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Effect of Blending Alcohol with Diesel on Performance, Combustion and Emission Characteristics of Four Stroke Diesel Engine– An Experimental Study

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Abstract: In day today's relevance, it is mandatory to devise the usage of diesel in an economic and environmentally benevolent way. In the present scenario, the very low combustion efficiency of compression ignition engine leads to poor performance of the engine and emission of incomplete combustion by-products. Hence it was planned to increase the combustion efficiency, and thereby to increase the performance and to reduce the exhaust emission by adding n-propanol at different proportions like 2%, 4%, 6%, 8% and 10% by volume with diesel. The performance, combustion and emission characteristics observed while using blended fuels were analyzed and compared with that of diesel as fuel without any additives. The performance analysis revealed that, at full load, the brake thermal efficiency is increases by 11.78% for 10% n-propanol blended with diesel. The emission test reported that, the CO and NO_x emissions decreased by 44.12% and 9.33% respectively at full load.
Keywords: Performance, Combustion, Emission, I.C. Engine, Blended fuel.

1. Introduction

In day today's applications, though the economic and environmental aspects do not agree to the usage of diesel, it is not promising to accomplish the habitual livelihood without the usage of diesel engine. Therefore it is enforced to formulate the usage of diesel in an economic and environmentally benign way. The emission from the diesel engines seriously disturbing the living beings. This work was aimed at to study the effect adding n-propanol with diesel on performance, combustion and emission characteristics of a four stroke diesel engine.

It was revealed that the combustion of diesel in a compression ignition engine occurs in three major stages like ignition delay, pre-mixed combustion phase and diffusion combustion phase. Of these, the ignition delay period which is the time period between the start of injection and the onset of combustion, have influence on all ignition processes[1].

Various studies have already been done to improve the performance and reduce the emission by using different types of neat bio-diesel, blending different bio-diesels with diesel at varying proportions, introducing some modifications in the fuel supply system and in the combustion chamber, and blending different additives with diesel.

Many researchers have done number of experimental investigations to use vegetable oils as fuel in diesel engines, and reported that the very high viscosity and low volatility of vegetable oils resulting in poor atomization slow burning, more smoke emissions and uncontrolled combustion. The exhaust gas temperature increase and NO_x reduction with a slight increase in CO emission while using diesel blended with vegetable oil as fuel were observed[2]. The usage of palm oil as fuel in diesel engines reported that the short term usage of palm oil augments the performance and emission levels considerably and the prolong usage causes carbon deposits and piston rings sticking[3]. The usage of preheated vegetable oil as fuel, reduces the problem of filter clogging, also increases the engine performance and reduces the carbon deposits[4].

The usage of cotton seed oil as fuel, without doing any modifications in the engine concluded that the engine parameters need to be readjusted in order to have the maximum power output and highest thermal efficiency[5]. The experimental investigation on a DI compression ignition engine by using honge, neem and sesame oil methyl esters as fuel resulted that the performance and emission characteristics are comparable[6]. The higher viscosity of biodiesel tends to reduce the engine power and engine torque, also the lower calorific value of biodiesel results in the increase in specific fuel consumption and decrease in combustion temperature[7]. The usage of biodiesel derived from rice bran oil concluded that there is an increase in NO_x emissions due to the presence of molecular oxygen in the bio-diesel[8]. The experimental investigation on the diesel engine while using linseed oil, rice bran oil and mahua oil with diesel, reported that blending of 50% linseed oil with diesel increases the smoke density and decreases the brake specific energy consumption, and also concluded that the mixing of 30% mahua oil with diesel reduces the smoke density and increases the thermal efficiency as compared to diesel[9].

The modifications in the engine received significant attention among the engine researchers in order to improve the performance and combustion characteristics, and reduce the emission from the diesel engine. The adoption of exhaust gas recirculation technique reduces the NO_x emission considerably[10]. The various experimental investigations by advancing the injection timing and increasing the injection pressure reported that there is an appreciable increase in brake thermal efficiency and decrease in CO, HC and smoke emissions[11].

In recent years, considerable attention is given on the usage of primary alcohols as additive or blend with diesel. The usage of alcohols in compression ignition engine has its own intricacy due to its high latent heat of vaporization and long ignition delay period[12]. However, it was reported that the usage of alcohols as blend in proper proportion with diesel reduces the exhaust emissions[13]. The bioethanol derived from vegetable oils was considered as the most suitable alternate fuel because of its better spark characteristics and higher cetane number values[14]. The blending of bioethanol with diesel appreciably reduces the green house gas emission[15]. The usage of biodiesel blended with alcohols reported that there is an increase in brake thermal efficiency and decrease in CO, HC and NO_x emissions[16].

In this experimental study, n-propanol was identified and blended with diesel as an additive in varying proportions like 2%, 4%, 6%, 8% and 10% by volume. The various performance, combustion and emission characteristics of the diesel engine while using diesel blended with n-propanol at different proportions and at different loading conditions were evaluated. The various performance, combustion and emission parameters of the engine thus evaluated were analyzed and compared with that of the engine while using normal diesel as fuel.

2. Methodology

2.1 Materials used

The normal diesel supplied by Indian Oil Corporation was procured from the local market. The n-propanol supplied by Kemphasol Limited, Mumbai was taken for blending with diesel. The blended fuels were prepared by mixing n-propanol at different proportions like 2%, 4%, 6%, 8% and 10% by volume with diesel. The complete mixing of n-propanol with diesel was done with the help of a mechanical stirrer. The various properties of diesel and n-propanol are given in Table 1.

Table 1: Properties of diesel and n-propanol used

S. No.	Property	Unit	Diesel	n- propanol
1.	Molecular formula	--	$C_{14}H_{22}$	C_3H_7OH
2.	Lower calorific value	kJ/kg	43,200	30,680
3.	Specific gravity	--	0.83	0.802
4.	Kinematic viscosity	cSt	2.6	2.8
5.	Latent heat of vaporization	kJ/kg	250	779
6.	Cetane number	--	49	15

2.2 Experimental set-up

For experimentation, a stationary single cylinder diesel engine with the specifications mentioned in Table 2 was used. The engine was coupled with an eddy current dynamometer. A computerized data acquisition system was attached with the engine to measure and record various performance and combustion parameters like fuel flow rate, speed of the engine, temperature of incoming air, temperature of exhaust gas, pressure inside the cylinder, heat release rate, etc. The AVL 444 Di gas exhaust gas analyzer manufactured by AVL, Austria was used to measure the amount of CO, CO_2, NO_x, O_2 and HC present in the exhaust emission. The AVL 413 smoke meter was used to measure the smoke density.

The layout of the experimental setup was shown in the figure 1.

Table 2: Engine specifications

S. No.	Parameter	Specification
1.	Engine model	Kirloskar TV-1, DI, Naturally aspirated, Water cooled
2.	Number of cylinders	1
3.	Bore	87.5 mm
4.	Stroke	110 mm
5.	Compression ratio	17.5
6.	Maximum power at rated rpm	5.2 kW
7.	Rated speed	1500 rpm
8.	Injection pressure	220 bar
9.	Injection timing	23° before TDC

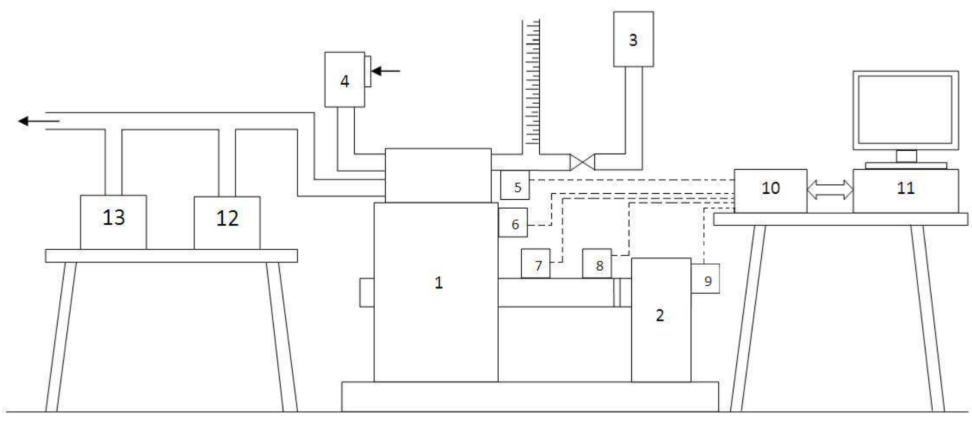

Figure 1. Experimental setup

1. Engine, 2. Eddy current dynamometer, 3. Fuel tank, 4. Air box, 5. Fuel flow sensor, 6. Pressure sensor, 7. Speed sensor, 8. Crank angle encoder, 9. Load cell, 10. Data acquisition system, 11. Computer, 12. Five gas analyzer, 13. Smoke meter.

2.3 Experimental procedure

Initially, at no load conditions, the engine was started using normal diesel as fuel. After getting warm-up, the load was applied on the engine at the rate of 20% of full load, through the eddy current dynamometer and the engine was allowed to run for a while. After the engine reaches equilibrium condition, the various performance, combustion and emission characteristic parameters were observed and recorded.

The load was then increased to 40% of full load and the engine was allowed to run for some time to reach the equilibrium condition. After reaching the equilibrium conditions, the various performance, combustion and emission parameters were noted as per the standard procedure. By adopting the same procedure, the various parameters were observed and recorded for higher loads such as 60%, 80% and 100% of full load. From the observed and recorded values, the various parameters such brake thermal efficiency, brake specific energy consumption were evaluated by using standard relations.

Then the diesel mixed with 2% n-propanol was used as fuel; by repeating the same procedure, the parameters related to performance, combustion and emission characteristics were observed and recorded at different load ranges. From the observed values, by using standard relations, the various performance, combustion and emission characteristics of the engine were calculated.

The same procedure was then repeated to evaluate the performance, combustion and emission characteristics of the engine for other blended fuels such as 4%, 6%, 8% and 10% of n-propanol blended with diesel. The various characteristics thus evaluated were compared and analyzed with that of the engine using normal diesel as fuel.

From the observed, recorded and calculated values, the various performance, combustion and emission characteristics of the engine while using diesel and different blended fuels at different loading conditions were presented in the form of graphs.

2.4 Uncertainty analysis

The operating environment and usage of various measuring instruments may cause some error and uncertainty in the observed and calculated values. The accuracy of the direct measuring instruments was given in the table 3. By using the uncertainty analysis based on Gaussian distribution method with a confident limit of ±2σ, the overall uncertainty for the performance parameters in the experiment was calculated as ±1.5%.

Table 3: Accuracy of the measuring instruments

S. No.	Name of the instrument	Parameters	Measuring range	Accuracy
1.	AVL 444 Digas - 5 gas analyzer	CO	0 – 10 % volume	<0.6 % vol: ±0.03% vol
		CO_2	0 – 20 % volume	<10 % vol: ±0.5% vol
		O_2	0 – 22 % volume	>2 % vol: ± 5% vol
		HC	0 – 20000 ppm volume	>200 ppm vol: ±0.5% of ind value
		NO_x	0 – 5000 ppm volume	>500 ppm vol: ±10% of ind value
2.	AVL 413 Smoke meter	Smoke density	0 – 100 % opacity	± 2%

3. Results and Discussion

The comparison and analysis made on the various graphs drawn resulted in the following discussions.

3.1 Comparison of Performance Parameters

3.1.1. Brake thermal efficiency

Figure 2 shows the relation between brake thermal efficiency and load. From the graphs, it was noted that the brake thermal efficiency is almost same at low and medium loads for all the blended fuels. But above 60% load, the brake thermal efficiency increases for all the blended fuels, moreover the increase in percentage of n-propanol with diesel increases the brake thermal efficiency. At full load, the brake thermal efficiency is increased by 3.907%, 6.541%, 7.773%, 9.462% and 11.784% for the addition of 2%, 4%, 6%, 8% and 10% n-propanol with diesel respectively. The increase in brake thermal efficiency due to the addition of alcohols was already evidenced[16]. The addition of alcohol with diesel decreases the viscosity of the blended fuels. The decrease in viscosity improves the spray characteristics and resulted in higher brake thermal efficiency[17]. The increase in brake thermal efficiency may also due to the higher premixed combustion of alcohol blended fuels because of low cetane number which lead the higher percentage of combustion at constant volume[18].

3.1.2. Brake specific energy consumption

Figure 3 indicates the graphs drawn between brake specific energy consumption and load. From the graphs, it was observed that the brake specific energy consumption at low and up to medium loads is almost same for all blended fuels, and decreases from medium load to full load for all blended fuels. The lower viscosity and higher volatility of n-propanol blended with diesel improves the spray characteristics, fast vaporization and thereby efficient combustion[19]. At full load, the brake specific energy consumption decreases by 3.763% for 2% addition, 6.139% for 4% addition, 7.214% for 6% addition, 8.646% for 8% addition and 10.543% for 10% addition. The decrease in brake specific energy consumption is also due to the lower calorific value of n-propanol[20].

3.1.3. Exhaust gas temperature

Figure 4 depicts the variation of exhaust gas temperature with respect to load. From the graphs, it was clear that at lower loads, the exhaust gas temperature for all blended fuels is slightly higher than that of diesel. But at medium and at full loads, the exhaust gas temperature for all the blended fuels is lower when compared with that of diesel. The reduction in exhaust gas temperature is due to the higher latent heat of vaporization and the quenching effect of n-propanol in the combustion chamber[21].

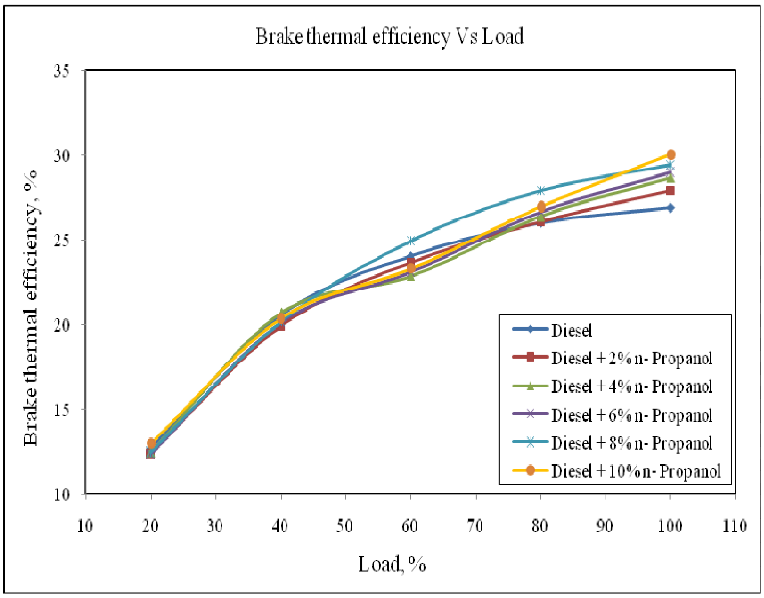

Figure 2: Brake thermal efficiency Vs Load

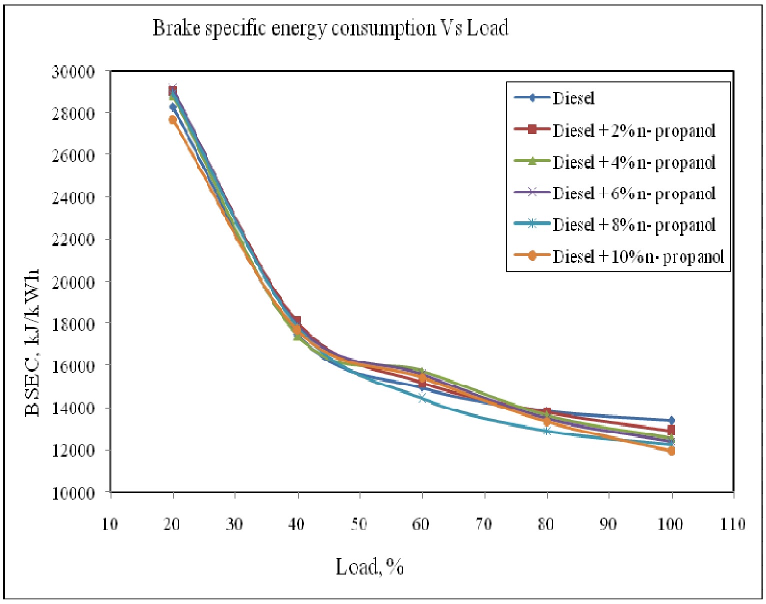

Figure 3: Brake specific energy consumption Vs Load

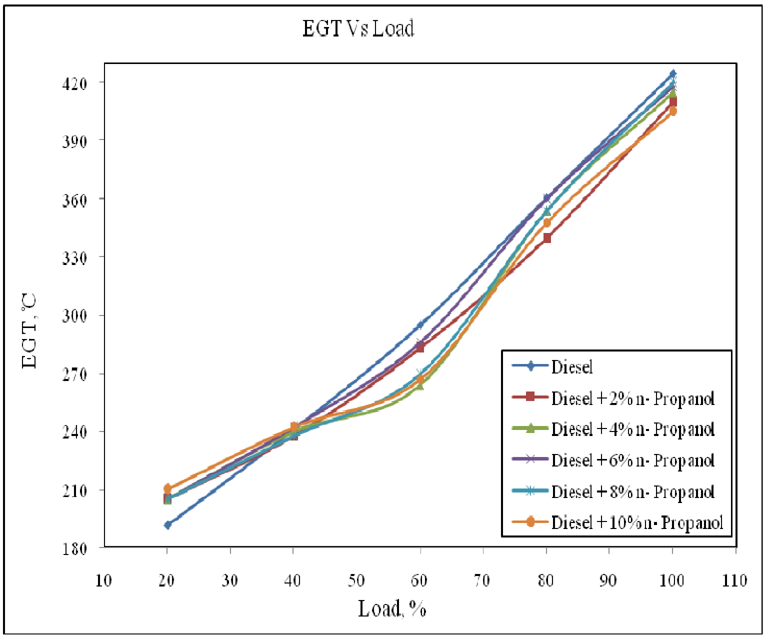

Figure 4: Exhaust gas temperature Vs Load.

3.2 Comparison of Combustion Parameters

3.2.1. Peak cylinder pressure

Figure 5 shows the magnitude and occurrence of peak cylinder pressure which were found from the history of cylinder pressure and crank angle at full load and at the rated speed of the engine. From the graphs, it was observed that the cylinder pressure curves for diesel and all blended fuels are similar in shape. The small variation in maximum cylinder pressure while using blended fuels was already reported by Gong Yanfeng et al[22].

3.2.2. Heat release rate

Figure 6 indicates the heat release rate of diesel and various blended fuels between the crank angle variation -30° and 90° when the engine is running at full load. From the graphs, it was observed that the maximum heat release rate is 138.6 kJ/$m^3$for diesel at 11° BTDC, 157.9 kJ/m^3 for 2% n- propanol with diesel at 10° BTDC, 154.2 kJ/m^3 for 4% n- propanol with diesel at 10° BTDC, 155.6 kJ/m^3 for 6% n- propanol with diesel at 10° BTDC, 148.1 kJ/m^3 for 8% n- propanol with diesel at 10° BTDC, 163.2 kJ/m^3 for 10% n-propanol with diesel at 10° BTDC. Here the increase in heat release rate for blended fuels is due to the increase in spray characteristics by the addition of n- propanol. The quenching effect and reduction in in-cylinder temperature due to the high latent heat of vaporization of n- propanol delayed the maximum heat release rate[19].

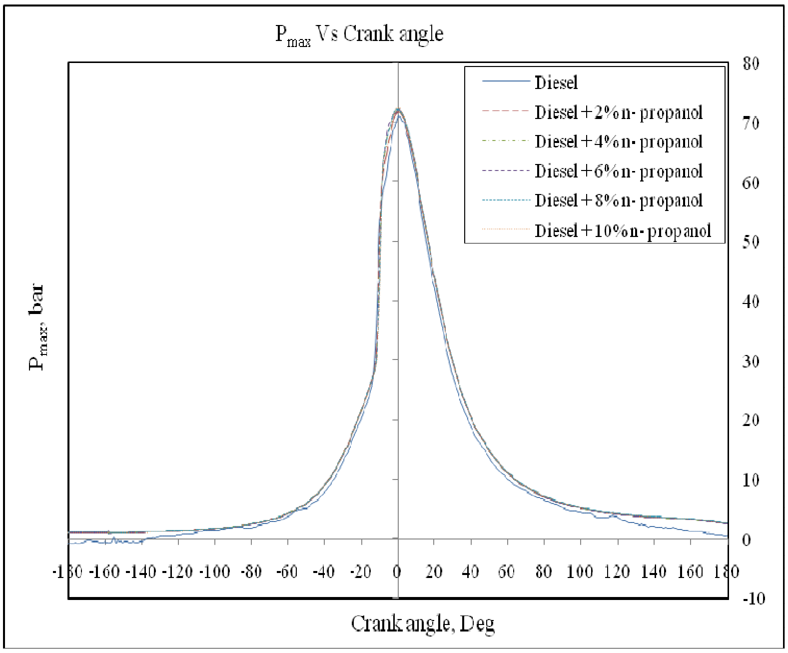

Figure 5: Peak pressure Vs Load

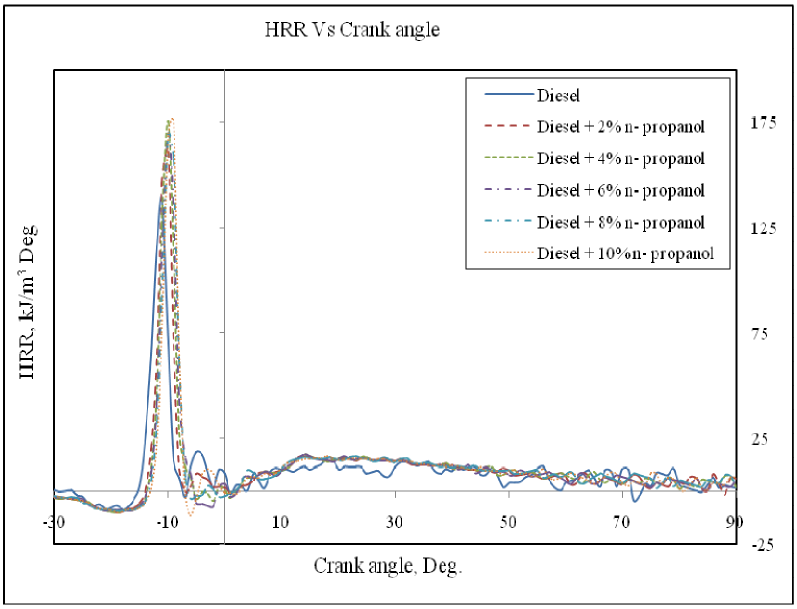

Figure 6: Heat release rate Vs Load

3.2.3. Total heat release

Figure 7 depicts the total heat release during a cycle. From the graphs, it was noted that the accumulated heat release rate is higher for all blended fuels at full load. This is due to the increase in flame travel speed and spray characteristics due to the addition of n-propanol which is supported by Gong Yanfeng et al[22].

Comparison of Emission

3.3.1. CO Emission

Figure 8 indicates the relation between the CO emission and load. From the graphs, it was observed that the CO emission decreases gradually up to 60% load on the engine for all fuels, then the CO emission gradually increases up to 80% load, and after that it increases rapidly up to full load. The sharp increase in CO emission at full load is because, when at high load the mixture supplied to the engine is rich[19]. It was also noted that, the CO emission for all blended fuels is greater than that of diesel at low and medium loads. This is due to the reduction in in-cylinder temperature by the higher latent heat of vaporization of n-propanol. But at full load, the reduction in CO emission is 5.882% for 4% addition, 14.706% for 6% addition, 23.529% for 8% addition and 44.118% for 10% addition of n-propanol with diesel. This is due to the fact that at full load, the in-cylinder temperature is high which makes better combustion[23].

3.3.2. CO_2 Emission

Figure 9 shows the graphs drawn between CO_2 emission and load. Since the CO_2 emission highly influences the green house effect and global warning, it is necessary to measure the CO_2 emission from the engine. From the graphs, it was observed that the CO_2 emission increases gradually as the load increases for all blended fuels and also the emission of CO_2 is almost less for all blended fuels at all load ranges. This is because of the lower operating temperature due to high latent heat of vaporization of n-propanol[21].

3.3.3. HC Emission

Figure 10 represents the graphs drawn between HC emission and load. The graphs concluded that the HC emission is more for all blended fuels and at all load ranges. Moreover the HC emission increases with the increases in load. Here it was noted that the increases in percentage of n-propanol with diesel increases the HC emission, the rate of increase in HC emission at medium load is higher than that at higher load. This may be due to the high latent heat of vaporization of n-propanol, which leads the development of quench layer, reduction in temperature inside the cylinder, slow vaporization and incomplete mixing[24].

3.3.4. NO_x Emission

Figure 11 indicates the presence of NO_x in the engine exhaust with respect to load. The NO_x formation is highly influenced by combustion temperature[25]. From the graphs, it was observed that the NO_x emission increases with the increase in load for diesel and various blended fuels. Moreover at all load ranges, the NO_x emission is less for all blended fuels when compared with that of diesel. At 60% load on the engine, the decrease in NO_x emission is 12.446% for 2% addition, 14.077% for 4% addition, 15.193% for 6% addition, 26.438% for 8% addition and 28.584% for 10% addition of n-propanol with diesel. The high latent heat of vaporization and lower calorific value of n-propanol reduces the in-cylinder temperature which in turn reduces the NO_x emission[26]. Since NO_x is the most harmful, the reduction of it holds a prime important in the engine research.

3.3.5. Smoke density

The graphs drawn between smoke density and load are given in figure 12. It was observed that the smoke density increases with the increase in load for diesel as well as all blended fuels. The smoke density is less for all blended fuels at low and medium loads and it is high for higher loads. For 2% addition of n-propanol with diesel, at 20% load, the smoke density gets decreased by 30.555%, and at 100% load the smoke density gets increased by 17.658%. This is due to the fact that the richness of the mixture increases with the increase in load and the reduction in in-cylinder temperature is due to high latent heat of vaporization of the blend[26].

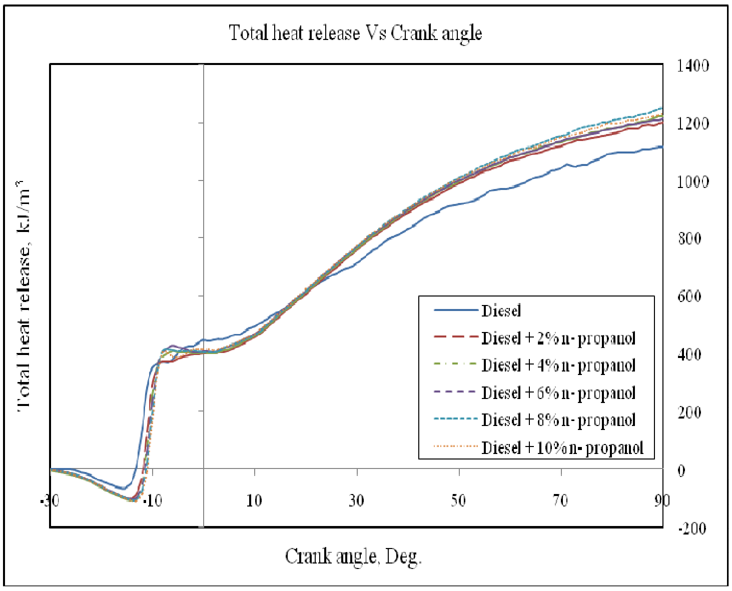

Figure 7: Total heat release Vs Load

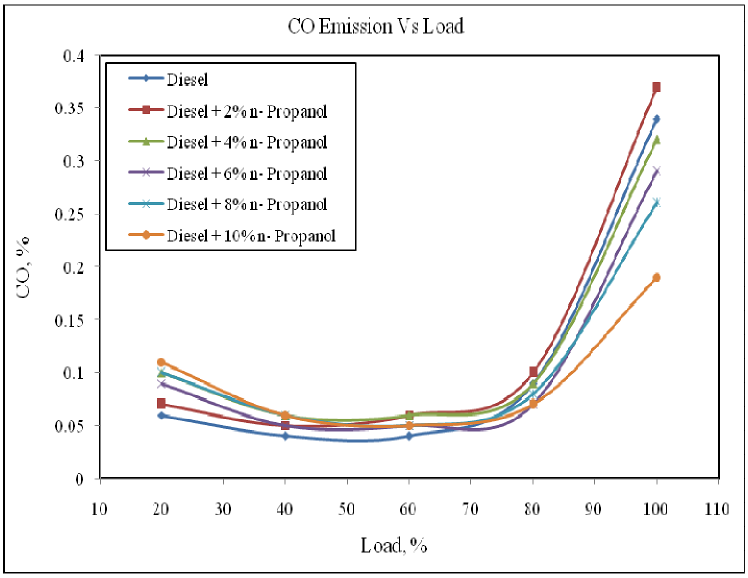

Figure 8: CO Emission Vs Load

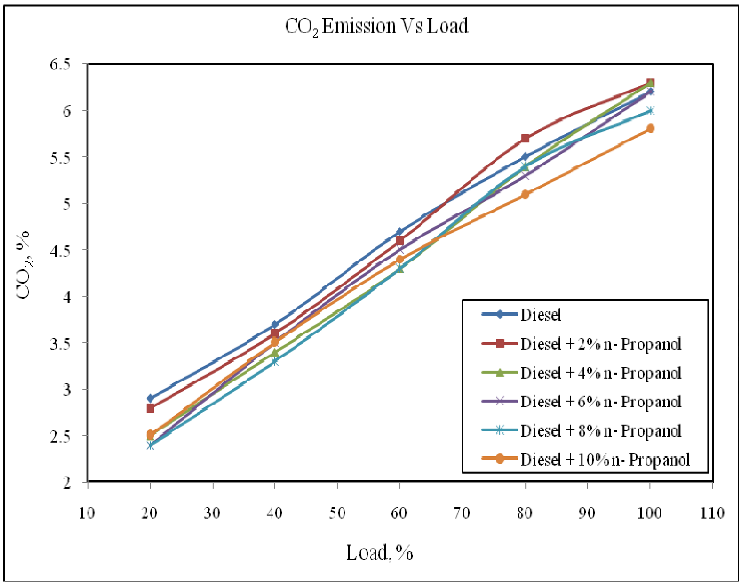

Figure 9: CO_2 Emission Vs Load

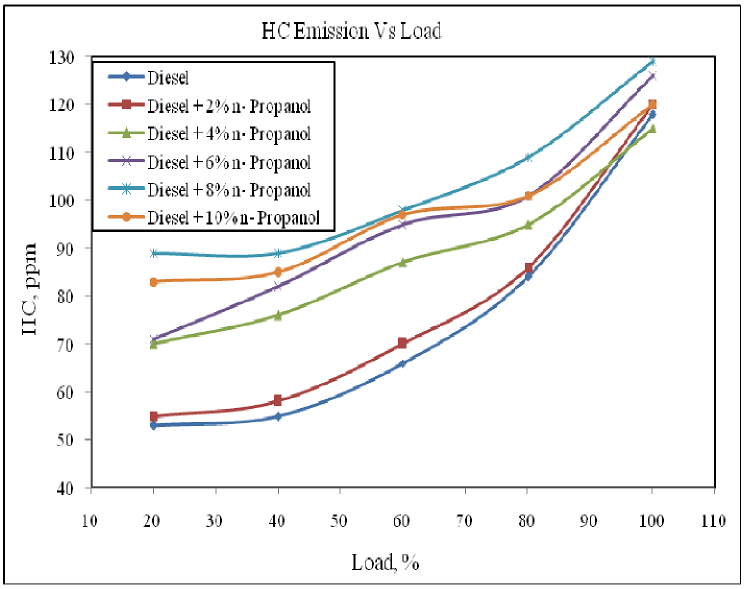

Figure 10: HC Emission Vs Load

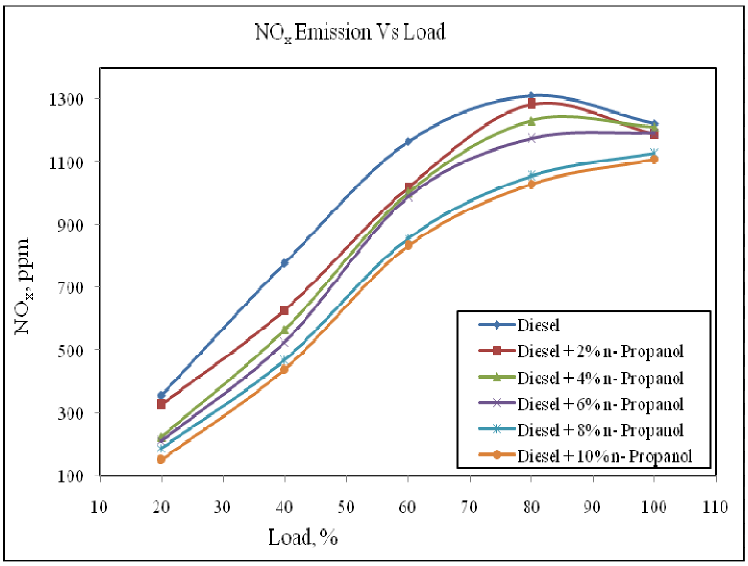

Figure 11: NO_x Emission Vs Load

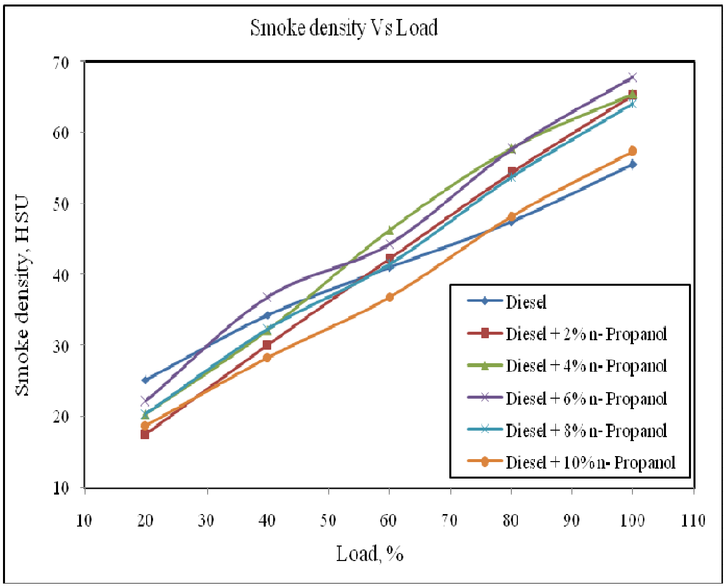

Figure 12: Smoke density Vs Load

4. Conclusion

This experimental investigation aimed at to enhance the performance of the diesel engine with the blend of n-propanol at different proportions like 2%, 4%, 6%, 8% and 10% by volume is attainable. In addition to this, within the scope of study, the NO_x emission from the engine could be minimized appreciably is another objective.

From the experimentation,

- The blending of n-propanol with diesel shows almost same brake thermal efficiency at low and medium loads, and higher percentage addition of n-propanol augments the brake thermal efficiency at high loads.
- The blending of n-propanol with diesel reduces the brake specific energy consumption at medium and high loads.
- The addition of n-propanol with diesel results in the reduction in engine operating temperature which in turn increases the life of the engine.
- The addition of n-propanol with diesel appreciably reduces the CO emission over the medium and high load ranges.
- The blending of n-propanol with diesel results in significant reduction in NO_x emission over the entire load ranges. Since NO_x is the most harmful, the reduction of it holds a prime important in the engine research.

From the above discussions, the blending of n-propanol with diesel at optimum percentage may be recommended and further researches in this area by advancing the injection timing and introducing the exhaust gas recirculation techniques will turn out to be highly efficacious.

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