EFFECT OF SOYBEAN OIL BIOFUEL BLENDING ON THE PERFORMANCE AND EMISSIONS OF DIESEL ENGINE USING DIESEL-RK SOFTWARE

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Abstract

The scope of the technology is to provide utility and comfort with no damage to the user or to the surroundings. For many years now, petroleum products and other fossil fuels have given us utility and comfort in a variety of areas, but causes environmental problems which threaten wild and human life**.** In this study, the performance and emissions of single cylinder, four stroke, direct injection diesel engine operating on diesel oil and different Soybean Methyl Ester (SME) blends have been investigated theoretically using the simulation software Diesel-RK. Based on the computed modeling results it's found that 41.3 %, 53.2 % & 62.6 % reduction in the Bosch smoke number obtained with B20% SME, B40 % SME and B100% SME respectively, compared to pure diesel operation. In addition a reduction in PM emissions is observed 47.2%, 60 % & 68% for the B20 % SME, B40 % SME, and B 100% SME respectively. On the average basis there is a reduction in the thermal efficiency, power, and SFC, for all SME blends by 2%, 3%, and 12% respectively compared to pure diesel fuel. All blending of SME produce higher NOx emissions more than 28% compared with pure diesel fuel. A parametric study of retarding injection timing, varying engine speed and compression ratio effects has been performed. It's observed that retarding the injection timing can reduce the increase in the NOx emissions to great extent. Among all tested fuels its noticed that B20% SME was the best tested fuel which gave the same performance results with good reduction in emissions as compared to pure diesel operation. A very good agreement was obtained between the results and the available theoretical and experimental results of other researchers.

Keywords: diesel engine, biodiesel, combustion, thermodynamic modeling, engine performance & emissions, Soybean oil methyl ester.

1. Introduction

One of the most important elements to effect world economy and politics is substantially of petroleum resources, which the main sources of world energy supply. In today's world, in order to meet the growing energy needs as a consequence of spiraling demand and diminishing supply, alternative energy sources mostly biofuels are receiving more attention. Petroleum products became very scare and expensive because of the environmental pollution from internal combustion engine; the increasing global concern has caused to focus on the oxygenated diesel fuels **Cengiz et al [5].**

Biodiesel is a non-toxic, biodegradable and renewable alternative fuel that can be used with no engine modifications. It can be produced from various vegetable oils, waste cooking oils or animal facts. The fuel properties of Biodiesel may change when different feed stocks are used. In general, if the fuel properties of Biodiesel are compared to petroleum diesel fuel, it can been seen that Biodiesel has higher viscosity, density, and cetane number. Also the energy content or net calorific value of Biodiesel is about 10-12 % less than that of conventional diesel fuel on the mass **Cengiz et al [5]**. In the last three decades, the impacts of Biodiesel produced from different feed stocks has been intensively investigated on the performance and emission characteristic of diesel engine by many researchers. These studies point put that Biodiesel exhibits similar results with very little performance differences and reductions on regulated emissions when compared to pure diesel baseline operation. Therefore, it may not require modifications on diesel engines to use biofuels. There is a general agreement on the Biodiesel fuel which provides substantial reduction in HC, CO, and smoke emissions but it increase the NOx emissions compared to pure diesel fuel. **Mustafa et al [9]** investigate the effect of the Biodiesel produced from high free fatty acid feed stocks on engine performance and emissions. Two different Biodiesel were prepared from animal fat-based yellow grease with 9% free fatty acids and from soybean oil. The pure fuels and their 20% blends with No. 2 diesel fuel were studied at steady-state engine operating conditions in a four-cylinder turbocharged diesel engine. Although both Biodiesel fuels provided significant reductions in particulates, carbon monoxide and unburned hydrocarbons, the oxides of nitrogen increased by 11% and 13% for the yellow grease methyl ester and soybean oil methyl ester, respectively.

Bryan [4] determined and compared the important fuel properties and emission characteristics of blends (20 vol. %) of soybean oil methyl esters (SME) and partially hydrogenated SME (PHSME) in ultra low sulfur diesel fuel (ULSD) with neat ULSD. The following changes were observed for B20 blends of SME and PHSME versus neat ULSD: improved lubricity, higher kinematic viscosity and cetane number, lower sulfur content, and inferior low-temperature properties and oxidative stability. With respect to exhaust emissions, B20 blends of PHSME and SME exhibited lower PM and CO emissions in comparison to those of neat ULSD. The PHSME blend also showed a significant reduction in THC emissions. Both SME and PHSME B20 blends yielded small increases in NOx emissions. The reduction in double bond content of PHSME did not result in a statistically significant difference in NOx emissions versus SME at the B20 blend level. The test engine consumed a greater amount of fuel operating on the SME and PHSME blends than on neat ULSD, but the increase was smaller for the PHSME blend**.** **Tamil et al [13]** studied the performance, emission and combustion characteristics of a single cylinder constant speed, direct injection diesel engine using methyl ester of sun flower oil – eucalyptus oil blend as an alternative fuel and the results are compared with the standard diesel fuel operation. Result indicated that 50% reduction in smoke, 34% reduction in HC emission and a 37.5% reduction in CO emission for the MeS50Eu50 blend with 2.8 % increase in NOx emission at full load. Brake thermal efficiency was increased 2.7 % for MeS50Eu50 blend.

2. Physical properties of bio-fuels

Physical properties of the fuel are necessary as input data for the calculation of the spray evolution dynamics, size of droplets and, consequently, the evaporation and heat release rates. Table (1) presents the properties of different blends of diesel oil and Soybean Methyl Ester. Properties of the blends are calculated using values of fractions of bio-fuel and diesel oil in the blend and their molar masses. The saturated pressure of the fuel is determined for each zone taking into accounts its temperature.

Table 1. Properties of diesel fuel and various blends of SME

Property	**Diesel B 0%**	**SME B20%**	**SME B40%**	**SME B60%**	**SME B80%**	**SME B 100%**
Mass composition of fuel						
C	0.87	0.8496	0.8297	0.8104	0.7915	0.7731
H	0.126	0.1245	0.1230	0.1216	0.1202	0.1188
O	0.004	0.0259	0.0473	0.0680	0.0883	0.1081
Sulphur fraction in the fuel	0.025	0.00105	0.00208	0.00308	0.00405	0.005
Low heating value (MJ/kg)	42.5	41.18	39.89	38.64	37.41	36.22
Cetane number	48	48.69	49.37	50.03	50.67	51.3
Fuel density (kg/m^3)	830	841	852	863	874	876
Surface tension (N/m)	0.028	0.03122	0.0344	0.03741	0.0404	0.0433

Dynamic viscosity (Pa.s)	0.003	0.00334	0.00368	0.004	0.00432	0.00463
Molar mass (kg/kmol)	190	211.5	232.5	252.9	272.8	292.2
Vapor pressure at 480 K	0.0477	0.0433	0.003822	0.03241	0.02567	0.01
Vapor pressure at critical T	1.616	2.408	3.609	5.549	8.956	15.760
Critical Temperature (K)	710	721.2	734	748.7	765.9	786
Coefficient A	5220	5768	6308	6877	7529	8372
Coefficient B	7.832	8.876	9.877	10.90	12.02	13.41

$$ln(p_s) = B - \frac{A}{T} \tag{1}$$

The theoretical analysis were carried out on a naturally aspirated, water-cooled, 1-cylinder, direct-injection diesel engine. The specifications of the engine are shown in Table 2

Table 2 Specification of Engine

Engine Make	Kirloskar AV-1
Engine Type.	(4-Stroke, Diesel Engine)
Number of Cylinder	1
Bore × stroke	85×110 mm
Cylinder capacity	0.624 L
Compression ratio	Variable (12-19)
Rated power	3.7 kW , 1500 rpm
Dynamometer	Electric AC-generator
Orifice diameter	0.15 mm
Injection pressure	(200-220) bar
Cooling system	Water cooling

Note: These specifications are taken from the research engine at IC engine laboratory of Andhra University, Visakhapatnam, India.

3. Scope of the software model

In the multizone combustion model, which was implemented in this work, the spray is split into seven characteristic zones, as shown in Fig. (1). In each zone specific evaporation and burning conditions exist and these are specified in the model.

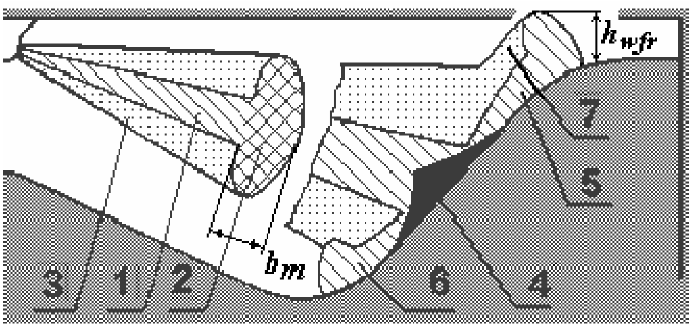

Fig.(1) Characteristic zones of the diesel spray

During the time prior to the jet impingement only three zones are considered in the spray. These are:

1. dense conical core of the free spray.
2. dense front of the free spray.
3. dilute outer sleeve of the spray.

The near wall flow (NWF) which is formed after the impingement has an inhomogeneous structure, temperature, and density and therefore it can be further subdivided into the following four zones:

4. dense conical nucleus of the NWF.
5. dense NWF on the piston surface.
6. dense front of the NWF.
7. dilute outer zone of the NWF.

If during its evolution, the spray reaches the surface of the cylinder liner and/or the head of the cylinder then it is necessary to introduce two further zones. The general principles of spray modeling and evaporating equations are mentioned in **A.S. Kuleshov [2] and Razleytsev N.F [12].**
The thickness of the front of the free spray is determined as;

$$b_m = A_m . l . F_s . We^{0.32} . M^{-0.7} . \rho^{0.5} \tag{2}$$

A_m=0.7 empirical coefficient

$F_s = 0.0075 - 0.009$

l : Is the current distance between the injectors nozzle and the contour portion of fuel

$$l = l_m \left[1 - \left(1 - \frac{t_k}{t_m} \right)^3 \right] \tag{3}$$

$$We = {U_{om}}^2 . dn \tag{4}$$

$$M = \frac{{M_f}^2}{\rho_f . d_n . \sigma_f} \tag{5}$$

h_{fwr} is the height of the dense front of the NWF.

The following two assumptions are made to calculate the heat release rate in the cycle.
(a) The speed of the fuel evaporation in each zone is equal to the sum of speeds of evaporation of the separate fuel droplets. The speed of evaporation of the droplet before and after ignition can be calculated as

$$d_k^{\,2} = d_o^{\,2} - K . t_u \tag{6}$$

The fuel injectors of modern diesel engines produce the fuel spray with approximately the same average size of droplets d_{32} which can determine using the Sauter's equation

$$d_{32} = E_k . d_n . M^{0.0733} . (W_e . \rho)^{-0.266} \tag{7}$$

Where $E_{k\,=}$ 1.7 is an empirical coefficient. The initial diameter of the droplet in equation (5) is taken to be equal to the average size of droplets

$$d_o = d_{32} \tag{8}$$

(b).The ratio $K/d_o^{\,2} = b_u$ is constant during the whole process of the fuel injection. The relative speed of the fuel evaporation in the specific i-zone now can be calculated as

$$\frac{d\sigma_{ui}}{dt} = \left[1 - (1 - b_{ui} . t_{ui})^{\frac{3}{2}} \right] . \frac{\sigma_{zi}}{t_{ui}} \tag{9}$$

Evaporation constants for the various specific zones in the combustion chamber can be calculated as

$$K_{ui} = 4 * 10^6 . Nu_D . D_p . \frac{ps}{\rho_f} \tag{10}$$

4. Heat release model

When calculating the heat release in the cycle, the combustion process of the fuel is split, as usual, into four stages, having unique physical and chemical features which limit the speed of the burning. These are as follows:
(a)Ignition delay period.
The auto ignition delay period is calculated by modified Tolstov's equation:

$$\tau = 3.8*10^{-6}\left(1-1.6*10^{-4}.n\right)\sqrt{\frac{T}{P}}.\exp\left(\frac{E_a}{8.312.T}-\frac{70}{CN+25}\right) \tag{11}$$

(b).Premixed combustion phase
During premixed combustion phase the heat release rate is:

$$\frac{dx}{dt} = \varphi_o P_o + \varphi_1 P_1 \tag{12}$$

Where

$$P_0 = A_0 \left(m_f / V_i\right)(\sigma_{ud} - x_0)(0.1\ \sigma_{ud} + x_0) \tag{13}$$

Where

$$P_1 = d\sigma_u / dt \tag{14}$$

x is the fraction of the heat released or fraction of the fuel burnt out; x_0 is the fraction of the fuel vapor formed during the ignition delay period.

(d). Mixing-controlled combustion phase.
In the mixing controlled combustion period the heat release rate can be determined as;

$$\frac{dx}{d\tau} = \varphi_1 P_1 + \varphi_2 P_2 \tag{15}$$

Where:

$$P_2 = A_2 \left(m_f / V_c\right)(\sigma_u - x)(\alpha - x) \tag{16}$$

(d).late combustion phase. In this phase the heat release rate is:

$$\frac{dx}{d\tau} = \varphi_3\ A_3\ K_T\ (1 - x)(\eta_b \phi - x) \tag{17}$$

In these equations it is assumed that $\varphi_0 = \varphi\ 1 = \varphi\ 2 = \varphi$ function describing completeness of fuel vapor combustion in the zones. ϕ is an equivalence ratio. During the simulation process the heat transfer in the cylinder is taken into account and heat transfer coefficients for its different zones are calculated using **Woschni's [14]**

5. Modeling of NOx formation

While nitric oxide (NO) and nitrogen dioxide (NO_2), are usually grouped together as (NO_x) emissions. NO is predominant in diesel engine **Heywood J.B [6]**. Therefore; only NO formation is considered and in simulation model all calculations are carried out with thermal mechanism. The same model adapted by **A.S. Kuleshov [3]** is adapted in this work. Features of the designed procedure are:

1. Step by step calculation of equilibrium composition of combustion products for eighteen species in the burnt gas zone.
2. Kinetic calculation of thermal nitrogen oxides formation with chain **Zeldovich** mechanism

The oxidizing of nitrogen is on the chain mechanism, basic reactions are:

$$O_2 \leftrightarrow 2O \tag{18}$$

$$N_2 + O \leftrightarrow NO + N \tag{19}$$

$$N + O_2 \leftrightarrow NO + O \tag{20}$$

A main reaction is in equation (20). Rate of this reaction depends on concentration of atomic oxygen. The Volume concentration of NO in combustion products formed in a current calculation step is defined with equation:

$$\frac{d[NO]}{d\theta} = \frac{p.2.333*10^7.e^{-\frac{38020}{T_Z}}\,[N_2]_e.[O]_e.\left\{1-\left([NO]/[NO]_e\right)^2\right\}}{R.T_Z.\left(1+\frac{2365}{T_Z}.e^{\frac{3365}{T_Z}}.\frac{[NO]}{[O_2]_e}\right)}.\frac{1}{rps} \tag{21}$$

6. Modeling of soot concentration

Soot is a fine dispersion of black carbon particles in a vapor carrier **Mohamed Fadhil, [10]**. The main source of soot is from the incomplete hydrocarbon combustion. Soot particles form, grow, and oxidize as a result of chemical reactions that occur during combustion. The detailed of soot modeling are mentioned by **Razleytsev N.F [12]** , Exhaust gas soot concentration related to normal conditions are as follows;

$$[C]_H = \int_{\theta B}^{480} \frac{d[C]}{dt}.\frac{d\theta}{6n}\left(\frac{0.1}{P}\right)^{\frac{1}{\gamma}} \tag{22}$$

Where γ =1.33 is an exhaust gas adiabatic exponent.

$[C]$ is a current soot concentration in cylinder

The Hartidge smoke level is calculated from the following equation;

$$Hartridge = 100[1 - 0.9545\,exp(-2.4226[C])] \tag{23}$$

An experimental curve was used to calculate the Bosch smoke level Bosch as a function of Hartidge equation.

Particulate Matter emission is calculated by equation of **Alkidas [1],** as a function of Bosch smoke number:

$$[PM] = 565\left(ln\frac{10}{10 - Bosch}\right)^{1.206} \tag{24}$$

Another important equation is the Emission Complex (Complex of air pollutant) which is the summary of PM and & NOx emissions which is denoted by SE.

$$\boldsymbol{SE = C_{PM}\left[\frac{PM}{0.15}\right] + C_{NO}\left[\frac{NO_X}{7}\right]} \tag{25}$$

C_{PM} is an empiric factor for PM 0.5 & C_{NO} is an empiric factor for NOx 1.0

7. Results and discussion

The first part of this section are the results of the effect of SME oil biofuel on the cylinder pressure, combustion zone temperature, hear release rate, and spray tip penetration of a diesel engine. The engine was run on 100 percent of diesel oil, (pure diesel operation), and 20 %, 40% and 100% SME biofuel blends which are shown in figures 1 to 4**.** It can be seen from **Fig.(1 & 2)** that the predicted cylinder pressure and combustion zone temperature for both diesel and SME biofuels are in close agreement especially in the region of peak values (i.e.) the global behavior for these figures is the same. **Fig.(3)** presents the computed heat release rate for diesel fuel and SME biofuels. It is evident from this figure that all the biofuels blends had earlier start of combustion, but slower combustion rate. The earlier start of combustion was caused by the earlier start of injection and shorter ignition delay; and the slower premixed combustion rate due to less energy released in premixed phase and also probably the lower volatility of biodiesel. In the diffusion combustion phases, the SME biodiesel fuels had rapid combustion as at this point most of fuels get vaporized.

In the second part of simulation results the engine performance and emissions with the base diesel fuel and the SME biofuels blends were evaluated. The variation of engine power & SFC with different fuels as pure diesel, 20% SME, 40% SME, 100% SME are presented in **Fig.(4), t**he engine power for all blends of SME was less than that of pure diesel by 3 %. This is due to lower heating value of biodiesel fuels compared to diesel fuel. On the other hand the SFC, in general, was found to increase with the increasing proportion of biodiesel in the fuels. SFC for all SME blending fuels is higher than pure diesel fuel by 12%. The increase in SFC is due to higher density and lower heating value, since the methyl esters have heating values that are about 12.4% less than pure diesel. These results are similar to those of **Monyem.A. [8] & Mustafa Canakci ,et al [9].**

Fig.(5) the variation of Sauter mean diameter as a function of SME blends, it can be seen that Sauter mean diameter increase with the increase in the SME fraction in the fuel. This could be explained by the rise in the coefficients of viscosity and surface tension when the bio-fuel content of the fuel is increased. With the increase of the diameter of the droplets it would be logical to expect that the evaporation velocity and the burning rate of the fuel vapor would decrease. The other part of this graph is the variation of ignition delay period with SME faction in the fuel. The physical and chemical properties of the fuels will affect the ignition delay period, and researchers have stressed that chemical properties are much more important than physical properties. The ignition quality of a fuel is usually characterized by its cetane number. Higher cetane number generally means shorter ignition delay. So high blending ratio of SME causes shorter ignition delay which causes earlier start of combustion, and less energy released in premixed phase. The same results were reported by **W. Yuan, [15].**

Fig.(6) discusses the relation between the Bosch smoke number data and NOx emissions data with different percentage of SME blending. The smoke levels for all SME percentages are lower than that of pure diesel. The average smoke level for B20, B40, and B100 were less than that of diesel fuel by 41.3 %, 53.2 %, and 62.6 % respectively. This is because smoke decreases with high oxygen content in the biofuel that contributes to complete fuel oxidation even in locally rich zones, so the oxygen within the fuel decrease the tendency of a fuel to produce soot, **Cengiz et al [5].** Another reason for smoke reduction when using biodiesel is the lower C/H ratio as compared to pure diesel fuel. This also indicated in the **Monyem.A.[8]**, he found a 60 % reduction in Bosch smoke number when fueling the engine with 100 % SME. The NOx emissions were higher for all blending of SME than pure diesel fuel. The NOx emissions for B20 % are higher than that of base line diesel operation by 22%. This is associated with the oxygen content in the biodiesel hence the fuel oxygen may provide additional oxygen for NOx formation. It is well known that NOx emissions are related to start of combustion timing and the energy released in premixed burning. Earlier start of combustion causes higher cylinder pressure and higher combustion temperature, which cause higher NOx emissions. **Mittelbach, M [11]** measured the exhaust emissions from a diesel engine fueled with SME and he found increase NOx emissions compared to pure diesel.

In **Fig.(7)** the specific particular matter and Summary of (PM and NOx emission) with diesel fuel and SME biofuels are shown. It can be seen that the level of PM for SME blending are less than that of pure diesel base line operation. There is a very good reduction in the emissions of PM, for B20 %, B40 %, and B100 % the emissions of PM is 47.23%, 60%, and 68 % less than pure diesel oil. The empirical equation for the emissions of PM and NOx emissions (SE) is computed for different percentages of SME biodiesel. There is a reduction in SE emission by 4% with 100 % SME as compared with pure diesel fuel, while 2% increase in SE with B20 %. Also in appendix (A) there is summary of the predicted results for the performance and emissions.

8. Parametric study

In this section the effect of injection timing, engine speed, and compression ratio on thermodynamic cycle parameters and emissions of diesel engine fueled with diesel fuel and different SME and its blending have been verified.

Figures (8-11) shows the effect of injection timing ranging from 7-12 ^{o} crank angle BTDC on the SFC, Bosch smoke level, NOx and PM emissions. There is no significant change in SFC with advancing injection timing. It is observed that as injection timing is advanced, cylinder pressure and temperature during the delay period become lower. Therefore, the ignition delay period becomes longer. This phenomena results in more fuel being burnt during premixed combustion phase following ignition delay period. With combustion rates during the premixed phase being much higher than those during diffusion phase, cylinder pressure increases faster as injection timing advanced, **Mohamed Fadhil [10]**. Higher cylinder temperatures with advancing injection timing result in an increase in NOx concentration. The decrease in the Bosch smoke level by (28.3-45.2) % and PM by (35-53.1) % has been observed. The injection timing has a significant effect on NOx. By retarding the injection timing, NOx of SME can be reduced to the same level of pure diesel. The computational model indicated that the advanced start of injection was one of the main reasons for increased NOx emissions of SME biodiesel. So retarding the injection timing from (12-7) CA^0 BTDC reduces in the emissions of NOx by 35% with diesel fuel, 36.85% with B20 SME, 38% with B40 SME, and 40% with B100 SME. These results are in good agreement with **W. Yuan [15].**

Figs.(12-15) represent the effect of engine speed on the SFC, Bosch smoke number, NOx and PM for the base diesel fuel and SME blends. Engine speed has been varied from (1000-3000) rpm. Peak pressure in the cylinder is increased with the increased engine speed and consequently led to higher rate of heat release due to the increase in air entrainment. Increasing engine speed caused increase in the engine power, and decrease in the SFC up to 2000 rpm, as a result of increasing rate of heat release, as the load on the engine become dominant. This causes an increase in the SFC. Smoke level, PM and NOx decreased with the increased engine speed. It is presumably due to the decrease in the period of time for reaction which overcomes the effect of increasing cylinder pressure and temperature, **Hiroyasu H., et al [7].**

Figs. (16-19) shows the effect of variable compression ratio on the performance which represented SFC and emissions which represented by Bosch smoke, PM, and NOx. The range of compression ratio was chosen from (12-19) which is the same range allowed in the VCR engine available in the lab. of IC engine at the Andhra university for all blends of SME As the compression ratio increasing there is a decrease in the engine power due to an increase in the load and thermal stresses on the engine, which causes an increase in the SFC. It is observed that increasing the compression ratio cause a reduction in the Bosch smoke number and PM as compared to diesel fuel, but increase the NOx emissions is observed due to increase in the combustion temperature.

The last part of this section is the present model verification with the other available researcher work. **Fig.(20)** shows a comparison between the computed thermal efficiency and SFC of diesel and SME blending with the results predicted by **Mustafa Canakci ,et al [9]**. It can be seen that the agreement between these results is fairly encouraging; the difference in the accuracy models is very small.

9. Conclusion

The objective of this paper was to study the effects of SME and its blends as an alternative source of energy on the performance and emissions of diesel engine by using the Diesel-RK software. The following conclusion can be drawn from the theoretical results of this study.

1. Using the SME and its blends was found to reduce the thermal efficiency, power, and SFC as compared to pure diesel fuel by 2%, 3% and 12% respectively. This is due to the lower heating of biodiesel compared to diesel fuel.
2. A very good reduction in the Bosch smoke number, and PM emission for all blends of SME relative to that of diesel fuel have been observed. This is due to the high oxygen content in the biodiesel, so reductions in the emissions made SME and its blends a suitable alternative fuel for diesel and thus could help in controlling air pollution.
3. For all blends of SME, the computed NOx was higher than that of pure diesel fuels. This is because Biodiesel fuels showed slower premixed combustion and faster diffusion combustion. In addition, higher overall cylinder temperatures were found for SME compared to D2, which was the direct cause of the higher NOx emissions of SME. The higher overall cylinder temperatures of SME were attributed to the earlier start of injection and decreased spray cone angle Therefore, the earlier start of injection and decreased spray cone angle of SME were concluded as the leading reasons for higher NOx emissions.
4. Retarding of injection was found to be the easiest strategy to reduce NOx from SME to great extent..
5. Its observed from all tested fuels, that B20 % SME was the best one which had same performance results with good reduction in the emissions as compared to base line diesel operation, also less increase in the NOx emissions was noticed in B20 % SME as compared with B40% & B100% SME biodiesel respectively this made B20% biodiesel a suitable alternative fuel for diesel engine, and could help in controlling air pollution problem.
6. The comparison of the theoretical results and experimental data published in the open literature shows very good agreement with each other.

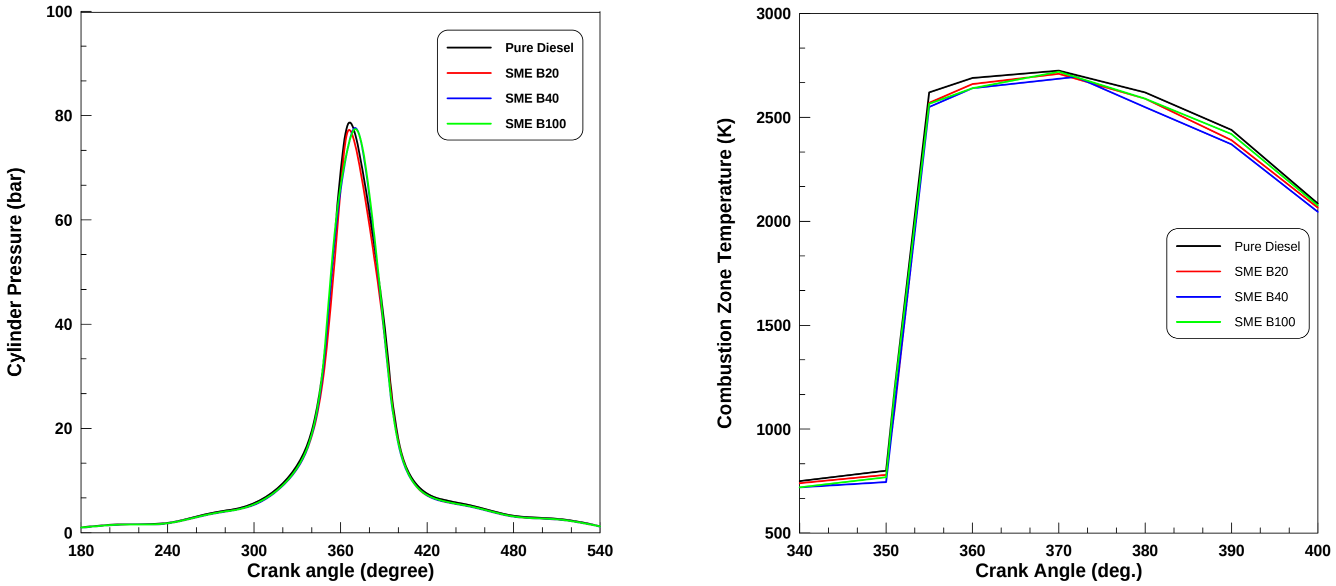

Fig.(1& 2) Variation of Cylinder Pressure and combustion zone temperature with crank angle for diesel fuel and different SME blending

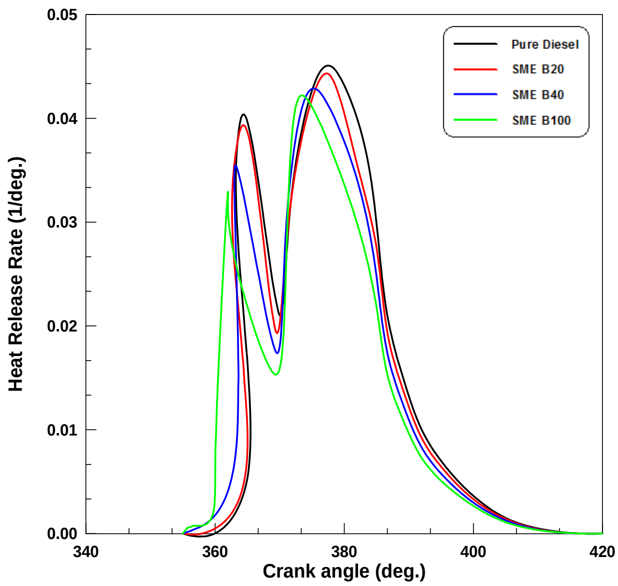

Fig.(3) Variation of Heat release rate with crank angle for diesel fuel and different SME blending.

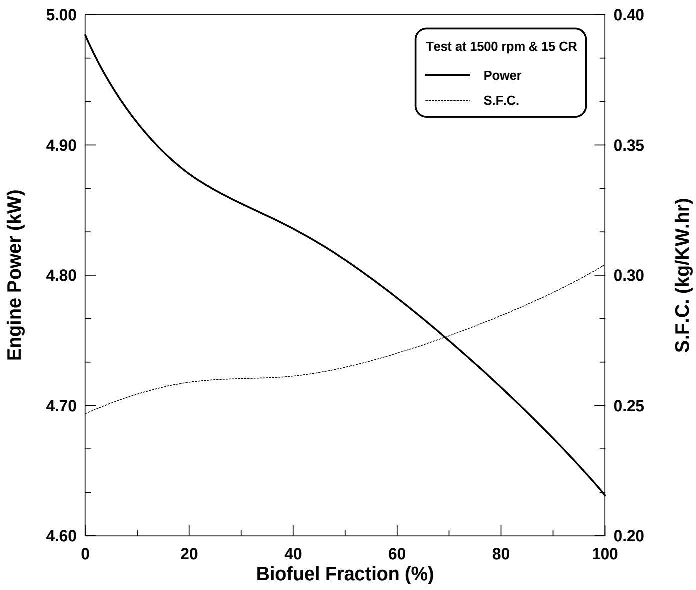

Fig.(4) Effect of SME blending on the engine power & SFC

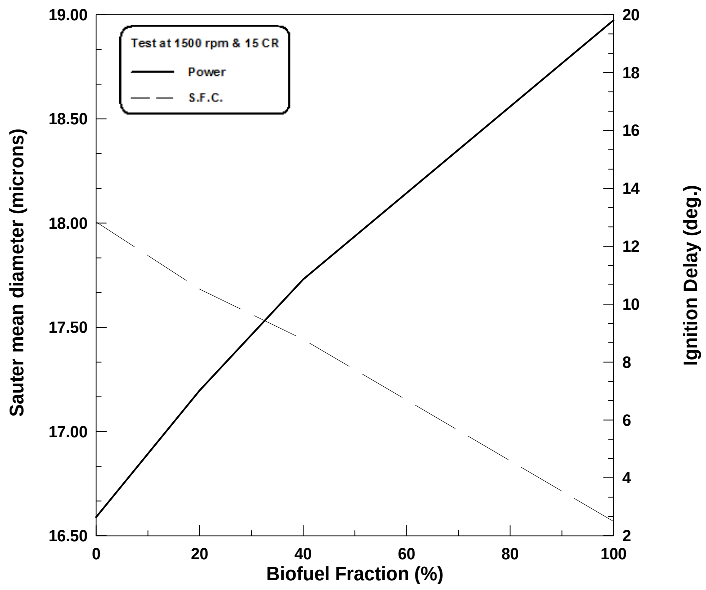

Fig.(5) Effect of SME blending on the Sauter mean diameter & Ignition delay

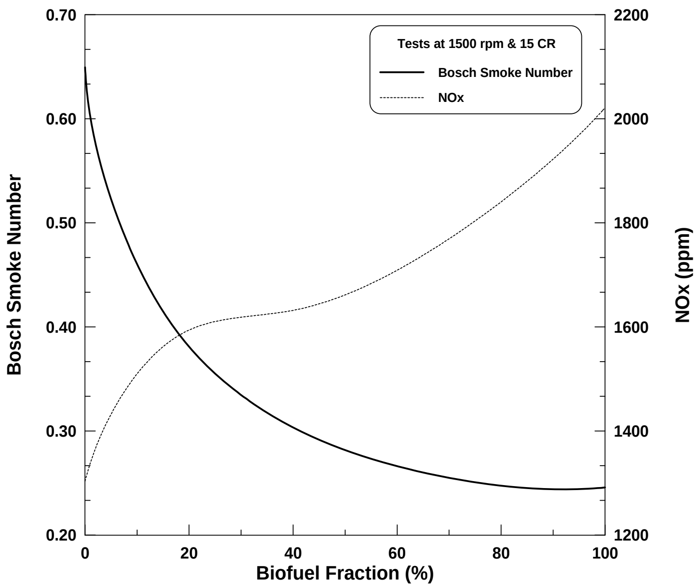

Fig.(6) Effect of SME blending on the Bosch smoke number & NOx emissions

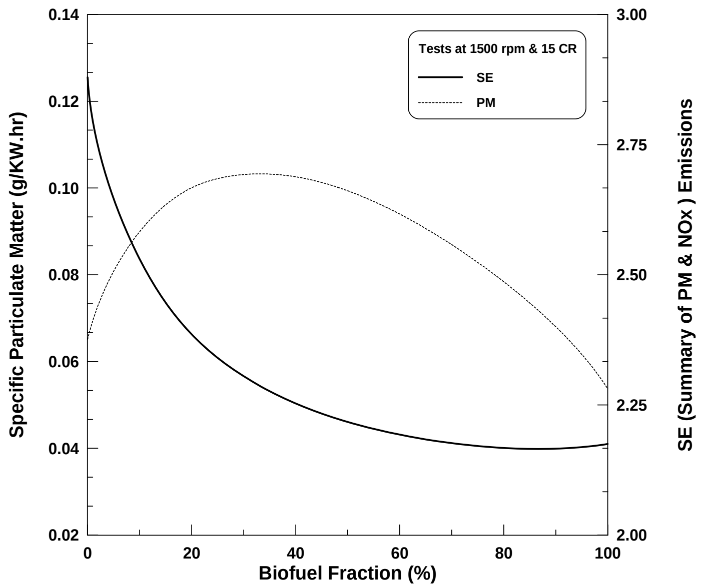

Fig.(7) Effect of SME blending on the Specific particular matter and air pollutant equation

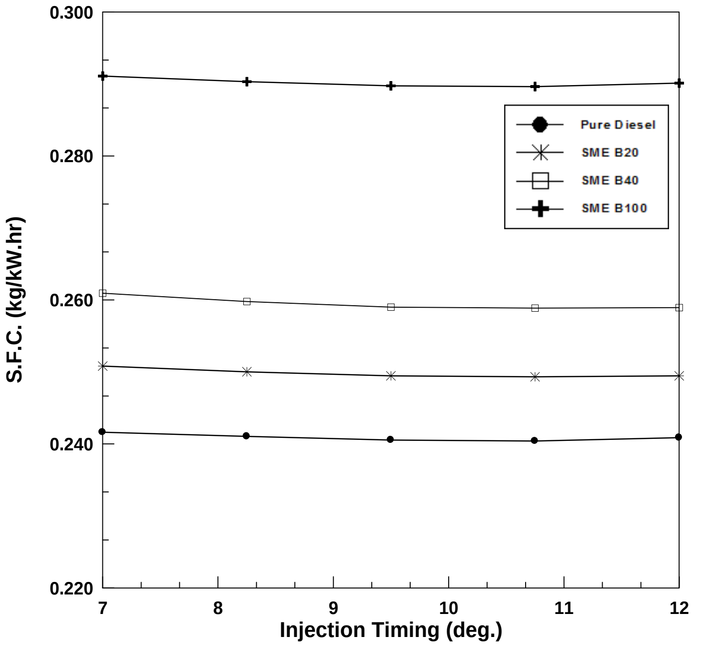

Fig.(8) Effect of Injection timing on the and SFC for the tested fuels

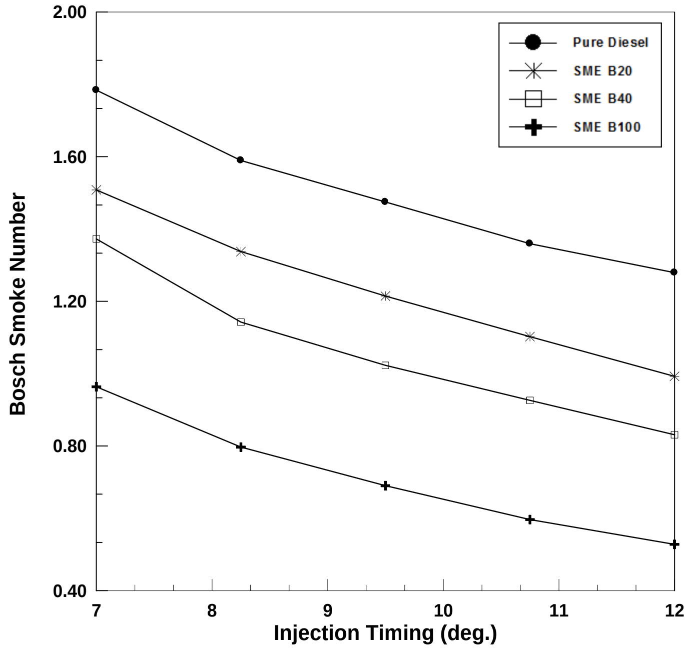

Fig.(9) Effect of Injection timing on the engine Bosch smoke number for the tested fuels

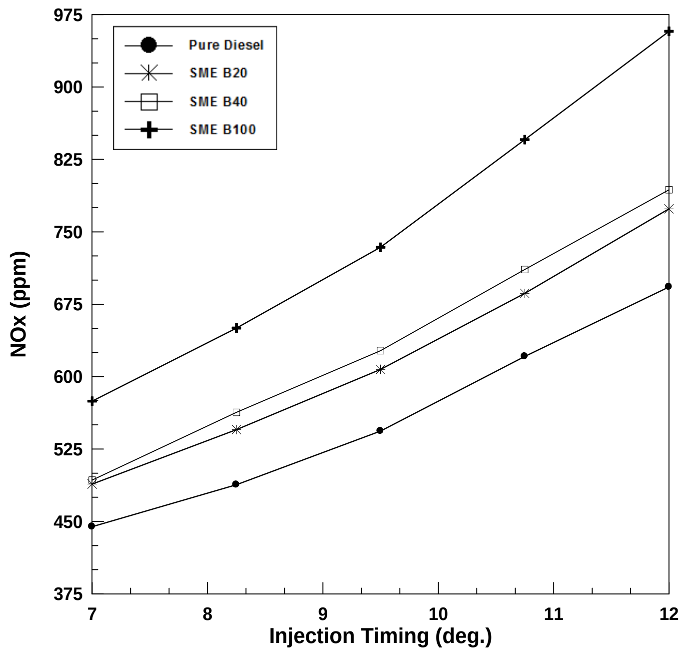

Fig.(10) Effect of Injection timing on the engine NOx emissions for the tested fuels

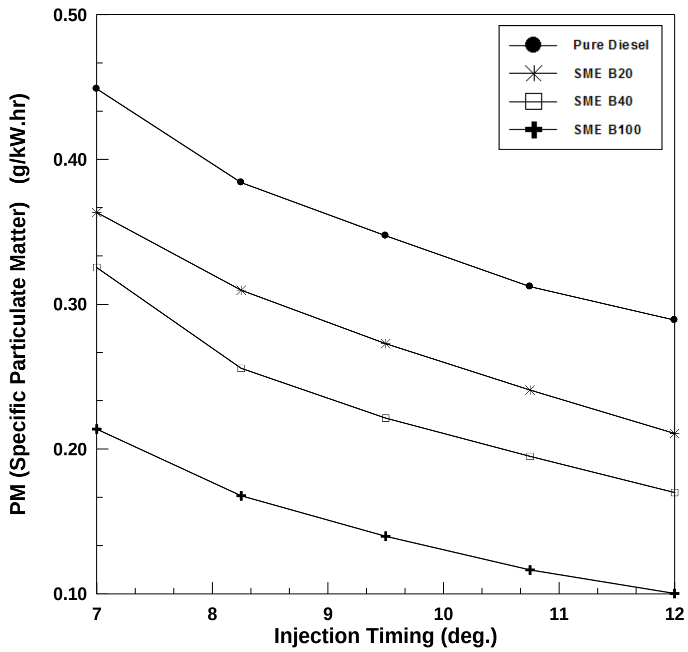

Fig.(11) Effect of injection timing on the PM emissions for the tested fuels

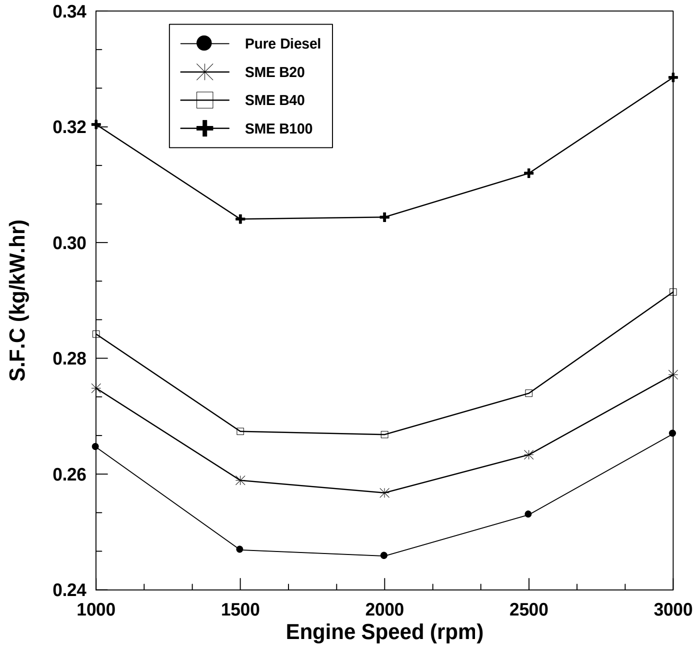

Fig.(12) Effect of engine speed on the and SFC for the tested fuels

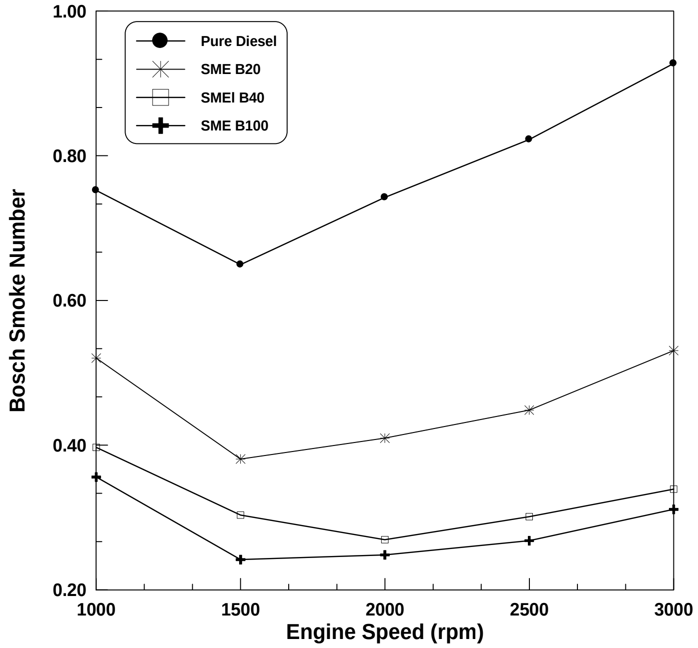

Fig.(13) Effect of engine speed on the engine Bosch smoke number for the tested fuels

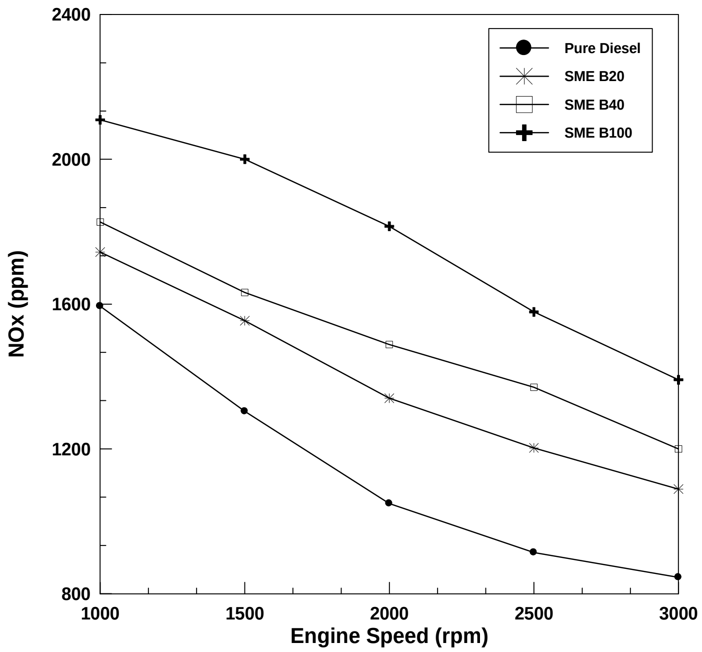

Fig.(14) Effect of engine speed on the engine NOx emissions for the tested fuels

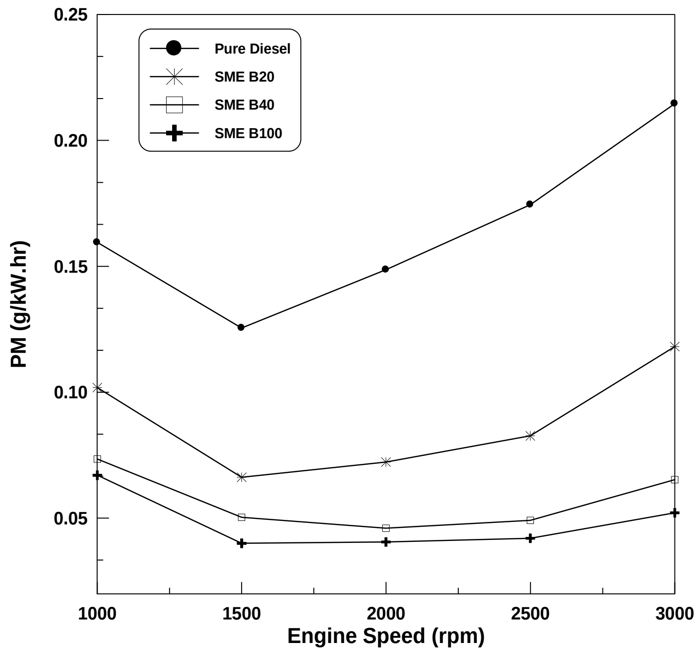

Fig.(15) Effect of engine speed on the PM emissions for the tested fuels

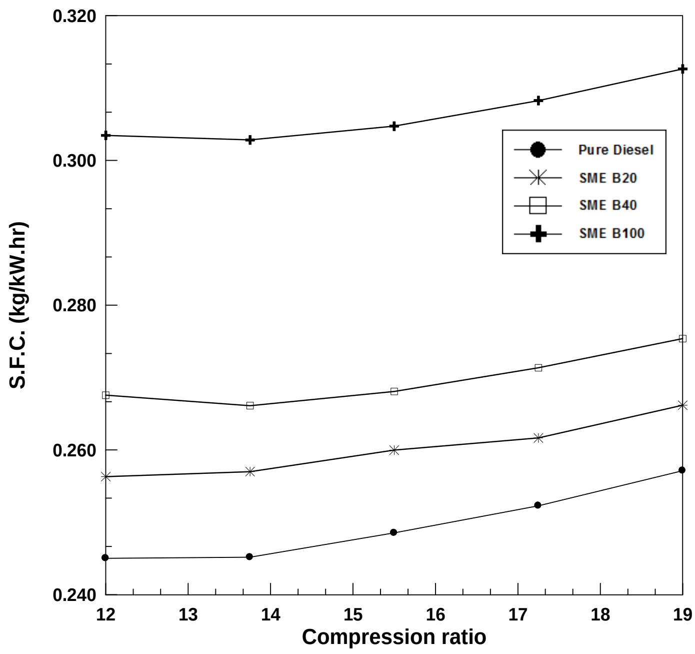

Fig.(16) Effect of compression ratio on the and SFC for the tested fuels

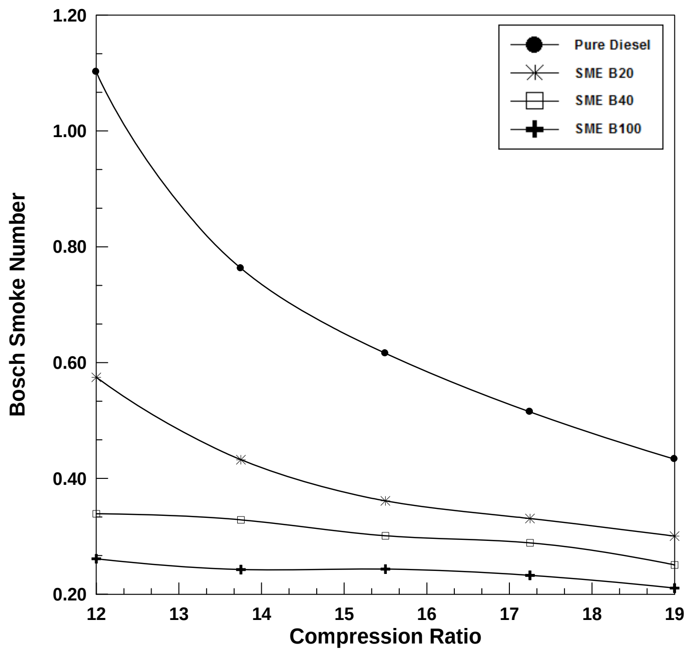

Fig.(17) Effect of compression ratio on the engine Bosch smoke number for the tested fuels

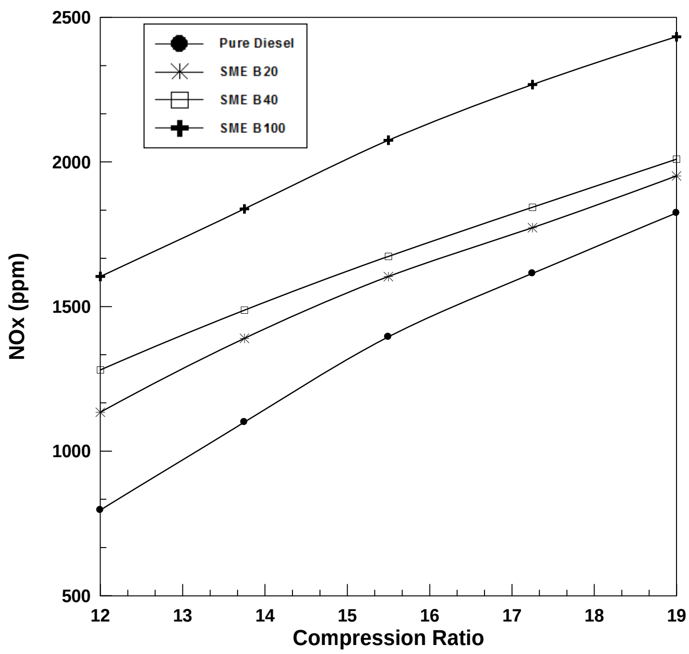

Fig.(18) Effect of compression ratio on the engine NOx emissions for the tested fuels

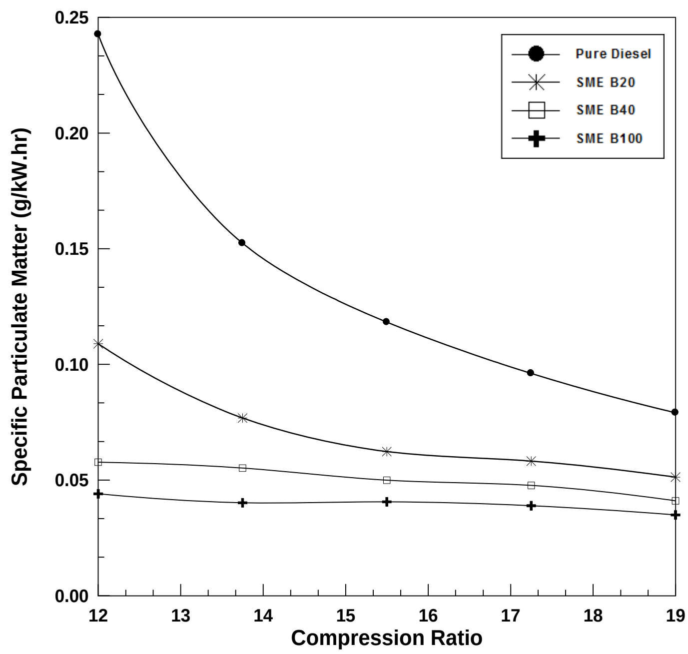

Fig.(19) Effect of compression ratio on the PM emissions for the tested fuels

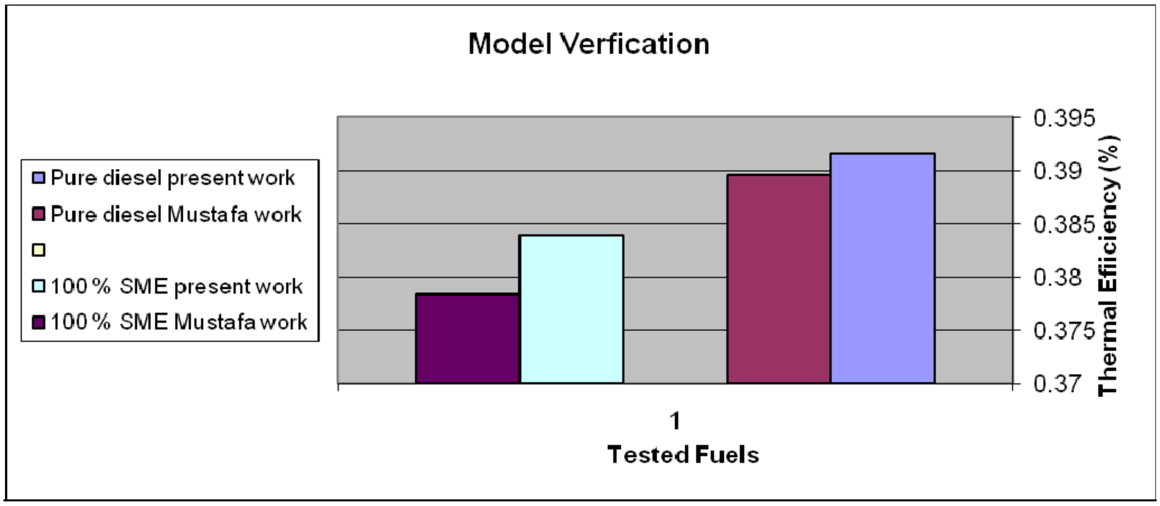

Fig.(20-a) A Comparison between Predicted thermal efficiency and other Model with level for pure diesel, and pure biofuel B100 SME,

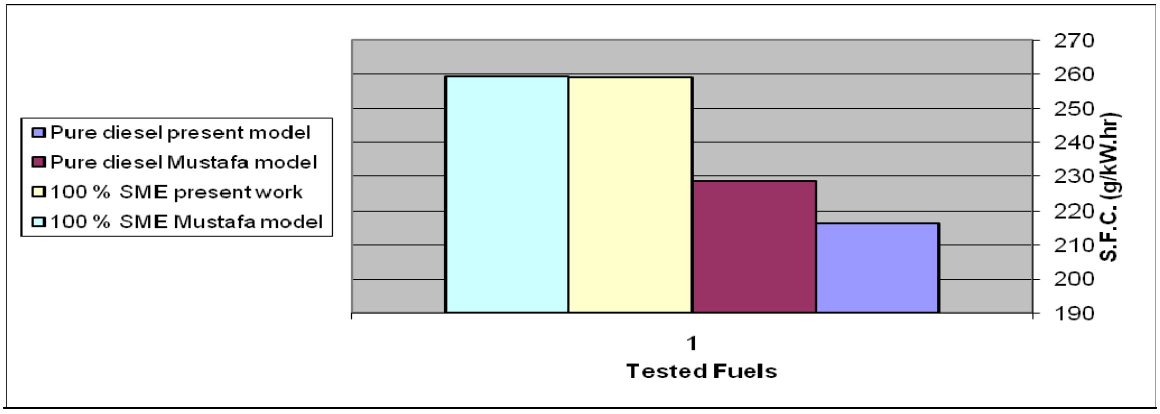

Fig.(20-b) A Comparison between Predicted SFC and other Model with level for pure diesel, and pure biofuel B100 SME,

Appendix A

The results from the software at 1500 rpm, 15 compression ratio, and 20^o CA BTDC injection timing.

Fuel	Power (kW)	SFC (kg/kW.hr)	Thermal Efficiency	Bosch smoke number	PM (g/kW.hr)	NOx ppm	SE (g/kW.hr)
Pure Diesel	4.9844	0.24689	0.42592	0.64942	0.12552	1303.7	2.3757
B20 SME	4.8777	0.25891	0.42163	0.38079	0.06623	1593.9	2.6671
B40 SME	4.83572	0.26127	0.42135	0.30329	0.05032	1631.8	2.6881
B100 SME	4.6311	0.30406	0.41248	0.24572	0.04099	2021.2	2.281

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Nomenclature

A, *B*, *b*u constants
*A*m, *E*k empirical coefficients
*A*S, *B*S, *C*S, *D*S, *E*S, *F*S coefficients
*B*m thickness of the front of the free spray (m)
B20 blend containing 80 per cent of diesel oil and 20 per cent SME
B40 blend containing 60 per cent of diesel oil and 40 per cent of SME
B100 blend containing 0 per cent of diesel oil and 100 per cent of SME (pure SME)
BSN Bosch smoke number
C carbon
CN Cetane number of fuel
DF Diesel fuel
*D*n diameter of the nozzle (mm)
*d*32 average size of droplets (Sauter mean diameter (microne)
*d*0, *d*k initial and current diameters of the fuel droplets (mm)
*D*p coefficient of diffusion of the fuel (m2/s)
E_a apparent activation energy for the auto ignition process(23000...28000 kJ/kmole)
IC E internal combustion engine
K evaporation constant
l current length of the spray (m)
*l*m penetration distance of the Control portion of fuel.
M, *We*, dimensionless parameters
m_f mass of fuel
Nu_D Nusselt number
NO*x* nitrogen oxides
p pressure in the cylinder (Pa)
*p*s saturation pressure of the fuel vapour (Pa)
PM particulate matter
R universal gas constant (8.3143)
rpm revolutions per minute
rps revolutions per second

SFC specific fuel consumption (g/kWh)
SE Summary of (PM & NOx emissions) (g/kW.hr)
SME soybean methyl ester
T temperature in the cylinder (K)
t time
t_u current time since the droplet entered the characteristic zone in the combustion chamber.
t_m travel time for the CPF to reach the spray front before stopping
t_k travel time for the CPF to reach the distance *l* from the injector's nozzle.
TDC top dead centre
U velocity of the CPF (m/s)
*U*0 initial velocity of the spray at the nozzle (m/s)
*U*0m mean velocity of the spray in the nozzle (m/s)
*U*m velocity of the spray's front (m/s).

Greek symbols

μ_f dynamic viscosity of the fuel (Pa s)
ρ_a density of air (kg/m3)
ρ_f density of fuel (kg/m3)
σ fraction of the fuel injected into the cylinder
σ_f surface tension coefficient of the fuel (N/m)
τ ignition delay (s)
θ crank angle (degree)
ψ function for description of the completion of combustion
ξ_b efficiency of air used.