Theoretical Study for The Influence of Biodiesel Addition on The Combustion, Performance and Emissions Parameters of Single Cylinder Diesel Engine

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Abstract

This study examines the characteristics of combustion, performance and emission of constant speed compression ignition engine fed with different percentages of diesel fuel and rapeseed methyl ester (RME) on a volume basis by using the well-known software simulation Diesel-RK. As the percentage of RME increased, the maximal pressure is noticed to be closer to top dead center (TDC). It was found that 47.27 %, 81.06 %, 82.56 % and 93.36 % reduction in the Bosch smoke number is obtainable with 10% RME, 20% RME, 50% RME and 100% RME respectively, compared with ordinary diesel. The blends of RME are noticed to emit higher NOx emissions. The result signals that 10% RME is the promising ratio of blending which reports less performance variations and reduced carbon emissions as well. The effect of variable injection timings is studied to moderate biodiesel NOx effects on the 10% RME and 18 degree crank angle before top dead center (BTDC) was recorded as the advisable injection timing which gives a promising reduction in NOx emissions.

Keywords: Rapeseed methyl ester, Diesel engine, Diesel-RK software, Engine emissions.

الخلاصة

في هذه الدراسة تم اختبار خصائص الاحتراق و اداء و ملوثات محرك اشتعال بالضغط ثابت السرعة يغذى بنسب حجمية مختلفة من وقود الديزل مع الوقود الحيوي (زيت بذر اللفت) باستخدام برنامج المحاكاة (الديزل–ار.كي). لقد لوحظ ان زيادة نسبة خلط الوقود الحيوي مع وقود الديزل الاعتيادي تؤدي الى اقتراب اعظم ضغط من النقطه الميته العليا. سجلت الدراسة انخفاضا في انبعاثات الدخان (السناج) وهي 47.27 %، 81.06 %، 82.56 % و 93.36 % عند خلائط 10% من زيت بذر اللفت و20% بالاضافه الى 100% على التوالي بالمقارنه مع وقود الديزل الاعتيادي. لوحظ ان خلائط زيت بذر اللفت تؤدي الى ارتفاع في انبعاثات اكاسيد النتروجين. اشرت النتائج ان 10% من الوقود الحيوي تمثل افضل نسبه خلط والتي تعطي اقل تاثير على اداء المحرك بالاضافة الى انحفاض كبير في الانبعاثات الكاربونية. تم دراسة تاثير تغير وقت الحقن للتقليل من زيادة انبعاثات اكاسيد النتروجين على الوقود الحيوي بنسبة 10% ووسجلت ان افضل وقت للحقن عند زاوية 18 قبل النقطه الميته العليا.والذي يعطي تخفيضا واعدا لانبعاثات اكاسيد النتروجين.

الكلمات المفتاحية :- بذر اللفت المثيلي المؤستر ، محرك ديزل ، برنامج نمذجة بلغة RK ، انبعاثات المحرك .

Nomenclature

Symbol	Definition and unit
A_o, A_2,A_3	Constants.
10% RME	Blend ratio of 90 % diesel and 10 % RME
20% RME	Blend ratio of 80 % diesel and 20 % RME
50% RME	Blend ratio of 50 % diesel and 50 % RME
100% RME	Blend ratio of 100 % RME
BTE	Brake Thermal Efficiency (%)
BSFC	Specific Consumption of Fuel (Brake) (g/kW.h)
[C]	Concentration of smoke
C^o	Centigrades (deg.)
CA^o	Crank shaft angle (deg.)
C.N	Cetane no.
E_a	Apparent energy of activation 23000 to 28000 kJ/kmole
K_T	Constant of evaporation
l	Spray length (meter)
l_m	Fuel penetration distance (meter).

m_f	Fuel mass (kg)
NOx	Oxides of nitrogen (ppm)
P	Cylinder pressure (pa)
p_s	Fuel saturated pressure (pa)
PM	Particulate Matter
R	Constant of gas (8.3143 kJ/kmol.^{o}K)
RME	Rapeseed methyl ester
RPM	Revolution Per Minute
rps	Revolution Per Second
T	Cylinder temperature (K^o)
t	Time (second)
TDC	Top Dead Centre
U	Fuel portion speed (meter/s)
U_0	Spray initial velocity (meter/s)
Um	Spray front velocity (meter/s).
V	Volume of cylinder (m^^{3}).
X	Heat release fraction
x_o	Fractional vaporization of fuel

Greek symbols

Symbol	Definition
σ	Injected fuel fraction
σ_u	Vapor fraction through delay time.
τ	Delay of ignition (sec.)
θ	The angle of crank (degree)
ϕ	Equivalence ratio
φ	Combustion function
γ	Specific heat ratio of gas
ξ_b	Air effectiveness.

1.Introduction

The rapid growth of internal combustion engines complies the current requirements of protecting environment, at the same time maintaining low consumption of fuel. Furthermore, reducing the pollutant exhaust gasses by selecting solutions and design parameters has a limited range (Wojciech *et.al.,* 2011). The goal of the research is to explore the scope of using alternative sources of energy. The scenario for such goal is characterized by many aspects, such as: the increasing energy demand, rapid petroleum source depletion, recent strategy for increasing the price of the traditional fuels and tough emissions regulations. The biodiesel, especially rapeseed oil and rapeseed oil methyl ester (RME), may serve much more to satisfy the quick requirement of future energy in additional to reduction of carbon emissions particularly the greenhouse effect and compounds of the combustion process (Dumitru, 2012).

On the other hand, the regulations for NOx emission from diesel engine are toughening, and reduction in greenhouse gas emissions also lifted essential matters on pollution emissions. (Yoon *et.al.*, 2011 ; Kim *et.al.*, 2008). According to this scenario biodiesel submitted to intense research works. Biodiesel is an attractive source renewable substitute which could be utilized in the absence of any improvements. It is produced from various feedstocks such as: various vegetable oils, waste cooking oil and animal fat (Roberto *et.al.*, 2011). Generally, biodiesel have properties are compared to original petroleum diesel fuel, it can be noticed that biodiesel has higher density, viscosity, and a cetane number as well.

(Lapuerta *et.al.*, 2008) reported that (NOx) emissions are observed to increase as a result, advancing of injection advancement while the carbon emissions are noticed to decrease because of the oxygen content of biodiesel. The oxygen content in the biodiesel provides additional oxygen, which participates for complete combustion. Same results are reported for greenhouse gas (CO_2).

(Canakci ,2007) investigated the impacts of various kinds of diesel fuel and soybean biodiesel on combustion and emission parameters. The experiments reported increase in the brake fuel consumption and NOx emissions as well as significant reduction in the carbon exhaust emissions.

(Labeckas *et.al.*, 2006) evaluate the effect RME blends with neat diesel on the performance and emissions of diesel engines. The results indicate higher values of fuel consumption for RME as compared to diesel fuel oil and low speed. Higher engine speeds result in lower consumption of fuel for 5% and 10% blends.

The motivation in this work focuses on the use of different blends on a volume basis, of RME biodiesel with diesel fuel in single cylinder, four stroke diesel engines and its effects on the combustion, performance and emission parameters using the simulation software Diesel-RK.

2.Physical Properties of Biodiesel

The properties of diesel and biodiesel under study are listed in Table1. The theoretical analysis was achieved on a direct injection diesel engine. The specifications of the engine used in the simulation process are shown in Table 2 (Mohamed *et.al.*, 2014).

Table1. Comparison of different proportions of RME with conventional diesel

Property	Diesel	B10% RME	B20% RME	B50% RME	RME
Chemical formula	$C_{13.77}H_{23.44}$	$C_{14.53}H_{24.74}O_{0.013}$	$C_{21.45}H_{26.04}O_{0.026}$	$C_{17.61}H_{29.94}O_{0.065}$	$C_{21.45}H_{36.44}O_{0.13}$
Density at 15 ^oC (kg/m^3)	830	834.4	838.2	852	874
Viscosity at 40 ^oC (pa.s)	0.003	0.003392	0.003784	0.00496	0.00692
Lower Heating value (MJ/kg)	42.5	42.195	41.85	40.975	39.45
Surface tension (N/m)	0.028	0.02835	0.0287	0.0297	0.0315
Cetane number	48	48.64	49.28	51.2	54.4

Table 2 Engine Technical data (Mohamed *et.al.*, 2014)

Engine Brand	Kirloskar - Compression ignition engine
Kind	Single cylinder, four Stroke
bore (mm)	87.5
stroke (mm)	110
Speed	1500 (RPM)
Ratio of compression	17.50
Power (kW)	4.4
Pressure of injection (bar)	220

3.Theoretical Analysis

In RK-model of combustion, the spray is classified into seven regions, as illustrated in Figure 1. Each region had a separate condition of evaporation and burning. Before the jet wall impingement just 3 regions are formed in the jet. These are:

1. Free spray core.
2. Front of the free spray.
3. Outer sleeve of the spray.

4. Near wall flow (NWF) nucleus.
5. NWF on the piston surface.
6. The dense front of the NWF and
7. NWF Outer zone.

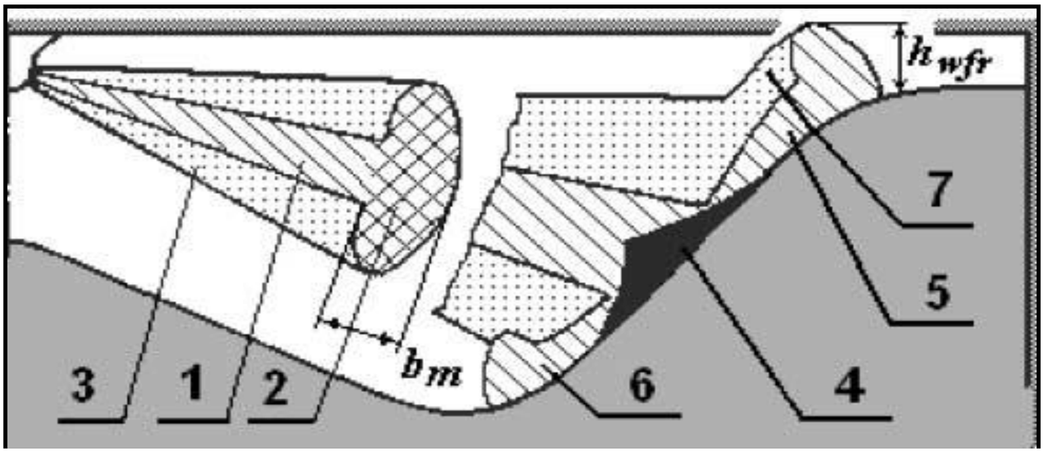


Fig. 1 Spray Characteristic (Mahkamov et al., 2007)

Once the spray is moving, the breakup of spray takes place. Movement of spray takes the surrounded gas along with it. The velocity of gas has low values in the environment as compared to the core of spray where it's accelerated rapidly and approaches the droplet velocity. The velocity as well as position of elementary fuel mass (EFM) is referred as:

$$\left(\frac{U}{U_o}\right)^{3/2} = 1 - \frac{l}{l_m} \tag{1}$$

Figure 2 presents the evolution of jet with respect to time. The detailed of mathematical analysis for evaporation are described in (Kuleshov ,2005).

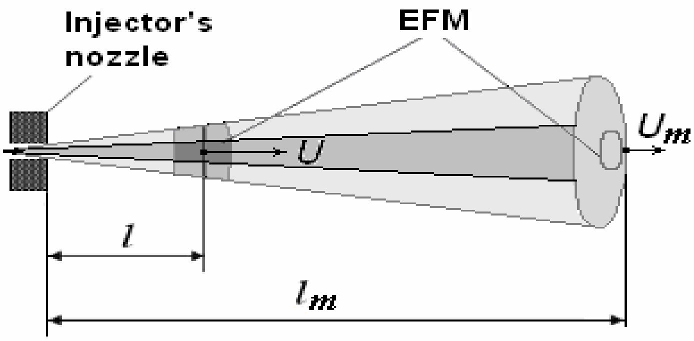


Fig. 2 Growth of spray v/s time.

3.1 Heat Release Model

The combustion process is divided into different phases. Below the detailed description of each one:

a) Delay of Ignition

$$\tau = 3.8*10^{-6}\left(1 - 1.6*10^{-4}.n\right)\sqrt{\frac{T}{P}}.\exp\left(\frac{E_a}{8.312T} - \frac{70}{C.N + 25}\right) \tag{2}$$

b) Uncontrolled pressure

$$\frac{dx}{dt} = \varphi_o\left(A_o\left(m_f / V_i\right).\left(\sigma_{ud} - x_o\right).\left(0.1\sigma_{ud} + x_o\right)\right) + \varphi_1\left(d\sigma_u / dt\right) \tag{3}$$

c) Controlled pressure

$$\frac{dx}{dt}=\varphi_1(d\sigma_u/dt)+\varphi_2\left(A_2\left(m_f/V\right).(\sigma_u-x)(\phi-x)\right) \tag{4}$$

d) End of combustion

$$\frac{dx}{dt}=\varphi_3 A_3 K_T(1-x)(\xi_b.\phi-x) \tag{5}$$

During simulation process Woschni's formula is used to predict coefficients of heat transfer in the cylinder (Woschni, 1967)

3.2 Modeling of NO_X Formation

The oxide of nitrogen reaction is depending on the oxygen concentration and it can be calculated as:

$$N+O_2 \leftrightarrow NO+O \tag{6}$$

The NO Concentration can be determined from below equation

$$\frac{d[NO]}{d\theta}=\frac{2.33*10^7\, p.e^{-\frac{38020}{T_z}}[N_2]_e[O]_e\left(1-\left([NO]/[NO]_e\right)^2\right)}{RT_Z\left(1+(2365/T_Z).e^{\frac{3365}{T_z}}.[NO]/[O_2]_e\right)}.\left(\frac{1}{rps}\right) \tag{7}$$

3.3 Smoke Modeling

Smoke is a thin particle of black carbon comes from rich combustion of hydrocarbon. During combustion, particles of soot oxidize because of chemical reactions. The smoke at exhaust can be calculated from equation below:

$$[C]_H=\int_{\theta_B}^{480}\frac{d[C]}{dt}.\frac{d\theta}{6n}\left(\frac{0.1}{p}\right)^{\gamma} \tag{8}$$

Bosch smoke number (BSN) is calculated from the equation of particulate matter (PM) emission (Alkidas, 1984) as follows:

$$[PM]=565\left(\ln\frac{10}{10-Bosch}\right)^{1.206} \tag{9}$$

4. Results and Discussion

The range of operating conditions was constant engine speed 1500 RPM, injection pressure 220 bars and 20^o BTDC injection timing. At full load condition the air-fuel ratio (A/F) is lower and higher smoke level; hence to distinguish the differences among the examined fuels, full load condition is chosen.

4.1 Combustion characteristics

In a compression ignition engine the pressure of the cylinder is a function of the rate of the burning fuel so that higher pressure of cylinder means smoother combustion and better heat release. Fig.3 displays the relation between cylinder pressure and crankshaft for diesel RME blends. It is noted that the peak pressure for diesel is 91.95 bar at 365^o which is nearer to TDC where for 100% RME maximum pressure was 78.88 bars at 367^o crank angle because of the supply of heat is reduced for tested biodiesel. The pressure behavior of 10% RME is observed to be closer to the neat diesel fuel.

Predicted zonal temperature is shown in Fig.4. Higher zonal combustion temperature is noticed for RME blend comparatively to the base line operation of diesel. Highly flame combustion temperature was an indicant for highly NOx emissions. The difference between diesel and RME is 50 K. The zonal combustion temperature is the main reason for increasing the emissions of NOx. Same observations were noticed in the results of (Yuan , 2005)

Figs. 5 and 6 present the computed heat release rate and fraction of heat release for diesel fuel and RME biodiesel respectively. The heat release fraction can be defined as the ratio of heat release to the lower heat value of the fuel. It is also observed from that RME blends had an earliest combustion start because of the time of injection advancement and the slowing rate of energy in the stage of uncontrolled pressure. In the stage of controlled pressure, all blends of RME biodiesel fuel had a rapid combustion as all fuel vaporized and burned.

Fig.7 shows the effect of blending ratio on the Sauter mean diameter (SMD) with crank angle. It can be seen that SMD increases with the increase of the RME fraction in the fuel. This is because of higher viscosity coefficients and surface tension. This reduction in rate of burning fuel and velocity is logical as the diameter of the droplets increased.

The variation of the delay period with RME biodiesel is shown in Fig.7. The quality of ignition is frequently influenced by a cetane number; therefore, highly cetane number indicates shortly ignition delay. So the higher blending ratio of RME causes shorter ignition delay. The same results were reported by (Hansen *et.al.,* 2009).

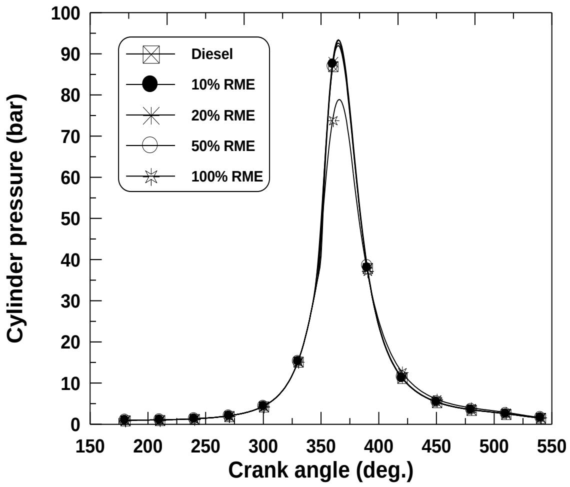


Fig. 3 Cylinder pressure v/s crank angle

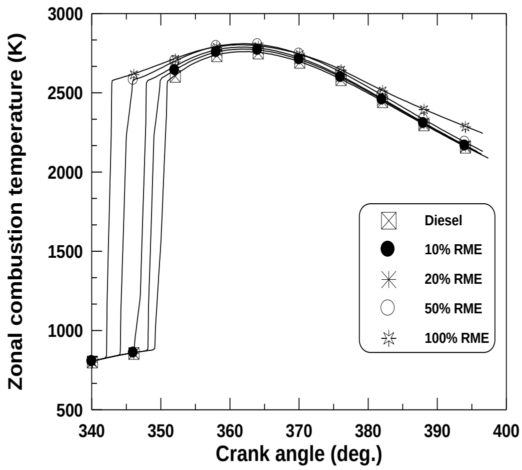


Fig. 4 zonal temperature v/s crank angle

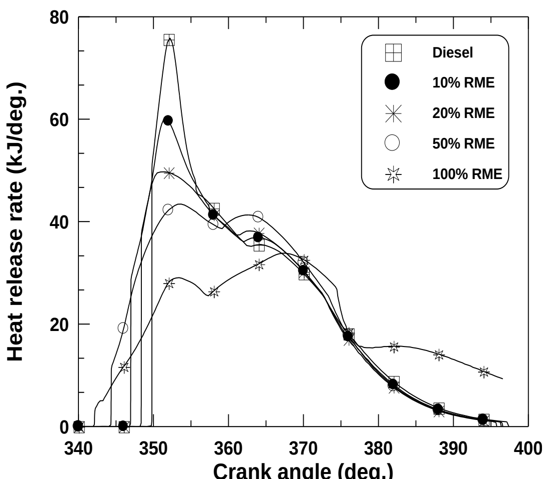


Fig. 5 Heat release rate v/s crank angle

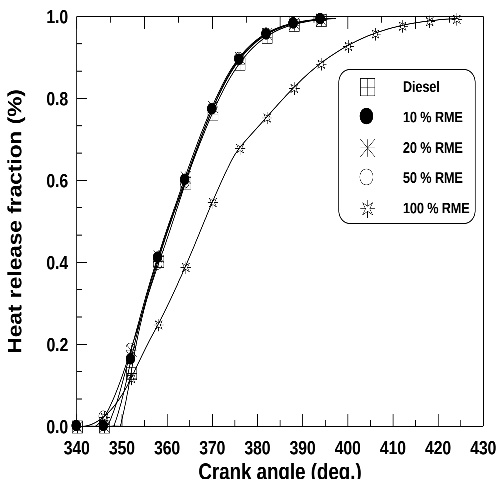


Fig. 6 Heat release fraction v/s crank angle

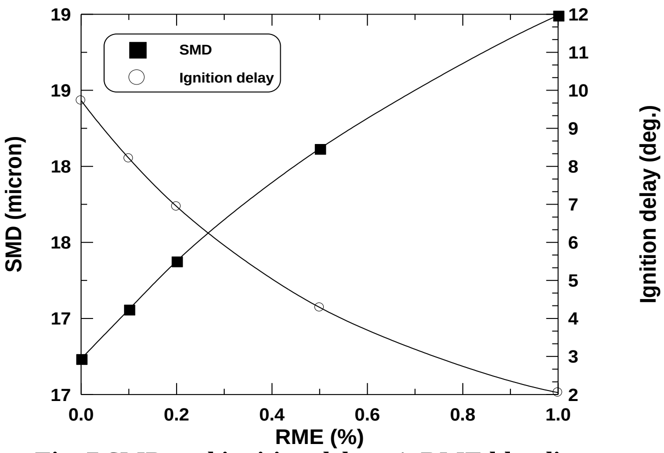


Fig. 7 SMD and ignition delay v/s RME blending

4.2 Performance parameters

Fig.8 examines the variant BSFC and the brake thermal efficiency (BTE) with RME biodiesel blends. It is found all blending of RME had lower BTE for the whole load by 0.5%. Decreasing trend of BTE due to the difference in heating values of the blended fuels.

The BSFC is increasable according to the increment in the blending ratio of RME. To obtain same power output and torque for each fuel examined, the BSFC was higher for RME and its blends. BSFC for B10 RME reported 2.763% rise, while it is higher by 12.03% for B100 RME. The difference in density and heating values are responsible for BSFC increment. The results are similar to those of (Mustafa et.al.,2001 ; Monyem , 1998).

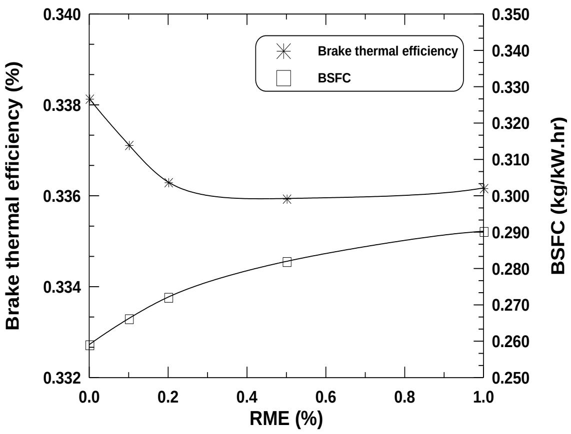


Fig. 8 BTE and BSFC v/s RME blending

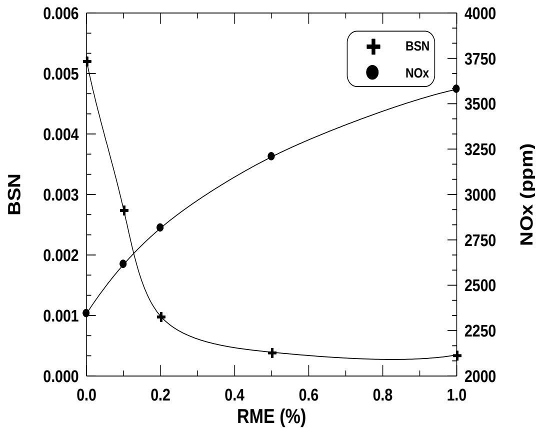


Fig. 9 BSN and NOx v/s RME blending

4.3 Emission parameters

Fig.9 explains the relationship between the Bosch smoke number (BSN) and NOx emissions with variable percentages of RME blend. The smoke levels for all RME percentages are less than pure petroleum. Average smoke data for 10% RME, 20%, RME, 50% RME and 100% RME was fewer comparable with diesel fuel by 47.27 %, 81.06 %, 82.56 % and 93.36 % respectively. The highly oxygen capacity of biodiesel is responsible for soot reduction which leads to complete oxidization, hence the oxygen molecules in the fuel decrease the trend of fuel for production of smoke, (Cengiz *et.al.,* 2009).

The emissions of NOx are higher for blends of RME biodiesel. This is because of the content of oxygen in the biodiesel hence it supplies extra oxygen for NOx formation.

The message from figure 9 says 10% RME was a best blending ratio as represents the intersection between two curves and even other percentages greater that 10% gives a good reduction in BSN but on the other side records a sharp rise in NOx emissions.

In Fig.10 it is noticed that the addition of RME from (0-100) % reports increase in the global NOx concentrations as compared to pure diesel operation. This is assigned to the zonal and global temperature increment. The zonal and global NOx concentrations increase sharply with time and reaches a maximum value and then drops slightly, this is due to increase in the oxygen concentration so the burned gas temperature is mixed with air (Van *et.al.,* 2000). Later, the concentration freezes due to temperature drop where the reactance cannot proceed.

Fig.11 presents the smoke concentration history with RME blending started from (10-100)%. It can be seen that as RME fuel ratio increases**,** the smoke level decreases, this is because of availability of oxygen in biodiesel as a result of biodiesel replacement and the high temperature which enhance soot oxidation.

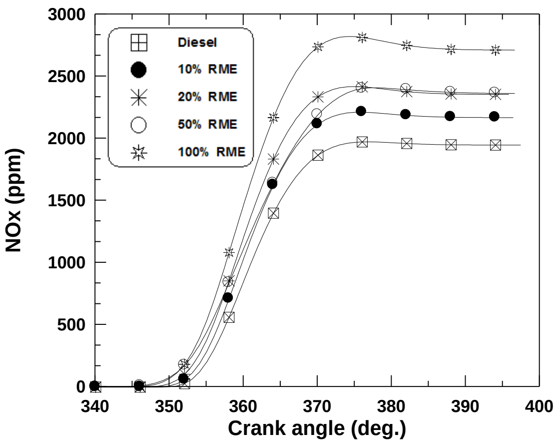


Fig. 10 NOx emission v/s crank angle

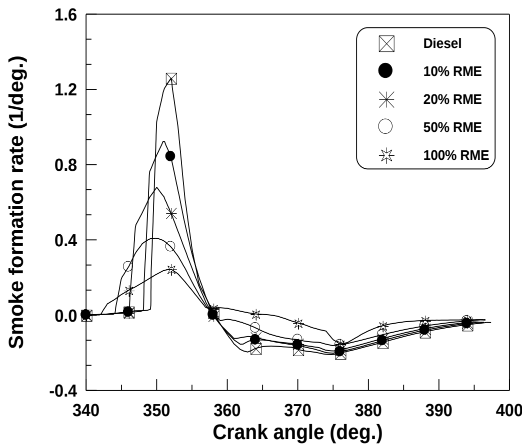


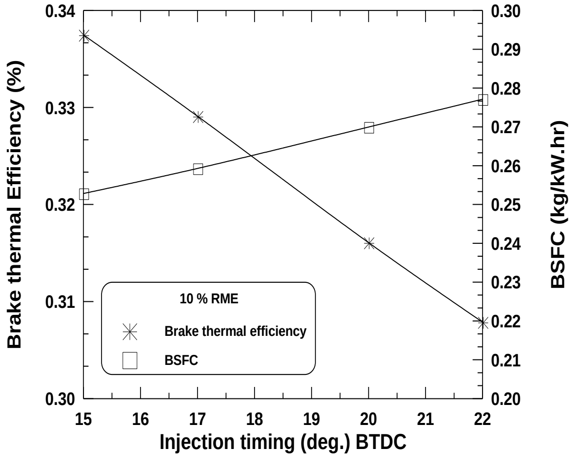
Fig. 11 smoke formation v/s crank angle

4.4 Parametric study

From the previous simulation, it was found that the best fuel combination is B10% RME which had same combustion results as well as promising emissions reduction. Also less increments in NOx emissions noticed with 10% RME as compared with other blends of RME biodiesel. This section deals with variation of injection timing to scale down smoke and NOx emissions from compression ignition engine working on 10% RME, at full load and constant speed of 1500 RPM.

4.5 Effect of injection timing

The results of simulation give an evidences of injection timing retarding from (22–15) ^{o}CA BTDC decreases fuel consumption and NOx by 6.36%, 22.43%, respectively, while increases BTE by 8.77% and smoke level (see Fig.12 and Fig.13). The time of injection has a considerable effect on NOx. Retarding injection timing, recorded a significant reductive in the nitrogen oxides. The results indicate that higher NOx emissions because of the earliness injection start. The best time of injection was noticed to be (18 degree BTDC) as representing the intersection between two curves in figures 12 and 13 respectively.



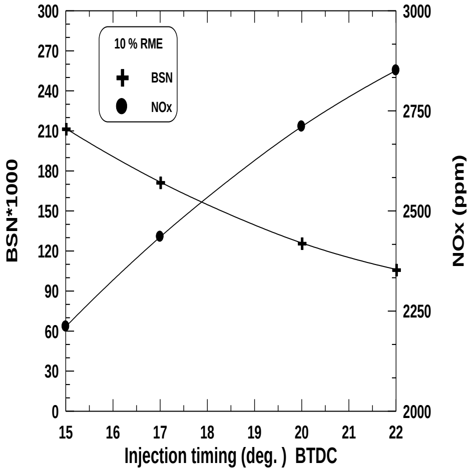


Fig. 12 BTE & BSFC v/s injection timing

Fig.13 BSN & NO_X v/s injection timing

5. Conclusions

1. Maximal pressure is observed to be nearer to top dead center whenever the ratio of RME is increased.
2. RME biodiesel has earliest combustion beginning comparative with original fuel.
3. Raising proportion of RME was reported a reduction in the BTE to a small extent and increased the fuel consumption.
4. A magnificent decrease in smoke levels of RME biodiesel comparing with baseline diesel.
5. RME biodiesel has highly NOx comparable with neat diesel.
6. B10% RME is the advisable mixing proportion that keeps the outcome of performance, reduces the emissions of carbon as well as a slight increment in the emissions of NOx comparable with other examined blends.
7. Retarding injection timing strategy comes with decrease NOx, fuel consumption and raise BTE and BSN respectively.
8. The best injection timing is 18 CA BTDC which is remarking little NOx increase for B10% RME

6. References

Alkidas A.C., 1984, Relationship between smoke measurements and particular measurements, J. SAE ,840412.

Canakci M. , 2007,Combustion characteristics of a turbocharged DI compression ignition engine fueled with petroleum diesel fuels and biodiesel. Bioresource Technology, Vol.98 (6), pp.1167-1175.

Cengiz O., Sehmus A., 2009. “Biodiesel production from inedible animal tallow and an experimental investigation of its use as alternative fuel in a direct injection diesel engine”. Applied energy, Vol.86(10), pp.2114-2120.

Dumitru bambuleac, 2012, Dynamical behavior of rapeseed oil and methyl ester of rapeseed oil during high-pressure injection, EPJ Web of Conferences 25,02002 .

Hansen A. W. Yuan, 2009 .“Computational investigation of the effect of biodiesel fuel properties on diesel engine NOx emissions”, International Journal of Agriculture and Biological Engineering, Vol.2 (2), pp.41-48.

Kim M.Y., Yoon S.H., Hwang J.W., Lee C.S., 2008,Characteristics of particulate emissions of compression ignition engine fueled with biodiesel derived from soybean, Journal of Engineering for Gas Turbine and Power 130 ,1–7.

Kuleshov A.S., 2005,Model for predicting air-fuel mixing, combustion and emissions in DI diesel engines over whole operating range, J. SAE -01-2119.

Labeckas, Stasys Slavinskas, 2006,Performance of direct-injection off-road diesel engine on rapeseed oil., Renewable Energy 31 ,849–863.

Lapuerta Magin,Octavio Armas, Jose Rodriguez-Fernandez,2008, Effect of biodiesel fuels on diesel engine emissions,Progress in Energy and Combustion Science 34,198–223

Mahkamov,A. Kuleshov ,2007, | Multi-zone diesel fuel spray combustion model for the simulation of a diesel engine running on biofuel", Proceedings of the Institution of Mechanical Engineers, Part A: J. Automobile Eng. 222 ,309-321.

Mohamed F. and Bhatti. S.K., 2014**,** Effect of Variable Compression Ratio on the Combustion, Performance and Emission Parameters of a Diesel Engine Fuelled with Diesel and Soybean Biodiesel Blending. World Applied Sciences Journal 30 (12): 1852-1858.

Monyem A. , 1998 "The effect of biodiesel oxidation on engine performance and emissions". Ph.D. thesis, Department of Mechanical Engineering, Iowa State University, Ames, IA.

Mustafa Canakci and Van Gerpen J.H. , 2001, "Comparison of Engine Performance and Emissions for Petroleum Diesel Fuel, Yellow Grease Biodiesel, and Soybean Oil Biodiesel". Presented as Paper No. 016050 at the 2001 ASAE Annual International Meeting Sacramento Convention Center Sacramento, California, USA July 30-August 1.

Roberto G., E.Oscar, D. Valdir, M. Dalni, T., 2011, Ednilton, Use of Soybean Oil in Energy Generation, Recent Trends for Enhancing the Diversity and Quality of Soybean Products,15-17.

Van Gerpen, Mustafa E. Tat, Jon H., Seref Soylu, Mustafa Canakci, Abdul Monyem, and Samuel Wormley. , 2000 "The Speed of Sound and Isentropic Bulk Modulus of Biodiesel at 21°C from Atmospheric Pressure to 35 MPa", JAOCS, Vol. 77 (12), pp. 1249-1256.

Wojciech Gis., A. towski, A. Bochska, 2011, Properties of the rapeseed oil methyl esters and comparing them with the diesel oil properties, J. of KONES Powertrain and Transport, 18 ,121-127.

Woschni, G., 1967,A Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine, SAE ,670931.

Yoon S. , Lee C., 2011, Experimental investigation on the combustion and exhaust emission characteristics of biogas–biodiesel dual-fuel combustion in a CI engine, J. Fuel precessing technology, 92 ,992–1000.

Yuan W.,2005,"Computational modeling of NOx emissions from biodiesel combustion based on accurate fuel properties", PhD thesis, Pennsylvania Avenue, Urbana .