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PERFORMANCE AND EMISSION CHARACTERISTICS OF A VARIABLE COMPRESSION RATIO DUAL FUEL ENGINE FUELLED WITH EUCALYPTUS OIL AND ITS BLEND

M. VENKATRAMAN[a*] and P. MADHANRAJ[b]

[a]Department of Mechanical Engineering, Aarupadai Veedu Institute of Technology, CHENNAI (T.N.) INDIA
[b]PG Full Time-Thermal Engineering, Aarupadai Veedu Institute of Technology, CHENNAI (T.N.) INDIA

ABSTRACT

The present work describes a experimental investigation concerning the performance of a four strokes compression ignition engine, which is powered by alternative fuels in the form of diesel and eucalyptus oil blend. Experiments was conducted in a single cylinder VCR diesel engine fuelled with a EB20 by volume with diesel fuel. It was observed that the simulated performance, combustion and emissions characteristics results are found satisfactory with the experimental results. Thus the developed computer simulation model can predict the performance, combustion and emission characteristics of any biodiesel blends with minimum inputs such as engine specifications and fuel properties.

Key words: CI engine, Performance, Combustion, Emission.

Notation:

BTHE	:	Brake thermal efficiency
BSFC	:	Brake specific fuel consumption
BSEC	:	Brake specific energy consumption
HRR	:	Heat release rate
EB	:	Eucalyptus oil blend
EB20	:	20% of eucalyptus oil and 80% of diesel
VCR	:	Variable Compression Ratio

*Author for correspondence; E-mail: mvramcet@yahoo.co.in, venkatraman@avit.ac.in

INTRODUCTION

Alternative fuels

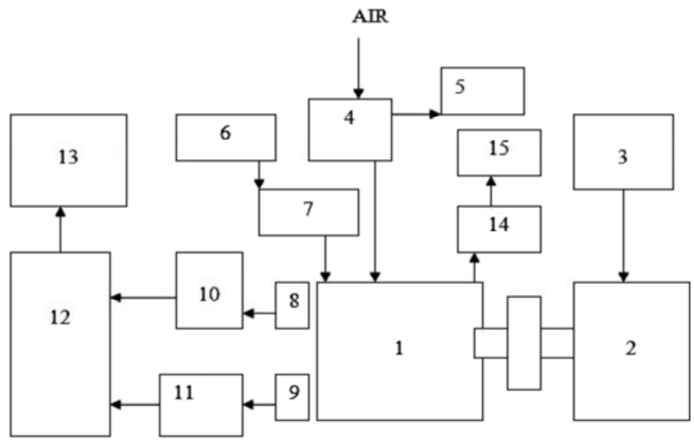
Many energy fuels such as eucalyptus oil, hydrogen, CNG, alcohols, Biogas, Producer gas and host of vegetable oils are being investigated as potential substitutes for the current high pollutant diesel fuel derived from diminishing commercial sources. With the fossil fuel depleting, biofuel as a renewable source of energy affords immense potential.

In India, our domestic production of fuel is simply not able to keep pace with the ever growing needs, compelling us depend heavily on imports. With about 70% of domestic demand for fuel to be met by imports, there is a huge outflow of valuable foreign exchange. Besides, the increasing use of fossil fuel has also been resulting in such seriously deleterious problems as Green House Effect affecting the entire humanity. It is therefore impending that a safe alternative is explored and exploited. Naturally, biofuel presents a most viable option as it can be obtained from renewable source of energy.

The fuels of Bio origin may be alcohols, edible and non edible vegetable oils, biomass, biogas etc. some of these fuels can be used directly while others need to be formulated to bring the relevant properties close to conventional fuels.

EXPERIMENTAL

Fig. 1 shows the schematic line diagram of the experimental set up and its specification are given in Table 2. A Electrical dynamometer was used to apply the load on the engine. A water rheostat with an adjustable depth of immersion electrode was provided to dissipate the power generated. Tests were carried out at various loads starting from no load to full load condition at a constant rated speed of 1500 rpm. At each load, the fuel flow rate various constituents of exhaust gases such as Hydrocarbon (HC), carbon monoxide (CO) and nitrogen oxides (NO_x), were measured with a 5-gas MRU Delta exhaust gas analyzer. The analyzer uses the principle of non-dispersive infrared (NDIR) for the measurement of CO and HC emissions while NO_x measurement was by means of electrochemical sensors. Combustion analysis was carried out by means of an AVL pressure pick-up fitted on the cylinder head and a TDC encoder fixed on the output shaft of the engine are given in Table 4. The pressure and the crank angle signals were fed to a pentium personal computer. Various combustion parameters like heat release rate, cumulative heat release rate and peak pressure and its accurance were obtained using data acquisition system. The engine was first operated with diesel oil to generate the baseline data followed by Methyl Esters of pongamia oil and their blend such as PME20.



1. Diesel engine
2. Electrical dynamometer
3. Dynamometer controls
4. Air box
5. U- Tube manometer
6. Fuel tank
7. Fuel flow meter
8. Pressure transducer
9. TDC position sensor
10. Charge amplifier
11. TDC amplifier circuit
12. A/D card
13. Personal computer
14. Exhaust gas analyzer
15. AVL smoke meter

Experimental setup

Specifications of engine

Make	Kirloskar
Model	TAF 1
Type	Direct injection, air cooled
Bore × Stroke (mm)	87.5 × 110
Compression ratio	17.5:1
Cubic capacity	0.661 lit
Rated power	4.4 KW
Rated speed	1500 rpm
Start of injection	24º bTDC
Connecting rod length	220 mm
Injector operating pressure	220 bar

Comparison of eucalyptus oil, methyl ester of eucalyptus with diesel

Properties	Diesel	Raw eucalyptus oil	Blend of eucalyptus oil
Specific gravity (g/cc)	0.834	0.9186	0.88
Kinematic viscosity @40uc (cst)	6.2	81.80	8.722
Flash point(uc)	50	240	170
Net-heatcontent (kj/kg)	43600	39774	38450
Cetane number	45-55	38	50

Fig. 1 shows that the effect of compression ratio on the variation of brake thermal efficiency with brake power for EB20. The maximum brake thermal efficiency obtained is about 30.9% for EB20 at compression ratio of 19:1. Increase in thermal efficiency is due to the increase in peak pressure and increases in combustion temperature. The other compression ratio of 16:1 and 17.5:1 offers relatively lower brake thermal efficiency than that of 19:1.

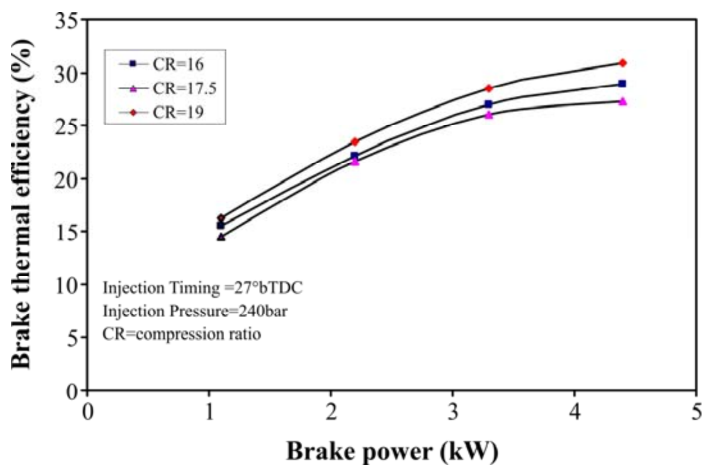


Fig. 1: Effect of compression ratio on the variation of brake thermal efficiency with brake power

Fig. 2 shows that the effect of compression ratio on the variation of brake specific energy consumption with brake power for EB20. From the results it is found that EB20 offers comparatively lower BSEC for compression ratio of 19:1 compare to other 16:1 and 17.5:1. This is due to better combustion of EB20 due to presence of high cetane of EB.

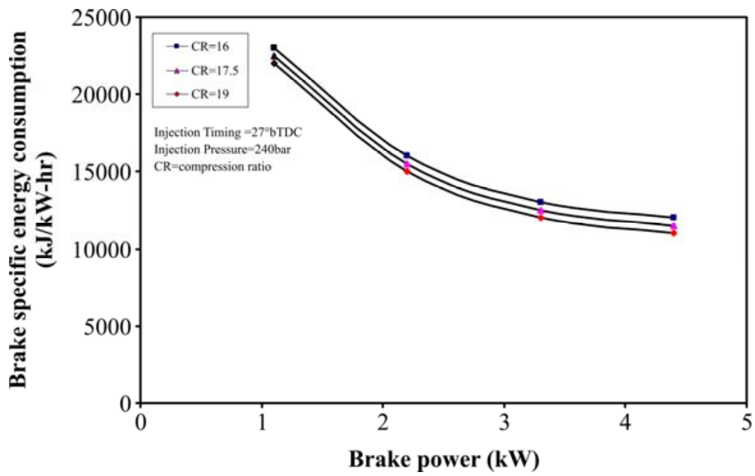


Fig. 2: Effect of compression ratio on the variation of BSEC with brake power

Fig. 3 Shows that the effect of compression ratio on the variation of hydrocarbon with brake power for EB20. It was observed that the maximum rate of hydrocarbon is 35 ppm for compression ratio at 19:1.It is also found that the hydrocarbon of 20 ppm for compression ratio at 17.5:1 decreases with increase in concentration of the biodiesel blend. This may be due to improved combustion because of increased in injection pressure and advanced injection timing.

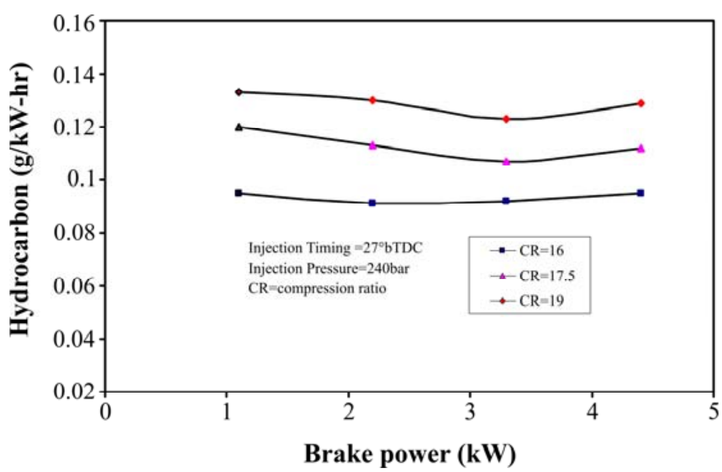


Fig. 3: Effect of compression on variation of hydrocarbon with brake power

Fig. 4 Shows that the effect of compression ratio on the variation of carbon monoxide with brake power for EB20. It was noticed that CO emission of 0.49%vol for compression ration at 19:1.CO emissions decreases with increase in EB in the blend had sufficient time for combustion process because of advanced injection pressure.

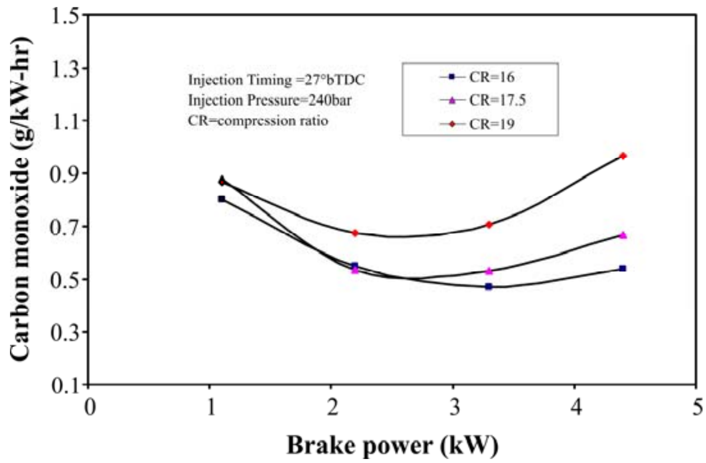


Fig. 4: Effect of compression ratio on the variation of carbon monoxide with brake power

Fig. 5 shows that the NOx emissions increases for EB20 (49.33 g/kW-hr)at compression ratio 19:1 compare to compression ratio at 16:1 (42.1 g/kW-hr).Due to the advancement of compression ratio and pressure all the injected fuel burnt as a result higher combustion temperature is attained. The higher temperature promotes NOx formation.

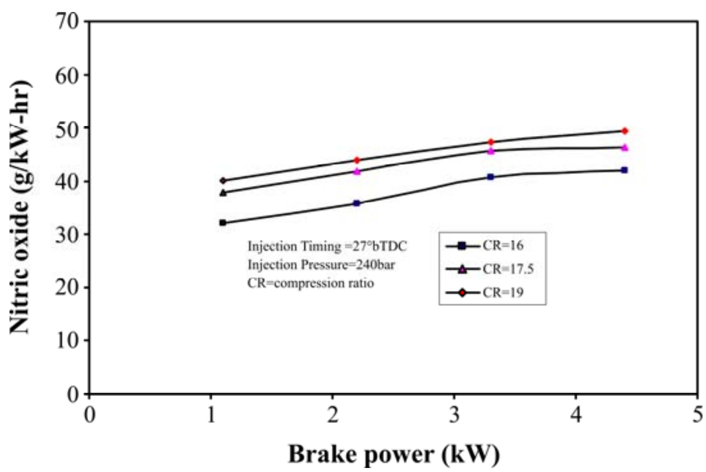


Fig. 5: Effect of compression ratio on the variation of nitric oxide with brake power

Fig. 6 shows that the peak pressure increases with increase in compression ratio. This is due to increased combustion temperature and shorter duration of combustion.

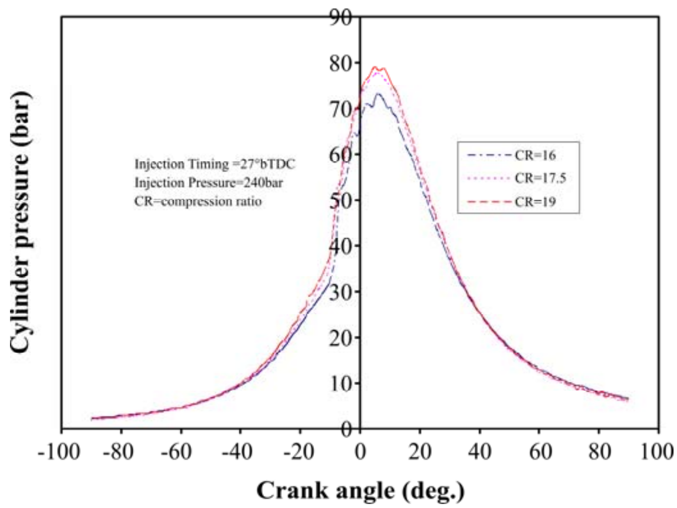


Fig. 6: Effect of compression ratio on the variation of cylinder pressure with crank angle

Fig. 7 shows that the height of premixed phase of combustion decreases with respect to increase in compression ratio. Also the compression ratio increases change the duration of combustion. Usually higher compression ratio offers shorter duration of combustion and cause better performance.

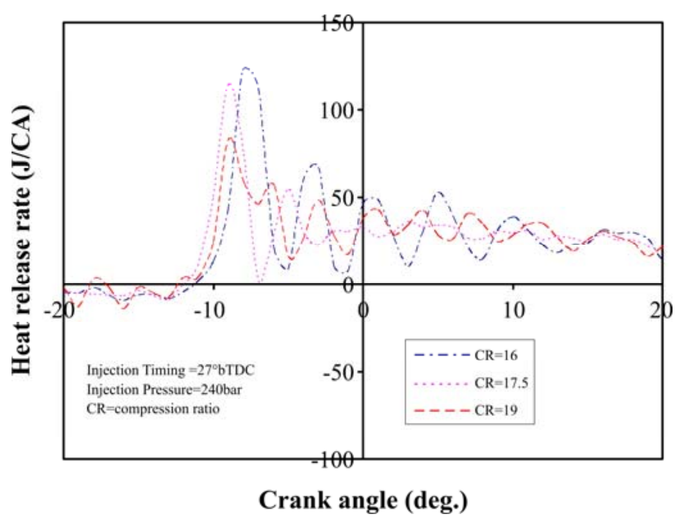


Fig. 7: Effect of compression ratio on the variation of HRR with crank angle

Fig. 8 shows that the height of premixed phase of combustion decreases with respect to increase in compression ratio. Also the compression ratio increases change the duration of combustion. Usually higher compression ratio offers shorter duration of combustion and cause better performance.

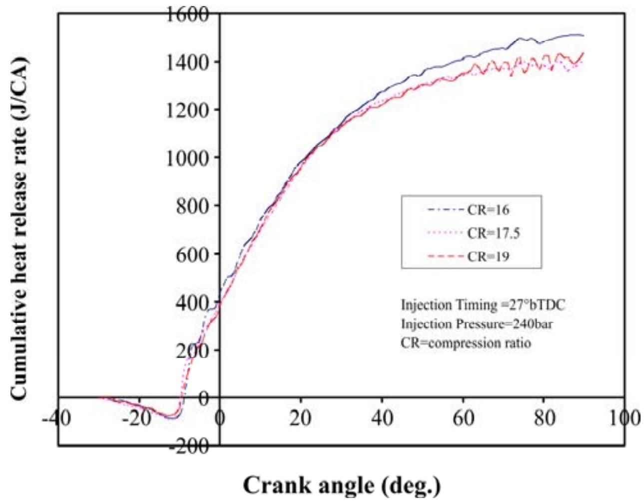


Fig. 8: Effect of compression ratio on the variation of CHRR with crank angle

Fig. 9 shows that the increase in injection timing increases the brake thermal efficiency and the maximum brake thermal efficiency occurs at 27°bTDC. The main reason for higher brake thermal efficiency at this particular timing is that the peak pressure occurs closer to TDC also fuel releases all the heat is shorter duration of combustion and resulting in improved performance.

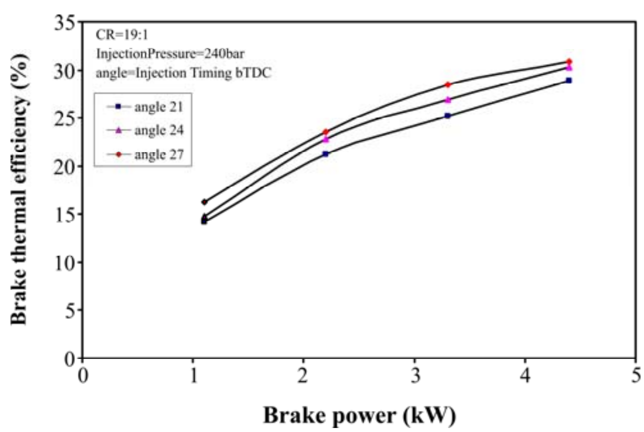


Fig. 9: Effect of injection timing on the variation of brake thermal efficiency with brake power

Fig. 10 shows that the change in injection timing changes the occurrence of peak pressure and changes the Duration of combustion. The injection timing 27°bTDC produces peak pressure closer to TDC and offers sufficient time to release heat, hence this particular timing offers lower BSEC compare to other injection timing 21°bTDC and 24°bTDC.

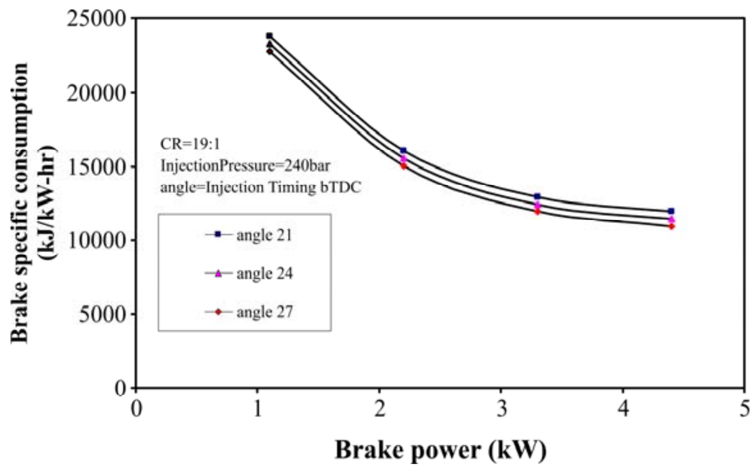


Fig. 10: Effect of injection timing on the variation of BSEC with brake power

Fig. 11 Shows that the maximum rate of hydrocarbon is 0.128g/kW-hr for injection timing at 27deg bTDC. The hydrocarbon of 0.112 g/kW-hr for injection timing at 24deg bTDC decreases with increase in concentration of the biodiesel blend. This may be due to improved combustion because of increased in injection pressure and increase in compression ratio.

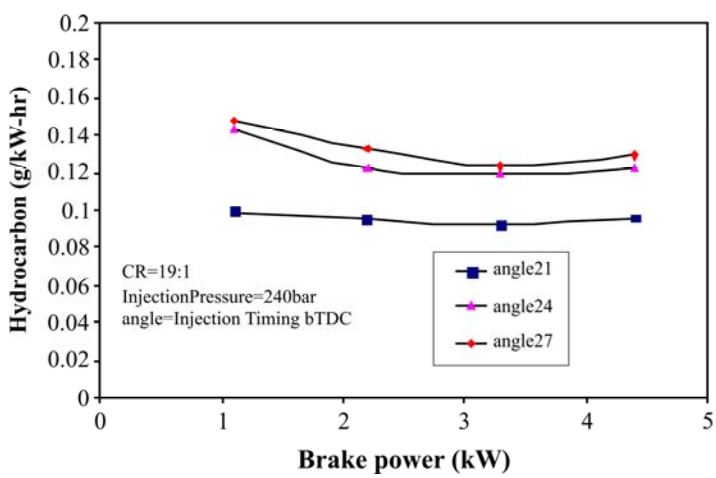


Fig. 11: Effect of injection timing on the variation of hydrocarbon with brake power

Fig. 12 shows that the CO emission of 1.343 g/kW-hr for injection timing 27deg bTDC because due to presence of oxygen in the biofuel.CO emissions decreases with injection timing at 21deg bTDC of 0.948g/kW-hr.

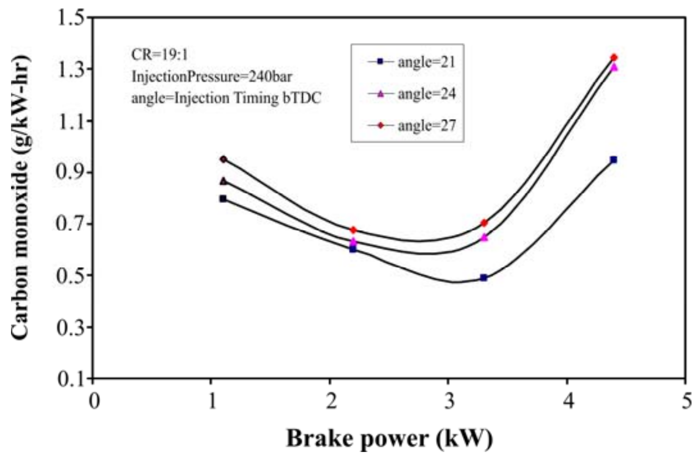


Fig. 12: Effect of Injection timing on the variation of carbon monoxide with brake power

Fig. 13 shows that the NOx emissions increases for EB20 (43.9 g/kW-hr) at injection timing 27 deg bTDC compare to injection timing at 21 deg bTDC (21.05 g/kW-hr). This is due to the advancement of injection timing and pressure all the injected fuel burnt as a result higher combustion temperature is attained. The higher temperature promotes NOx formation.

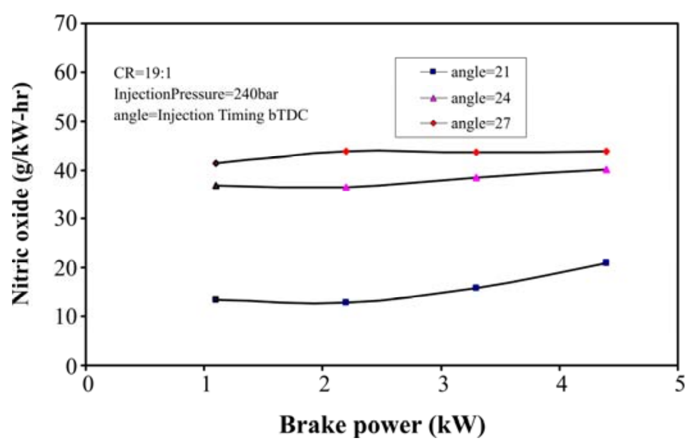


Fig. 13: Effect of injection timing on the variation of nitric oxide with brake power

Fig. 14 shows that the change in injection timing changes the occurrence of peak pressure and combustion duration. The injection timing of 27°bTDC produces peak pressure few degree before TDC and utilizes the heat energy well before the completion of power stroke and hence the timing offers maximum peak pressure compare to other timing. This is the main reason for higher brake thermal efficiency of 27°bTDC.

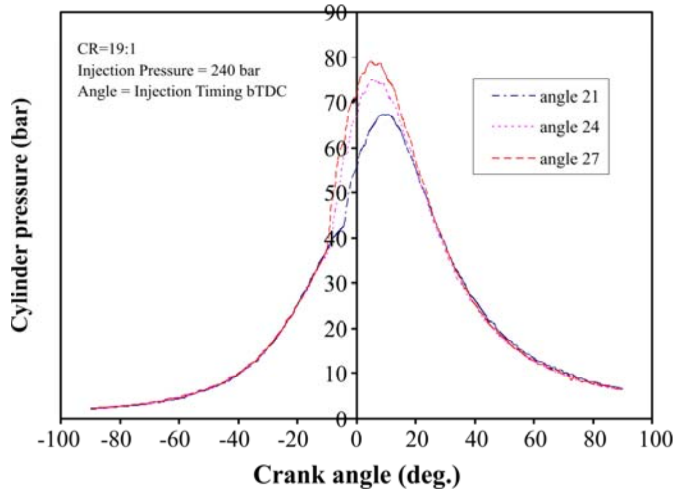


Fig. 14: Effect of injection timing on the variation of cylinder pressure with crank angle

Fig. 15 shows that the injection timing of 27°bTDC offers comparatively shorter duration of combustion and produces peak heat release rate closer to diesel baseline operation. Hence this particular timing offers better performance compared to other injection timing of 21°bTDC and 24°bTDC.

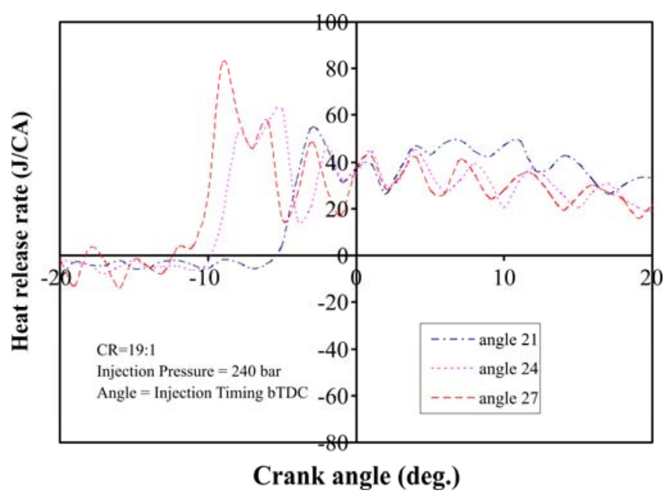


Fig. 15: Effect of injection timing on the variation of HRR with crank angle

Fig. 16 shows that the injection timing of 27°bTDC offers comparatively shorter duration of combustion and produces peak heat release rate closer to diesel baseline operation. Hence this particular timing offers better performance compare to other injection timing of 21°bTDC and 24°bTDC.

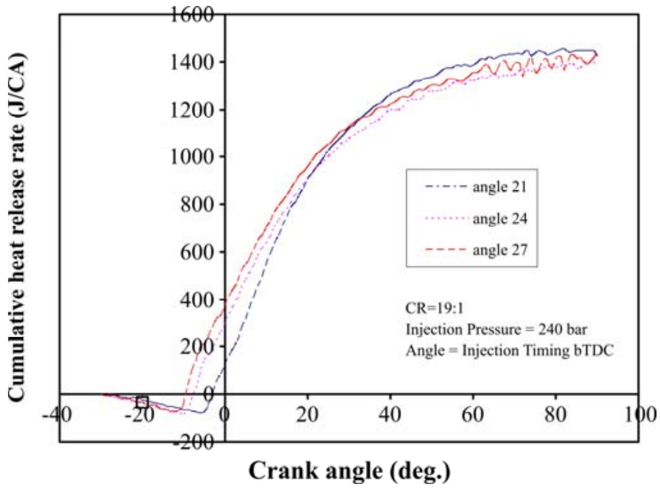


Fig. 16: Effect of injection timing on the variation of CHRR with crank angle

CONCLUSION

Following are the conclusions based on the experimental and simulation results obtained while operating single cylinder diesel engine fuelled with biodiesel from pungam oil and their blends.

(i) Eucalyptus oil blend can be directly used in diesel engine without any modifications. The maximum brake thermal efficiency is found in EB20 at injection timing of 27ºbTDC, injection pressure of 240bar and compression ratio of 19:1.

(ii) It is found that the combined increase of compression ratio, injection timing and injection pressure increases the BTHE and reduces BSEC with lower emissions.

(iii) The predicted results of performance and emissions have been in close agreement with experimental results. The developed simulation model appears to be a useful tool for analyzing the diesel engine combustion accurately.

(iv) Thus the developed computer model can predict the various performance and emission parameters of any vegetable oilesters with minimum inputs such as density, calorific value, chemical formula and engine specification.

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