 9 Brake Thermal Efficiency for different blends with PM inside combustion chamber at an RPM of 1500

Type of Fuel	Efficiency %
Diesel	17.8
DK20	18.2
D80K15E5	18.3
D80K15B5	18.7
D80K15DE5	19.1

Table-8 It shows that the brake thermal efficiency (BTE) of D80K15DE5 and D80K15B5 are higher than DK20 and diesel fuel can be easily explained. Consequently they showed this kind of higher BTE, as BTE is simply the inverse of the multiplication of BSFC and calorific value. Though its calorific value was the lowest among the blends for D80K15E5, it showed lower BTE for its higher BSFC which depicts its lower combustion efficiency than D80K15DE5 and D80K15B5

Table 10 Quantification of NOx formation for different blends without PM inside combustion chamber at an RPM of 1500

Type of Fuel	Nox(g/kg-f)
Diesel	33.71
DK20	34.93
D80K15E5	32.39
D80K15B5	31.62
D80K15DE5	30.14

Table 11 Quantification of NOx formation for different blends with PM inside combustion chamber at an RPM of 1500

Type of Fuel	Nox(g/kg-f)
Diesel	21.43
DK20	22.02
D80K15E5	21.19
D80K15B5	20.93
D80K15DE5	20.13

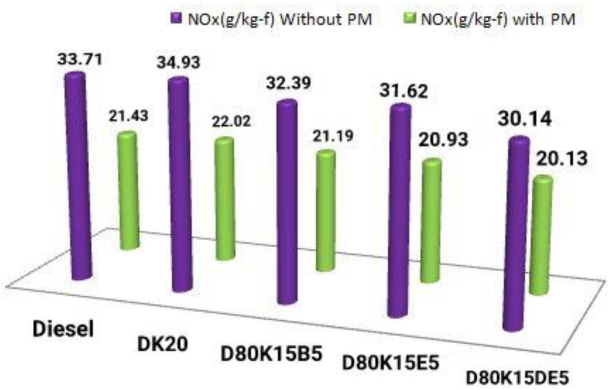

Figure 1 Comparison of NOx formation for different blends with and Without PM inside combustion chamber at an RPM of 1500

Table 12 Quantification Soot formation for different blends without PM inside combustion chamber at an RPM of 1500

Type of Fuel	Soot (g/kg-f)
Diesel	3.18
DK20	3.09
D80K15E5	2.46
D80K15B5	3.03
D80K15DE5	3.05

Table 13 Quantification Soot formation for different blends with PM inside combustion chamber at an RPM of 1500

Type of Fuel	Soot (g/kg-f)
Diesel	3.07
DK20	2.41
D80K15E5	2.37
D80K15B5	2.56
D80K15DE5	2.74

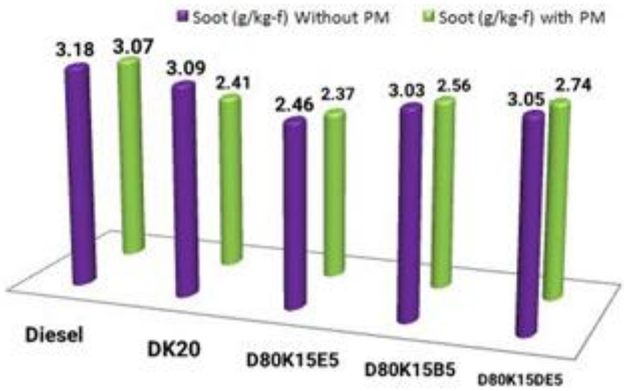

Figure 2 Comparison of Soot formation for different blends with and Without PM inside combustion chamber at an RPM of 1500

Figure (1) and Figure (2) illustrates the improvement of porous media cylinder inside the combustion chamber. Though this process ethanol and *n*-butanol having higher oxygen content D80K15E5 and D80K15B5 showed lower NO this can be explained by their lower calorific value and higher heat of evaporation which resulted in lower in-cylinder temperature. In the case of D80K15DE5, lower NO can be attributed to reduced part of premixed combustion where NO is mainly formed. None the less, D80K15E5 showed the highest amount of NO which can be attributed to the comparatively higher oxygen content of ethanol among the blends with additives.

It can be observed that, oxygenated compounds available in the biodiesel made the soot emission lower in the case of DK20 it can be observed. It showed higher amounts of soot emission in spite of higher oxygen content of ethanol, *n*-butanol and diethyl ether blends. This behavior can be the effect of addition of additives like ethanol, *n*-butanol and diethyl ether which make it easier to evaporate the fuel and slipped into the cylinder especially at low speed during expansion stroke. Another reason can be mentioned here is the increase of 'lean outer flame zone'. This actually means the envelope of the spray boundary where the fuel is already beyond the flammability limit because of over mixing. However, the comparative emission of soot among the blends with additives can be explained easily with the oxygen content of the additives mentioned earlier.

Table 14 Improvement in performance of different parameters with the introduction of Porous Media inside combustion chamber

Type of fuel	Improvement in Engine Brake Power (%)	Reduction in BFSC (%)	Improvement in Thermal Efficiency (%)	Reduction in Nox (%)	Reduction in Soot (%)
Diesel	7.342	-6.35	2.17	-40.00	-6.04
DK20	6.386	-6.07	2.45	-38.64	-14.5
D80K15E5	6.430	-6.09	1.51	-37.54	-8.74
D80K15B5	6.647	-6.13	1.57	-37.02	-7.54
D80K15DE5	7.234	-6.43	1.59	-36.04	-5.84

It can be observed from the discussion there is a marked improvement in the performance of the different blends in the presence of the porous media inside the combustion chamber. It can also be inferred from these discussions the performance of the blend D80K15DE5 containing diesel, biofuel and diethyl ether as additive is much better when compared to other fuel blends

III. Conclusion

The performance of PM inside a combustion chamber for different types of fuel blends with additives simulated in this paper.The PM and the IC engines as whole was modeled using modified KIVA-3V code. In terms of brake power, fuel consumption rate, the performances of the engines were studied to illustrate their capability.

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