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SOME STUDIES ON THE EFFECT OF ALCOHOLIC FUEL BLENDS ON VCR DIESEL ENGINE CHARACTERISTICS

Pasi Ketansai, Teeda Chakradhar Rao
Department of Mechanical Engineering, MVGR College of Engineering, Vizianagaram-535005, AP, India

ABSTRACT

The alcoholic fuel used in this study is ethanol, which is also known as ethyl alcohol (C2H5OH). Ethanol is used as an alternative fuel because it is renewable and produced from biomass. Different experiments are conducted to observe the effect of ethanol-diesel blended fuels and pure diesel on Variable Compression Ratio Diesel engines. In this study, experiments have been carried out to find out the outperformance, combustion, vibration, and emission characteristics of various blends such as E0 (diesel 100%), E5 (diesel 95%, ethanol 5%), E10 (diesel 90%, ethanol 10%), E15 (diesel 85%, ethanol 15%), and E20 (diesel 80%, ethanol 20%). The experiments were on single-cylinder, four-stroke, variable compression ratio, compression ignition engines. Engine tests are carried out at a constant rated speed of 1500 rpm for different loading conditions at different compression ratios (15, 16, 17, and 17.5) for various fuel blends. When fully loaded, brake thermal efficiency is higher by 13.33% for E20 blend over E0 fuel at CR 17.5. The maximum rate of pressure rise is higher for the E15 blend compared to other blends and pure diesel at CR 17.5 full load conditions. At CR 15, the maximum net heat release rate is higher for E15 blends compared to other blends and pure diesel. The E10 blend's smoke density (HSU) at CR16 with a 25% load is lower compared to other blends. E5 has lower vibrations under full load conditions compared to other blends. Therefore, from this study, E5 and E10 are recommended for a diesel engine as an alternative fuel from the perspective of reduced engine vibrations and emissions.

Keywords: Alternative Fuel; Ethanol; Combustion; Vibrations; Emissions; Diesel Engine.

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1. INTRODUCTION

The world is on the verge of a global energy catastrophe, owing mostly to the depletion of conventional energy sources; reducing reliance on fossil fuels is advised. The main reason behind the development of alternative fuels for the IC engine is to reduce the emissions of gasoline and diesel engines. Alcohol is now utilized as an alternative fuel, combined with diesel and gasoline. Many studies have been conducted on the usage of ethanol in spark ignition (SI) engines. In comparison to spark-ignition engines, research on the use of alcohols in compression ignition (CI) engines is less extensive. The first studies exploring the use of ethanol in diesel engines were conducted in South Africa in the 1970s [1] and continued in Germany and the United States during the 1980s [2, 3]. The diesel engine is the most widely used, accounting for more than 80% of the global transportation industry and making an irreplaceable contribution to the development of society [4]. However, due to a lot of particulate matter (PM), hydrocarbons (HC), nitrogen oxides (NO_x), sulfur oxides (SO_x), and other toxic or harmful substances produced by the diesel engine [5], The objective of this work was to find the maximum possible and optimum replacement of diesel fuel with ethanol and compare the performance of diesel engines fueled with ethanol-diesel blended fuels [6]. In order to blend ethanol with diesel fuel, phase separation is a major issue, especially at low temperatures. The emulsifier or co-solvent is required to keep the mixture as a homogeneous emulsion or solution, called diesohol or E-diesel [7–9].

Internal combustion engines use fossil-based fuels. Implementing alternative fuels in internal combustion engines has the potential to reduce the dependency on fossil-based fuels. Further, the development of efficient and eco-friendly combustion systems and alternative fuels becomes increasingly important. Alcohol has been considered an alternative fuel for diesel engines [10]. With the advancement of technology in recent years, a lot of researchers have explored the ethanol-diesel blend fuel utilized in CI engines [11]. The modification of fuel's Physico-chemical properties changes the spray characteristics, combustion, performance, and emissions of the engine. Several approaches have been evaluated to allow the use of E-Diesel in CI engines. They are alcohol-diesel fuel blends, alcohol-diesel fuel emulsions, alcohol fumigation, and dual injection [12].

Table-1 Properties of Ethanol and Diesel

Properties	**Ethanol**	**Diesel**
Molecular Weight	46	170-198
Calorific Value (KJ/Kg)	26700	42600
Octane Number	105-110	-
Cetane Number	0-5	50
Density (Kg/m³)	789	846
Latent heat of vapourization (KJ/Kg)	904	700
Viscosity (mPa-s)	1.074	3.546
Flash Point(ºC)	13	70
Oxygen content(mass%)	34.78	0

2. EXPERIMENTALSETUP

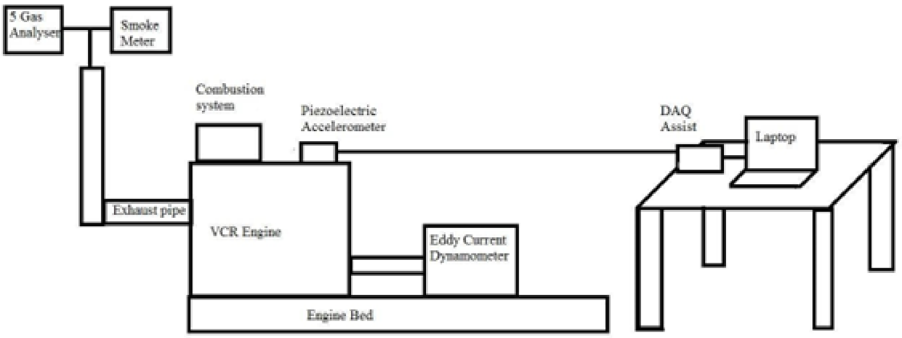


Figure1. Schematic of engine setup

The engine setup consists of a single-cylinder, four-stroke, water-cooled direct injection, variable compression ratio (VCR) diesel engine, which is coupled with an eddy current dynamometer. The output shaft of the eddy current dynamometer is connected with a strain gauge for load measurement. Fuel flow rate is measured using a 10-cc burette and stopwatch. The air box is used for airflow adjustment and measurement. A piezoelectric pressure sensor with a sensitivity of 1 mV/PSI and a range of 5000 PSI is mounted on the engine head for combustion pressure measurements. An average of 10 consecutive combustion cycles is used for pressure measurement. RTD, PT100, and K-type thermocouples are used to measure temperature. For detecting engine vibrations, a PCB single-axis accelerometer sensor is installed vertically on the cylinder head. This is a non-intrusive accelerometer that operates on the piezoelectric effect theory. A PCB single-axis accelerometer sensor measures the vibrations in the time domain. The time-domain data is converted into the frequency domain using a fast Fourier transform (FFT). The vertical-axis vibration sensor is more sensitive for combustion diagnosis. The accelerometer is mounted on the engine head using Loctite adhesive. This accelerometer is connected to the NI9234 (A/D converter) data acquisition card, which provides a connection to an input channel with an input range of ***5 v.*** The input channel of the accelerometer is connected to AI0 of the A/D converter. At last, the collected data is transferred to the laptop for recording. A program is written in NI LabVIEW software for measuring engine vibrations. This data is used to draw the graphs between acceleration vs. time and acceleration vs. frequency. The AVL Di Gas 444 5-gas analyzer is used for the measurement of exhaust gases such as CO, HC, carbon dioxide (CO_2), NOx, and oxygen (O_2). An AVL 437C smoke meter is used for measuring smoke in Hartridge Smoke Units (HSU). All experiments were performed at a constant speed of 1500rpm. In this experiment, the compression ratios employed were 15, 16, 17, and 17.5.

SPECIFICATIONS OF ENGINE SETUP:

Engine type	Single cylinder, four-stroke, VCR, water-cooled, direct injection, diesel engine
Stroke	110mm
Bore	87.5mm
Speed	1500rpm
Range of compression ratio	12–18
Injection timing	23 degrees before top dead center
Injection pressure	200 bar
Power	3.5 KW

Calorific value refers to the quantity of heat produced by a unit volume of a substance upon full combustion. The higher the calorific value of the fuel, the greater its efficiency. A calorimeter was used to determine the calorific values of pure diesel and alcoholic blends (E05, E10, E15, and E20). Piezoelectric accelerometer It works on the principle that when a mass exerts a force on a piezoelectric material (crystal or ceramic), the ensuing stress on the material produces a charge proportionate to the force. Following that, the data was given to DAQ assistance, which translated the form and provided the vibration amplitude in LabVIEW. Carbon monoxide, hydrocarbons, carbon dioxide, nitrogen oxide, and oxygen are all measured by the AVL Di-gas 444 5-gas analyzer.

When a thin layer of a solid or liquid, or a thicker layer of a gaseous material, is exposed to all wavelengths of infrared light, only the wavelengths that correspond to the vibrational frequencies of its molecules are absorbed; all other wavelengths are transmitted. The operating idea of smoke metres is that the amount of light obscured in the smoke produced by engines in cars is detected and quantified using opacity metres. Smoke density, which is an indicator of combustion efficiency, is displayed by the smoke metre measurement. Reading makes use of the Beer-Lambert notion. It shows how much smoke is in the exhaust. A reasonable and required percentage of ethanol is blended with Diesel using a stirrer for homogeneous mixing. Table 2 shows the calorie value of each ethanol-Diesel combination as determined with a Digital Bomb calorie metre. Each mix is now poured into the petrol tank, and the engine is operated at various loads and CRs (15, 16, 17, and 17) using a hydraulic dynamometer. The fuel consumption of the engine is tracked using volumetric analysis, which is utilised to compute performance metrics. The vibration characteristics are recorded by a piezoelectric accelerometer placed over the engine head. At various blends, loads, and speeds, the vibrations are recorded using LabVIEW software. The proportion of smoke in the exhaust gases is measured using a smoke metre. Exhaust gases are measured using five gas analysers (HC, CO, CO2, NO, and O2).

3. RESULTS

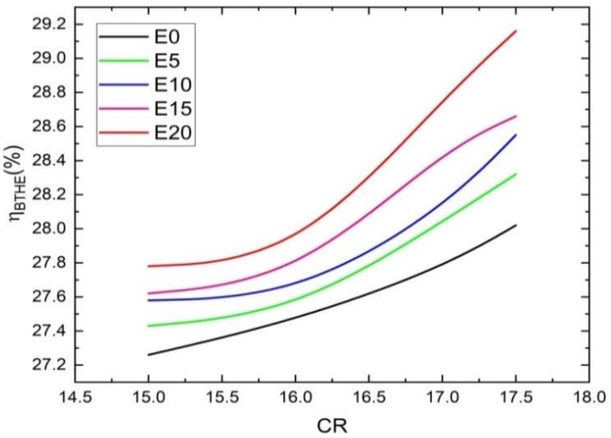
Table-2 properties of Ethanol-Diesel blends

Blended Fuel	Calorific Value(kJ/kg)	Density(kg/m^3)
E0	45,728.11	835
E5	44,162.38	900
E10	43,837.38	920
E15	41,768.20	942
E20	40,847.28	974

3.1. Performance Curves

From Fig. 1, which shows the variation of brake thermal efficiency and the load at CR17.5, it is observed that as brake thermal efficiency increases, the load increases. This is because as the load increases, the suction The pressure generated will be higher, which may have resulted in efficient combustion. When compared to diesel, ethanol blends have higher thermal efficiency. Despite the fact that ethanol blends have a lower calorific value than diesels due to their higher oxygen content, ethanol blends have more effective combustion (i.e., complete energy conversion) than diesel. When compared to E0, the increases in BTHE with load for E5, E10, E15, and E20 are 1.05%, 3.75%, 6.39%, and 13.33%, respectively. According to Fig. 2, which depicts the relationship between brake thermal efficiency and compression ratio, as the compression ratio increases, so does brake thermal efficiency. Because a higher compression ratio results in more braking power, the thermal efficiency of the brake increases.

The ethanol blends have higher brake thermal efficiency at the corresponding compression ratio, while the E20 blend has higher brake thermal efficiency at all compression ratios, with a peak of 29.1% at CR17.5. At CR15, the E20, E15, E10, E5, and E0 had brake thermal efficiencies of 27.8%, 27.63%, 27.6%, 27.4%, and 27.2%, respectively.



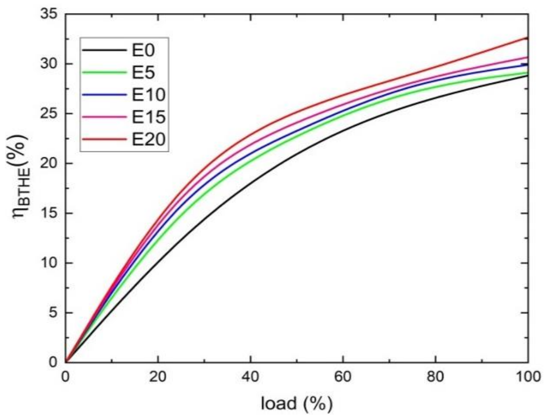
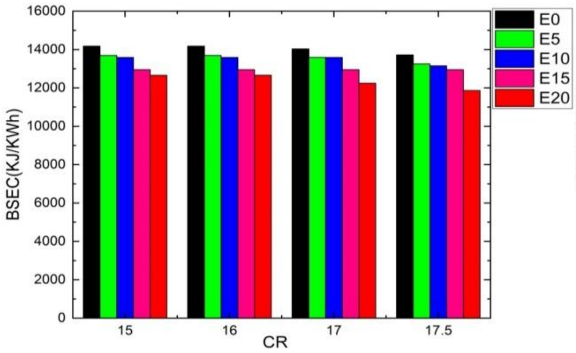


Fig.1 Variation of brake thermal efficiency with CR **Fig.2** Variation of brake thermal efficiency with load

Figure 3 depicts the brake specific energy consumption for Diesel and Ethanol blends at various loads at CR17.5. As the load increases, the brake specific energy consumption decreases. Brake-specific energy expenditure is the amount of heat required to generate one unit of power. The discrepancy in ethanol blends is due to ethanol blends' poor calorific value as mixing increases. When compared to E0, the reductions in brake-specific energy consumption for E5, E10, E15, and E20 at maximum load are 3.42%, 4.13%, 5.61%, and 7.69%, respectively. The fluctuation of brake-specific energy consumption and compression ratio is depicted in Figure 4. As the compression ratio increases, so does the brake specific energy consumption, which in some cases can be practically comparable. Because blends have a lower calorific value than pure diesel, the brake specific energy consumption decreases as the quantity of ethanol blending increases. For E20, E15, E10, E5, and E0, the brake specific energy consumption (KJ/KWh) at CR15 is 12254.18, 12948.14, 13589.58, 13690.33, and 14175.71.



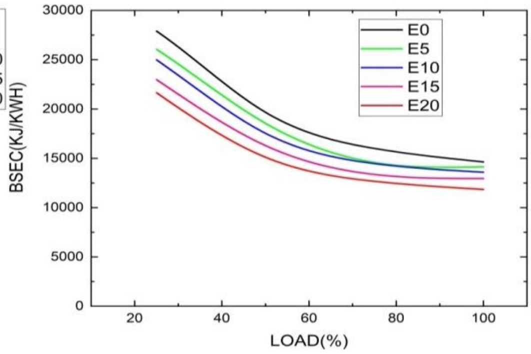


Fig 3 Variation of brake specific energy consumption with CR **Fig 4** Variation of brake specific energy consumption with load

3.2. Combustion characteristics

The graph in Fig. 5 illustrates the variation of the maximum rate of pressure rise and the various fuels at CR17.5 full load conditions. It shows that, with a slight increase in comparison to the starting blends, the rate of pressure rise decreases as the ethanol content and blend ratio increase. Because RPR analysis shows how smoothly the combustion process moves through the cylinder, it is essential to engine studies.

According to the graph, the rate of pressure rise (bar) for E0, E5, E10, E15, and E20 is 5.4, 5.2, 5.0, 4.9, and 4.9, respectively. The graph presented in Figure 6 illustrates the variation in the compression ratio and maximum rate of pressure rise for each blend under full load conditions. The graph shows that for blends, the rate of pressure rise falls at any given compression ratio while it increases as the compression ratio increases. The maximum rate of pressure rise (bar) for blends E0, E5, E10, E15, and E20 is 5.2, 5.1, 5, 5,5 at CR15, 5.4, 5.3, 5.2, 5.1 at CR16, and 5.5, 5.4, 5.3, 5.2, 5.2 at CR17, in that order. Reduce the maximum rate of pressure rise in order to prolong engine life and reduce noise.

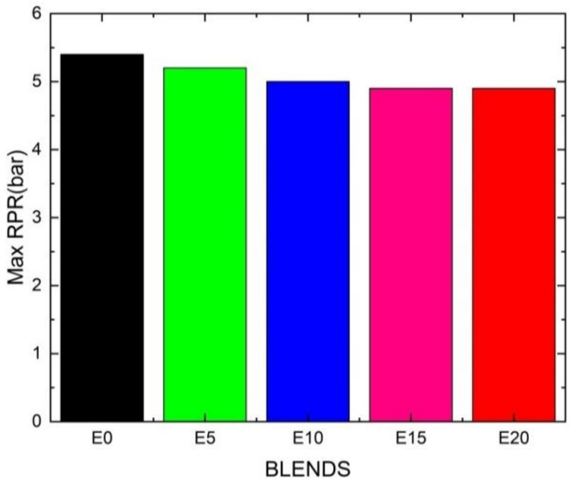


Fig 5. Variation of rate of pressure rise with blends

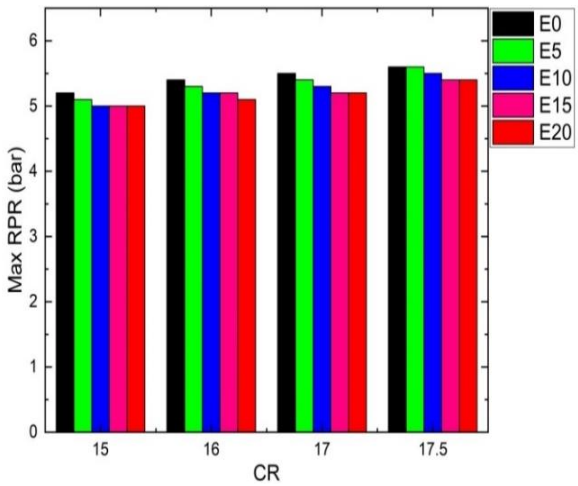
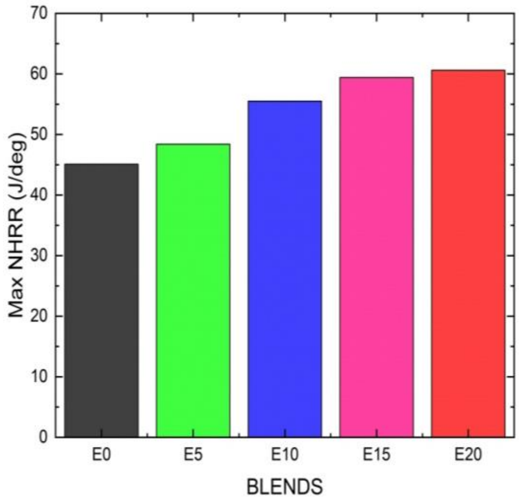


Fig 6 Variation of rate of pressure rise with CR

Under CR17.5 full load conditions, Figure 7 shows the net heat release rate and mixes for pure Diesel and Ethanol blends. The net heat release rate rises when comparing mixes to diesel. This is because the combustion phase proceeds more effectively and the chemical delay diminishes as mixing increases. Consequently, the quantity of heat produced per degree of crank angle rises as blending continues. The NHRR for E5, E10, E15, and E20 is 1.02%, 14.55%, 23.9%, and 25.98% higher than Pure Diesel. The E20 blend has a higher net heat value than the E0 blend. Figure 8 shows a graph that shows how the compression ratio affects the net heat release rate. The net heat release rate falls as the compression ratio rises. Blending the ethanol content increases the net heat release rate at a given compression ratio. The net heat release rate is influenced by both the combustion process and the chemical delay. Higher mixes result in a higher chemical delay, which raises the net heat release rate because the combustion phase proceeds more efficiently. As indicated by the above chart, the net heat release rate (J/deg) for E0, E5, E10, E15, and E20 is 58, 59.4, 61.3, 65.4, 65.4 at CR15, 51.6, 54.6, 58.6, 58.8, 63.3 at CR16, and 49.1, 49.6, 55.1, 60.1, 61.7 at CR17.



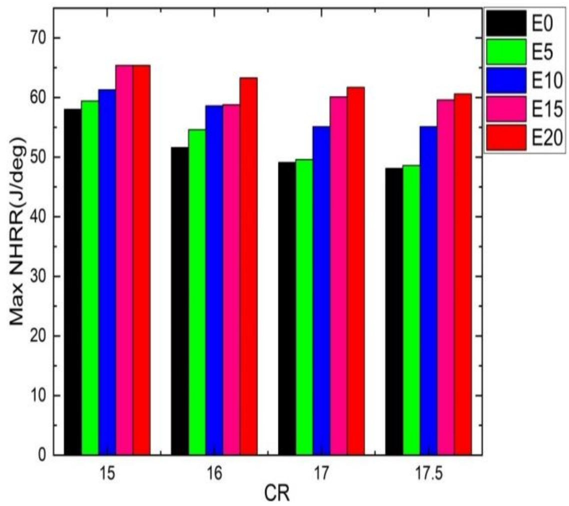
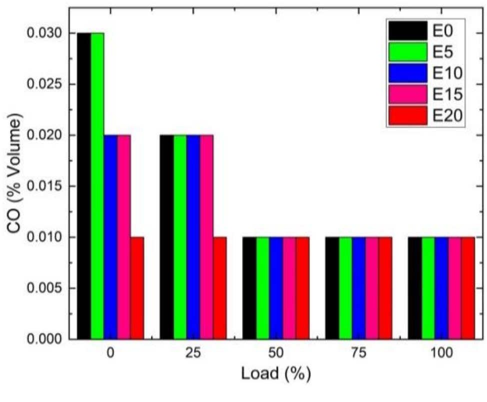


Fig 7 Variation of Net heat release rate with blends **Fig 8** Variation of Net heat release rate with CR

3.3. Emission characteristics

CO emissions and load for the different mixes (E0, E5, E10, E15, and E20) are shown in the graph above (Fig. 9). When there is not enough oxygen present in the fuel to completely burn all of the carbon, carbon monoxide is created during the combustion process. Rising combustion chamber temperatures, the physical and chemical properties of the fuel, the air-fuel ratio, a lack of oxygen at high speeds, and a shorter time allowed for full combustion are some of the possible causes of the above pattern of increased CO emissions. It would be anticipated that as oxygen concentration rose, CO would decrease due to the effect of fuel viscosity on fuel spray quality. Because of an increase in oxygen content, the blends' CO emissions decrease with increasing load compared to diesel. The change in CO emissions and compression ratio at full load is shown on the graph in Figure 10. When the compression ratio rises, CO emissions decrease until they are nearly equal to the blends' full load CO emissions at higher compression ratios. For E0, E5, E10, E15, and E20, the CO (% Volume) emissions are 0.03 at CR15, 0.02 at CR16, 0.02 at CR17, and 0.02 at CR15, 0.02, 0.02, 0.02, 0.02, and 0.02. The graph shows that CO emissions at CR17 and CR17.5 are almost equal.



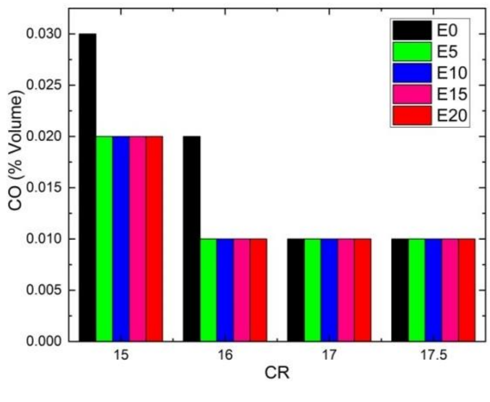


Fig 9 Variation of CO with load **Fig 10** Variation of CO with CR

The graph in Fig. 11 depicts the fluctuation of CO2 and load at CR17.5 (increased compression ratio). As indicated in the graph above, CO2 drops as ethanol blending increases. The combustion process produces carbon dioxide when rich fuel mixes are used and when there is insufficient oxygen to completely convert all of the carbon in the fuel to CO2.

The above pattern of increased CO2 emissions could be attributed to growing combustion chamber temperatures, the physical and chemical qualities of the fuel, the air-fuel ratio, a lack of oxygen at high speeds, and a shorter amount of time allowed for full combustion. When operating at full load, the CO2 emissions for E0, E5, E10, E15, and E20 are 0.5, 0.5, 0.4, 0.4, and 0.4, respectively. The compression ratio and CO2 fluctuation are shown in the above chart (Fig. 12). Because ethanol contains more oxygen than E0, which results in a richer air-fuel mixture and an efficient amount of oxygen needed for fuel combustion, CO2 emissions decrease as the compression ratio increases and at any given compression ratio for different blends. For E0, E5, E10, E15, and E20, the carbon dioxide emissions at CR15 are 0.53, 0.51, 0.5, 0.5, and 0.5, respectively, and the values are almost the same for higher blends.

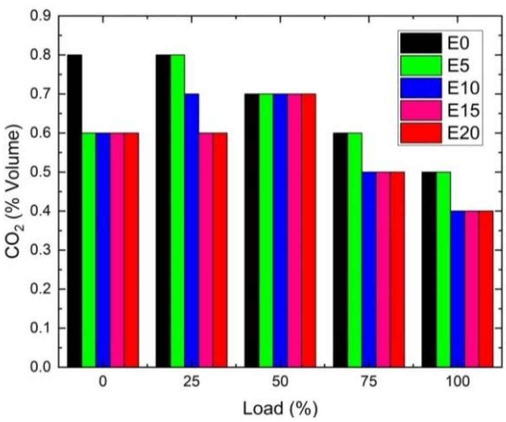


Fig 11 Variation of CO_2 with load

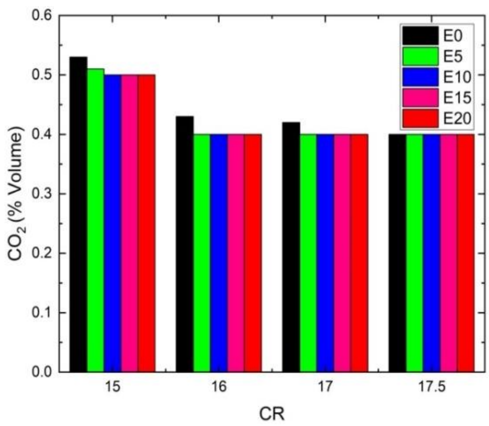


Fig 12 Variation of CO_2 with CR

The variation of hydrocarbon emissions and load at CR17.5 is shown in Fig. 13, which indicates that HC emissions rise with increasing ethanol content. This is because higher temperatures are reached and incomplete combustion of the hydrocarbons occurs after the afterburning stage in the combustion chamber. The fact that HC emissions increase sharply with increasing ethanol content is one of the main disadvantages of using ethanol as an alternative fuel. At maximum load, the HC (ppm) emissions for E0, E5, E10, E15, and E20 are 1, 5, 7, and 9, in that order. A comparison of unburned hydrocarbons (HC) and the compression ratios for blends and diesel at full load is shown in Figure 14. The temperature of the gas is the main factor that determines how much HC is released. Although the formation of mixed gas and fuel combustion are accelerated by rising temperatures in the combustion chamber walls and gas, the combustion process leaves behind unburned hydrocarbons, which raises HC emissions. Compared to diesel, all of the blends have higher HC emissions. In addition to the aforementioned causes, the mixes' increased oxygen concentration is another factor. At CR17, the corresponding HC emissions for E0, E5, E10, E15, and E20 are 5, 6, 6, 7, and 8.

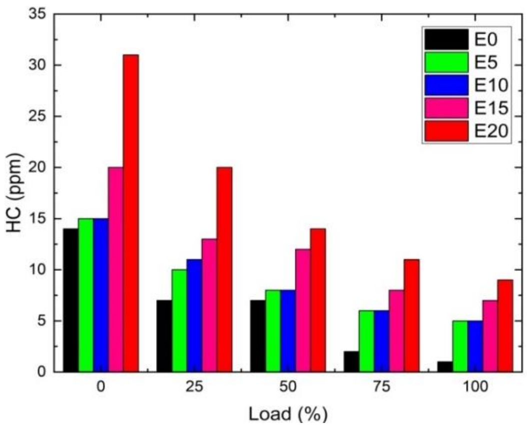


Fig 13 Variation of HC with load

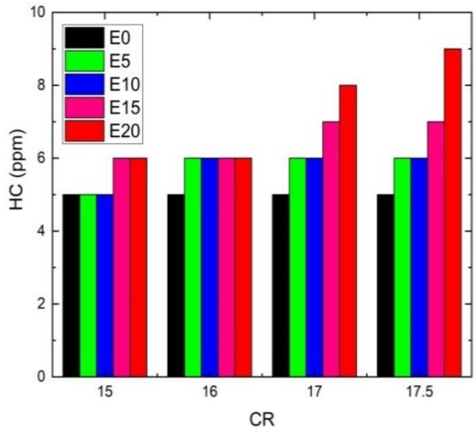


Fig 14 Variation of HC with CR

The nitrogen oxide emissions and load for E0, E5, E10, E15, and E20 are shown in Figure 15. Nitrogen dissociation, which is the result of fuel unburning at high temperatures, releases oxides of nitrogen, which are dangerous pollutants. The test findings show that nitrogen oxides are extremely low at low loads and increase with load from E0 to E20. Nitrogen oxide values are lowered when ethanol and diesel are combined, as the graph shows. The engine temperature rises with higher loads, which raises the amount of NOx exhaust gases. As shown in the chart, blending ethanol with pure diesel raises the heat of vaporisation value, which lowers the blends' nitrogen oxide levels. The variation of nitrogen oxides with compression ratio for multiple blends at maximum load is shown in Figure 16. NOx emissions at CR15 are 41, 35, 31, 31, and 30 for E0, E5, E10, E15, and E20, respectively. For E0, E5, E10, E15, and E20, respectively, NOx emissions at CR17.5 are 31.7%, 42.85%, 51.61%, 45.61%, and 43.33% higher than at CR15. Blends have a higher heat of vaporisation value than single-source blends, so blends have lower oxides of nitrogen emissions even at higher temperatures. This is because increasing the compression ratio reduces the clearance volume, leaving little volume left for combustion. The viscosity and calorific value of the blends may also have an impact on NOx emissions.

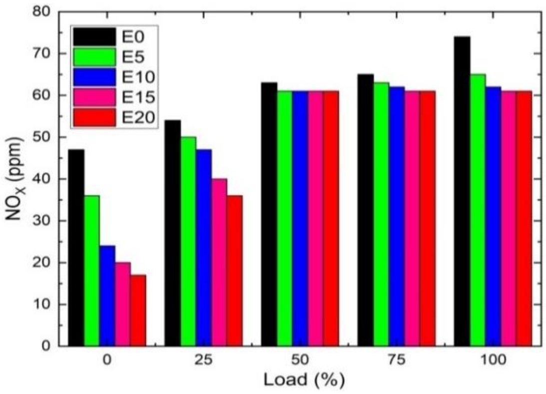


Fig 15 Variation of NO_x with load

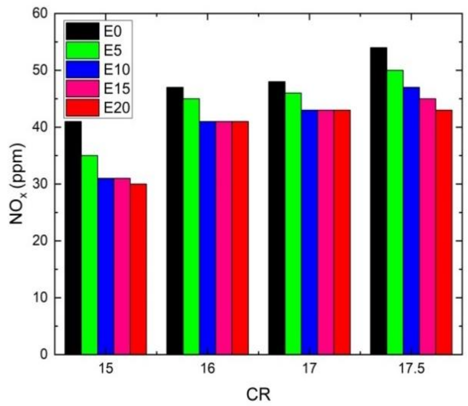


Fig 16 Variation of NO_x with CR

3.4. Vibration analysis

According to Fig. 17, which depicts the superimposed time waves, acceleration variation, and time at a compression ratio of 17.5 and full load circumstances for E0 and E20, the maximum acceleration peaks produced for E20 blend are less than those obtained for E0 blend. This is because, as previously stated, the rate of pressure rise slows as the ethanol content of the fuel increases. The fundamental cause is that ethanol-blend fuels have a lower cetane number than pure diesel, resulting in a longer ignition delay for the blended fuels and a reduction in vibrations.

Peak pressures in the cylinder during the combustion process are represented by the peaks in the time-domain signals. The frequency spectra curves for E0 and E20 at full load and CR17.5 are superimposed in Figure 18. Peak amplitudes for diesel fuel are 2.62 m/s2 and 2.52 m/s2 at 63.32 Hz and 163.49 Hz, respectively. The E20 fuel has peak amplitudes of 2.13 m/s2 at 164 Hz and 2.37 m/s2 at 63.67 Hz, respectively. The reason for the decrease in E20's peaks when compared to E0 is that ethanol has a lower cetane number than diesel. As a result, blended fuels have a lower cetane number than diesel, which results in fewer peaks when compared to E0 fuel, but higher peaks in the graph when compared to pure diesel. This may be because vibrations are mostly produced by reciprocating and rotating parts, and when mixed petrol is used, the crank's rotational speed increases, causing the peaks in certain frequencies to be marginally larger for E20 than for E0.

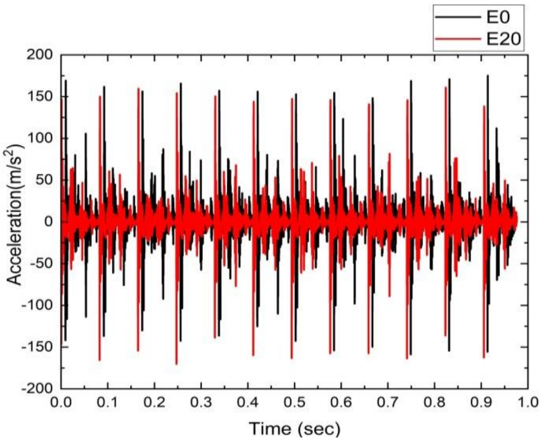


Fig 17 Variation of Acceleration with time

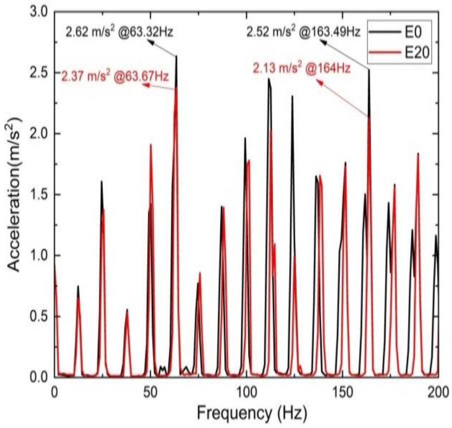


Fig 18 Variation of Acceleration with frequency

Figure 19 displays the maximum acceleration values and load variation for all mixes at increasing compression ratios. The peaks of the acceleration values rise with the load because the added weight increases the amount of fuel needed to maintain the constant speed, which in turn causes vibrations. Blended fuels have lower maximum acceleration values than pure diesel. As mentioned earlier, when it comes to blended fuels, a lower Cetane number results in an increase in ignition delay, which lowers vibrations and speeds up the rate of pressure rise. Maximum acceleration values at full load for E5, E10, E15, and E20 are lower than E0 by 5.37%, 27.7%, 29.03%, and 30.10%, respectively. The maximum acceleration and compression ratio at full load for each blended fuel are shown in Figure 20. Unidirectional combustion forces resulting from changes in cylinder gas pressures, structural resonances, and alternating inertia forces concentrated on engine components are what cause vibrations in a diesel engine. The graph indicates that engine noise and vibration were decreased when the compression ratio was increased. The ethanol-blend petrol vibrates less when the compression ratio rises. The blends for E5, E10, E15, and E20 fall by 1.89%, 9.76%, 12.44%, and 14.74% at compression ratio 16 in comparison to E0. Comparing E5 to E0, blend decreases are 2.43%, 5.37%, 7.77%, and 9.94% for E5, E10, E15, and E20 at compression ratio 17. The CR15 and CR17.5 for E0, E5, E10, E15, and E20 are 34.5%, 34.8%, 48.3%, 47.2%, and 45.83%.

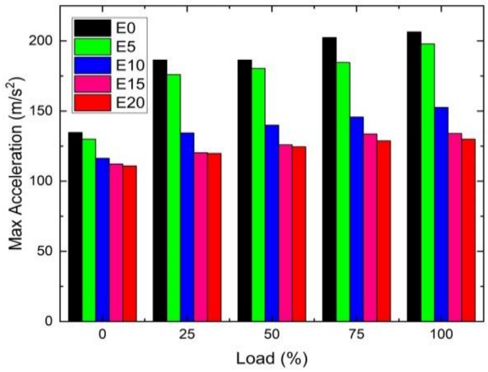


Fig 19 Variation of Acceleration with load

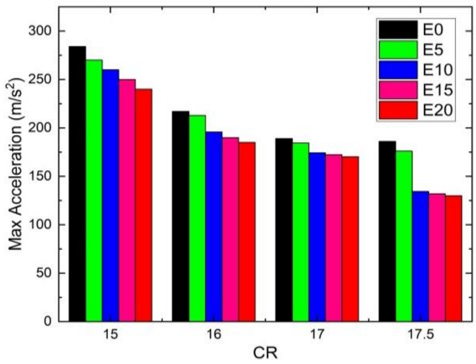


Fig 20 Variation of Acceleration with CR

4. CONCLUSION

The goal of this research is to identify the ideal mixture for meeting the performance requirements for vibrations, emissions, and combustion. For all blends, thermal efficiency rises with load, but it is greater for the E20 blend. At CR17.5, the E20 blend's thermal efficiency is 13.33% greater than that of pure diesel. Compared to pure diesel, the rate of pressure rise for E15 is slower at full load. Compared to pure diesel, blended fuel has a higher NHRR. All blends have lower CO and smoke density emissions than pure diesel. Both the emissions and the vibrations are reduced in the E5 and E10. Therefore, E5 and E10 blends are advised for the engine's optimal operation based on the engine's performance, combustion, emissions, and vibrations.

5. NOMENCLATURE

E0	-	100% Diesel	BSEC	-	Brake Specific Energy Consumption
E5	-	Diesel 95% Ethanol 5%	BSFC	-	Brake Specific Fuel Consumption
E10	-	Diesel 90% Ethanol 10%	CR	-	Compression Ratio
E15	-	Diesel 85% Ethanol 15%	BTE	-	Brake Thermal Efficiency
E20	-	Diesel 80% Ethanol 20%	CI	-	(Compression Ignition)engine

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editor@iaeme.com