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EFFECT OF COMPRESSION RATIO ON THERMAL CHARACTERISTICS OF VCR DIESEL ENGINE USING *NICOTIANA TABACUM L.* SEED OIL METHYL ESTER

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Abstract

The use of diesel engines has been increased in the field of transport sector and agriculture due to their high power output and fuel economy. Though the concept of bio-fuel is not a new one, no vehicle is running on road in India. The selection of oil for biodiesel production and stringent emission norms are being the important factors. On the other hand the modern engine technology has to address the challenges of high power output, fuel economy, control of peak cylinder pressure and cold start ability against the fixed compression ratio diesel engines when fuelled with both conventional and bio-fuels. The above mentioned challenges can be addressed by employing the non-edible oils along with the variable compression ratio technology. Different kinds of edible/non-edible oils and their methyl/ethyl esters have been tested in C.I. engines with fixed compression ratio and few of them with variable compression ratio (VCR) also. However, *Nicotiana Tabacum L.* Seed Oil (NTSOME) has not been tested in VCR diesel engines, yet. Tobacco seed oil is non-edible oil and it is the by-product of tobacco leaves process. In this direction, the concern of the present experimental investigation is the assessment of NTSOME as an alternative fuel in a VCR diesel engine. The use of NTSOME in a VCR engine at higher compression ratio results in shorter ignition delay, smaller rate of pressure rise, higher heat release rate and insignificant decrease in mass fraction burnt when compared to that of diesel operation. At high compression ratio, the NTSOME was found to exhibit maximum thermal efficiency. Also, the results show that there is significant reduction in NOx and smoke compared to diesel fuel.

Keywords: *Nicotiana Tabacum L.*seed oil; Methyl ester, Variable compression ratio, Combustion, Emission, Diesel engine.

Abbreviations

ASTM	American Society for Testing and Materials
BIS	Bureau of Indian Standards
BSEC	Brake Specific Energy Consumption
BTE	Brake Thermal Efficiency
CA	Crank Angle
CHR	Cumulative Heat Release rate
CP	Cylinder Pressure
CR	Compression Ratio
NOx	Oxides of Nitrogen
NTSOME	*Nicotiana Tabacum L.* Seed Oil Methyl Ester
ROPR	Rate of Pressure Rise
ROPR	Rate of Pressure Rise
VCR	Variable Compression Ratio

1. Introduction

The rapid environmental changes and the uncertainty concerning future energy contributions are the consequences of conventional engine technologies. The immediate development of high efficient technologies and the availability of those technologies are the indispensable actions for the present vehicles as quickly as possible. The new technologies like hybrid vehicles, enrichment of oxygen and turbo-charging are well known for the improvement of the fuel economy and output of the engine. Variable Compression ratio is one of the most promising internal combustion engine concepts for the future. The VCR engines in their design include different methods for varying the compression ratio such as Tilting the cylinder head, Lifting the cylinder head [1], Auxiliary combustion chamber in cylinder head, varying the volume of the Chamber, piston with variable compression height, modifying the connecting rod geometry, crankpin adjustment within the crankshaft and Eccentric the crankshaft axis [2].

Among all the methods mentioned above, tilting the cylinder block method is one of the arrangements where the compression ratio can be changed without any change in the combustion geometry. Moreover, this method facilitates in changing of compression ratio within designed range without bringing the engine to stop. On the other hand, rapid changes in the availability of conventional energy sources made the researchers attentive in search of alternative energy sources. In this view, a large number of studies were reported but only few vehicles have adopted the gaseous fuels as alternate energy source in India and none of the vehicles have looked into alternate fuels like the fuels from vegetable oils. Also, these biodiesels have not been implemented for the vehicles in India.

In this concern, the planning commission of India has successfully launched the bio fuel project in 200 districts of the country to bridge the gap identified in utilizing straight vegetable oils. If the biodiesels produced from the non-edible oils are used on a mass scale, the cost of biodiesels will be reduced greatly and also the food industry will remain unaffected. A variety of first and second generation fuels like poon oil [3], cottonseed oil [4], Mahua oil [5, 6], Jatropha oil [7, 8], Terebinth oil [9], Soybean oil [10], pungamia oil [11], Palm oil [12], Rice bran oil [13], Annona oil [14], diesel and ethanol [15], Rapeseed oil [16], Biogas [17],Castor and karanja oils [18], Mahua oil and methanol [19], ethanol [20] have been tested in C.I. engines with fixed compression ratio to get best suitable alternative to diesel fuel.

The works like Effect of varying 9-octadecenoic acid (oleic fatty acid) content in biofuel [21], combustion study of castor oil and its blends [22], Alvar cycle engine with Iso-octane [23], performance and emission characteristics of cator oil [24], operating characteristics of tamanu oil [25] with variable compression ratio facility have been studied but simultaneous study of performance combustion and emission characteristics of the engine with variable compression ratio not reported extensively.

India is the world's 2nd largest producer of tobacco with an estimated annual production of 800 million kg. Tobacco occupies a meagre 0.24% of the country's total arable land area. It is grown largely in semi-arid and rain-fed areas where the cultivation of alternative crops is economically unviable [26]. The seeds of the tobacco plant are very small in size, but they come in an extremely large quantity per plant. They can be preserved for a long time if they are stored in dry conditions, resistant to rather high humidity at ordinary temperatures, and have a strong shell.

The tobacco seeds endosperm contains thin walled cells and affluent in oil. The oil extracted from tobacco seed is non-edible oil. The physical, chemical and fuel related properties of tobacco seed oil showed that tobacco seed oil may be an appropriate substitute for diesel fuel [27, 28]. To address the concern, an attempt has been made on a compact, constant speed, direct injection diesel engine with variable compression ratio to run on pure biodiesel (NTSOME).

The effect of compression ratio on engine combustion and emission characteristics has not been studied extensively. Also, studies on variable compression ratio using NTSOME was not found in the open literature to the best of the knowledge of the authors. In this study experiments are carried out for combustion and emission characteristics of VCR engine using NTSOME and diesel at compression ratios 15:1, 16:1, 17:1, 17.5:1 and 18:1 from zero load to full load conditions and the combustion parameters such as variation of cylinder pressure, maximum rate of pressure rise, heat release rate and mass fraction burnt are discussed with reference to the crank angle for different compression ratios.

2. Transesterification of *Nicotiana Tabacum L.* Seed Oil (NTSO)

Transesterification (also called alcoholysis) is the reaction of a fat or oil with an alcohol to form esters and glycerol. In the present work transesterification was carried out using both acid treatment and base treatment [28, 29]. Sodium hydroxide is used as a Catalyst to improve the reaction rate and yield. Raw oil is filtered and heated to 100 ^{o}C and cooled to 60 ^{o}C to remove any moisture content in the oil before acid treatment. Methanol was added to the oil at this temperature and stirred for 5 min maintaining the same temperature. Then acid catalyst (H_2SO_4) was added to this mixture and stirred for 1.5 hours constantly and maintaining the temperature of mixture between 50 ^{o}C to 60 ^{o}C.

The stirred mixture was allowed to be stable about 2 hours to get settled the oil and glycerin. Glycerine was settled at the bottom of the separating funnel and the upper layer was the ester. The glycerine layer was removed at the end of the settling. After the acid treatment process, again the oil was heated up to 50 ^{o}C and the methoxide (NaOH + methanol) was added at this temperature. The mixture was allowed to form two layers i.e. glycerin and ester after which the glycerin was removed. The ester was washed with pure water along with phosphoric acid proportionately and heated to 100 °C to remove any water from the oil left in the ester.

Table 1 shows the comparison of the various properties of the biodiesel derived from NTSO properties according to ASTM standards [30] compared with the diesel.

Table 1. Properties of raw NTSO, NTSOME, and diesel.

Property	Raw NTSO	NTSOME	ASTM* standards [30]	Diesel
Iodine value (g I_2/100)	471	110	Max 130	-
Acidvalue (mgKOH/g)	36.6	0.49	Max 0.8	-
Cetane index	30	35	45	40-55
Saponification number	192.5	95	195	-
Kinematic Viscosity at 38°C (cSt)	41.2	0.04125	1.9-6.0	0.01483
Density (Kg/m³)	901	870	860-900	840
Flash Point (°C)	252	174	>130	56
Pour point (°C)		≤15	-15 to 10	-40
Calorific Value (MJ/kg)	32.5	37.5	-	42.5

***ASTM: American Society for Testing and Materials**

3. Experimental Setup

Figure 1 shows the experimental setup of variable compression ratio diesel engine. Table 2 indicates the specifications of the variable compression ratio diesel engine and mechanical parameters adopted for the present experiment. Engine compression ratio was changed by cylinder head tilting method. This was changed by rotating the adjuster which tilts the cylinder block so that the compression ratio can be set to the desired position. Measuring and noting down the centre distance between two pivot pins of the CR indicator can be used to know new CR. Loading of the VCR engine was done the help of an eddy current dynamometer.

The test setup also has a provision for measuring the combustion pressure, crank angle, airflow, fuel flow, temperatures, and load measurements by using different sensors. The rotameter was used for the measurement of cooling water and the water flow in calorimeter. AVL 5 Gas analyser has been used for the measurement of NOx. AVL Smoke meter has been employed for the measurement of smoke. A constant engine speed of 1500 rpm was maintained to conduct all the tests.

4. Error Analysis

Errors will creep into all experiments regardless of the care which is exerted. Errors and uncertainties in the experiments can arise from instrument selection, condition, calibration, environment, observation, reading and test planning. Uncertainty analysis is needed to prove the accuracy of the experiments [31]. The percentage uncertainties of various parameters like brake power and brake thermal efficiency were calculated using the percentage uncertainties of various instruments given in Table 3. An uncertainly analysis was performed using the following equation [32]:

Total percentage of uncertainty = Square root of (uncertainty of $BSEC^2$+ uncertainty of BTE^2+uncertainty of NOx^2+ uncertainty of Smoke $opacity^2$ uncertainty of pressure pickup) = $\pm$1.11%

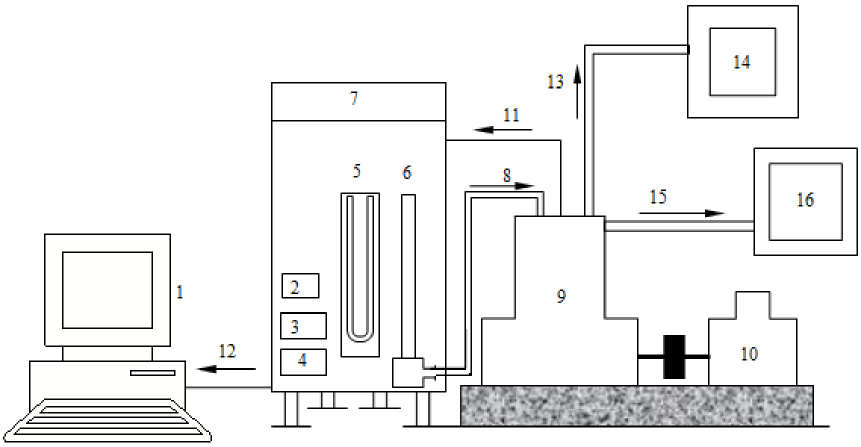
Table 2. (a) Engine specifications and (b) Mechanical parameters adopted for the present experiment.

(z)Make	:	Kirloskar Oil Engines
Model	:	TV1
Type	:	1 cylinder
No. of strokes	:	4 stroke
Type of cooling	:	water cooled
Stroke	:	110mm
Bore	:	87.5 mm
Capacity	:	661 cc
Power	:	3.5 KW at 1500 rpm
CR range	:	12:1-18:1
Injection variation	:	0- 23 Deg before TDC

Sl. no	Parameters	Specifications	Sl. no	Parameters	Specifications
(b)	Specific gas constant (kJ/kgK)	1.00	8.	Orifice coefficient of discharge	0.60
1.	Air density (kg/m^3)	1.17	9.	Dynamometer arm length (mm)	185
2.	Adiabatic index	1.41	10.	Fuel pipe dia (mm)	12.40
3.	Polytrophic index	1.45	11.	Ambient temp. (°C)	27
4.	Number of cycles	10	12.	Pulses Per revolution	360
5.	Cylinder pressure reference	3	13.	Fuel type	Diesel
6.	Top dead center (TDC) reference	0	14.	Fuel density (kg/m^3)	840
7.	Orifice diameter (mm)	20.00	15.	Calorific value of fuel (MJ/kg)	42.5

Table 3. List of instruments and its accuracy and percentage uncertainties for the present experiment.

Measurement	Accuracy	Percentage uncertainty
Engine speed	$\pm$ 30 rpm	$\pm$ 0.2
Temperatures	$\pm$ 1°C	$\pm$ 0.2
Nitrogen oxides	$\pm$10 ppm	$\pm$ 0.2
Smoke	$\pm$ 1%	$\pm$ 1
Crank angle encoder	$\pm$ 0.5°CA	$\pm$ 0.2
Load	$\pm$ 0.1kg	$\pm$ 0.2
Burette	$\pm$ 0.1 cc	$\pm$ 1
Time	$\pm$ 0.1 s	$\pm$ 0.2
Manometer	$\pm$ 1 mm	$\pm$ 1



1. Computer with data Acquisition system 2. Temperature Indicator 3. Load Indicator 4. Speed Indicator 5. Manometer 6. Burette 7. Fuel Tank 8. Fuel Supply from burette 9.VCR Diesel Engine 10. Eddy current Dynamometer 11. Data sent from the engine to data acquisition system 12. Data transfer from acquisition system to computer 13. Exhaust gases to Exhaust Gas Analyser 14. Exhaust Gas Analyser 15 Exhaust gases to Smoke meter 16. Smoke meter

Fig. 1. Layout of VCR Engine connected with gas analyser and smoke meter.

5. Results and Discussion

The graphs are plotted for different performance parameters with respect to different compression ratios. The parameters considered are brake thermal efficiency, cylinder pressure, heat release rate, mass fraction burnt, NOx emissions and smoke opacity. The performance data of NTSOME was compared with baseline data of diesel.

5.1. Brake specific energy consumption

Figure 2 depicts the variation of brake specific energy consumption (BSEC) at different loads and at different compression ratios. It can be clearly observed that the brake specific energy consumption of the diesel fuel at CR-15 is slightly higher than the considered remaining compression ratios. Also, the experimental results divulge that as the load on the engine increases, BSEC decreases for both diesel and NTSOME.It is lucid that for NTSOME, the lower energy consumption is 3.71×10^4 kJ/kWh corresponding to the compression ratio of 18 whereas it is 3.06×10^4 kJ/kWh for diesel. It is observed that the BSEC decreases with the increase in compression ratio.

This can be attributed to the high compression ratio and energy required per kW is less than that of lower compression ratio. Due to increase in compression temperature, complete combustion of the fuel occurs. Consecutively it minimizes the self-ignition temperature of the fuel.

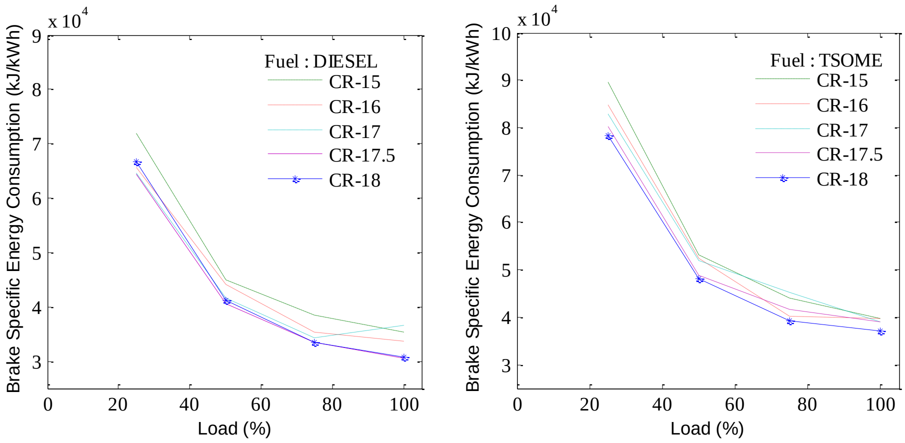


Fig. 2. Variation of brake specific energy consumption with load for Diesel and NTSOME at different compression ratios.

5.2. Brake thermal efficiency

Figure 3 shows the effect of compression ratio on brake thermal efficiency of the engine. It is observed from the graph that the BTE of both diesel and NTSOME increase with the increase in CR of the engine. The brake thermal efficiency for diesel and NTSOME for the compression ratio18 is observed to be 31.73% and 31.62% respectively. At compression ratio 18, the use of NTSOME produced much higher BTE than all the other conditions in which it was tested. Moreover, at higher compression ratios the BTE of the NTSOME is closer to the diesel operation which clearly indicates that the BTE varies with the increase in compression ratio [33].

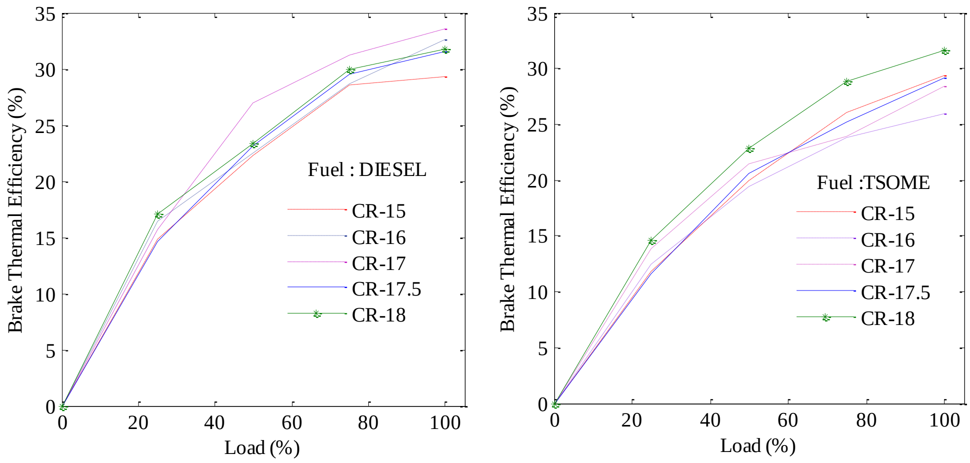


Fig. 3. Variation of brake thermal efficiency with load for Diesel and NTSOME at different compression ratios.

Cylinder pressure

Figure 4 indicates the variation of cylinder pressure with respect to crank angle for different compression ratios. It has been observed that due to shorter ignition delay and lower energy content, the NTSOME delivers low cylinder pressure compared to that of standard diesel. In a CI engine, the rise in pressure depends on the combustion rate in the initial stages. Further, it is influenced by the amount of fuel contribution in the uncontrolled combustion phase and the delay period [33]. The maximum amount of heat energy is absorbed by the cylinder as soon as the fuel gets injected into it [3]. This ultimately results in longer ignition delay.

The maximum pressures are 50.38bar, 52.43, bar, 54.75bar, 56.62 bar and 59.85 bar for standard diesel and 47.98bar, 47.87bar, 49.28bar, 53.65bar and 54.84bar for NTSOME at compression ratios 15:1, 16:1, 17:1, 17.5:1, 18:1 were observed respectively. The rise of pressure follows an increasing trend with the increase in compression ratio. Moreover, at a compression ratio 18:1, peak pressure rise of the diesel is 14.08% higher than the NTSOME. The reason for this is accounted to the delay period which results in faster and complete combustion of diesel fuel within the combustion chamber.

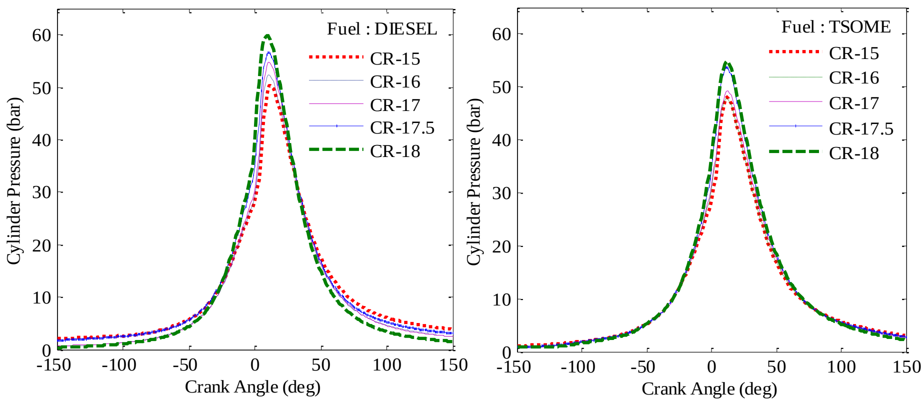


Fig. 4. Variation of cylinder pressure with crank angle for Diesel and NTSOME at different compression ratios.

5.3. Cumulative heat release

The Cumulative Heat Release (CHR) is shown in Fig. 5. It is observed that with the rise in the engine load, the CHR increased continuously. The reason for this is the increased quantity of fuel injected into the cylinder. The maximum CHR is found to be 1.67 kJ occurring at 82^{o}CA aTDC for NTSOME at CR 18 and 1.36 kJ at 116^{o}CAaTDC for the diesel. The minimum CHR was 1.12 kJ occurring at 102 ^{o}CA aTDC for diesel fuel and 1.2 kJ at 98 ^{o}CA aTDC for NTSOME at compression ratio 16. It has been accepted that only at the end of combustion, the maximum CHR occurs. Moreover, for all compression ratios using both the fuels, the last part of combustion changed within the range of 82-122^{o}CAaTDC.

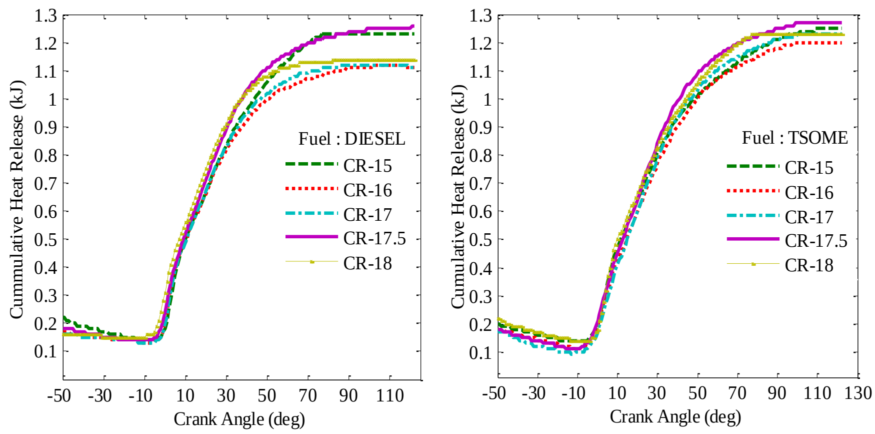


Fig. 5. Variation of cumulative heat release with crank angle for Diesel and NTSOME at different compression ratios.

5.4. Rate of pressure rise

Figure 6 shows the ROPR for the NTSOME and Diesel. Varying the compression ratio shows effect on maximum pressure of the cycle. Compression ratio change would results in decreased delay period and consecutively decreases the ROPR in the cycle [2]. In comparison with diesel, NTSOME showed lower ROPRs as the premixed combustible mixture would make less fuel to be burned in the premixed burning phase. Higher viscosity and lower volatility of NTSOME are the reasons for less premixed combustible mixture which in turn decreases the CP and ROPR [34, 35]. With the rise in engine load and compression ratio, for both the test fuels, ROPR increased subsequently due to enrichment in the quantity of fuel that is injected into the cylinder per CA, improved combustion rate and increased burnt fuel in the uncontrolled combustion phase. Also, the higher ROPR indicated that a greater proportion of the fuel injected is being burnt in the premixed combustion phase [36]. Due to the reasons similarly mentioned above, the increasing of CR was also the most effective factor on the ROPR. In all the tests carried out, the maximum ROPR was observed to be 3.9bar which occurred at 4^{o}CAbTDC for diesel at CR-18. Also, the minimum ROPR was 2.56bar which occurred at 4^{o}CAbTDC for NTSOME at CR-15.

5.5. Mass fraction burned

The mass fraction burned variation with respect to degree crank angle for the Diesel and NTSOME is shown in Fig. 7. At higher compression ratios, the mass fraction burnt of NTSOME is closer to the standard diesel. Also the engine is found to be operating at rich mixtures and reaches the stoichiometric region at higher compression ratios. The combustion is sustained in the diffusive combustion phase due to the oxygen content of NTSOME. The availability of more fuel in the combustion phase causes rapid heat release [37]. The shorter combustion duration is observed at all compression ratios NTSOME comparatively with diesel. The mass fraction burnt for the NTSOME is ranges from 24^{0} CA aTDC - 37 ^{0}CA aTDC and 21^{o} CA aTDC - 26^{0}CA aTDC for Diesel. The mass fraction that is burnt for i-th interval can be calculated as the ratio of is the burned mass (m_b) and total mass ($m_{b(total)}$) in the cylinder and it is given by Eq. (1) [34]:

$$MFB = \frac{m_{b(i)}}{m_{b_{total}}} \tag{1}$$

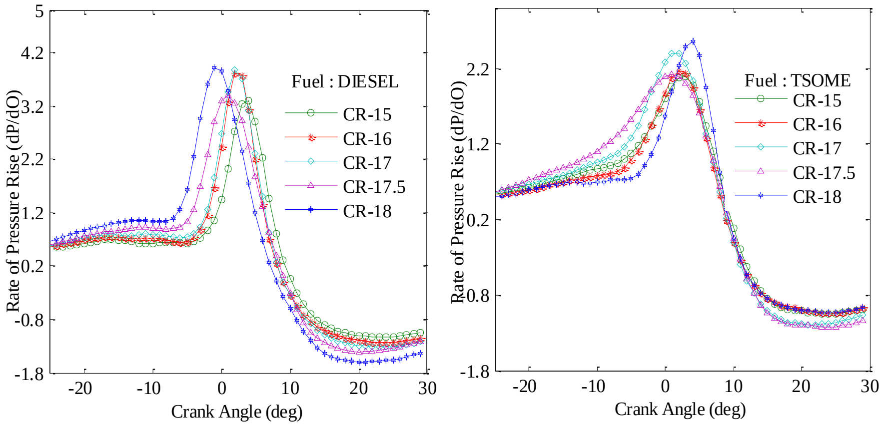


Fig. 6. Variation of rate of pressure rise with crank angle for Diesel and NTSOME at different compression ratios.

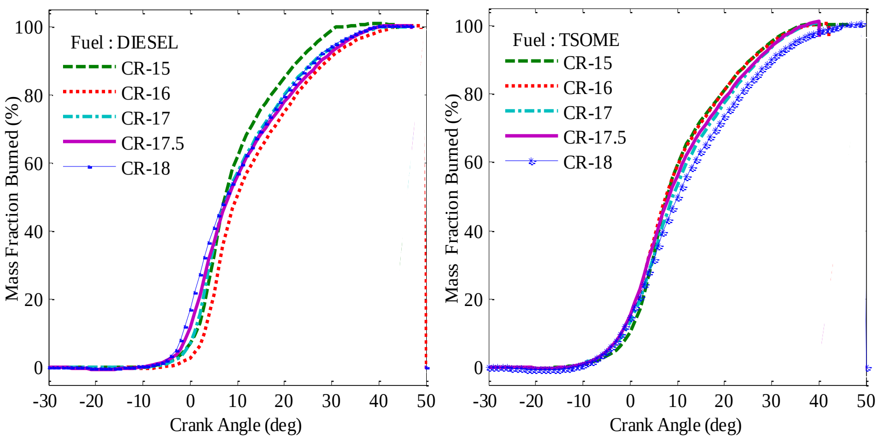


Fig. 7. Variation of mass fraction burned with crank angle for Diesel and NTSOME at different compression ratios.

5.6. Nitrogen oxides (NOx) emission

Figure 8 shows the variations in the emission of nitrogen oxides (NOx) with respect to load for diesel and NTSOME at different compression ratios. From the Fig. 8, it can be seen that increase in the compression ratio increases the NOx emissions for NTSOME and diesel. The emissions of NTSOME are comparatively less than the diesel. NOx emissions for diesel reach a maximum value of 110 ppm whereas it reaches to 60 ppm for NTSOME. At CR 18:1 NOx emissions for NTSOME are 45% less than that of pure diesel. As known already, the increased compression ratio increases the maximum temperature during the combustion, which contributes towards NOx production.

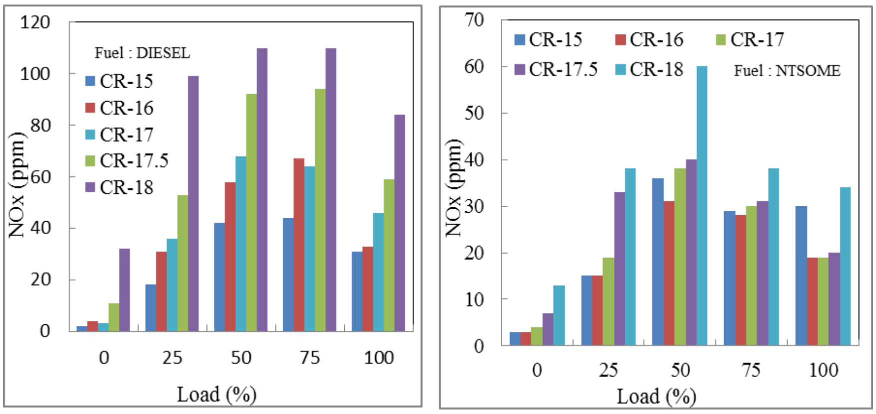


Fig. 8. Variation of NOx with load for Diesel and NTSOME at different compression ratios.

5.7. Smoke opacity

The formation of smoke occurs due to air deficiency. As shown in Fig. 9, the smoke content is observed to be decreasing with compression ratio. The smoke opacity at CR 15:1 for diesel is 97% and for NTSOME is 45% at full load operation. Smoke opacity varies from 35% to 45% for NTSOME and 80% to 97% for diesel when the engine operated from no load to full load condition. The reasons for this may be accounted to the better air-fuel mixing which helps in superior combustion of the fuel inside the cylinder and may be due to late burning in the expansion and exhaust. Lower the C/H ratio lower is the smoke. As compared to pure diesel, biodiesel has lower C/H which results in low smoke.

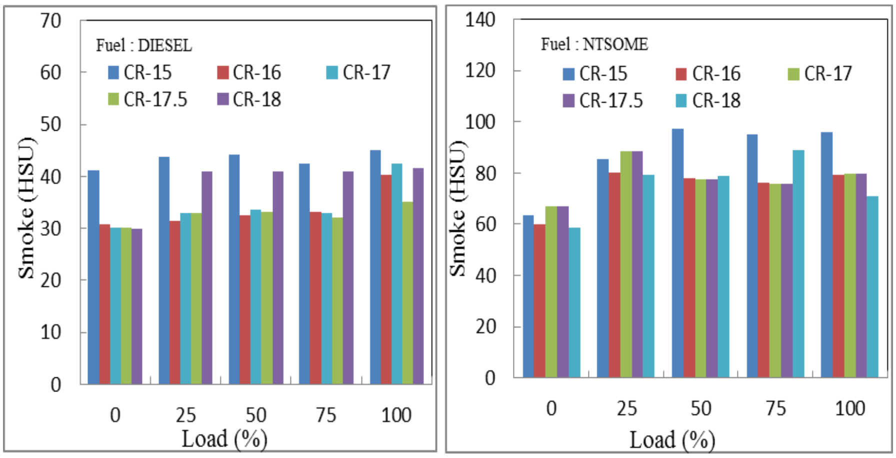


Fig. 9. Variation of smoke with load for Diesel and NTSOME at different compression ratios.

6. Conclusions

The combustion characteristics (along with performance and emission characteristics) of a variable compression ratio engine fueled with NTSOME are investigated. The potential of using NTSOME as an alternative source for the energy supply in diesel engine is evaluated in this process. The results obtained from the experiment confirm that the BSEC and BTE of variable compression ratio engine are a function of load and compression ratio. Also, the engine performance is varied by increasing the compression ratio thus becoming comparable with standard diesel operation. The following conclusions are drawn from this study:

- The BTE of the NTSOME is on par with diesel at compression ratio 18. The reason for this may be due to anticipated combustion, and lower energy content of the biodiesel.
- The cylinder pressure of VCR engine with NTSOME approaches the diesel values at all compression ratios. This is due to the rapid and unabridged combustion of fuel inside the combustion chamber.
- Also, rate of heat release is minimal at the inception of combustion and increases further at all compression ratios. Viscosity of the NTSOME, combination of air entrainment with lower air/fuel mixture might be the reason for this. At full load, the declination in the mass fraction burnt for NTSOME in comparison with diesel at full load is insignificant.
- NO_x from the NTSOME is low at low compression ratio than that of high compression ratio. Remarkable reduction of NO_X emission was observed at high compression ratio and it was about 45 % compared to diesel.
- From the experimental results, at lower compression ratios, the smoke opacity is higher and vice versa. The exhaust gas analysis also gives acceptable emission values for NTSOME used in the VCR engine.

The experimental results prove that the NTSOME can be used in diesel engine with VCR technology in view of combustion and consumption, but an additional technology is required for the reduction of NOx at high compression ratios.

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