Chapter

An Attempt in Blending Higher Volume of Ethanol with Diesel for Replacing the Neat Diesel to Fuel Compression Ignition Engines

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Abstract

Alcohols are renewable in nature and can be manufactured from biomass. Butanol a higher alcohol, can be utilized as co-solvent to prevent the phase separation of diesel-ethanol blends as per the previous researches. This experimentation has been conducted with the blends of diesel-ethanol with various proportions of n-butanol followed by the solubility test in the temperature range of 5–25°C. The results indicate that 45% of ethanol can be blended with diesel by the assistance of 10% of n-butanol to make the final blend stable up to a temperature of 5°C for 20 days, which met the requirements of the essential properties (ASTM). Existing diesel engine has been modified as per the optimal level of parameters such as intake air temperature (IAT), fuel injection timing (FIT), nozzle opening pressure (NOP) and compression ratio (CR) obtained using Taghuchi method of L_9 orthogonal array. Arrived out parameters are 75°C of IAT, 29° before top dead centre of FIT, 210 bar of NOP and 19: 1 of compression ratio. The implementation of these parameters in diesel engine and fueling with diesel-ethanol butanol blend containing 45% ethanol produced closer performance and emissions characteristics compared to that of diesel. However, the emissions of smoke, hydrocarbon and carbon monoxide produced by the optimal blend are found to be marginally higher compared to that of diesel. These can be ratified by the introduction of after treatment systems modifications.

Keywords: bioethanol, biobutanol, Taghuchi method, low temperature

1. Introduction

Renewable sources are the major available resource to replace the dependency of diesel fuel to internal combustion engines (CI) engines. This motivates the researchers to rapid up research in finding out a replacement to fossil by alcohols or biodiesels from various oils to blend with diesel, which leads in low dependency of diesel alone to fuel CI engines. Alcohols are better than biodiesels, as most researchers reported that the higher the volume of biodiesel in diesel-biodiesel blends the higher oxides of nitrogen emissions (NO_X). The utilization of biodiesel from edible resources will dictate limited usage of edible oils as food material. Out of the alcohols ethanol [1] can be blended into diesel for fueling in diesel engine which can be manufactured from biomass. The author utilized ethanol into diesel engine in a dual fuel mode up to 80%. Author concluded that although 80% blending of ethanol is possible for blending, the increase in the ethanol content increased the ignition delay and decreased the thermal efficiency. This also resulted in misfire for the blend containing higher volume of ethanol (higher than 30%) The limitations in using biodiesel [2] as fuel was stated by the author and recommended for low volume of biodiesel (up to 20%) along with diesel in diesel engine. The author also stated that there was a significant decrease in power by the utilization of biodiesel into CI engine. This tuned the researchers to the focus on fueling the CI engines with diesohol instead of biodiesel-diesel blends. The utilization of methanol is not found attractive as this is meant as poisonous. This paved a way to utilize diesel-ethanol to fuel CI engine. Ethanol was started its attempt as fuel for CI engines from 1980s onwards. Previous researches with respect [3] to diesel-methanol, diesel-ethanol and diesel-ethanol on the solubility and performance was compiled by the author. **Table 1** shows the standard properties of diesel, ethanol and butanol [4]. From the table the research octane number of ethanol is very much higher, this will lead to higher rate of combustion and hence, ethanol has been chosen.

The author concluded that further research on the utilization of higher volume of ethyl alcohol in ethanol-diesel and higher volume of butanol in butanol-diesel blends are further progressed in low temperature analysis. Speed of the flame in incylinder [4] of CI engine using ethanol-diesel (containing 5% ethanol) was compared with diesel. The author stated that the flame spread speed was found decreased monotonously at the initial phase and remains unchanged after certain height of ullage. This can be attributed to efficiency in combustion and the oxidizing rate when fueled with ethanol-diesel blends. A study was conducted by fueling 20% of ethanol along with Jatropha methyl ester and diesel blends on the evaporation characteristics when fueled in CI engine. It was reported that the liquid penetration of the ethanol blended biodiesel-diesel blends and the vapor penetration were found matching with that of diesel. The improvement in the evaporation rate of the fuel blend was due to the higher heat of vaporization of ethanol in the blend and the higher boiling point of the biodiesel in the blend. Diesohol containing up to 19% of ethyl alcohol was studied [5] for the essential properties such as cetane number, calorific value and flash point. The author reported that the properties are found to be closer with respect to that of diesel fuel and suitable to fuel CI engine. The author also studied the characteristics of CI engine when fueled with this blend. The report indicated that a significant increase in BTE, decrease in the emissions

S.No.	Property	Diesel	Ethanol	Butanol
1.	Density (kg/m^3)	829	785	809
2.	Kinematic Viscosity (mm^2/s)	4.04	1.07	2.6
3.	Calorific Value (MJ/kg)	42.8	26.9	33.1
4.	Heat of Vaporization (MJ/kg)	—	0.92	0.43
5.	Flammability Limits, volume (%)	—	19	11.2
6.	Flash Point (°C)	74	13	35
7.	Cetane Number	50	8	25
8.	Research Octane Number	—	129	96
9.	Energy Density (MJ/L)	—	19.6	29

Table 1.
Properties of fuels standard.

and exhaust temperature by utilizing ethanol-diesel blend in CI engine. Cyclic irregularities of diesel-ethanol [6] and diesel- butanol blends were compared when fueled in CI engine as fuel to replace diesel. The author stated that the cyclic variations produced by diesel-ethanol blends were found to be a bit stronger compared to those produced by diesel-butanol blends. The author stated that the reason for this activity was by the oxygen possessed by ethanol. Most researchers attempted diesel-ethanol blends as fuel; however, attempts are limited for the fuel blend [7–10] possessing higher volume of ethanol and for low temperatures. Hence this experimental study considers the objective as utilizing higher volume of ethanol under low temperature up to 5°C with the assistance of n-butanol as co-solvent.

2. Materials and methods

2.1 Fuels used and preparation of blends

Diesel used in this study is Bharat Stage VI low sulfur diesel procured from market. Ethanol is procured from bioethanol producer who produces bioethanol [11] from waste vegetables cut wastes. These wastes are generally not utilized properly and thrown into garbage and causing land pollution to a greater extent. Butanol is also procured from a bulk manufacturer who produces butanol from food [12] wastages. This is the novelty in this study. To start with biobutanol has been blended in proportions (**Table 1**) ranging from 0–10% in increments of 1% and kept separately. **Table 2** lists the different proportions of diesel, ethanol and butanol.

Percentage of butanol		**Fuels in percentage by volume**									
1	D	94	89	84	79	74	69	64	59	54	49
	E	5	10	15	20	25	30	35	40	45	50
2	D	93	88	83	78	73	68	63	58	53	48
	E	5	10	15	20	25	30	35	40	45	50
3	D	92	87	82	77	72	67	62	57	52	47
	E	5	10	15	20	25	30	35	40	45	50
4	D	91	86	81	76	71	66	61	56	51	46
	E	5	10	15	20	25	30	35	40	45	50
5	D	90	85	80	75	70	65	60	55	50	45
	E	5	10	15	20	25	30	35	40	45	50
6	D	89	84	79	74	69	64	59	54	49	44
	E	5	10	15	20	25	30	35	40	45	50
7	D	88	83	78	73	68	63	58	53	48	43
	E	5	10	15	20	25	30	35	40	45	50
8	D	87	82	77	72	67	62	57	52	47	42
	E	5	10	15	20	25	30	35	40	45	50
9	D	86	81	76	71	66	61	56	51	46	41
	E	5	10	15	20	25	30	35	40	45	50

Percentage of butanol	Fuels in percentage by volume										
10	D	85	80	75	70	65	60	55	50	45	40
	E	5	10	15	20	25	30	35	40	45	50

D-Diesel, E- Bioethanol B – Biobutanol.

Table 2.
Various proportions of diesel-ethanol blends by varying butanol from 0 to 10%.

These biobutanol-diesel blends were taken in a beaker for blending of bioethanol. Bioethanol was filled in burette and slowly added into biobutanol-diesel blends in the proportions ranging from 0 to 50% in increments of 5% of bioethanol assisted with magnetic stirring. This was carefully handled such that bioethanol will not evaporate during the process of blending. The magnetic stirrer (**Figure 1**) has been operated at a speed of 1500 rpm and for a set cycle of 2 minutes.

Each blend has been stirred for three to five times and the prepared blends were kept in a temperature controlled box (**Figure 2**) for five different temperatures 5, 10, 15, 20 and 25°C. This temperature range has been chosen by considering the climatic conditions of India. In India most part of the country [13] will attain 5°C during the winter season. The fuel blend found by this study has to be suitable to fuel CI engine for most places in our country. Fuel blends after the temperature stability tests are presented in **Figure 3**. **Figure 3** show three representative samples which are kept at 5°C (**Figure 3**) for a period of 20 days.

Periodical monitoring has been performed and the statuses of the blends were recorded. This is to find out the homogeneity of the fuel blend and to ensure that there is no phase separation between diesel and alcohols.

2.2 Property test

Prepared fuel blends (100 blends) were tested for the essential properties required as per the ASTM standards and comparison made [14] with respect to the diesel fuel as base. The instruments used for the properties along with the accuracy and ASTM standards are listed in Table. **Table 3** lists the properties of 5 representative fuel blends [15] containing 15, 25, .35, 45 and 50% of ethanol in comparison to that of diesel.

Figure 1.
Magnetic stirrer used for the blend preparation.

Figure 2.
Temperature control box for storing the prepared blend in various temperatures.

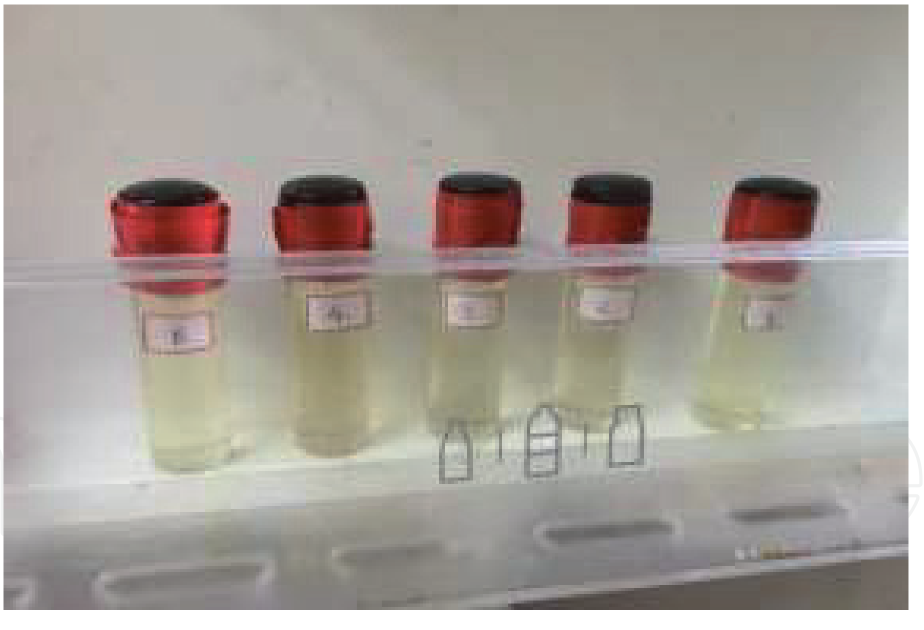

Figure 3.
Samples kept at 5°C after 20 days.

2.3 Experiment set up

A water-cooled, direct injection, Kirloskar make diesel engine [16] of 4.4 kW capacity at the rated speed 1500 rpm was used for testing the fuel blends. The engine (**Figure 4**) is coupled with eddy current dynamometer and electrical loading. Fuel flow was measured using burette and digital stop watch. Intake air flow was monitored by manometer and orifice plate. The displacement volume was 661.5CC, compression ratio was 17.5:1 and the recommended nozzle opening pressure was set at 200–205 bar. AVL pressure sensor has been used to capture the

S.No.	Property	Unit	Instrument Used	Accuracy	Percentage of Uncertainty	ASTM Standard
1.	Flash Point,	°C	Pensky-Martens Closed cup	± 0.1 °C	± 0.05%	ASTMD93-16a
2.	Kinematic Viscosity	mm^2/sec	Red wood viscometer	0.01 Centi Stokes	±0.02%	ASTMD445/446
3.	Calorific Value	kJ/kg	Bomb Calorimeter	1 J/grams.	±0.1%	ASTMD4868
3.	Cetane Number	No Unit	Ignition delay	± 0.1	±0.07%	ASTMD976/ ASTMD4737

Table 3.
List of instrument used for property testing.

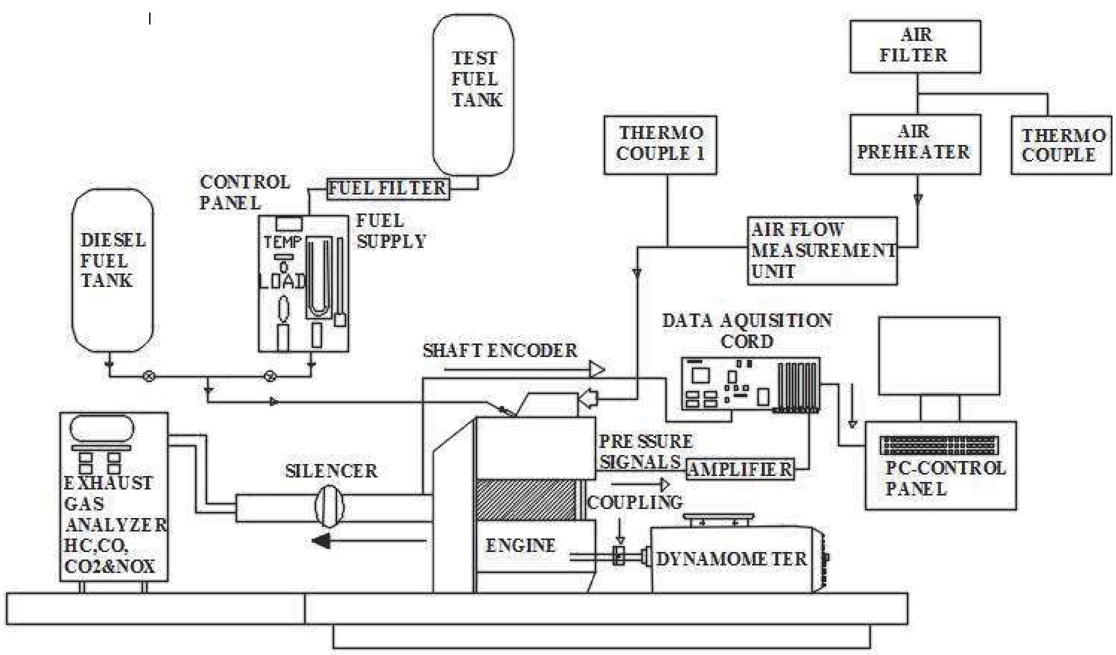

Figure 4.
Schematic layout of experimental set up.

pressure during the cylinder operation and to feed the captured signals to the data acquisition device. The fuel system of the experimental set up was mechanically controlled type and this was periodically cleaned and calibrated as per the recommendations of the manufacturers. Air preheater for heating the incoming air was used in the suction side. A heater of coil type of 1.0kVA capacity has been deployed heating the incoming air. The input electrical supply has been varied by a power regulator installed with the heater to obtain temperature difference. The input and out condition of the air has been measured by two separate thermocouple enabled with electronic readout. The injector used for fuel injection is a jet injector and is of mechanical type with a proper calibration. The average nozzle opening pressure has been set at 200 bar. To vary the nozzle opening pressure a washer of 0.20 mm has bee deployed in the vicinity of nozzle and the spring in the injector. The nozzle opening pressure has been verified with a dial type calibrating gauge for ensuring the pressure. The statndard injection timing of the injectore has been set as 23 deg. before top dead centre. To obtain the varied injection timing a washer has been added which is of 0.25 mm thickness. This was provided in vicinity of the engine and the fuel feed pump. This washer has been procured from the manufacturer and deployed as per the guidelines dictated by the supplier. Data acquisition system used for the present study consists of a computer, programmed with AVL 621 IndiModul system, which is receiving the signals amplified by a charge amplifier from a water cooled pressure transducer of KISTLER piezo electric transducer.

This system was controlled by IndiCom software. Specifications of the pressure transducer are given in Appendix 4. This device was programmed for generating the combustion data according to the pressure input. The encoder captures the position of the crank angle of the respective pressure signal and was duly connected to the engine. Specifications of the encoder are tabulated. 200 cycles of pressure data were captured and recorded for the analysis of combustion characteristics in the data acquisition system. This combustion parameters calculations were performed from the input received from the pressure transducer, crank angle encoder and intake air measurement. This also receives the input from the thermocouples for the temperature of the intake air, exhaust gases and incylinder. AVL-444 Di-Gas analyzer is used in this study for capturing the emissions from the test engine fuelled by the blends during the experiment. This measures CO, HC, NOx, and CO2 and oxygen concentration. It uses non-dispersive infrared (NDIR) sensor for measuring CO, $CO_2$and HC. Also, it measures NOx and oxygen concentrations by electrochemical sensors. All the emissions are recorded and converted to g/kWh for further analysis. This device is auto calibrated periodically as per the manufacturer advice. The measured values from the exhaust gas analyzer are in ppm [17] and the following conversion equations depict the onversion of ppm to g/kWhr which are standard equations (assuming 5% residual oxygen).

1000 ppm of NOx corresponds to 6.60 g/kWh.

100 ppm of HC corresponds to 0.20 g/kWh.

100 ppm of CO corresponds to 0.36 g/kWh.

2.4 Experimental uncertainty

Any experiment has its own uncertainty and the overall uncertainty has been arrived from individual uncertainties of the various instruments used. In the present study various instruments have been used and each one has different level of uncertainty. Hence, a detailed uncertainty analysis was carried out by the method of [18]. The total was arrived as ±1.3%. The uncertainty in any measured parameter was estimated based on Gaussian distribution method with confidence limits of ±2σ (95% of measured data lie within the limits of 2σ of mean). Thus the uncertainty (Eq. (1)) was estimated using the following equation:

Uncertainty of any measured parameter

$$(\Delta x_i) = \left(2\sigma_i/\overline{X_i}\right) * 100 \tag{1}$$

From the uncertainties of the measured parameters, the uncertainties in computed parameters are evaluated by using an expression, which is derived as follows. If an estimated quantity, R depends on independent variable like (x_1, x_2, x_3 … … . x_n) then the error in the value of “R” is given by Eq. (2).

$$\Delta R = \left[\left(\frac{\partial R}{\partial x_1}\Delta x_1\right)^2 + \left(\frac{\partial R}{\partial x_2}\Delta x_2\right)^2 + \dots \dots \dots \dots \dots \dots + \left(\frac{\partial R}{\partial x_n}\Delta x_n\right)^2\right]^{1/2} \tag{2}$$

The estimated uncertainty values at a typical operating condition are given below:

Speed: ± 0.12% Load: ± 0.49%
Mass flow rate of air: ± 0.62% Mass flow rate of diesel: ± 0.87%
Brake power: ± 0.25% Brake thermal efficiency:± 0.27%
NO_X: ± 1.1% Hydrocarbon: ± 0.01%
CO: ± 0.8% Smoke: ± 1.3%

There are various methods available to reduce the errors observed in the instruments such as selecting the instruments according to the measurement level required (range of measurement), Accuracy of the instrument used, sensitivity etc., this experiment was conducted by deploying the appropriate instruments within the range of measurement, accuracy and sensitivity requirement.

3. Results and discussion

From **Figure 5** it is seen that the two blends D75E15B10 and D65E25B10 offers higher incylinder pressure in comparison to that of diesel. This can be attributed to the improvement in the physicochemical properties of the blends up to a certain extent of ethanol into diesel. This is due to the improved complete combustion of the blends by the addition of ethanol till 25% by volume (**Figure 6**).

However, fuel blends D55E35B10 and D45E45B10 produce lesser in cylinder pressure compared to diesel [18]. This is due to the suppression of combustion by the higher volume of ethanol in the blends, which is due to higher heat of vaporization. It can also be observed from figure that the peak pressure from D75E15B10 and D65E25B10 are 6.4% and 15.2% higher than diesel. **Figure 7** shows the variation of incylinder peak pressure versus brake power for the blends. It is seen that the addition of ethanol (up to a volume of 25%) into diesel increases the incylinder peak pressure. Significantly. Also, the increase in the incylinder peak pressure is found proportional to the increase in brake power. This is due to the improvement in the physico-chemical properties of the blends by the addition of ethanol. The improvement in kinematic viscosity [19], density results in better atomization which leads

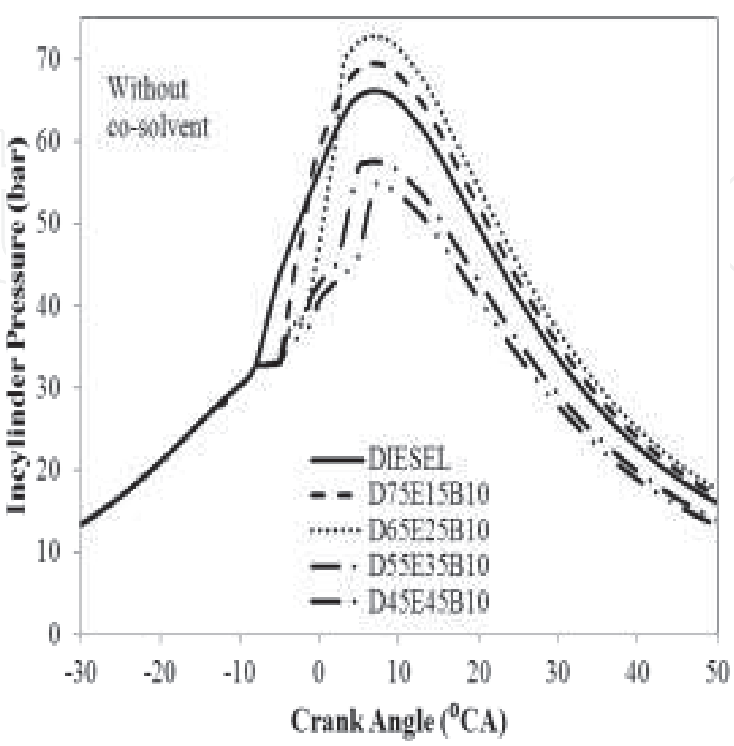

Figure 5.
Variation of Incylinder pressure versus crank angle.

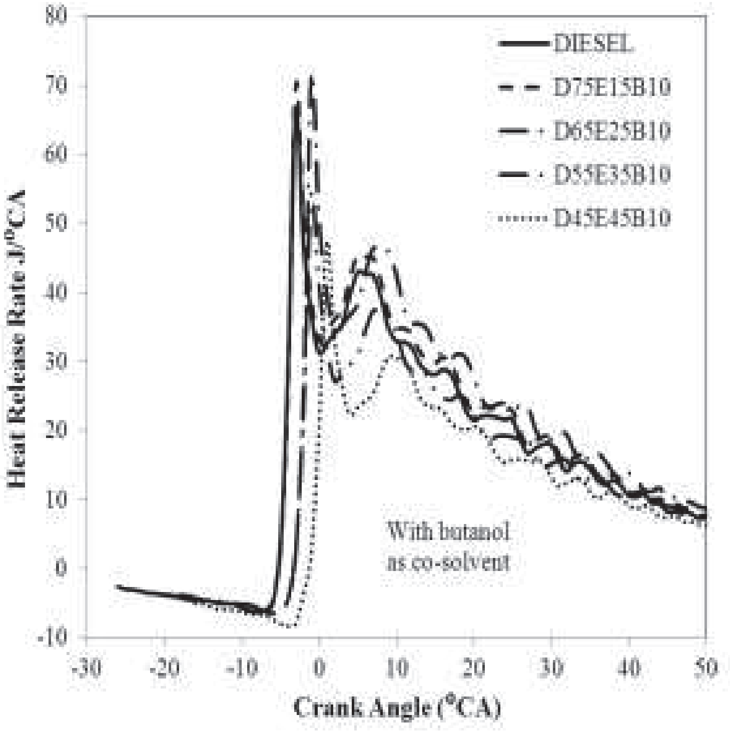

Figure 6.
Variation of HRR of fuel blends with crank angle at rated power.

to the more complete combustion. It is also seen that the addition of ethanol into diesel higher than 25% by vol. reduces the incylinder peak pressure significantly. This is due to the dominance of heat of vaporization of the blends with the increase in the volume of ethanol in the blend. This produces a cooling effect which results in poor atomization and lesser rate of oxidation which results in lesser incylinder peak pressure. The increases in incylinder peak pressure of D75E15B10 and D65E25B10 significantly. Also, the increase in the incylinder peak pressure is found proportional to the increase in brake power. This is due to the improvement in the physico-chemical properties of the blends by the addition of ethanol.

Heat release rate is an indicator of combustion efficiency and these parameters is helping for explaining the BTE, EGT, increase in the incylinder pressure, emissions from the engine and the pressure during the engine operation. **Figure 8** depicts the heat release rate diagrams generated during the engine operation fueled with the blends with and without butanol. The representation of a diagram generated at rated power condition is presented. HRR graphs are generated at all loads and for representation HRR at full load condition is presented. It can be seen from figure that ethanol addition up to 25% increases the HRR to a greater extent due to the enhanced combustion behavior resulted from better atomization. However, the increase in ethanol content beyond 25% decreases HRR of the blends as poor atomization resulting in lesser heat release rate (**Figure 9**). It is also seen that the volume of ethanol are directly having impact in this parameter. Increase in the volume of ethanol decreases the heat release rate. Longer crank angle has been necessary for the blend to start up the heat release rate compared to diesel. Also, these blends produced low combustion duration compared to diesel [20, 21]. It can be observed from the **Figure 10** that two blends D75E15B10 (22 Wang *et al*. 2018) offers a significant increase in BTE with respect to diesel under all brake power conditions. This can be attributed to the increase rate of spray characteristics which is a result of increase in the volatility of the fuel. This is the main reason for the increase in BTE. Beyond 25% of ethanol a decrease in heat release rate has been observed (**Figure 11**). This is due to the decrease in the self-ignition property which

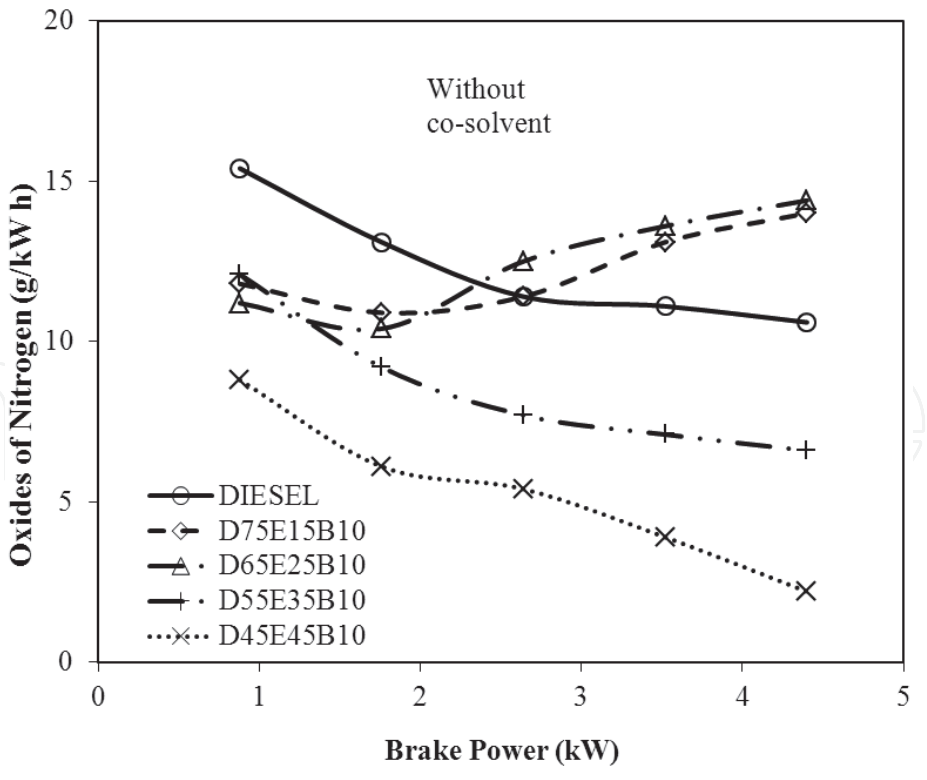

Figure 7.
Variation of NOx emissions of fuel blends versus brake power.

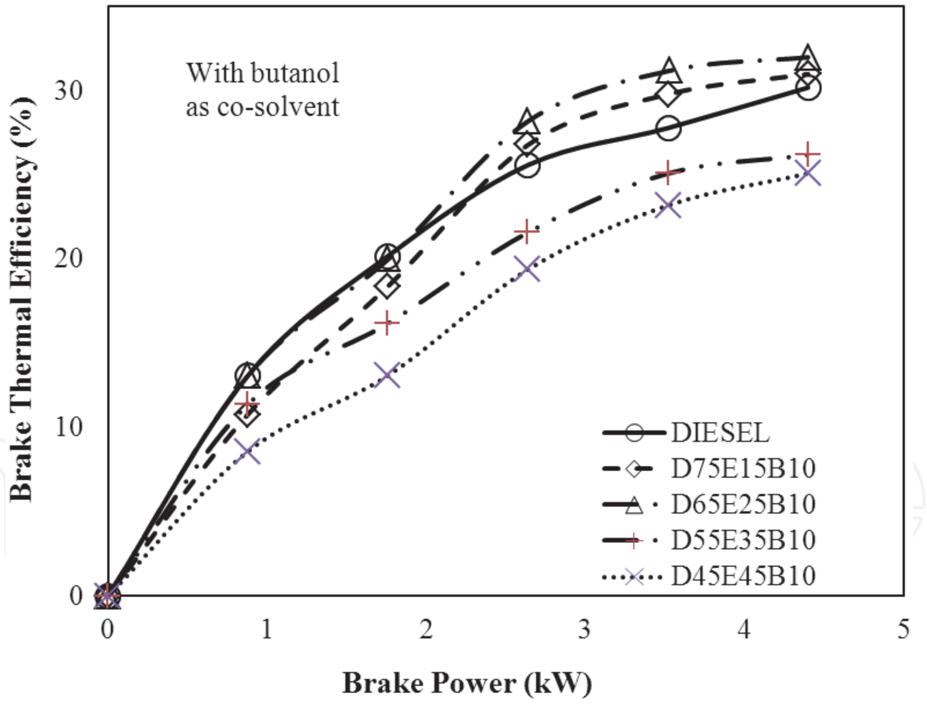

Figure 8.
Variation of brake thermal efficiency versus brake power.

is suppressed by the higher volume of ethanol. Variations of EGT versus brake power for the blends are as shown in **Figure 12**. It is seen that the EGT of the blends containing combustion characteristics by the dominance of heat of vaporization of the final blends. This produces a cooling effect in the incylinder which reduces the rate of reaction of fuel particles with oxygen available and results in lesser BTE and lesser EGT. This is the main reason for the reduction of EGT of D55E35B10 and D45E45B10 [22]. The increases of EGT for the blends D75E15B10 and D65E25B10

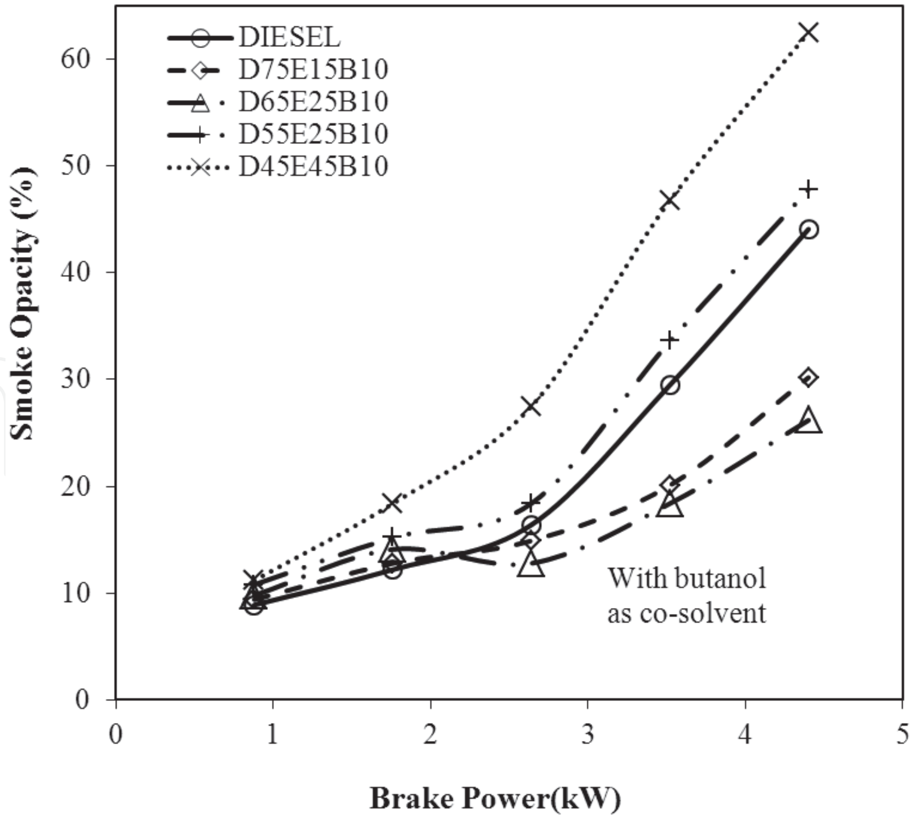

Figure 9.
Variation of smoke emissions versus brake power.

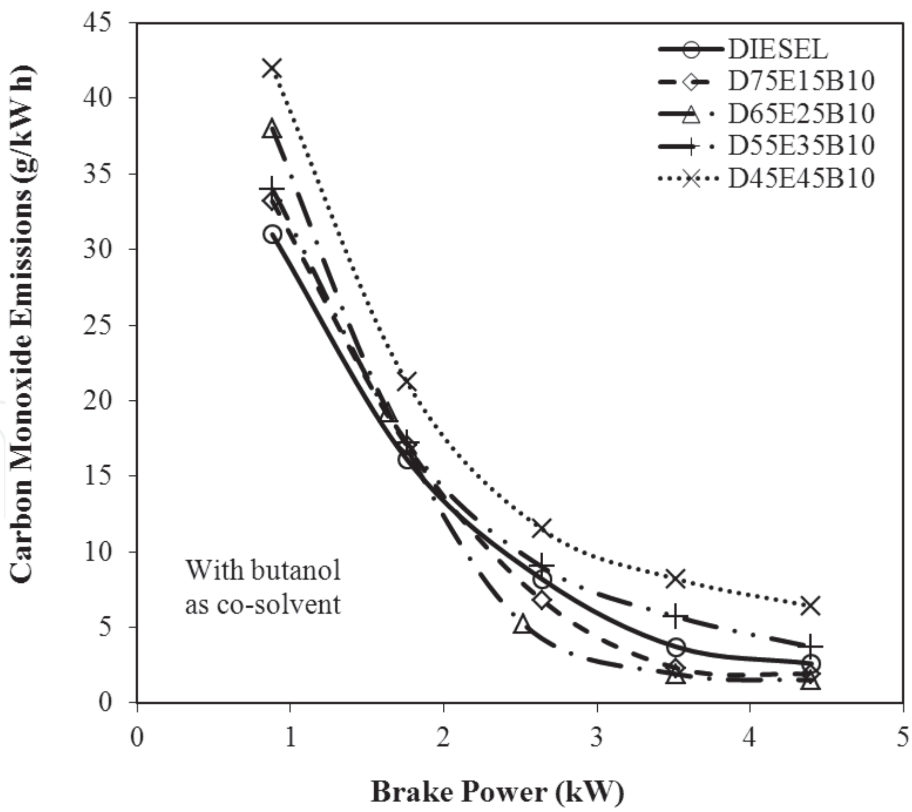

Figure 10.
Variation of CO emissions versus brake power.

are 16.9% and 22.6% respectively in comparison to diesel at rated power. The decrease of EGT for the blends D55E35B10 and D45E45B10 are 13.6% and 20.4% respectively in comparison to diesel at rated power. From **Figure 13** it is observed

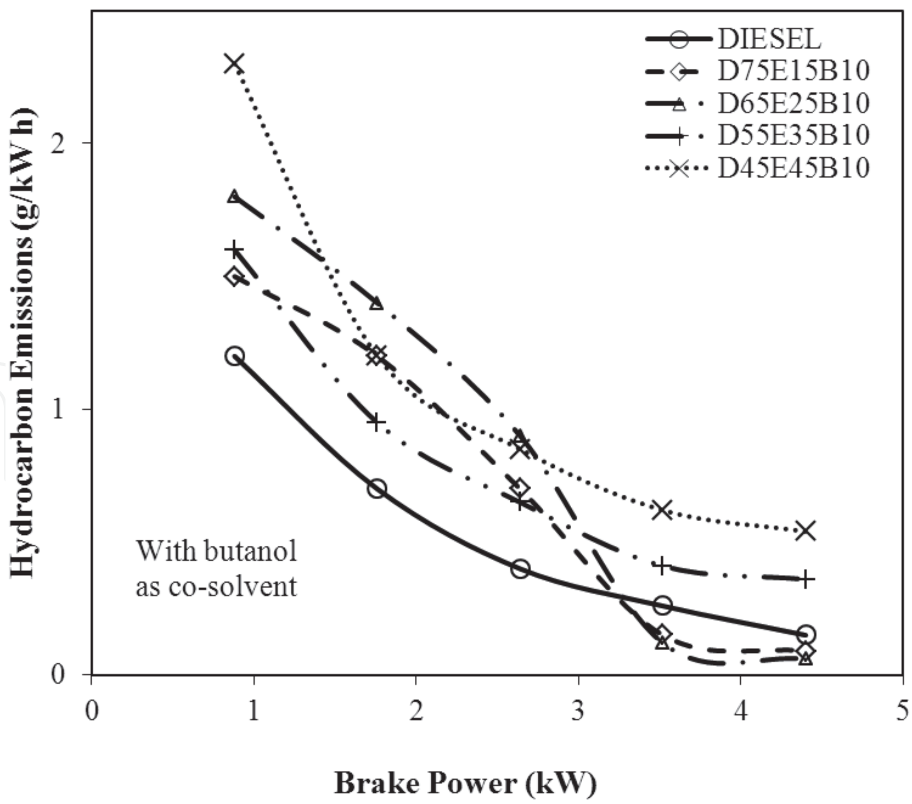

Figure 11.
Variations of HC emissions of fuel blends.

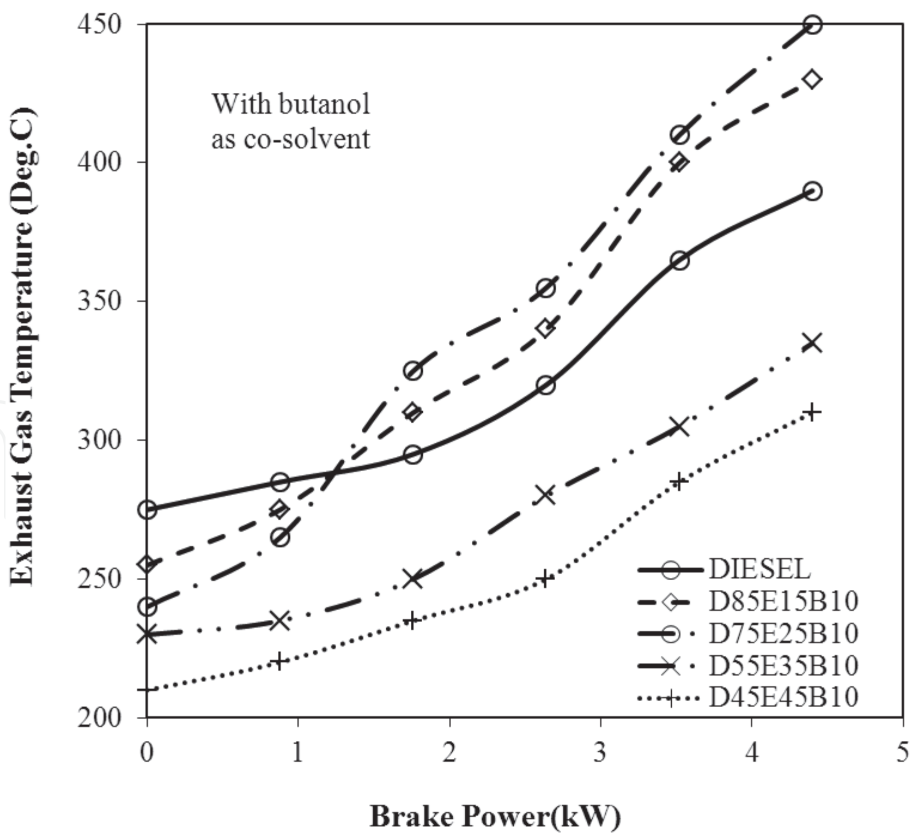

Figure 12.
Variations of exhaust gas temperature versus brake power.

that the blends containing lower ethanol content (lower than 25%) producing higher oxides of nitrogen and higher volume of ethanol content (higher than 25%) produce low oxides of nitrogen with respect to diesel. The ethanol addition

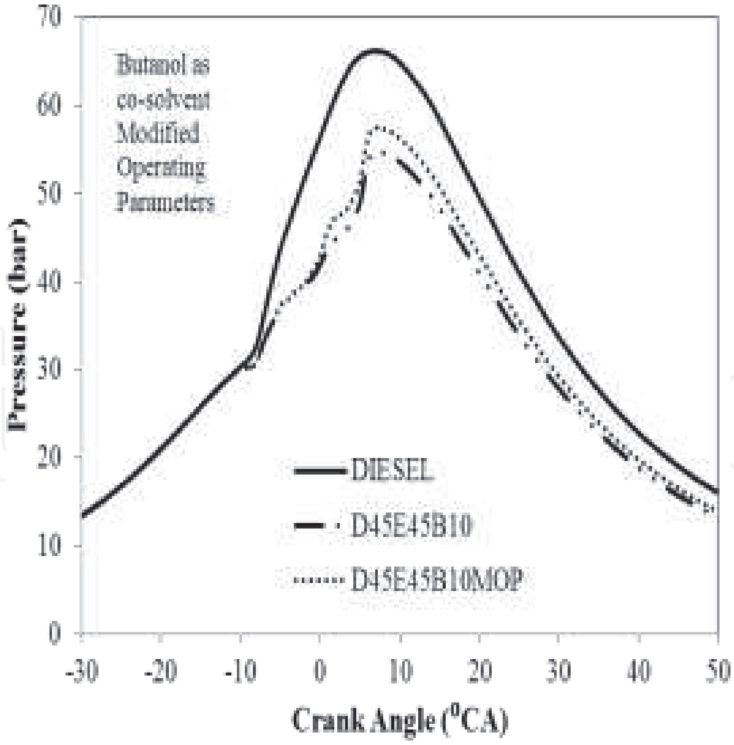

Figure 13.
Variation of cylinder pressure with crank angle at rated power.

improves the physico chemical properties such ease of evaporation, rate of atomization and a decrease in viscosity [23].

Figure 13 depicts the emissions of smoke for the engine fueled with various proportions of ethanol. Out of the fule blends the blend containing 25% ethanol produces the lowest smoke emission. This can be attributed to the enhanced physicochemical properties of the fuel blend up to this volume of ethanol. This increases the rate of combustion and resulted in a significant decrease in smoke emissions. Beyond this volume the dominance of the heat of vaporization suppresses the incylinder temperature and results in a decrease of oxidation rate and thereby higher smoke emissions. The present work utilizes various proportions of ethanol into diesel as fuel. **Figure 14** indicates that higher volume of ethanol produces higher smoke emissions and lower volume of ethanol (lower than 25%) produces lower smoke emissions. This can be attributed to increase in the rate of combustion up to 25% of ethanol and suppression of temperature above 25% of ethanol content in the blend The decreases of CO emissions from D75E15B10 and D65E25B10 are found as 27% and 46.1% respectively at rated power in comparison with diesel. The increases CO emissions from D55E35B10 and D45E45B10 are found as 37.2% and 58.2% respectively at rated power in comparison with diesel. From the **Figure 15** it can be seen that containing 15% of ethanol offers lower HC emissions compared to diesel. This is mainly due to the increase in combustion characteristics of the blends containing lower volume of ethanol and decrease in combustion characteristics of the blends containing higher volume of ethanol. From the **Figure 15**, it can be observed that D75E15B10 and D65E25B10 produce 28% and 7.6% lesser than diesel. The blend containing 45% of ethanol offers higher HC emissions and the blends. However, D55E35B10 and D45E45B10 produce significantly higher emissions compared to diesel at load conditions. However, D55E35B10 and D45E45B10 offer 8.2% and 12.6% lesser in cylinder pressure compared to diesel at full load. Also it can be observed that start of pressure rise of all fuel blends are away from that of diesel. This is due to lesser cetane number of the final blend compared to diesel. The previous phase of the present study indicated that D45E45B10 blend is containing

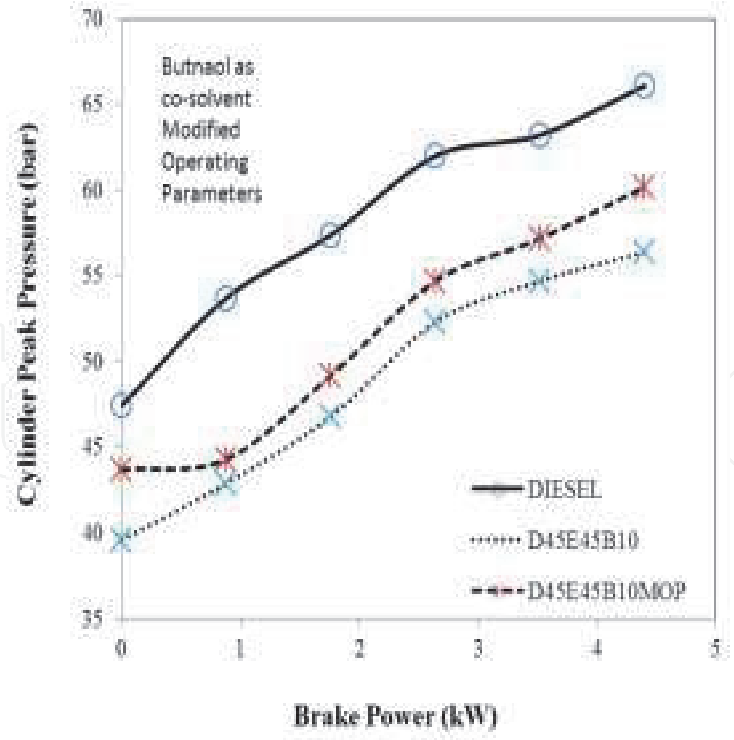

Figure 14.
Variation of Incylinder peak pressure versus brake power.

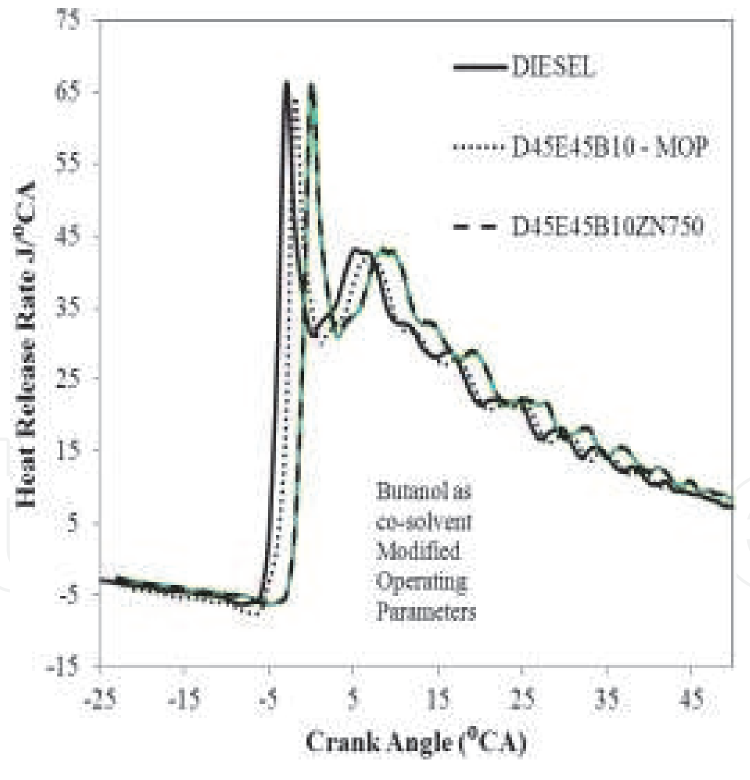

Figure 15.
Variation of HRR with crank angle at rated power.

higher possible volume of ethanol has failed to produce better performance and emission characteristics. Hence, it is decided to modify the parameters of the test engine to improve the performance and emission characteristics of the fuel blend. Also in the first phase this blend has not suffered phase separation which is the major limitation of utilizing ethanol diesel blends in CI engine up to a temperature of 5°C. The suitable parameters for fuelling CI engine by D45E45B10 have been determined by Taguchi method on ANOM approach (Analysis of mean) (**Table 4**).

Blend	Flash Point	Energy Content	Density	Kinematic Viscosity	Oxygen content	Cetane Number
Units	**^{0}C**	**MJ/l**	**kg/ m^3**	**mm^2 /s**	**wt..%**	
Diesel	74	42.8	829	4.04	0	50
Ethanol	13	26.9	790	1.37	34.8	8
n-butanol	35	33.1	809	3.2	21.58	25
D75E15B10	64	40.24	823	3.7	5.64	43.3
D65E25B10	57.9	38.65	818	3.45	9.12	39.1
D55E35B10	51.8	37.06	813	3.19	12.59	34.9
D45E45B10	47.5	37.13	807	2.94	17.16	30.7
D40E50B10	39.6	33.88	805	2.62	19.6	26.5

Table 4.
Properties of diesel-ethanol-butanol blends.

This part of the work used Taguchi method for designing experimental layout and rank matrix to attain optimum level of parameters.

The steps involved in the optimization process are:

- Selection of operating parameters and their levels
- Selection of Orthogonal array by Taguchi method
- Preparation of experimental layout
- Conducting the experiments using the experimental layout
- Observation of response parameters
- Listing the results and formation of Rank matrix
- Suggesting optimal level of parameters
- Conducting engine experiment using optimal parameters

Present investigation has considered four operating parameters viz. injection pressure (IP), injection timing (IT), compression ratio (CR) and intake air temperature (IAT) for optimization. The range and level of parameters are decided with literature support and preliminary engine experiments. **Table 5** shows the level of operating parameters.

S.No.	Symbol	Parameters	Level 1	Level 2	Level 3
1	A	Injection Pressure (IP) (bar)	190	200	210
2	B	Injection Timing (IT) (^{0}BTDC)	26	29	32
3	C	Compression Ratio (CR)	17.5	19	21
4	D	Intake Air Temperature(IAT) (°C)	50	75	100

Table 5.
Parameters involved in the optimization and their levels.

Using this parameters and their levels a suitable orthogonal array, experimental layout and number trials of the experiments have been arrived from Taguchi method of optimization.

Taguchi method of optimization offers a systematic approach to arrive at the level of performance parameters involved in the response parameters. The Taguchi method uses an orthogonal array for designing the experimental layout. The selection of orthogonal array is arrived from the degrees of freedom of the parameters involved. The minimum number of experiments (trials) for selecting the optimum level of parameters can be determined using the relation:

$$N = (L - 1) * P + 1 \tag{3}$$

Where, N = Total number of test runs, L = Number of levels of parameters and P = Number of control parameters.

The present study uses (**Table 5**) four parameters and three levels and hence, the total degrees of freedom of control parameters are 8. Therefore, L_9 is suitable OA for the total degrees of freedom of involved parameters.

Analysis of Mean (ANOM) This is used after attaining the experimental results as per the L_9 orthogonal array of nine experiments containing 3 sets of reading in each setting. A rank matrix table is utilized for the analysis of captured data (**Table 6**). A rank matrix **Table 7** has been constructed to arrive at the optimal level of parameters. Average of the sum of the each level outcome has been obtained and the rank is tabulated for the maximum of the outcome. Assuming that Y as output parameter and the level summation has been obtained as:

Trial No.	Column No.			
	A	B	C	D
1.	1	1	1	1
2.	1	2	2	2
3.	1	3	3	3
4.	2	1	2	3
5.	2	2	3	1
6.	2	3	1	2
7.	3	1	3	2
8.	3	2	1	3
9.	3	3	2	1

Table 6.
L_9 orthogonal array.

Rank	A_1 (Level 1)	B_2 (Level 3)	C_3 (Level 3)	D_4 (Level 3)
Level/Parameter	**A**	**B**	**C**	**D**
1	31.3	31.6	30.6	30
2	31.05	29.86	31.36	31.6
3	31.1	31.96	31.66	31.7

Table 7.
Rank Matrix (for BTE).

$$A_1 = Y_1 + Y_2 + Y_3 \text{ (in which the level 1 is denoted in the orthogonal array)} \tag{4}$$

Similar calculation has been done for three levels and for four parameters, from which the rank matrix table has been constructed:

From the **Table 7** it can be concluded that IP 190 bar (LEVEL1), IT 29^0bTDC (LEVEL3), CR 19 (LEVEL3) and IAT 100 (LEVEL3) are the optimal parameters by comparing the rank. The same sets of readings are captured for NOx to match with the brake thermal efficiency. The optimized levels of operating parameters are as shown in **Table 7**. Blend D45E45B10 have been tested under the modified operating parameters and the results are compared with diesel and D45E45B10 under normal operating parameters. The same engine has been used for the testing of the blends under modified operating parameters. The results of the experiment are presented in graphical form. The variation of cylinder pressure with crank angle at rated power for the blend D45E45B10 under standard operating parameters and modified operating parameters are presented in **Figure 13**, it is seen that the modified engine operating parameters increased the cylinder pressure significantly compared to diesel. This is due to the increased heat energy release in the combustion chamber with increase in compression ratio and intake air temperature. Also, the advancement in the injection timing improves the precombustion phase and results in more complete combustion. This shows the suitability of the modified engine operating parameters for the blend D45E45B10. The increase in pressure of D45E45B10MOP is found as 7.1% higher than diesel at rated power. However, the cylinder pressure is found lesser than diesel. This is due to the lesser essential properties of D45E45B10 in comparison to diesel. Variation of incylinder peak pressure versus brake power for D45E45B10 under modified engine operating parameters is shown in **Figure 14**. It is seen that the incylinder peak pressure increases by fuelling D45E45B10 under modified operating parameters compared to that of normal operating parameters. This is due to the suitability of the modified operating parameters for the blend D45E45B10. Also, the increase in the incylinder peak pressure is found proportional to the increase in brake power. This increase is due to the improved rate of combustion by the increase in compression ratio and intake air temperature. Also, the advancement of injection timing improved the precombustion phase which suppresses the dominance of heat of vaporization of the blend. However, the incylinder peak pressure of D45E45B10MOP is found lesser than diesel at all load conditions. This is due to the lesser energy content of D45E45B10 in comparison to diesel. The increase in the incylinder peak pressure of D45E45B10MOP is found as 6.3% higher than D45E45B10. Heat release rate is an indicator of combustion efficiency and these parameters is helping for explaining the BTE, exhaust gas temperature, rate of pressure rise, emission parameters and cylinder pressure.

Figure 15 shows the It can be seen from figure that ethanol addition up to 25% increases the HRR to a greater extent due to the enhanced combustion behavior resulted from better atomization. However, the increase in ethanol content beyond 25% decreases HRR of the blends as poor atomization resulting in lesser heat release rate.

From the **Figure 16** it is observed that the target blend D45E45B10 offers higher BTE with modified operating parameters compared to that of BTE with normal operating parameters. However, this blend offers lesser BTE compared to that of diesel (**Figure 17**). The reason for the increase in BTE is due to the increase in heat content of the combustion chamber resulted from the enhanced combustion triggered by the modified operating parameters. Ignition quality, which decreases the combustion temperature and thereby lesser BTE compared to diesel. Similar

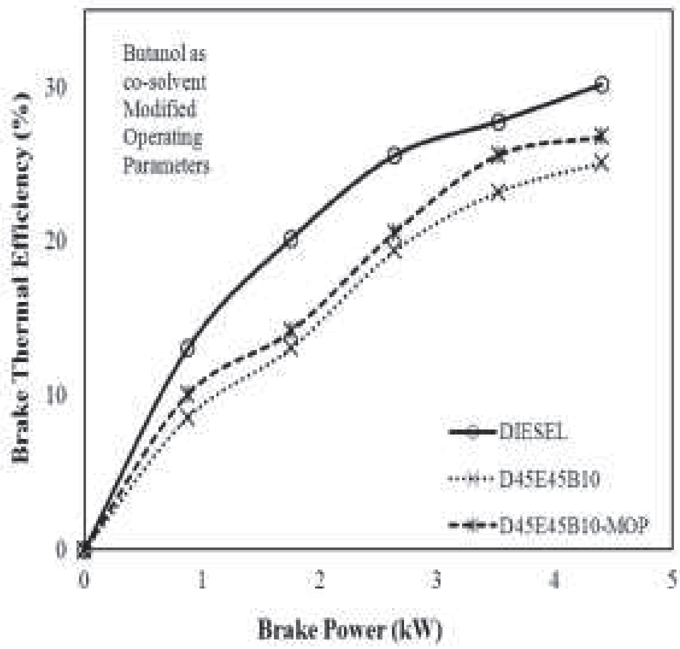

Figure 16.
Variation of brake thermal efficiency versus brake power.

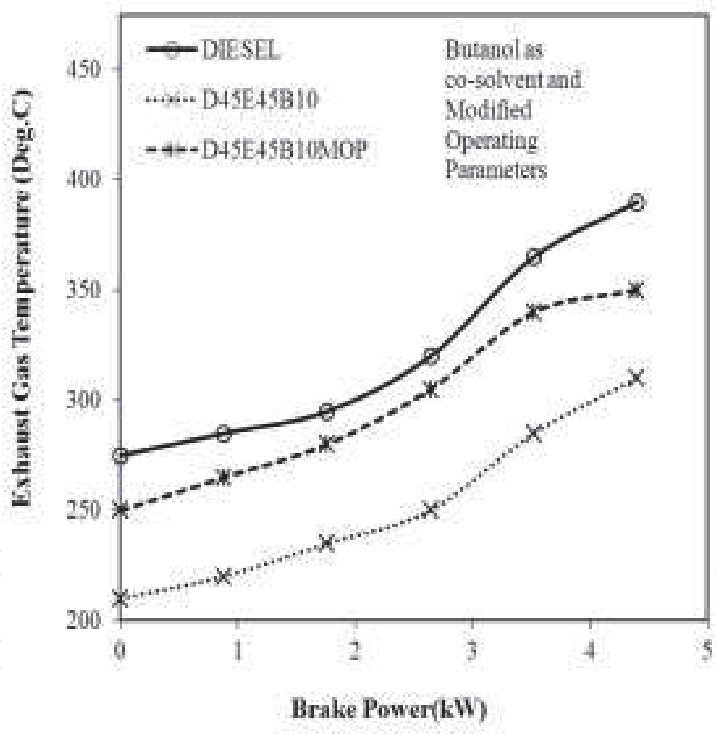

Figure 17.
Variation of EGT versus brake power.

observation was presented by previous researchers [24]. The increase in BTE by the modification of operating parameters is 6.7% compared to those in normal operating parameters, which indicates the suitability of the parameters for the target blend. The decrease in BTE of the target blend at modified operating parameters is only 2.1% compared to diesel. Variation of EGT with respect to brake power is as shown in **Figure 18**. The quantity of ethanol in the blend determines the performance of the blend as the increase in ethanol volume results in poor to brake power for the blend D45E45B10 operated under normal operating parameters and modified operating parameters in comparison to diesel. It is seen that there is a

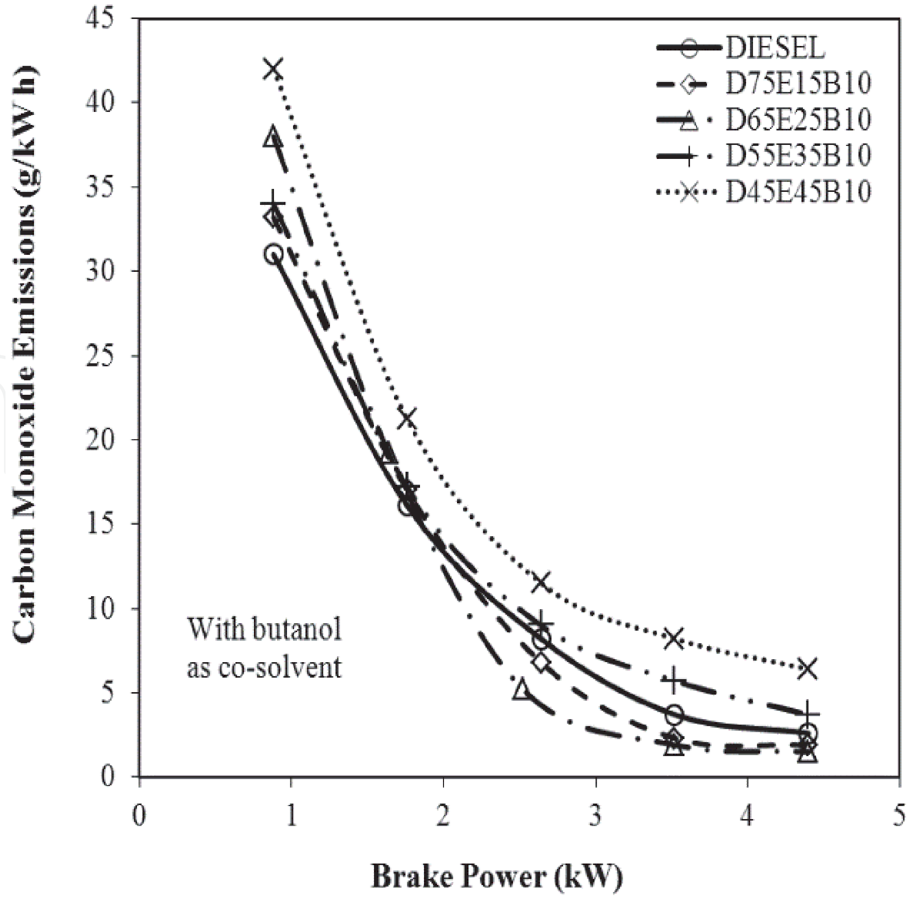

Figure 18.
Variations of CO emissions versus brake power.

significant increase in EGT of D45E45B10MOP in all load conditions compared those under normal operating parameters. This is due to the higher heat energy release by the blend operated under modified operating parameters. This is due to the suppression of the dominance created by the heat of vaporization of the higher volume of ethanol by the modified parameters to a certain extent. However, the EGT of D45E45B10MOP is found lesser than diesel. The increase of EGT of D45E45B10MOP is found 13.1% higher than D45E45B10 at rated power.

Any engine producing higher emissions of oxides of nitrogen is an indication of higher temperature of the in cylinder which is the result of complete combustion. **Figure 19** shows the emissions of oxides of nitrogen from the engine fueled using D45E45B10 with and without modification of parameters along with diesel for comparison. It can be observed that there is an increase in NOx emissions from the target blend when fuelled in CI engine which shows the suitability of the modified parameters. This is due to the increase in heat content of the target blend operating with modified operating parameter and compressed air, which helps to combust the fuel by reducing the ignition delay. However, the emissions of NOx are lesser than diesel as the higher volume of ethanol suppresses the temperature of the in cylinder. The increase in NOx emissions due to the modification of operating parameters is 100% (approximately double) compared to that of operating under normal operating parameters. The decrease in NOx emissions of D45E45B10 –MOP is 40.5% compared to that of diesel at full load condition. **Figure 20** shows the smoke opacity of the target blend under modified operating parameters at all load conditions. It can be observed that there is a significant reduction in smoke emissions from the target blend under modified operating parameters compared to that under normal operating parameters. This is due to the reason of increased temperature of the in

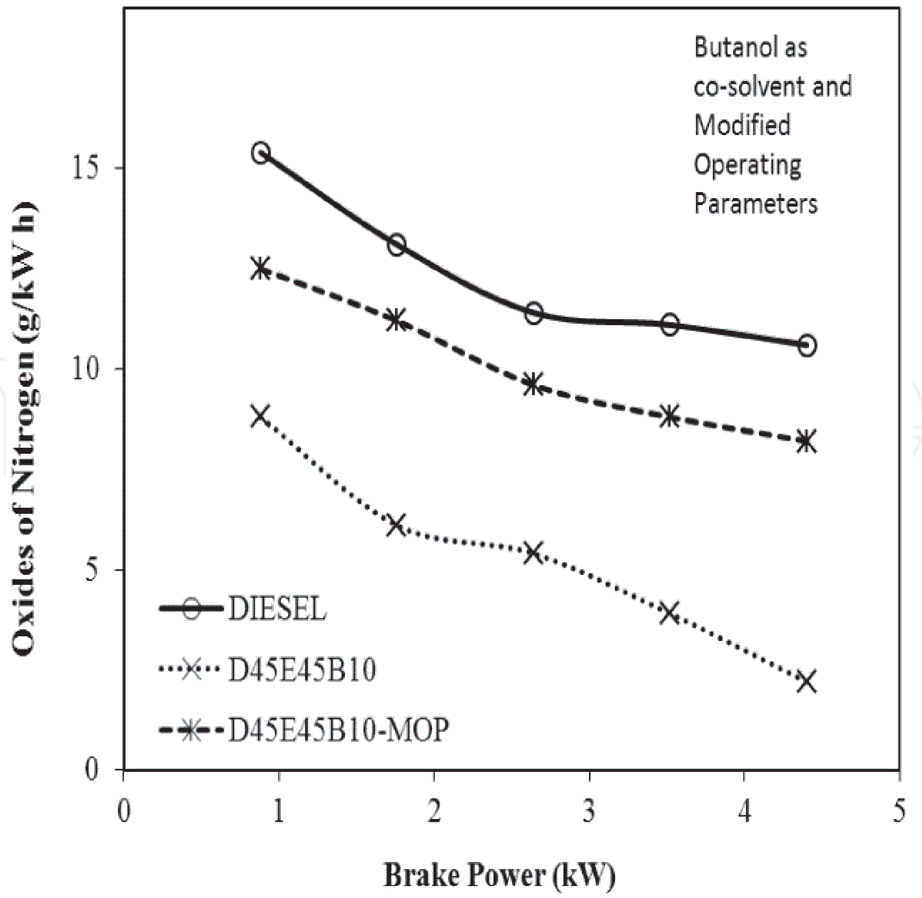

Figure 19.
Variation of NOx emissions versus brake power.

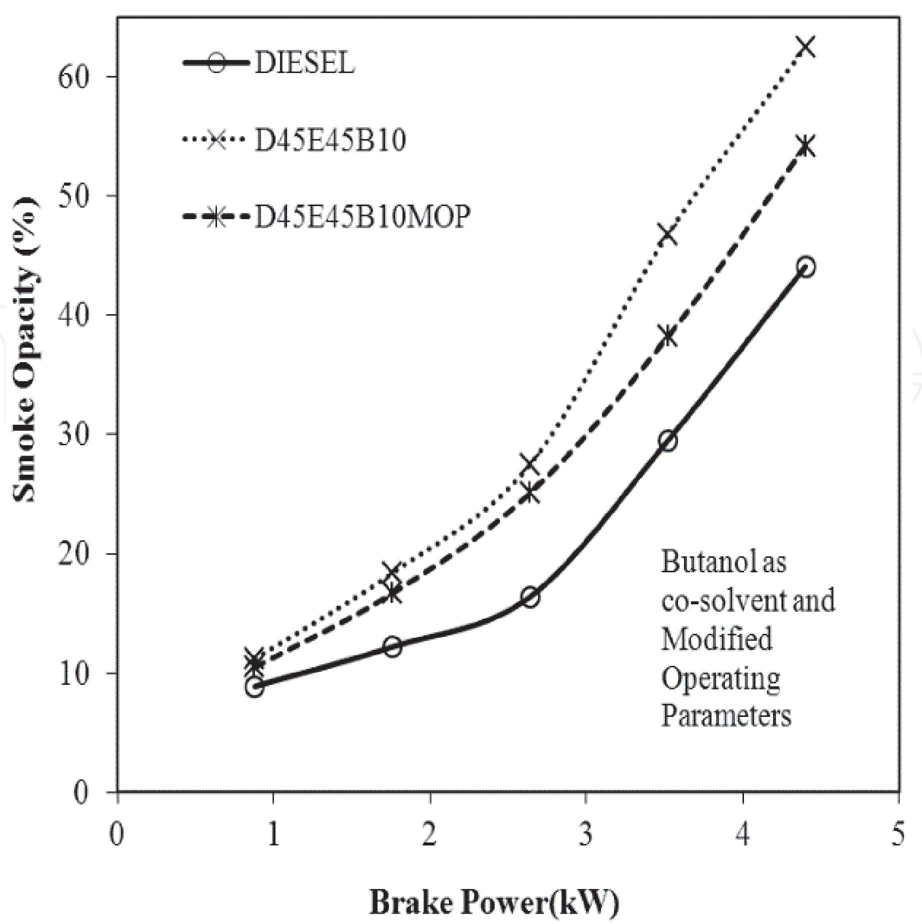

Figure 20.
Variation of smoke opacity versus brake power.

cylinder by the modified operating parameters which enhances higher heat release resulted from compressed air. However, the higher heat of vaporization of the blend still suppresses the temperature and hence there is an increase in smoke emissions compared to that of diesel. The decrease in smoke emissions is 21.2% compared to D45E45B10 operated under normal operating parameters. The increase in smoke emissions of D45E45B10-MOP is 16.5% higher than diesel at full load condition. Similar results were observed by previous researchers [25]. From the **Figure 18** it can be seen that there is a significant reduction of CO emissions due to the modification of operating parameters to the target blend. This is due to impact of the modified parameters on the combustion characteristics to a certain extent. However, the higher ethanol content increases the heat of vaporization of the final blend, which results in poor ignition quality which results in lesser temperature of the in cylinder shows the variation of CO emissions of D45E45B10 fuelled in the test engine under modified operating parameters compared to that of diesel. This reduces the BTE of the blend lesser than diesel.

The increase in BTE of the blend at modified operating parameters is 29.6% compared to that operated under normal operating parameters. However, the increase in CO of the blend is 19.3% higher than diesel. Higher ethanol content affects the self-ignition property; hence it reduces reaction rate, combustion temperature and heat release rate [26].

4. Conclusion

Different phases of study have been followed to utilize diesel ethanol blends as fuel in compression ignition (CI) engine in this study. Experiments were conducted with diesel ethanol without co- solvent and with butanol as co-solvent. The effects of engine operating parameters such as Injection Pressure (IP), Injection Timing (IT), Compression Ratio (CR) and Intake Air Temperature (IAT) on engine performance, combustion and emission were studied.

- Solubility test indicates that ethanol can be blended with diesel up to a volume of 50% with 10% butanol as co-solvent. This blend is found as stable up to a lower temperature of 5°C for 20 days.

- Property testing show that properties of the blend containing 45%of ethanol and 10% butanol as co-solvent is found suitable for replacing diesel to fuel CI engine.

- However, blend containing 50% ethanol and 10% butanol is found not suitable as the cetane number is less than 30 which is a minimum requirement as per ASTM standards.

- Improved physicochemical properties, Better ignition quality, higher combustion temperature and higher oxygen content increase the NO_x emission by 13.2% in the case of D80E20 whereas 2.9% increase is observed for D90E10 compared with diesel.

- The smoke level of D80E20 49.2% lesser than that of diesel but D90E10 results in 32% lesser smoke emission.

- The HC and CO emissions are reduced by about 66% and 9.6% respectively in D80E20 operation compared to diesel. Improved atomization of fuel in the incylinder in lesser HC and CO emission than diesel.

- The lower cetane number of D45E45B10 retard the combustion by 4°CA compared to diesel operation.

- The peak pressure is lower for D45E45B10 in the entire load range when compared to diesel operation.

- The lower energy content and higher heat of vaporization of D45E45B10 leads a lower peak heat release rate compared to diesel fuel operation.

- The D45E45B10 shows a significantly lower brake thermal efficiency compared to diesel operation and is found 16.8% lesser than diesel at rated power.

- The NO_x emission for D45E45B10 is 22.5% lesser than diesel operation due to lower energy content and higher heat of vaporization.

- The increase in smoke emission is about 49.2% for D45E45B10 operation compared to diesel.

- HC is increased by 6.7% in the case of D45E45B10 operation compared to diesel operation. The CO emission follows the same trend as that of HC emission.

- Even though this phase gave adverse effects in performance and emissions, higher volume of ethanol is utilized without any phase separation.

- D45E45B10 can be used as a fuel for CI engine with modified operating parameters. This enhanced improved complete combustion and shows significant improvement in performance at all load conditions.

- D45E45B10MOP operation advances the combustion and improves premixed combustion compared to D45E45B10 under normal operating parameters. However, D45E45B10 shows lower peak heat release rate and peak pressure at rated power compared to diesel operation.

- D45E45B10MOP operation improves the BTE significantly compared to D45E45B10 at all load conditions. The increase in BTE at rated power is 6.8% higher than D45E45B10 fuelled under normal operating parameters. However, BTE of D45E45B10MOP is found lesser than diesel at rated power.

- There is a significant increase in NO_x emission in D45E45B10MOP operation compared to D45E45B10. The increase in NOx emissions of D45E45B10MOP is found thrice that of NOx emissions from D45E45B10 fuelled under normal operating parameters. However, NOx emissions of D45E45B10MOP are found lesser than diesel at rated power.

- The smoke emission is reduced by 15.4% in fueling D45E45B10MOP compared to D45E45B10 fuelled under normal operating parameters. However, the smoke emissions from D45E45B10MOP are found higher than diesel at rated power.

- The HC and CO emissions are reduced by 22.5% and 9.2% respectively in fuelling D45E45B10MOP compared to D45E45B10 fuelled under normal operating parameters. However, this is found higher than diesel at all load conditions which is due to physicochemical properties of D45E45B10 compared to diesel.

As a sum up, although the efficiency produced by D45E45B10 is found to be marginally lower and the emissions of smoke, HC & CO produced are found to be marginally higher compared to that of diesel. The utilized ethanol and butanol are manufactured from waste products and the emissions of oxides of nitrogen produced are found to be significantly lower compare to that of diesel. Hence, higher volume of ethanol can be utilized and a saving of 55% of diesel fuel can be achieved by the implementation of this modification in fuel and in engine. This in turn reduces the dependency of other countries for import of crude oil.

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