SRTTU

JCARME
ISSN: 2228-7922
Journal of Computational and Applied Research in Mechanical Engineering
Vol. x, No. x, xxxx
jcarme.sru.ac.ir

Effect of fuel injection pressure of microalgae spirulina biodiesel blends on engine characteristics

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Article info:

Received: 00/00/2019
Accepted: 00/00/2019
Online: 00/00/2019

Keywords:
Numerical simulation,
CI engine,
Pressure,
Performance,
Combustion,
Emission,

Abstract

The unvarying condition diesel engines used for commercial applications, transportation and industries also lead to the crisis of petroleum fuel diminution and ecological squalor caused by due to exhaust gases. Therefore, in this paper optimize the use of MSB in naturally aspirated, direct injection diesel engines, parameters of pure diesel (D100), 80% diesel + 20% microalgae spirulina (B20), 60% diesel + 40% microalgae spirulina (B40) and pure microalgae spirulina biodiesel (B100) were investigated at various fuel injection pressures (FIP) of 18 to 26 MPa and stationary injection timings (23.5° b TDC). The result shows that optimum effect can be obtained in 22 MPa FIP, with B20 bio-diesel without compromising the performance against diesel. B20 blend presented lesser NO_X and smoke emissions by 13.7% and 22.2% respectively with no significant change in engine performance when compared to diesel at full load operating condition. The simulation and experiment results are verified at the same operating conditions.

Nomenclature

BN	Bosch number
BSN	Bosch smoke number
b TDC	Before top dead centre
CN	Cetane number of fuel
CP	Cylinder pressure (bar)
CPP	Cylinder peak pressure (bar)
CPHRR	Cylinder peak heat release rate (J/deg.)
ECD	Eddy current dynamometer
EGT	Exhaust gas temperature *(K)*
FM	Fuel measuring
HVM	Heat value measuring
FIP	Injection pressure (bar)
IT	Injection timing (deg.)
LI	Load indicator
m	Total mass (kg)
MSB	Microalgae *spirulina* biodiesel
n	Engine speed (rpm)
P	Pressure (bar)
PPM	Parts per million
PS	Pressure sensor
PTS	Pressure transducer sensor
SOI	Start of injection
SM	Smoke meter
SS	Speed sensor

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T	Temperature (K)
TS	Temperature sensor
x	Fraction of fuel burnt
W_U	Overall percentage of uncertainty (%)
AF_C	Air-fuel equivalence ratio at combustion
L_C	Cycle work (kj)
q_c	Cycle fuel mass (kg)
ν	Specific volume (m^3/kg)
E_a	Activation energy constant (kj/k mole)
m_f	Mass flow rate per cycle (kg/h)
V_C	Volume of cylinder (cm^3)
K_T	Evaporation constant
$\frac{dx}{dt}$	Heat release rate (J/deg.)
α	Constants
τ	Time (second)
ω	Angular crank velocity (rpm)
ξ_b	Cylinder air charge usage efficiency
$^\circ CA$	Crank angle degree
ϕ	Crank angle (deg.)
X_0	Burning fuel at ignition delay
$\frac{dx}{d\tau}$	Heat release rate (J/sec)
ρ	Density (kg/m^3)
CO_2	Carbon dioxide (g/kwh)
NO_X	Nitric oxide (ppm)
B0	100% diesel + 0% spirulina
B20	80% diesel + 20% spirulina
B40	60% diesel + 40% spirulina
B100	0% diesel + 100% spirulina

1. Introduction

Internal combustion engines were a necessary component in the modern era. Diesel engines were widely used due to its ability to produce higher torque, which in turns carries heavy loads. But over time, various methodologies have been adopted in extracting waste heat [1] and reduction of various pollutants for using diesel and other biofuels [2]. Biofuels from numerous sources have been continuously produced and extracted for use in diesel engines. One of the significant areas and latest biofuels production is the biodiesel extraction from microalgae [3, 7, 13].

Canola-safflower biodiesel with fuel additives such as solketal, ethanol has proven to reduce emissions such as CO, CO2, HC, while increasing NO_X [4]. The addition of hydrogen in the diesel engines has also been widely acclaimed to reduce emissions from the engine [5]. Homogenous charged compression ignition engines were also proven to have better combustion than standard diesel engines [6]. The structures of the nozzle have been studied further to check its effect on the mixture of fuel and air in a diesel engine. It was found that multi-hole nozzle have better homogenous mixing than the single hole nozzle [8].

As the future holds the path for clean energy [9, 11], the technologies under study by various researchers were to be deemed necessary. One such example of emission control is the capturing of CO2 by integrating the evaporative gas turbine with Oxy-fuel combustion [10]. The study for gas turbines can be effectively used for cogeneration plants using diesel engine where maximum power is produced and emission of minimum pollutants. Other parameters have also been checked for reduction of emission from diesel engines. Changing the piston bowl geometry to check the combustion and emission characteristics using biofuels was conducted. The study acknowledges that using shallow depth combustion chamber increases the emission of NO_X at low engine speed, as compared to the hemispherical combustion chamber and Omega combustion chamber [12].

Addition of alcohol with biodiesel and diesel [16] have reported reducing NO_X with an increase in fuel consumption and thermal efficiency. Emulsifiers such as Sorbitan monooleate and polyoxyethylene Sorbitan monooleate were also used for emulsification of biodiesel fuels. The study showed an increase in BTE and the reduction of exhaust gas temperature (EGT) [15]. Alternative fuels such as tire pyrolysis oil [17], waste cooking oil [18, 22], Mahua oil methyl ester [19], etc. have been successfully tested on a diesel engine for their engine characteristics.

Researchers have also predicted and concluded that among all types of alternative fuels, the combination of diesel, biodiesel, and alcohol have provided to be the most optimal case which can reduce maximum emission [20]. Another methodology such as reduction of emission using kapok methyl ester through the combined coating of partially stabilized Zirconia (PSZ) with B25 and B50 blends was also studied. The study revealed that using the thermal layer and B50 reduces the emissions of the diesel engine while showing significant increases in BTE [21].

Numerically investigated effects of first, second and third generation fuels on a diesel engine and evaluated combustion and emission characteristics at different engine load and compression ratio (16.5 to 18.5). The results showed that the reduction in soot emission with an increased compression ratio [36].

In this study, the effect of diesel, spirulina biodiesel and its blends with the variation of fuel injection pressure. Up to the authors best of knowledge, the impact of various fuel injection pressure (VFIP) using microalgae biodiesel has not been reported. In this present study usage of microalgae spirulina biodiesel (up to 40%) blend with diesel has been presented. In this study concentrates on the effect of VFIP (18 to 26 MPa) at different engine loading with CR17.5.

2. Experimental setup and procedures

2.1. Experimental Procedure

Experiments were performed on a single cylinder, liquid cooling, diesel engine. The technical specification of the naturally aspirated engine is given in Table 1. The single cylinder coupled with eddy current dynamometer and crank angle encoder coupled with engine shaft opposite to dynamometer as shown in Fig. 1. Measurement of engine parameters of combustion pressure with the help Kistler piezoelectric pressure sensor and mounted at the cylinder head, exhaust gas temperature using K-type temperature sensor at different position of engine setup. The engine speed and brake power ware about 1500 rpm and about 3.7 kW, respectively.

Table 1. Specification of engine

Parameters	**value**
Engine type	single cylinder four stroke
Fuel injection pressure	180 to 260 bar
Rated speed	1500 rpm
Engine cylinder/stroke	1/4
Bore x stroke	80 x 110 mm
CR	17.5
Connecting rod length	235 mm
Standard injection timing	23.5° b TDC
Eddy current dynamometer	model: power mag, 3. 7 kW
Fuel injection pressure	18-26 MPa

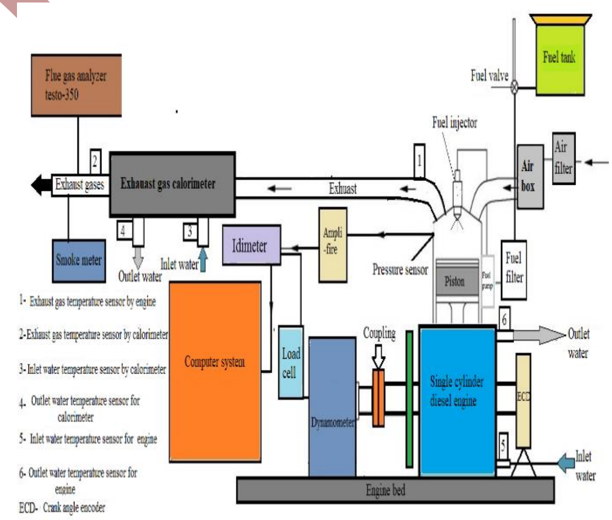


Fig.1. Multi-fuel single cylinder engine

2.2. Fuel properties

The fuel properties of diesel and microalgae spirulina alternative fuel for a diesel engine in the present study are taken from previous studied [23, 32, 30, 35]. The processor of biodiesel was produced from microalgae oil by using trans-esterification processing and properties were calculating according to ASTM standard [42].

The essential fuel properties of diesel and spirulina biodiesel are given in Table 2.

Table 2. Microalgae *spirulina* biodiesel and diesel properties

Properties	Microalgae spirulina biodiesel			
	B0	B20	B40	B100
Calorific value (MJ/kg)	42.5-45	41.4	40.9	41.36
Flash point (˚C)	52-65	61	78	130
Viscosity (mm^2/s) at 40 ˚C	2.4-4.59	3.66	4.26	5.66
Density (kg/m^3)	815-837.7	832.1	838.3	860

2.3. Error analysis

Calculating the total percentage of uncertainty analysis within the experimental setup by the well-known method of standard deviation are shown in equation (1-5). The percentage of uncertainty analysis all instrument as shown in Table 3. The calculating percentage of uncertainty (Wu) by standard deviation is found to be ± 2.21% and equation are given in Appendix 1.

Table 3. Accuracies and uncertainty of the instruments

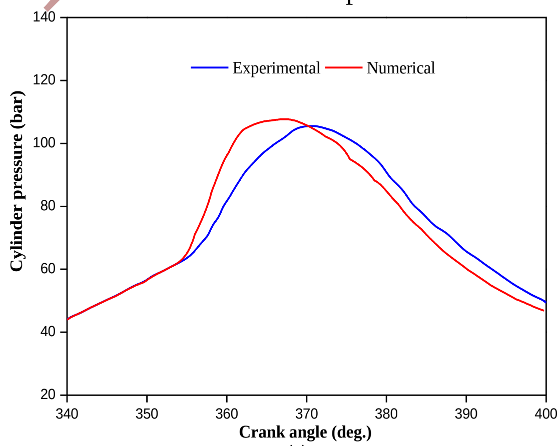
Instrument	parameter	uncertainty (%)
Eddy current dynamometer	load	±0.15
Speed sensor	rpm	±1.0
Load indicator	load	±0.2
Pressure sensor	cylinder pressure	±0.5
Crank angle encoder	angle	±0.2
Fuel measuring	heights of the liquid column	±0.5
Temperature sensor	temperature	±0.15
Heat value measured	heat value	±1.0
Smoke meter	smoke	±1.0
Testo 350 gas analyzer	CO_2 NO_X	± 1.0 ± 0.5

2.4. Model description

The numerical simulation with multi-zone diesel fuel spray combustion software Diesel-RK [14, 16, 24, 28, 29, 30, 36, 39]. Diesel-RK software based on zeldovich mechanism for NOX emission calculating. The Diesel-RK software based on the first law of thermodynamics and used to analyze the different characteristics of a compression ignition engine. The basic model equation is given in Appendix 1.

2.5. Validation of experimental and numerical results

The validation of Diesel-RK results against experiment results for cylinder pressure, thermal efficiency and NO_X emission using diesel fuel. The simulation results and experiment results have been depicted in Fig. 2 (a, b and c). The accuracy within the results as shown in Table 4. The maximum deviation was found to be 2.4% for cylinder pressure, 0.73% for thermal efficiency and 0.43% for NO_X emission. The input initial boundary condition is given in Table 1 for the simulation and experimental results.



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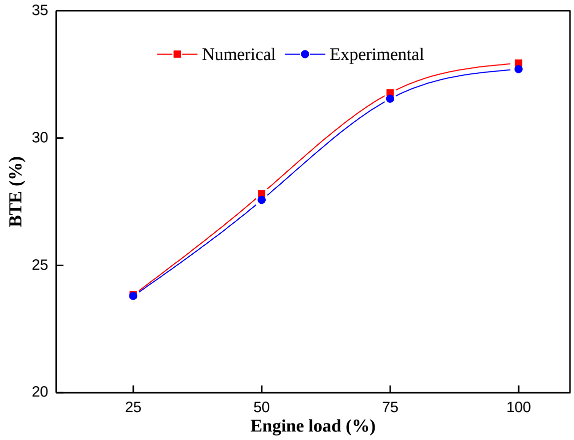

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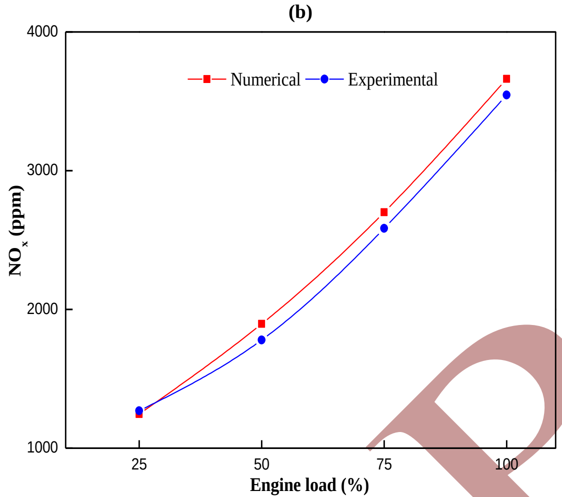


(c)

Fig. 2. Comparison of numerical and experimental results of at various engine loads at a) cylinder pressure versus crank angle, b) BTE versus engine load, and c) NO_X of diesel fuel at various engine loads at 220 bar

Table 4. Comparison between experimental data and numerical values at 100% load

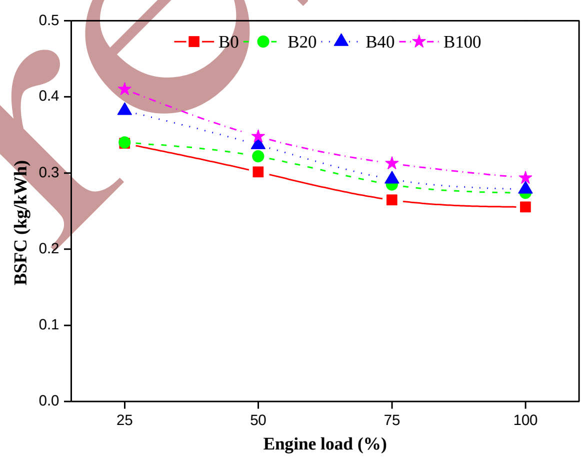
Parameters	numerical values	experimental results	error deviation
CPP (bar)	108.2	105.6	2.40%
BTE (%)	32.94	32.7	0.73%
NO_X (ppm)	3661	3645	0.43%

3. Results and discussion

3.1. Performance parameter of research engine

3.1.1. Brake specific fuel consumption (BSFC)

The small droplets size leads to better atomization of fuel. Lesser droplets size formed at higher fuel injection pressure (FIP) and gradually vaporizes small depicts. Further, to be increasing fuel consumption when increase FIP from 18 to 26 MPa due to deprived combustion and lower penetration length, deprived dispersion of the fuel rate and weaker air entrainment [23, 27, 34]. But with the increase in engine load, the BSFC decreases with increased engine load for all blend ratios of spirulina biodiesel and its blend ratio. The better combustion indicated lower fuel consumption due to the higher engine load. Fig. 3 variation of BSFC at engine loads by varying FIP of 18, 20, 22, 24 and 26 MPa. At full load, the BSFC (kg/kWh) was found to be 0.257 for D100, 0.274 for B20, 0.281 for B40, and 0.328 for B100 at 22 MPa. The comparisons of BSFC for spirulina (B20) and diesel (D100), results in higher BSFC (4.1%) at full load with FIP 22 MPa due to higher density and viscosity of microalgae spirulina biodiesel, as compared to diesel and higher droplets size than the increase in the amount of BSFC.



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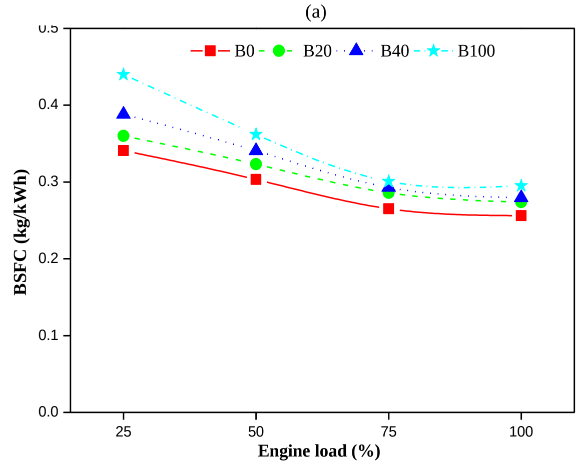

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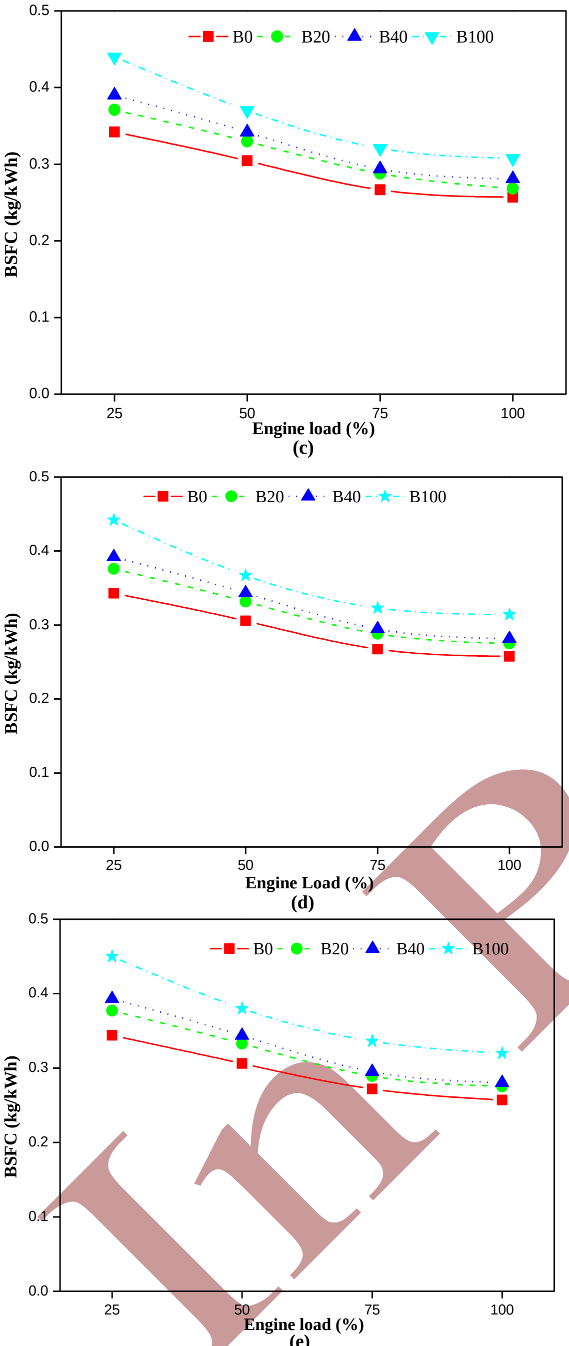
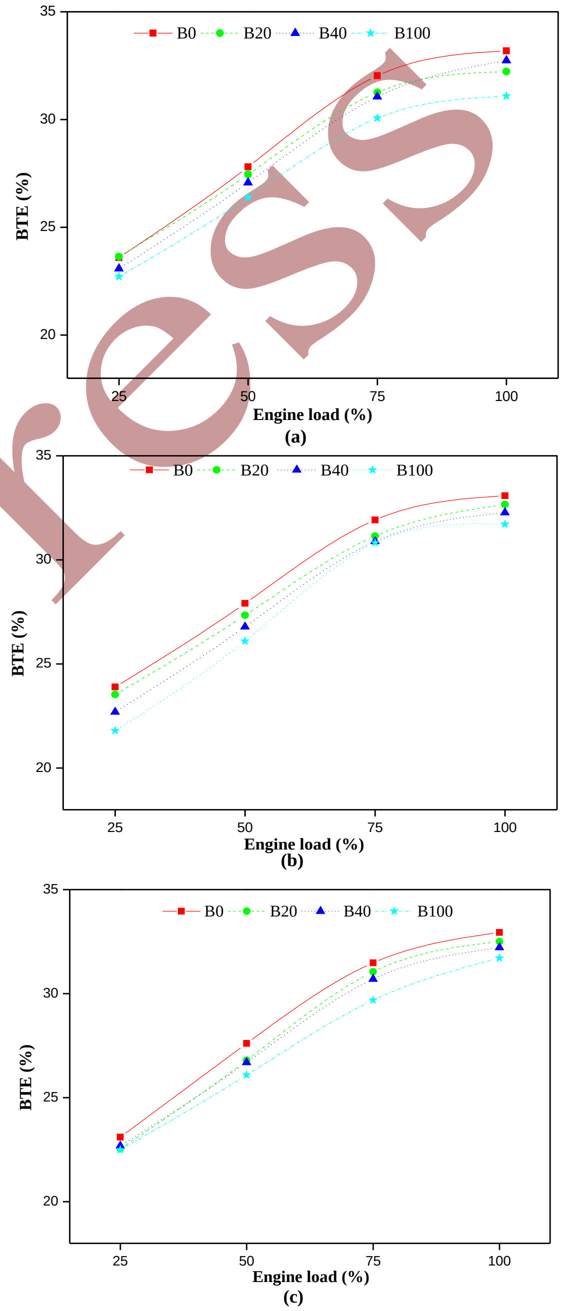


Fig. 3. BSFC of different blend ratio at various engine loads for a) 180 bar, b) 200 bar, c) 220 bar, d) 240 bar and e) 260 bar

3.1.2. Brake thermal efficiency (BTE)

The converting fuel chemical energy into engine output power is called BTE of compression ignition engine. The BTE is higher with an increase in higher engine load [23-24, 35]. Fig. 4 describe the variation of BTE with variation FIP and loads for microalgae spirulina biodiesel and its blends. The BTE (%) was found to be 33.19, 33.08, 32.94, 32.87 and 32.97 for D100; 32.22, 32.66, 32.5, 32.41 and 32.54 for B20; at FIP of 18, 20, 22, 24 and 26 MPa respectively at 100% load. At 100% engine load, was found to higher for diesel fuel as compared to microalgae biodiesel and its blends. The BTE of microalgae biodiesel (B20) biodiesel was found to be close with diesel.



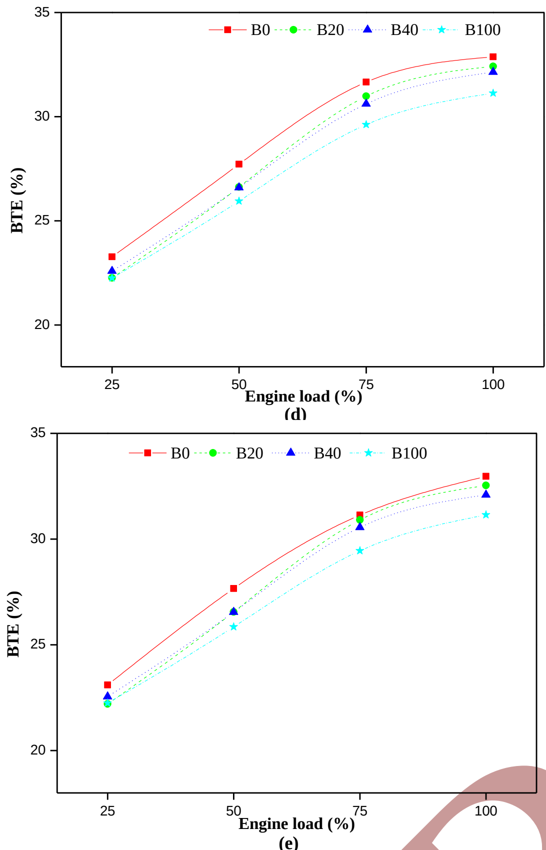


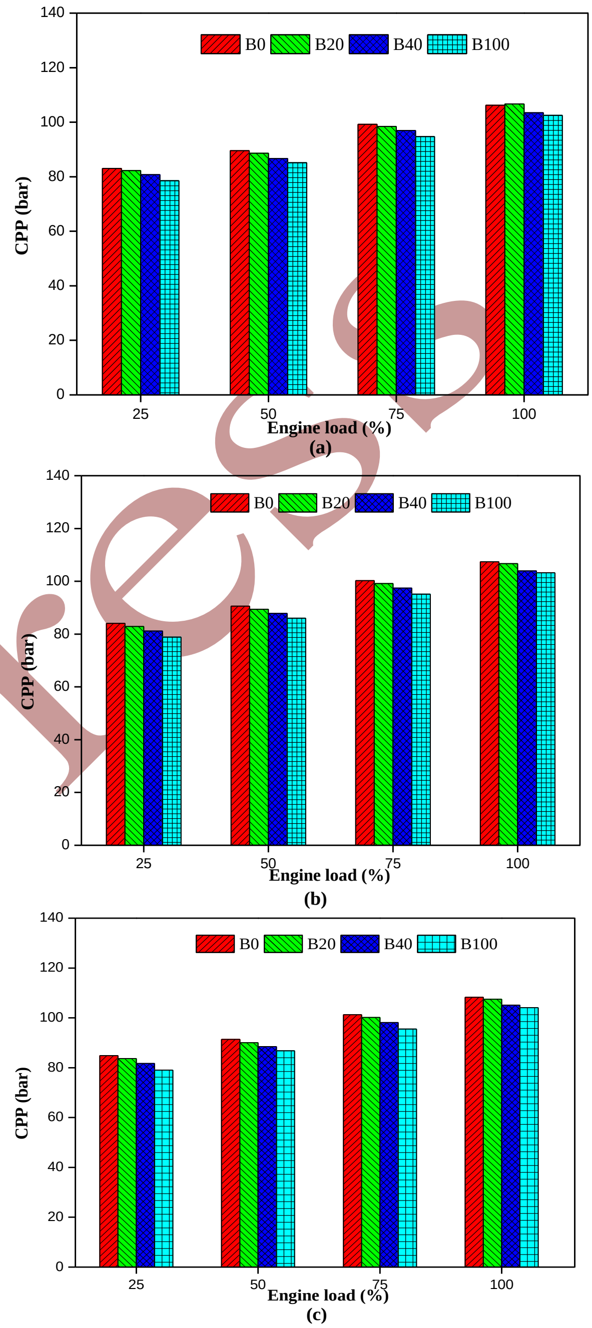
Fig. 4. BTE of different blend ratio at various engine loads for a) 180 bar, b) 200 bar, c) 220 bar, d) 240 bar and e) 260 bar

3.2 Combustion parameter of research engine

3.2.1. Cylinder pressure

The cylinder peak pressure (CPP) increases with increases in FIP and engine load due to better fuel atomization. The higher latent heat of biodiesel vaporization lead to poor fuel atomization and leads to lower CPP. CPP and cylinder peak temperature higher with high FIP due to the latent heat of vaporization fuel decreased [30-31, 37]. The CPP depends on fuel injected into the combustion chamber, ignition delay, and fuel consumption [23-24]. Fig. 5 (a-e) shows cylinder pressure versus engine load at different injection pressure. At higher loads, diesel displays higher CPP as compared to spirulina biodiesel and its blend fuels. At 100% load, it was observed that the CPP (bar) was found to be 106.21, 107.4, 108.2, 108.9, and 108.4 for diesel (D100); 106.1, 106.7, 107.5, 107.9 and 107.8 for spirulina biodiesel (B20) at 18, 20, 22, 24 and 26 MPa FIP respectively for full load condition. The CPP

while using B100 was found to be lower (2.4%) than that of D100 for 220 bar FIP, with CR17.5.



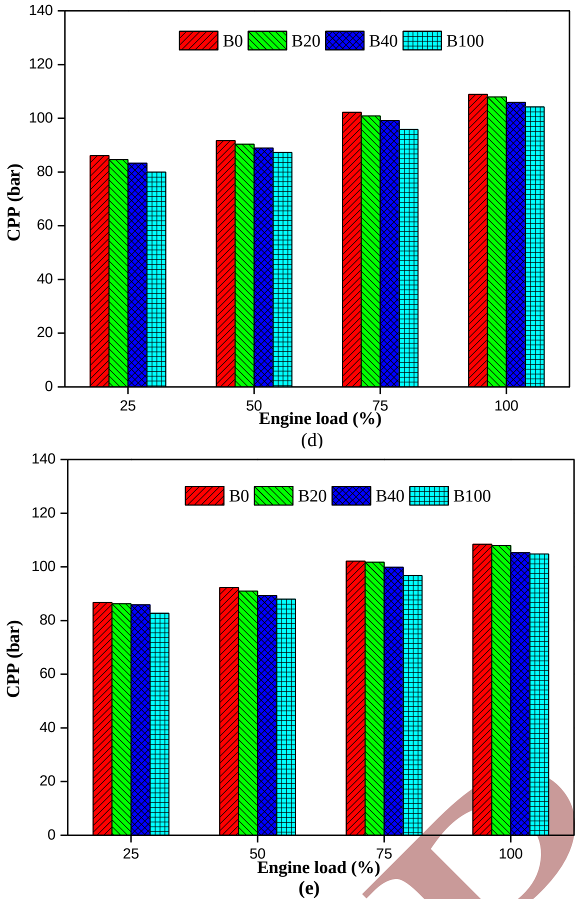


Fig. 5. CPP of different blend ratio at various engine loads for a) 180 bar, b) 200 bar, c) 220 bar, d) 240 bar and e) 260 bar

3.2.2. Heat release rate

Maximum heat release rate (MHRR) values and their locations were very near about to each other. The differences between in areas of MHRR were near about 2° CA for diesel and microalgae spirulina and its blends [4, 38, 41]. Fig. 6 (a-b) shows cylinder pressure versus crank angle at full and partial engine load for diesel, spirulina biodiesel and its blend with diesel at 220 bar FIP. At 100% load, diesel displays higher MHRR as compared to spirulina biodiesel and its blend fuels. At 50% and 100% load, it was observed that the MHRR (J/ deg.) was found to be 66.8 and 94.7 for diesel (D100) respectively. The value of 59.9, 57.8, 54.2 for spirulina biodiesel (B20, B40, B100) at partial load and 93.0, 89.5, 69.7 for spirulina biodiesel (B20, B40, B100) at at full load with FIP of 22 MPa. The MHRR while using B20% was found to be lower by1.8%, as compared to D100 for FIP of 22 MPa, with CR17.5.

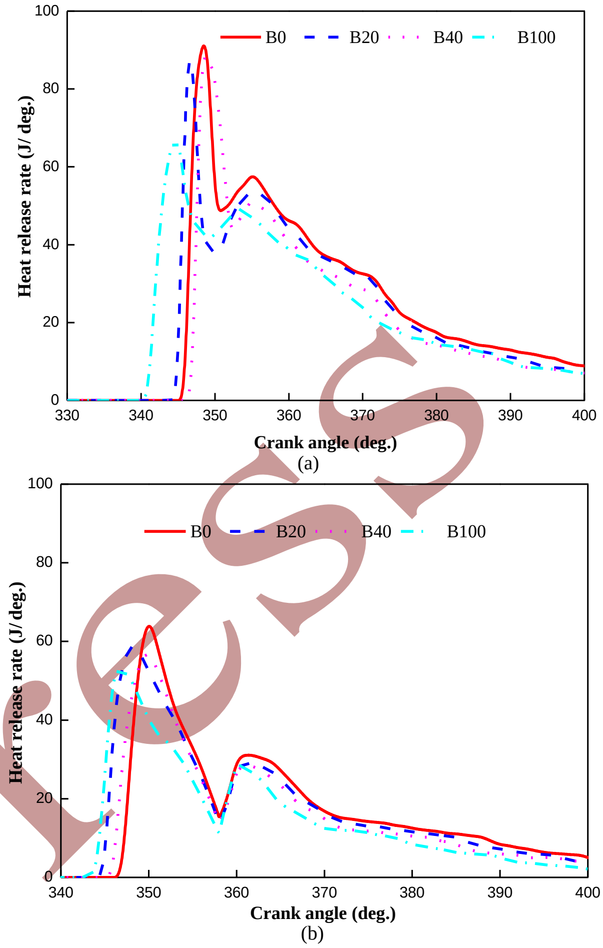
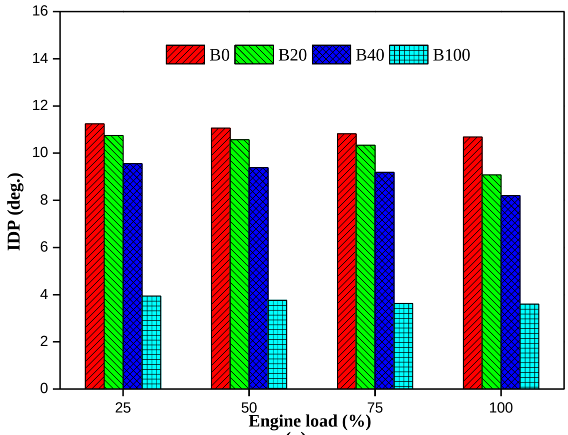


Fig. 6. HRR with crank angle at a) 100% and b) 50% load at 22 MPa

3.2.3. Ignition delay period (IDP)

The IDP, which is an outstanding design and performance parameter of the CI engine, is defined as the period difference between the beginning of fuel injection at the start of combustion. The ignition delay period is affected by different parameters of CI engine like CN, FIP, CR, RPM, and intake temperature and the air-fuel ratio [33, 30, 37]. The IDP of biodiesel and its blend was found to be lower compared to a diesel with higher FIP upon modification of the engine. Air-fuel mixing rate is better within the combustion cylinder described shorter ignition delay period. The longer IDP leads to extended air, and fuels mixing rate and it results in higher sudden heat release rate [23, 30, 31]. Fig. 7 (a-e) depicts ignition delay with engine loads for biodiesel and its blends. The ignition delays (degree) was found to be 10.68, 10.67, 10.66, 10.65 and 10.6 for diesel (D100), 9.07, 9.06, 9.03, 9.02 and 9.0 for spirulina biodiesel (B20) for FIP of 18, 20, 22, 24 and

26 MPa respectively, at 100% load condition. There is closeness in the IDP of diesel with spirulina biodiesel (B20).



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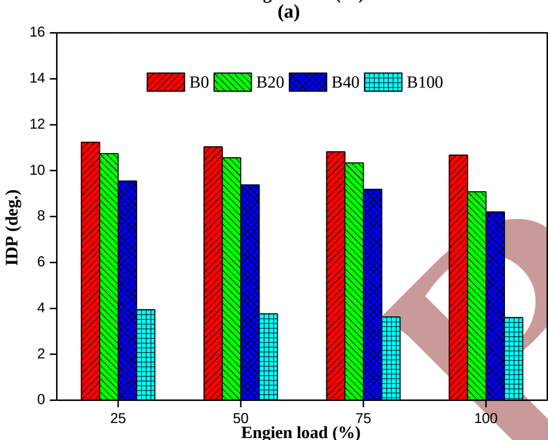

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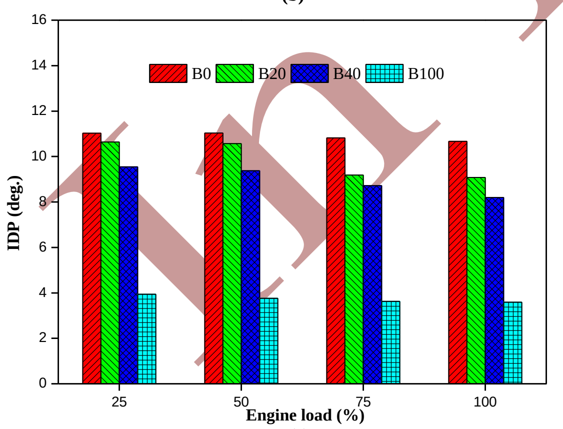

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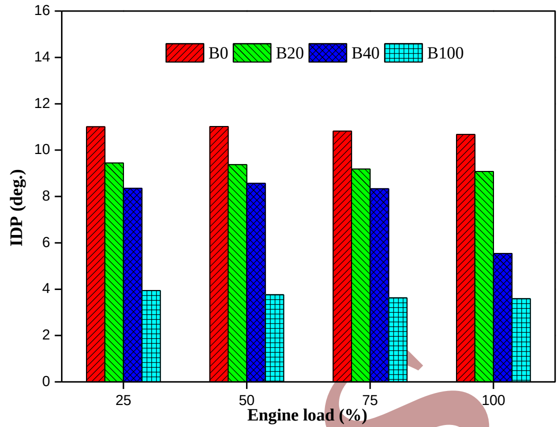

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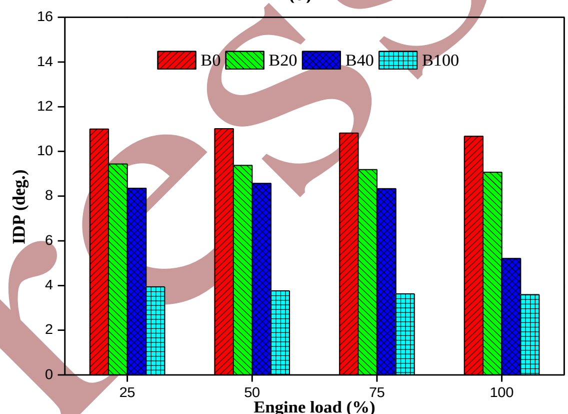


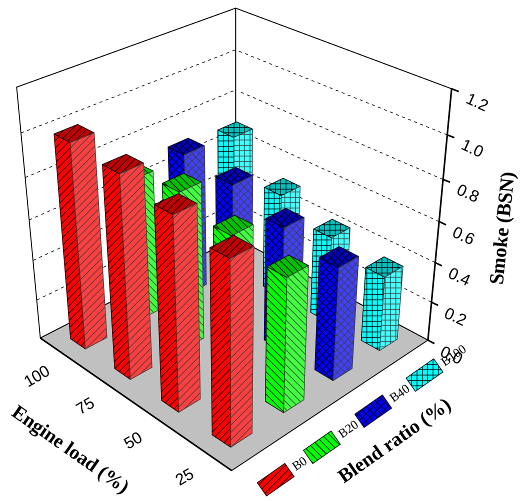
(e)

Fig. 7. Ignition delay period of different blend ratio at various engine loads for a) 180 bar, b) 200 bar, c) 220 bar, d) 240 bar and e) 260 bar

3.3 Emission parameter of research engine

3.3.1. Bosch smoke number

Bosch smoke number (BSN) at various injection pressures for the different biodiesel blends of microalgae spirulina biodiesel and its blends and diesel fuel for engine loads as shown in Fig. 8. At 100% load, the smoke emission (BSN) was found to be 1.02, 0.93, 0.90, 0.88 and 0.81 for diesel; 0.72, 0.71, 0.70, 0.68 and 0.62 for spirulina biodiesel (B20) at various FIP (18, 20, 22, 24 and 26 MPa), at 100% load condition respectively. BSN for spirulina biodiesel (B20) lower by 22.2% compared to diesel (D100) at FIP of 22 MPa with full load condition. Thus, smoke emission higher for diesel fuel compared to microalgae spirulina biodiesel and its blends due to the higher percentage of oxygen [23, 24]. It is clear from the figure with an increase in engine load smoke emission increasing.



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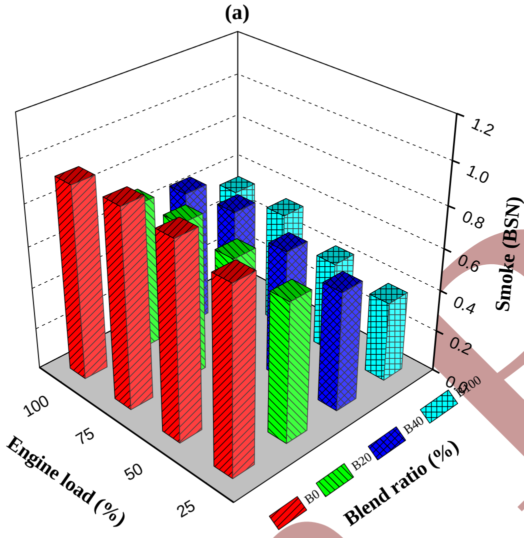

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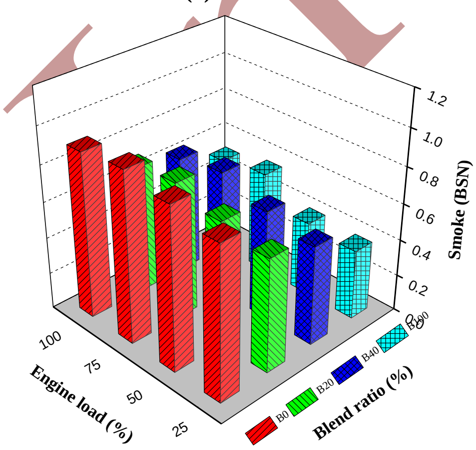

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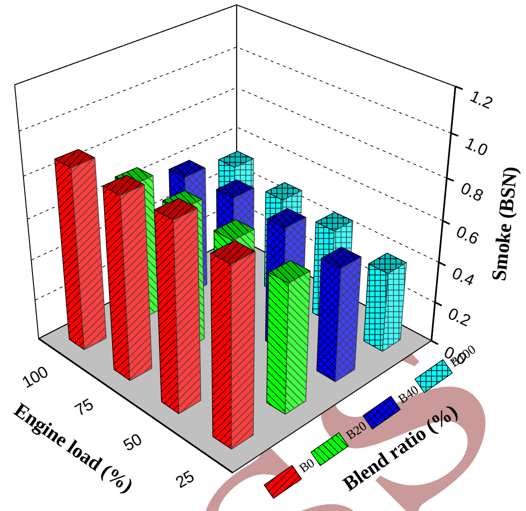

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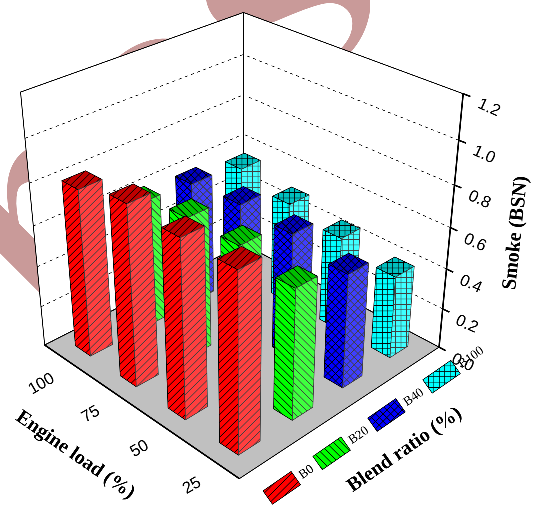

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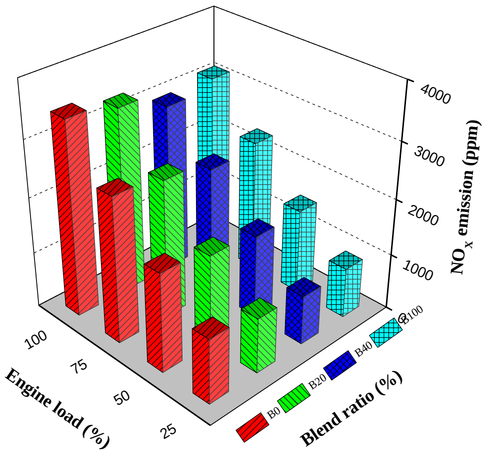
Fig. 8. Smoke level of different blend ratio at various engine loads for a) 180 bar, b) 200 bar, c) 220 bar, d) 240 bar and e) 260 bar

3.3.2. Fraction of wet NO_X in exhaust gas

The NO_X emission at various injection pressures for the different biodiesel blends of spirulina biodiesel with diesel fuels and different loading are shown in Fig. 9. At 100% load, the NO_X (ppm) was found to be 3473.6, 3576, 3661, 3715 and 3774 for diesel; 3304.6, 3305, 3389, 3456 and 3497 for spirulina biodiesel (B20) at various FIP (18, 20, 22, 24 and 26 MPa) respectively. The fraction of NO_X emission is lowered by 7.4% for spirulina biodiesel (B20) compared to diesel (D100) at FIP of 22 MPa with full load condition. Thus, NO_X emission was found to be higher for diesel fuel compared to spirulina biodiesel and its blends. The NO_X increases with increasing load due to higher combustion temperature. It can be observed that the NO_X emission is dependents on the combustion temperature, oxygen contents. Low combustion heat release rate led to low NO_X emission [16, 30].



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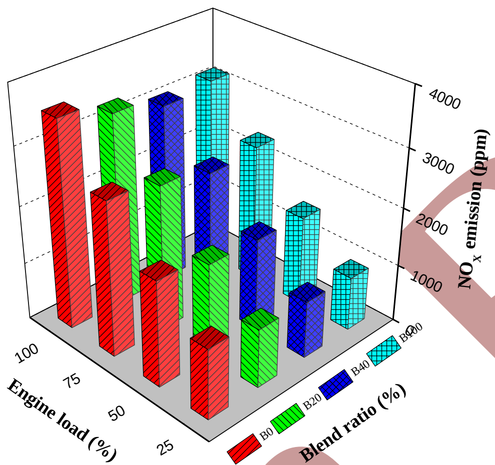

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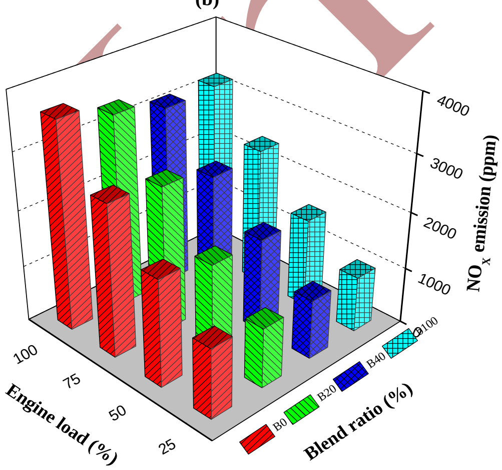

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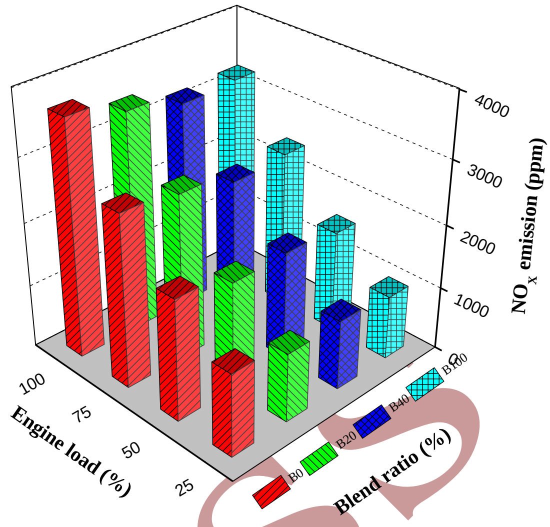

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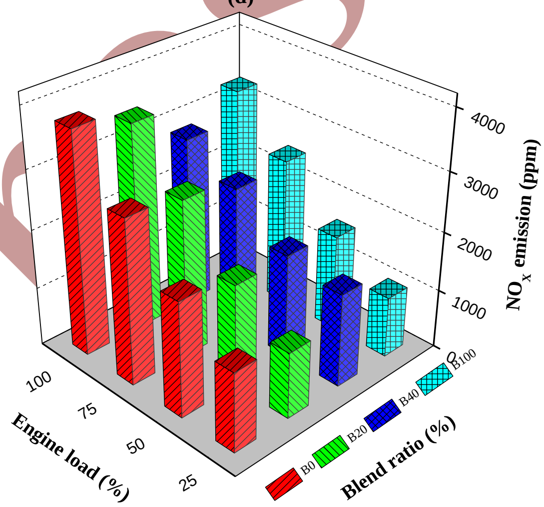

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Fig. 9. NO_X emission of different blend ratio at various engine loads for a) 180 bar, b) 200 bar, c) 220 bar, d) 240 bar and e) 260 bar

4. Conclusion

A summary of the present study are following:

- Microalgae spirulina biodiesel has been used as an alternative fuel in place of diesel.
- In all the blend ratio of spirulina biodiesel, IDP is short as compared to D100, due to a higher CN of spirulina biodiesel.
- While snowballing the percentage of biodiesel, there is a decrease in BTE and EGT, while the increase in BSFC.
- At 100% load, BTE was highest for the diesel. The BTE was found to be lower by 1.33% for spirulina biodiesel (B20) compared to diesel at full load conditions with FIP (22 MPa).

- At full loading, BTE is lowered by 1.33%, but BSFC was higher by 4.1% for spirulina biodiesel (B20) as compared to diesel (D100) at FIP of 22 MPa.
- The MHRR while using spirulina biodiesel (B20) was found to be lower by 1.8%, as compared to diesel fuel (D100) for FIP of 22 MPa, with CR17.5.
- With an increase in engine load, it shows a potential reduction in CO_2 emission.

B20 (80% diesel and 20% spirulina microalgae biodiesel) can be used as an alternative fuel in diesel engines; in this approach emissions values can be reduced.

Acknowledgements

The authors are grateful to the National Institute of Technology Manipur supporting research work.

Appendix 1
Uncertainty analysis

The uncertainty in each individual amount (Xi) leads to accuracy of results for any variable ''U'' that is computed from n independent measurement of the following relation [22, 27, 28, 30, 36, 40].

$$U = a_1 x_1 + a_2 x_2 + \cdots\cdots\cdots a_n x_n = \sum a_i x_i \quad (1)$$

$$\frac{\partial U}{\partial x_i} = a_i \quad (2)$$

Total percentage of uncertainty in the result may then be following equation as

$$w_U = \left\{ \sum \left[\left(\frac{\partial U}{\partial x_i} \right)^2 w^2_{x_i} \right] \right\}^2 \quad (3)$$

$$w_U = \left\{ \sum \left[(a_i)^2 w^2_{x_i} \right] \right\}^2 \quad (4)$$

$$w_U = \sqrt{\left((0.15)_{ECD}{}^2 + (1)_{SS}{}^2 + (0.2)_{LI}{}^2 + (0.5)_{PS}{}^2 + (0.2)_{CAE}{}^2 + (0.5)_{FM}{}^2 + (0.15)_{TS}{}^2 + (1)_{HVM}{}^2 + (1)_{SM}{}^2 + (1)_{CO_2}{}^2 + (0.5)_{NO_X}{}^2 \right)} \quad (5)$$

Diesel-RK model equation

The governing equation for Diesel-RK model are given in equation (6-14) [16, 27, 30].

Conservation of energy

$$\frac{d(mu)}{dt} = -p\frac{dv}{dt} + \frac{dQ_{ht}}{dt} + \sum_j \dot{m}_j h_j \quad (6)$$

Heat model

(a) Ignition delay period model

$$\tau = 3.8\times10^{-6}(1-1.6\times10^{-4}.n)\sqrt{\frac{T}{p}}\exp\left(\frac{E_a}{8.312T} - \frac{70}{CN+25}\right) \quad \mathbf{(7)}$$

(b) Premixed combustion period model

$$\frac{dx}{d\tau} = \Phi_0 \times \left(A_0 \left({}^{m_f}\!/_{v_i} \right) \times (\sigma_{ud} - x_0) \times (0.1\times\sigma_{ud} + x_0) \right) + \Phi_1 \times \left(\frac{d\sigma_u}{d\tau} \right) \quad (8)$$

(c) Controlled combustion period model

$$\frac{dx}{d\tau} = \Phi_1 \times \left(\frac{d\sigma_u}{d\tau} \right) + \Phi_2 \times \left(A_2 \left({}^{m_f}\!/_{v_c} \right) \times (\sigma_u - x) \times (\alpha - x) \right) \quad (9)$$

(d) Burning period model

$$\frac{dx}{d\tau} = \Phi_3 A_3 K_T (1-x)(\xi_b \alpha - x) \quad (10)$$

NO_X formation model

The calculating NOX emission based on Zeldovich mechanism. The NOX emission grouped of nitric oxide (NO) and nitrogen dioxide (NO2). The zeldovich mechanism shown in equation (11-14) [14, 24].

$$[O_2] \leftrightarrow [2O] \quad (11)$$

$$[N_2] + [O] \leftrightarrow [NO] + [N] \quad (12)$$

$$[N] + [O_2] \leftrightarrow [NO] + [O] \quad (13)$$

$$\frac{d[NO]}{d\theta} = \frac{P\times 2.333\times10^{-7}.e^{-\frac{38020}{T_b}} [N_2]_e \cdot [O]_e \cdot \left\{ 1 - \left({}^{[NO]}\!/_{[NO]_e} \right)^2 \right\}}{R.T_b \cdot \left(1 + \frac{2365}{T_b}.e^{\frac{2365}{T_b}} \cdot \frac{[NO]}{[O_2]_e} \right)} \cdot \frac{1}{\omega} \quad (14)$$

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