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Research Article

Combustion and Emission Characteristics of Hydrous Ethanol–Gasoline Blends with Co-solvents in Spark-ignited Single cylinder Engine

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Abstract

In Brazil, hydrous ethanol is blended with gasoline and is successfully implemented in flexi fuel engines and it has the potential to directly compete with fossil fuels. However, the Ethanol Blending Program of India has not been successful due to non-availability of sugarcane molasses and could not achieve its 5% blending target based on the literature review study. Hydrous ethanol with co-solvent is proposed in this work as a promising blending oxidant instead of energy-intensive anhydrous ethanol in gasoline. This research work also delves into the challenges of using hydrous ethanol in gasoline blended fuel such as water tolerance, fuel properties, and fuel selection. The selected fuel sample 2EW30TBA10, which contains 30% hydrous ethanol and 10% TBA (t-butyl alcohol), and gasoline was selected based on water tolerance of the blended sample and stability. The combustion and emission characteristics of the selected fuel sample 2EW30TBA10 are studied in 4-stroke, single cylinder, water cooled engine, and the performances are compared to base fuel (E0) and reference fuel (E10). The significant contributions of the present research work are the development of combustion model using MATLAB using: Apparent heat release model (Model-1) and Combustion pressure method (Model-2) and the results are validated using modified Wiebe function. There was an increase in brake thermal efficiency by 5% compared to base fuel (E0) and the specific fuel consumption (sfc) for 2EW30TBA10 fuel sample is 290 g/KWh as compared to E10, which is 300 g/KWh and for Petrol (E0), sfc is 360 g/KWh. There was a reduction in CO and HC emission compared to base fuel (E0) and an increase in NOx emissions. The cyclic variations of experimental results are validated using a non-linear regression model.

Keywords: *Combustion, Emission, Hydrous Ethanol, Ethanol-Gasoline, Co-solvents, Combustion modelling, Ethanol Blending Program*

1. Introduction

India produced 2.2 billion litres of fuel ethanol from its 162 distilleries across the country. Ethanol is produced in India from sugar cane molasses through the process of fermentation. (Gain Report, 2017) Ethanol Blending Programme (EBP) has been effectively implemented in Brazil, the United States, and the European Union. The Ethanol blending program in India has not achieved 5% blending target due to non-availability of molasses for production of ethanol and price of ethanol. (Gain Report, 2017; S.Ray *et al*,2011) Considering that the conventional cars will continue using gasoline in the near future, ethanol blending program will complement gasoline vehicle fleet as a potential octane booster and could be used for reducing emissions. (Gain Report, 2017; S.Ray *et al*, 2011; A.S.Ramadhas, 2016; U.Larsen *et al*, 2016; E.F. De Almeida *et al*, 2007 ; T.N.Sreenivasa *et al*, 2015 ; A.Kyriakides *et al*, 2013)
Ethanol is produced through a fermentation process from various feedstocks classified as sugars (sugarcane, sugar beet, sweet sorghum, and molasses), starch (grains, tapioca, cassava) and cellulose (wood, straw, forest feedstock). (A.S.Ramadhas, 2016)

The process of converting biomass to ethanol is by converting starch to sugar using enzymes, fermenting sugar with yeast yielding a mixture of ethanol and water, followed by distillation and dehydration. The cost of dehydration of hydrous ethanol or rectified spirit accounts for 14% of the total cost of production. (U.Larsen *et al,* 2016) The hydrous ethanol or rectified spirit is of 95% purity.

The objective of this research work is to use hydrous ethanol as an ethanol-petrol blended fuel which drastically reduces the cost of ethanol.(U.Larsen *et al,* 2016) And the use of hydrous ethanol-petrol

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blended fuel in 4 stroke petrol engine to study performance and emission reduction. Also, to address the challenges of using hydrous ethanol.

Hydrous ethanol is economic to anhydrous ethanol and it has been very competitive in the Brazilian markets for more than a decade.(E.F. De Almeida *et al*, 2007) Hydrous ethanol can be produced in the sugar industry with the simple distillation method considerably reducing the energy cost. Hydrous ethanol has been successfully used as blended fuel in the Brazilian market without or with minor modification in Flexi Fuel vehicle.(E.F. De Almeida *et al*, 2007) However, the use of hydrous ethanol has the following challenges. (A.S.Ramadhas, 2016)

- **Phase separation:** Separation of petrol and ethanol in blended fuel due to water, resulting in a reduction in octane number of the blended fuel.
- **Volatility properties:** Ethanol addition can significantly affect the volatility properties of gasoline resulting in off-specs gasoline.
- **Material compatibility:** Plastics, elastomers, and non-metals are not recommended for use with ethanol-gasoline blended fuel due to swelling and the problem of corrosion due to wet corrosion due to hydrous ethanol. (T.N.Sreenivasa *et al*, 2015)
- **Performance of engine and emission:** Tuning of the engine specifically for the hydrous ethanol-gasoline blend. Engine behaviour of hydrous ethanol-gasoline blends is different from anhydrous ethanol-gasoline blends. (A.Kyriakides *et al*, 2013 ; R.Munsin *et al*,2013 ; R.C.Costa *et al*,2010 ; R.C.Costa *et al*,2011 ; T.C.C de Melo *et al*, 2012)

The concerns described above motivated this research work and an attempt has been made to address some of these challenges as well. This research work covers some of the issues listed above and a study has been conducted to address some of the problems in the following areas:

- Hydrous ethanol-gasoline blends miscibility characteristics
- Fuel properties
- Atmospheric distillation curves study
- Combustion and emission characteristics

This research work does not include materials compatibility, corrosions and wears related to ethanol-gasoline blended fuels. The miscibility characteristics are not presented here and are available in the published paper. (T.N.Sreenivasa *et al*, 2016)

2. Literature review

Kyriakides *et al.* (2015) evaluated the use of gasoline-ethanol-water ternary mixture as a fuel in Otto engine and tested the ternary mixture for stability at three different temperatures (2°C,10°C and 18°C), three water qualities (distilled, bottled and seawater), two gasoline compositions (commercial gasoline and formulated gasoline without TAME and MTBE) and three additives (isopropanol, 2-butanol and palmitic acid) were burnt in a stationary Otto engine without catalytic converter. (A.Kyriakides *et al*, 2013) In the experiment that was conducted, three different fuels were tested: E0 normal gasoline (96 RON commercial), E40 - a mixture of 60% gasoline and 40% ethanol (99.9 pure), E40h - a mixture of 60% gasoline and 40% ethanol (10% hydrous). The experimental results showed an impressive reduction of NOx emissions by 300-800ppm for hydrous ethanol mixture (E40h) in the λ range from 0.87-0.99 at 20% throttle in comparison with E0 (commercial gasoline) due to the water content in hydrous ethanol, which lowers the peak temperature during burning and reduces NOx formation.

Munsin *et al*. studied the effects of hydrous ethanol with high water content up to 40% on the performance and emissions of a small spark ignition engine for a generator.(R.Munsin *et al*,2013) The result showed that for ethanol with 5% water content (Eh95), CO, HC and NOx emissions after the catalytic converter were lower than the EPA (Environmental Protection Agency) limit. However, for emission before the catalytic converter, only CO emission was lower than the EPA limit, while HC + NOx were higher. HC + NOx emission for hydrous ethanol with water content up to 40% by volume (Eh60) can meet HC + NOx limit of EPA model year 2007 to 2010 (EPA limit: 12 g/kWh) without a catalytic converter. But HC + NOx emission was 10 g/kWh before the catalytic converter, which is higher by 3-4 g/kWh above the EPA model year 2011 limit of 8 g/kWh.

Costa *et al*. compared the performance and emissions from a four-stroke engine fuelled by hydrous ethanol (6.8% water content in ethanol) vs. 78% gasoline- 22% ethanol blend. (R.C.Costa *et al*,2010) The results showed that at high engine speeds, higher torque and BMEP (Brake Mean Effective Pressure) were achieved when hydrous ethanol was used; and at all speed ranges investigated, hydrous ethanol produced higher thermal efficiency reaching a maximum improvement of 14.1%. SFC (Specific Fuel Consumption) for hydrous ethanol was higher by around 54% than gasoline-ethanol blends. Hydrous ethanol reduced CO and HC but increased CO_2 and NOx emission. There was an increase in CO_2 emission in case of hydrous ethanol by 1-2% in comparison with gasoline-ethanol blended fuel for all engine speeds and reduction in CO emission by 3-5% in hydrous ethanol in comparison with gasoline-ethanol blended fuel. This is due to higher water content in the hydrous ethanol molecules, which converts CO produced during combustion into CO_2. Therefore, the use of hydrous ethanol is beneficial with respect to emission control. HC emission in the case of gasoline-ethanol blended fuel was in the range of 300-500 ppm depending on the engine speed. At lower engine speed (2500 rpm), the HC emission was 500 ppm and at higher engine speed

(6000 rpm), the HC emission was 300 ppm. But, in the case of hydrous ethanol fuel, HC emissions are reduced in comparison with gasoline-ethanol blended fuel and were in the range of 40-120 ppm. At lower engine speed (2500 rpm), the HC emission was 120 ppm and at higher engine speed (6000 rpm), the HC emission was 40 ppm. The reason behind the reduction of HC emission in the case of hydrous ethanol in comparison with gasoline-ethanol blended fuel is due to the chemical structure of the latter, which has a higher presence of carbon and hydrogen resulting in un-burnt HC than hydrous ethanol. NOx emissions are higher by 100-1000 ppm in the case of hydrous ethanol than the gasoline-ethanol blended fuel for various speed ranges and it is due to the faster flame speed of hydrous ethanol along with the more advanced ignition timing, which results in higher peak pressure and, therefore, higher peak temperature in the combustion chamber.

Costa *et al*. also studied the effect of compression ratio on an ethanol/hydrous ethanol-gasoline blended fuel and its engine performance. The results showed that by increasing compression ratio the engine performance substantially improved, significantly decreased SFC and increased thermal efficiency when using hydrous ethanol as fuel.(R.C.Costa *et al*,2011)

Cordeiro de Melo *et al*. investigated hydrous ethanol-gasoline blends on a Flex-Fuel Otto Engine to study fuel consumption, emissions, and in-cylinder pressure data. Flex Fuel Vehicle represented over 90% of new light-duty vehicles sold in 2009 in the Brazilian market and these vehicles used gasoline blended with anhydrous ethanol (20 to 25% v/v), 100% of hydrous ethanol (contains from 6.2 to 7.4 % w/w of water) or any blend of these fuels.(T.C.C de Melo *et al*, 2012 ; T.C.C de Melo *et al*, 2010) de Melo *et al*. also measured aldehydes, unburnt ethanol and total hydrocarbons using FTIR (Fourier transform infrared spectroscopy) and the emission results showed reduction trend in CO, THC, and NOx, a trend of increase in aldehydes and unburnt ethanol and no significant changes in CO_2. At lower speeds (1500 and 2250 rpm), and at stoichiometric condition (60Nm), the CO emission reduced from 15 g/kWh at 0% hydrous ethanol to 13 g/kWh at 100% hydrous ethanol, NOx emission reduced from 12.5 g/kWh at 0% hydrous ethanol to 10 g/kWh at 100% hydrous ethanol, THC emission reduced from 2.5 g/kWh at 0% hydrous ethanol to 0.5 g/kWh at 100% hydrous ethanol. On the other hand, there was increase in Aldehyde emission from 0.15 g/kWh at 0% hydrous ethanol to 0.38 g/kWh at 100% hydrous ethanol and also increase in un-burnt ethanol emission from 0.5 g/kWh at 0% hydrous ethanol to 2.5 g/kWh at 100% hydrous ethanol. (T.C.C de Melo *et al*, 2011)

Venugopal *et al* investigated combustion characteristics of a port-injected engine fuelled with hydrous ethanol gasoline blend and found that higher flame velocity and wider flammability limits of the blend resulted in lower cycle-by-cycle variation in IMEP in comparison to neat gasoline. NOx emissions were lower in hydrous ethanol gasoline blend due to the higher heat of vaporization of the ethanol-gasoline blend and the presence of water reduces the in-cylinder temperature. (T.Venugopal *et al*, 2013)

Schifter *et al* performed the test in a single cylinder engine using mid-level (0-40% volume) hydrous ethanol in lieu of traditional anhydrous ethanol-gasoline blends. The results showed that higher pressure and lower intake temperatures were achieved with hydrous ethanol fuel blend. (I.Schifter *et al*, 2013) Gautam *et al* conducted a test on a single cylinder

Waukesha cooperative research engine (CFR) using higher alcohols (propanol, butanol, and pentanol)-gasoline blends. The test results showed that higher alcohol-gasoline blends have greater resistance to knock than gasoline and the ignition delay and combustion interval data showed that higher alcohol-gasoline blends have faster flame speed. (M.Gautam *et al*, 2000)

Fagundez *et al* studied wet ethanol energy balance from production to fuel and summarised that small increase in water quantity in the distillate can contribute in the net energy gain, making the use of wet ethanol more attractive. (J.L.S.Fagundez *et al*, 2015)

3. Research Methodology

The motivation of this research work is based on the current blending mandate by the Government of India and the prospects of increasing the ethanol blending target presently at 5% to 20% in the near future based on the Government Bio-fuel policy and the forecast highlighted in the World Energy Outlook 2016 report related to use of ethanol as a blended fuel in India.(World Energy Outlook 2016, 2016) Based on literature review study, we have identified hydrous ethanol as a potential alternative to the present anhydrous ethanol, both in terms of cost as well as in terms of availability of hydrous ethanol from sugar plants in India through simple distillation process instead of 30% more energy intensive dehydration process to produce anhydrous ethanol. In Brazil, hydrous ethanol is blended with gasoline and is successfully implemented in flexi fuel engines and it has the potential to directly compete with fossil fuels based on the literature review study.

Figure 1 shows the research methodology adopted in this research work starting with the research problem statement mentioned above supported by government mandate and literature review study conducted in this research work. One of the important challenges of using hydrous ethanol as a blended fuel is phase separation of petrol and ethanol in presence of water. This problem is addressed in the miscibility study of the water-ethanol-gasoline mixture without and with co-solvents. (T.N.Sreenivasa *et al*, 2016) The miscibility sampling study was conducted for various proportions of water (1-5% vol) and anhydrous ethanol (5-25% vol) in gasoline without co-solvents at various temperatures.

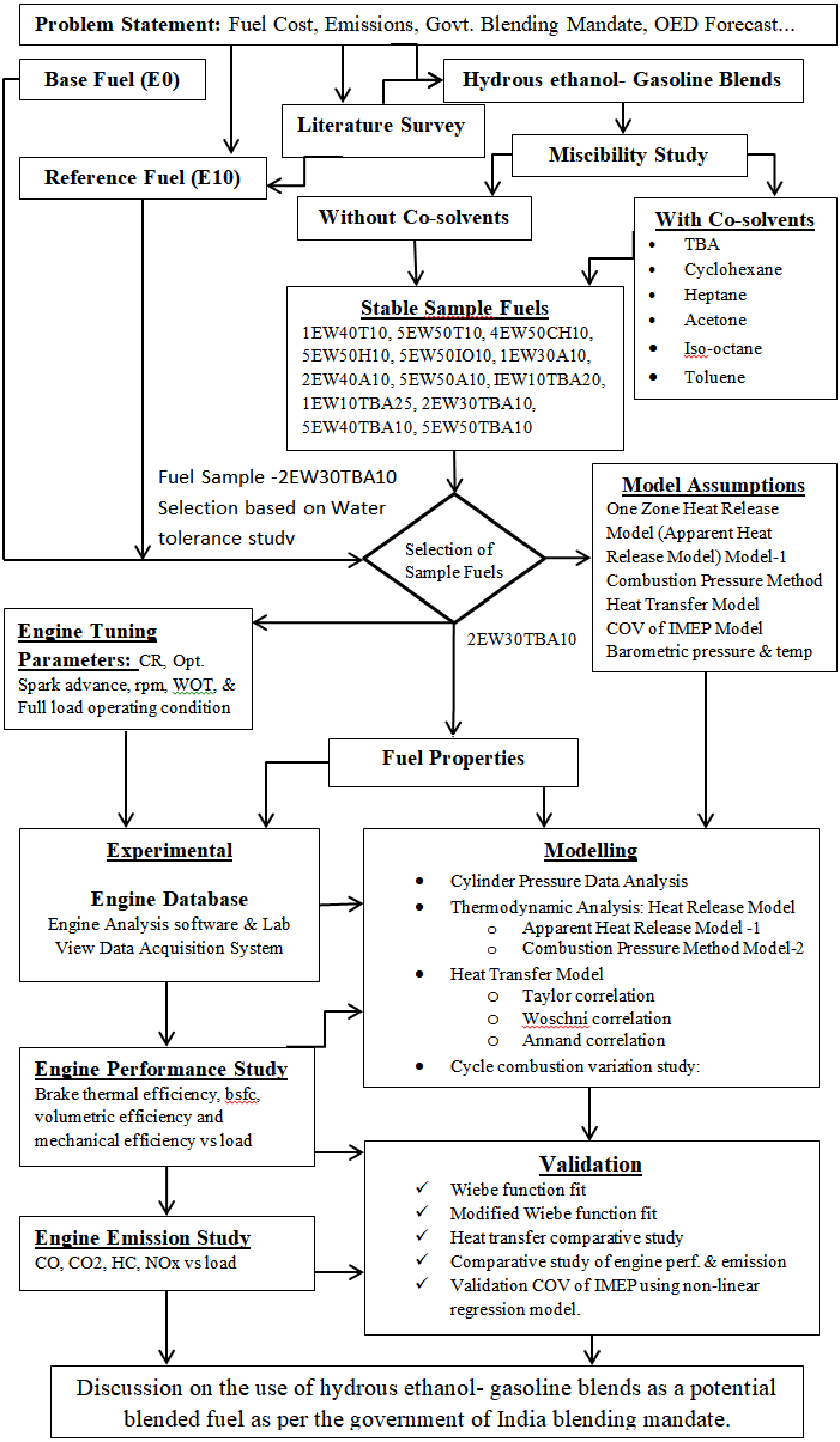

Figure 1 Research Methodology

Similarly, the miscibility sampling study was conducted for various proportions of water (1-5%) and anhydrous ethanol (10-50%) in gasoline with co-solvents (1-25%) that is, TBA (t-butyl alcohol), Cyclohexane, Heptane, Acetone, Iso-octane and Toluene at room temperature (300 K). List of stable samples with water tolerance and without phase separation at room temperature was identified (T.N.Sreenivasa *et al*, 2016) and the sample fuel blended with TBA was selected in terms of water tolerance as well as being proven co-solvent / blending fuel based on literature review as well as based on IS 2796:2008, (Bureau of Indian Standards, 2008) where TBA is used as oxygenate. The fuel sample 2EW30TBA10 (30% hydrous ethanol, 10 % TBA and 60% gasoline) has been specifically selected with water tolerance of 2 % to test spark ignition engine with higher alcohol content for performance and emission as well as to achieve higher blending target and to reduce total fuel cost without any modification in the engine. The selected fuel sample has been tested as per ASTM standards (ASTM.ORG) and the properties are compared to base commercial petrol E0 and the reference fuel E10, which is a blend of 10% anhydrous ethanol with commercial gasoline.

4. Fuel Selection and Fuel Properties

The fuel sample 2EW30TBA10 is tested for the following properties as per ASTM standards (ASTM.ORG): Density, RVP, MON, RON, LHV, Moisture content, Distillation curves, CHN analysis, Fuel composition especially oxygenates in the fuel sample, and hydrocarbon analysis (PIONA test), see Table 1.

Table 1: Fuel properties of Gasoline, Ethanol, TBA, and 2EW30TBA10

Property	Unit	Gasoline (IS 2796; I.Schifter *et al*, 2016)	Ethanol (A.K.Thakur *et al*, 2017)	2EW30TBA10	ASTM Standard
Chemical formula	-	C_5-C_{12}	C_2H_5OH	-	
Molecular weight	kg/kmol	100-105	46.07	-	
C	w%	87.4	52.2	83.5	ASTM D-5291
H	w%	12.6	34.7	11.9	ASTM D-5291
O	w%	0	13	4.6	ASTM D-5291
Specific Gravity	-	0.7-0.78	0.794	0.770	
Density	kg/m^3	750-765	785-809	770	ASTM D4052
Reid Vapour Pressure	kPa	53-60	17	56.73	ASTM D-5191
RON	-	91-100	108.61-110	107	ASTM D2699
MON	-	82-92	92	98.8	ASTM D2700
Fuel Sensitivity		9	18	8.2	
Anti-knock Index		87-96	101	102.9	
Lower Heating Value	MJ/kg	44	26.9	35.94	ASTM D4809
Latent of vaporization	kJ/kg	380-400	900-920	-	
Distillation					ASTM D-86
IBP	^{0}C	45	78	40	
10%	^{0}C	54	78	55	
50%	^{0}C	96	78	75	
90%	^{0}C	168	79	145	
FBP	^{0}C	207	79	182	
Oxygenates	vol%	5 max	-	29.47	ASTM D-4815
Olefins	vol%	18 max	-	8.98	ASTM D6729
Aromatics	vol%	42 max	-	16.71	ASTM D6729
Paraffins	vol%	11.4*	-	6.98	ASTM D6729
Isoparaffins	vol%	52.6*	-	21.12	ASTM D6729
Naphthenes	vol%	4.7*	-	4.92	ASTM D6729

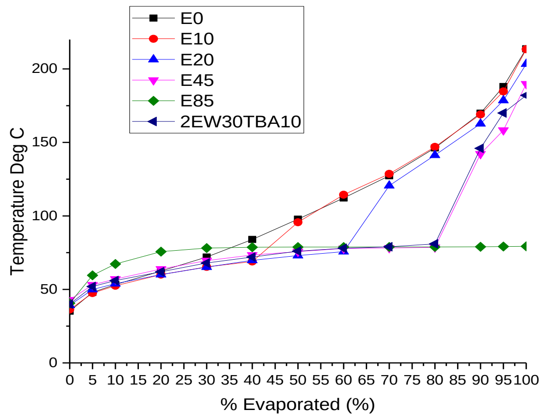

Fig 2 Atmospheric Distillation Curves (I.Schifter *et al*, 2018)

The properties of the base fuel E0 are either taken from the source of purchase of commercial petrol as per IS 2796 and the properties of reference fuel E10 are taken from the literature for comparison.

Atmospheric distillation curves of the fuel sample have been specifically studied as per ASTM D86 standards (ASTM.ORG) for the fuel sample 2EW30TBA10 and the results are compared with the literature (I.Schifter *et al*, 2018) along with base fuel E0, reference fuel E10, E20, E45, and E85, see Fig 2.

5. Engine tests

The selected fuel samples are tested in variable compression 4-stroke water cooled multi-fuel oil engine of 3HP rated power fitted with spark plug and carburettor system connected to eddy current dynamometer, see table 2 for engine specification. The engine analysis software and the data acquisition system from the National Instruments – LabView acquires data from the engine and logs data into the database. See Fig 3 A for the experimental setup.

The engine is tuned to operate for the fuel sample 2EW30TBA10 by optimizing the operating parameters such as: operating the engine at fixed rpm (1000 rpm), running the engine at wide open throttle (WOT), adjusting the spark timing advance to an optimum value and running the engine at maximum load. The fuel sample 2EW30TBA is tested at various compression ratios and the results are compared with the base fuel E0 and the reference fuel E10 for engine performances and emissions. The compression ratios are changed by increasing/ decreasing the clearance volume of the cylinder head. See Fig 3 B. An exhaust gas analyser measures the exhaust gases as shown in Fig 3 C.

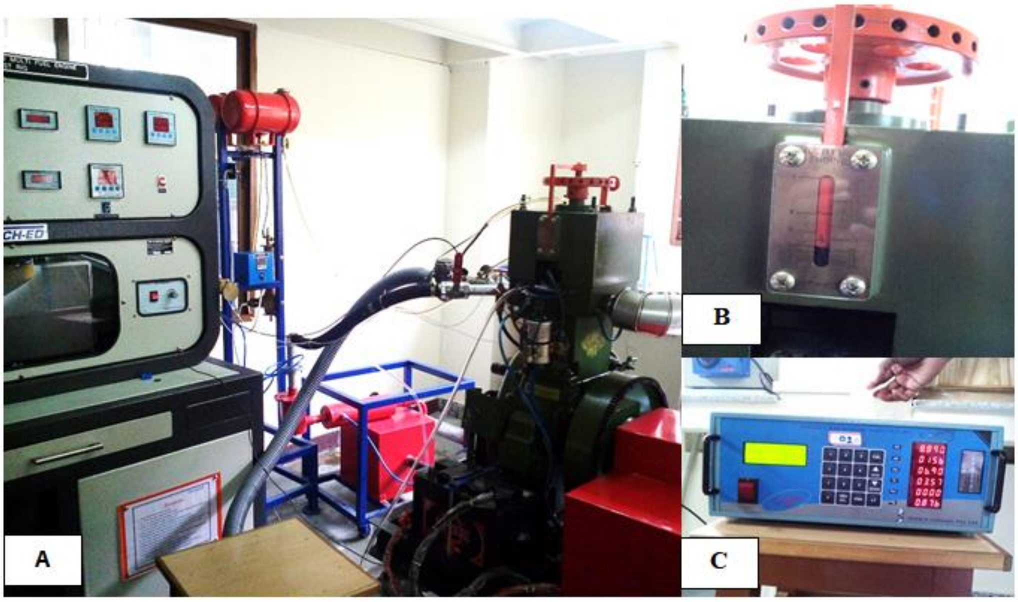

Fig 3 Experimental Setup: A- Engine with Data acquisition system, B- Variable Compression Ratio and C- Exhaust Gas Analyser

Table 2: Engine Characteristics: 4 stroke single cylinder water cooled multi-fuel variable compression ratio engine

Make	Kirloskar AV1
Rated power	3 HP (Petrol) @ 1500 rpm
Bore Dia	80 mm
Stroke Length	110 mm
Connecting rod length	234 mm
Swept Volume	552 cc
Compression Ratio	6:1 to 12:1 (Petrol)
Rated speed	1500 rpm
Rated Torque	24 N-m
Inlet valve opens BTDC	4.5 Deg.
Inlet valve closes ABDC	35.5 Deg.
Exhaust valve opens BBDC	35.5 Deg.
Exhaust valve closes ATDC	4.5 Deg.

6. Results & Discussion

The modelling and analysis of cylinder pressure data, heat release and heat transfer are done using MATLAB, which reads data from the engine database for the fuel sample 2EW30TBA10, base fuel (E0) and reference fuel (E10), and plots of P-θ, P-V, Log P-LogV, Heat release based on Apparent heat release model and Combustion Pressure Method. And it also uses a smooth function to generate smooth plots of heat release. MATLAB code also prompts the user to enter specific heat ratios and generates a comparative plot of heat release based on Apparent heat release model (J.A.Gatowski *et al*,1984; H.M.Cheung *et al*,1993; K.M.Chun *et al*, 1987) and Rassweiler-Withrow Method (Combustion pressure method) (M.F.J.Brunt *et al*,1998; B.M.Grimm *et al*, 1990) for various specific heat ratios, refer equation 2. The modelling of heat release are based on certain assumptions for one-zone heat release model (Gatowski Model) (J.A.Gatowski *et al*,1984; J.H.Grau *et al*, 2002), which is further simplified by not taking into account heat transfer and the effect of crevice flow. This model is also called as Apparent heat release model refer equation 1. The heat release data based on Apparent heat release model (Model-1) and Combustion pressure method (Model-2) are compared statistically using regression analysis.

$$\frac{dQ}{d\theta} = \left(\frac{\gamma}{\gamma-1}\right) p \frac{dV}{d\theta} + \left(\frac{1}{\gamma-1}\right) V \frac{dp}{d\theta} \quad (1)$$

$$\frac{dQ}{d\theta} = \frac{1}{\gamma-1} V \frac{dP}{d\theta} \quad (2)$$

Where, Q - Energy released, γ- specific heat ratio, p,v,θ – pressure, volume & crank angle.

The specific heat ratios are estimated based on the slope for compression and expansion processes using LogP-LogV diagram and the comparative plots of the fuel sample 2EW30TBA10 with the base fuel (E0) and the reference fuel (E10) are studied at various compression ratios, spark advance timing, engine speed etc. Fig 4 shows a P-θ diagram and Log P- Log V diagram for the fuel samples E0, E10 & 2EW30TBA10 at compression ratio 9 and 1000 rpm.

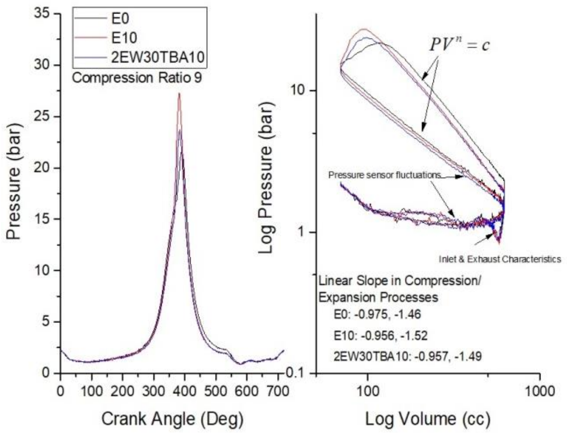

Fig 4 Comparative P-θ and Log P- Log V diagram for E0, E10 & 2EW30TBA10 at CR9

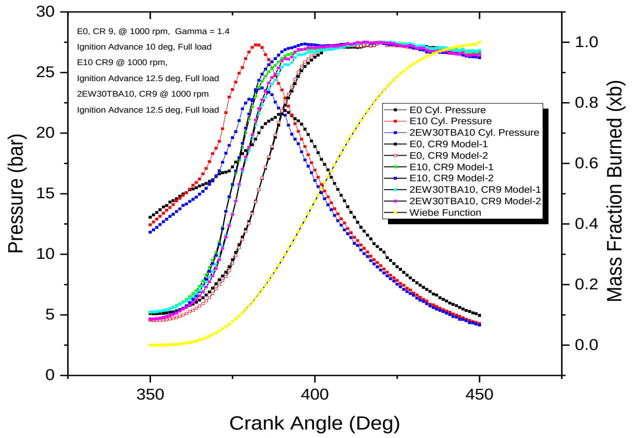

Fig. 5 Cylinder Pressure and Mass Fraction Burned versus Crank Angle of the samples

The mass fraction burnt xb versus crank angle is illustrated in Fig 5 based on experimental data vis-à-vis Wiebe function (J.B.Heywood, 1988) for a = 5 and m=2. The mass fraction burnt, xb is found for Model-1 using equation 1 (Apparent heat release) and for Model-2 using equation 2 (Combustion Pressure Method), integrated and normalized using equation 3 for start of combustion, θo=350, and duration of combustion, Δθ=100.

$$xb = \frac{\int_{\theta_0}^{\theta} \frac{dQ}{d\theta} d\theta}{\int_{\theta_s}^{\theta_f} \frac{dQ}{d\theta} d\theta} \quad (3)$$

The Wiebe function (4) and the modified Wiebe function (5) are presented below for the fuel sample 2EW30TBA10 and the equations are specific to the engine studied and for the operating conditions (CR 9 at 1000 rpm, Spark Advance $+10^0$, WOT, and Full load condition).

$$xb = 1 - \exp[-23.34 \left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{2.581}] \quad (4)$$

$$xb = 1 - 0.887 \exp(-59.34 \left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{3.493}) \quad (5)$$

The results of brake thermal efficiency are presented below in Fig 6b for compression ratio 9 for the fuel sample E0, E10, and 2EW30TBA10. It is apparent from the results that brake thermal efficiency is 4-5% higher in the fuel sample 2EW30TBA10 at higher load condition compared to petrol; similarly, for E10, brake thermal efficiency is 2-4% higher in comparison to petrol sample (E0).

Specific fuel consumption (sfc) of the fuel samples is shown in Fig 6a for compression ratio 9. For this size of the engine with 552 cc cylinder, the specific fuel consumption for 2EW30TBA10 fuel sample is 290 g/KWh in comparison to E10, which is 300 g/KWh and for Petrol (E0), SFC is 360 g/KWh. (J.B.Heywood, 1988) Volumetric efficiency is higher at full load condition for the fuel sample 2EW30TBA by 10-14% compared to E0 as shown in Fig 6c for compression ratio 9.

Mechanical efficiency which is a ratio of brake power and indicated power is higher for E10 by 5-15% compared to Petrol as seen in Fig 6d for compression ratio 9, and at full load condition for the fuel sample, 2EW30TBA10 is higher by 5% compared to petrol (E0).

In comparison to typical design and operating data for an internal combustion engine and this is an oil engine converted to operate as SI engine with spark plug installed in the injection port and a carburetor for supplying pre-mixed air-fuel mixture. In comparison to typical design, the specific fuel consumption of 300 g/KWh achieved in full load condition is comparable to a standard design for this cylinder size and compression ratio. (J.B.Heywood, 1988) However, Power per unit volume (KW/dm3) achieved for fuel sample 2EW30TBA10 at full load condition is 3.35 KW/dm3, which is considerably less for an engine of cylinder size of 552 cc. (J.B.Heywood, 1988) As well as the BMEP (brake mean effective pressure) achieved in this engine for these fuel samples at full load conditions at compression ratio 9 and 10 are tabulated in table 3 for comparison with the standard design.

Table 3: Brake Mean Effective (bmep) at the Full Load condition

bmep, bar	E0	E10	2EW30TBA10	Typical Design
CR 9	2.98	3.74	3.75	7-10 bar
CR 10	3.69	3.69	3.65	

As seen in table 3, the brake mean effective pressure developed by this engine is comparably less than the standard design of this size; one of the reasons for lower bmep could be a limitation in dynamometer or could be due to not operating the engine at a rated capacity of 3HP. (J.B.Heywood, 1988)

The other stable fuel samples using co-solvents such as cyclohexane, heptane, acetone, iso-octane, and toluene fuel properties, engine performance, and emissions are not presented here in this research work. Only 30% hydrous ethanol and 10% TBA has been chosen based on water tolerance and stability of the sample, and its engine performance and emissions are studied. The results are not compared with E30 or E40 but only with existing blending proportion i.e. E10.

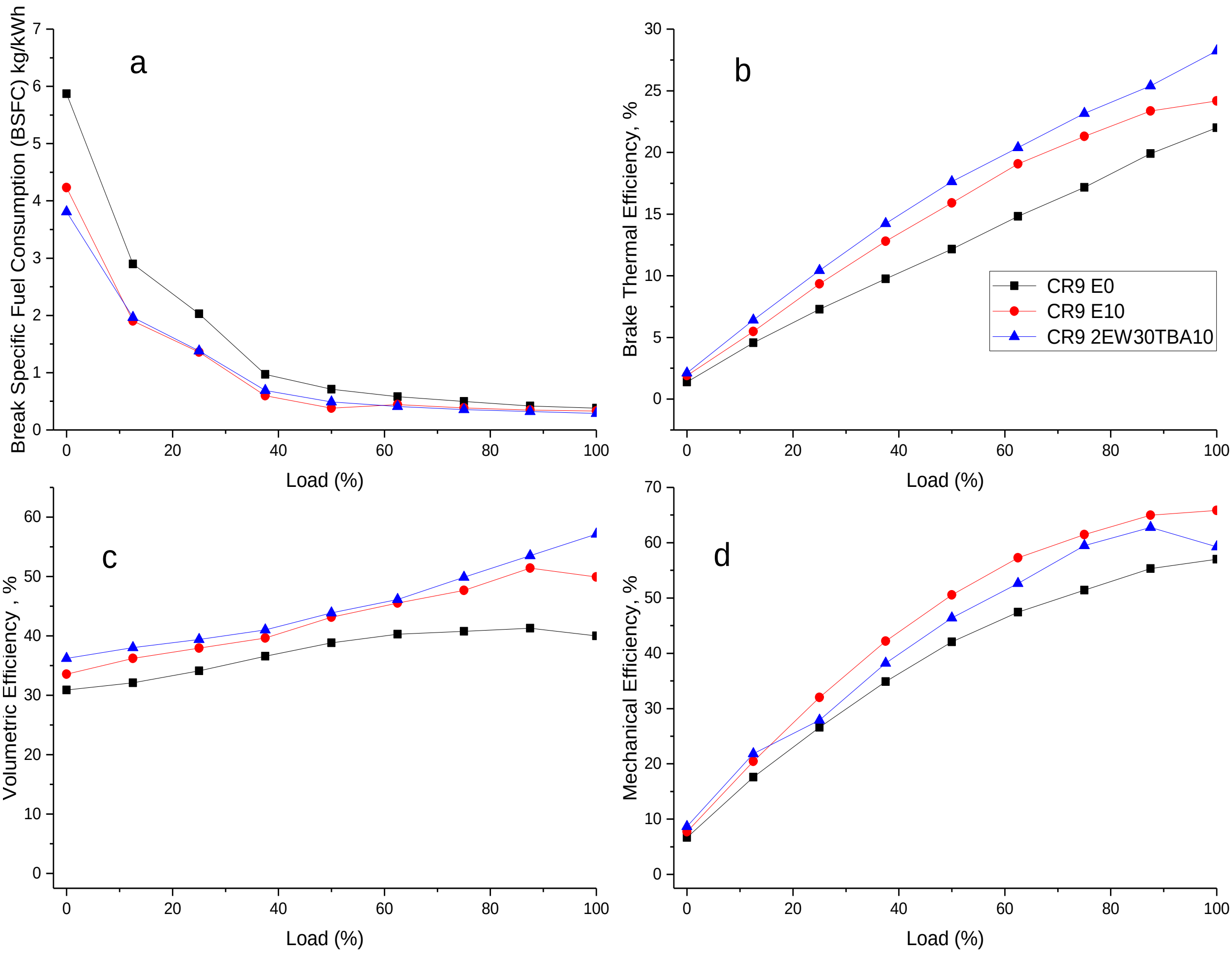

Fig 6 BSFC, BThE, VE and ME versus Load (Nm) of blended fuels at CR9 1000 rpm

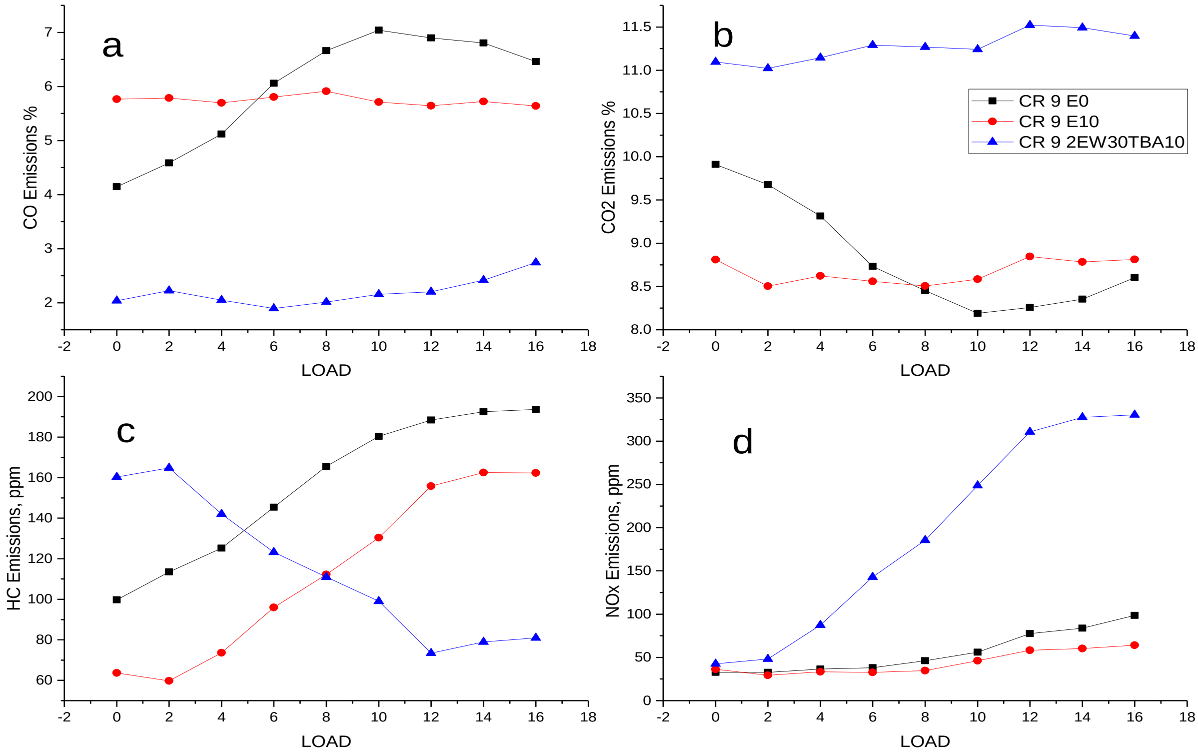

Fig 7 Emissions versus Load (N-m) of blended fuels at CR9 1000 rpm

Emissions were measured at the exit of exhaust gas calorimeter in the experimental setup using Indus Five Gas Analyser Model PEA 205 measuring exhaust emissions of Carbon monoxide (CO), Carbon dioxide (CO_2), Oxides of Nitrogen (NOx) and Hydrocarbon (HC). The gas analyzer measures CO, CO_2, and HC using non-dispersive infra-red sensor and NOx and O_2 were analyzed using the electrochemical sensor. The accuracy of measurement for CO is +/-0.06%, +/-0.5% for CO_2, +/-12 ppm for HC and +/-0.1% for O_2. And it is the certified instrument by Automotive Research Association of India. The catalytic converter is not used in this experimental setup.

The results of the exhaust gas analysis of CO, CO_2, HC, and NOx are presented here for fuel samples E0, E10, and 2EW30TBA10 at compression ratios 9 and 1000 rpm, see Fig 7.

Carbon Monoxide Emission: CO emissions in the exhaust are due to lack of oxygen and in a rich mixture the CO emission increases and in a lean mixture the CO decreases. CO emissions are directly influenced by Air-Fuel ratios. In the results shown in the Fig 7a, CO emission of the fuel sample 2EW30TBA10 is less by 2-3% at compression ratio 9 and the A/F (Air/Fuel ratio is around 19-20 and being a highly lean mixture coupled with leaning effect of alcohol in the fuel the CO emissions are considerably less compared to E0 or E10. The CO emissions of E10 is also less compared to E0 (Petrol) due to the presence of oxygen molecules in Ethanol as seen in Fig 7a at Compression ratio 10. (C.R.Ferguson *et al*; H.Bayraktar *et al*, 2005)

Carbon dioxide Emission: CO_2 emissions increases in the fuel sample 2EW30TBA10 by 2-3 % compared to E0 /or E10 as seen in Fig 7b. CO_2 emissions decrease with load for the base sample E0, and reference sample E10.

Hydrocarbon Emission: HC emissions are due to the presence of unburnt hydrocarbon fuel in the exhaust of an engine and also due to oil layers within an engine cylinder and the solubility of the fuel in the oil. HC emissions are also due to carbon deposits build up on the valves, cylinder, and piston heads of an engine. And another possible reason for hydrocarbon emissions is due to unburnt gas trapped in crevice regions in the combustion chamber. Fig 7c shows the HC emissions of the blended fuels. As seen in Fig 7c, the HC emission of the fuel sample 2EW30TBA10 decreases by 100 ppm from no load to full load condition at compression ratio 9.

Oxides of Nitrogen Emission: Nitrogen oxides are formed in the combustion chamber when oxygen and nitrogen react at very high temperature. The formations of NOx are temperature dependent and are directly proportional to engine load. At engine start-up the NOx concentration is relatively low and as the engine heats up with load the NOx concentration increases as seen in Fig 7d at compression ratio 9. The oxides of nitrogen formed in SI Engine are dominantly nitric oxide (NO) against the concentration of nitrogen dioxide (NO_2), which are relatively small in percentage of 1-2%. (C.R.Ferguson *et al*)

Cyclic Variations: In-cylinder pressure varies from cycle to cycle in a spark-ignition engine due to the variation of the combustion process in the cylinder. These variations are caused by movement and mixing of gases within the cylinder, spark timing-misfire, variation in Air/Fuel ratios per cycle, and variations are also due to fresh charge inducted into the cylinder as well as due to residual gases within the cylinder per cycle. Vehicle driveability has a direct correlation to variation in brake torque which in turn is related to cylinder pressure. Variation in-cylinder pressure and mass fraction burned (MFB) for the fuel sample E0, CR7 is shown in Fig 8 along with variation in P_{max} versus COV (Coefficient of Variation) and IMEP. Higher the P_{max} , faster the burning rate; lower the $P_{max,}$ lower the burning rate. Rapid burning is seen in the mass fraction burnt per cycle along with the partial burning of fuel. P_{max} variation with respect to variation in IMEP also shows variation in the cycle. Cyclic variations can be broadly divided into two distinctive groups: prior-cycle effects and same cycle effects (J.B.Heywood, 1988). The prior cycle effects are due to residual gas, misfire, partial burning etc. and the same cycle effects are due to random variation of in-cylinder flow. The dominance of one group over another depends on operating conditions.

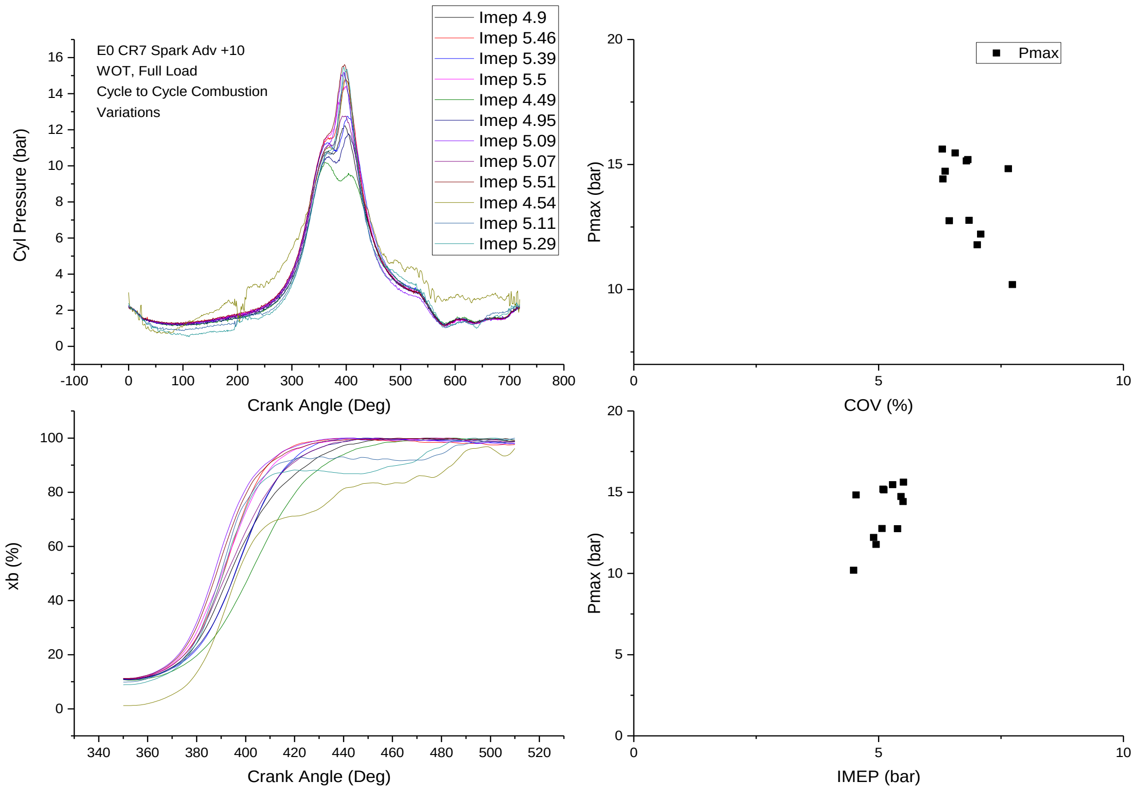

Fig 8 Cyclic variation of cylinder pressure, MFB versus crank angle, Pmax vs COV, IMEP for E0 CR7, Spark Adv +10, WOT, Full load condition

The cyclic variations are measured using pressure parameters, combustion parameters, flame front parameters, and exhaust gas parameters. In this study, cyclic variations using pressure parameters i.e. P_{max}, Peak pressure and IMEP per cycle were studied by comparing experimental data with Non-Linear Regression Model for engine COV of IMEP. (W.Dai *et al*, 2000)

Figure 9 shows the variation in COV versus IMEP and Air-Fuel Ratio (AFR), based on experimental and modelled data, COV of experimental data is varying from 6.3 to 7.73, well within 10%. If COV is more than 10% then it can have drivability problems, which isn't the case in the engine we are studying here. As we can see comparing COV Experiment and COV Modelled, the COV % is well within range though there isn't any correlation between the data.

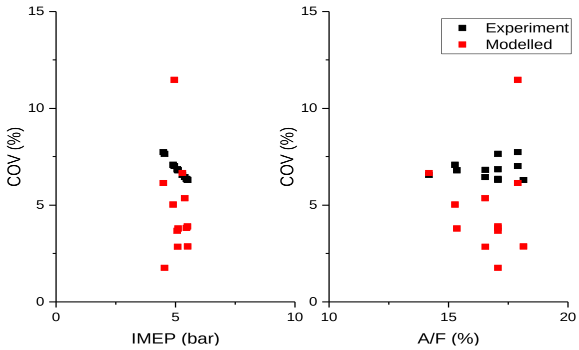

Fig 9 COV versus IMEP & AFR based on Experimental and Modelled Data. (W.Dai *et al*, 2000)

The non-linear regression model used here is of a polynomial form, COV of IMEP is a function of engine speed, equivalence ratio, residual fraction, tail side burn duration (0-10%), burn duration (10-90%) and location of 50% MFB. This model is based on 6000 data points collected and processed from engines of 1.6 litres to 4.6 litres and has been successfully implemented in GESIM, which is an engine cycle simulation model. (W.Dai *et al*, 2000)

Conclusions

The main contributions and significant findings of this research work are:

- The use of hydrous ethanol as a potential blending fuel based on literature review, which can contribute to India's Ethanol Blending Program by considerably reducing the fuel cost.
- The selection of fuel sample: 2EW30TBA10 is based on water tolerance and stability study. (T.N.Sreenivasa *et al*, 2016) The properties of the blended fuel sample: 2EW30TBA10 are determined as per ASTM standards and the sample fuel properties are compared with the base fuel E0 and reference fuel E10.
- Development of combustion model with predictive capability specifically for the engine studied using MATLAB. Combustion Model-1 based on one-zone heat release model (Apparent heat release) and Model-2 based on Rassweiler-Withrow Method (Combustion pressure method) are studied in this research work using a modified oil fired 4-stroke engine with spark plug and the results of the combustion model-1 and model-2 were found to be consistent.
- Development of Wiebe function and modified Wiebe function and determination of parameters using MATLAB for the fuel samples E0, E10 and 2EW30TBA10 at compression ratio 9 specific to the engine studied and operating conditions. The results are validated for repeatability using parameters determined in Wiebe function and modified Wiebe function and found consistent with the experimental data.
- COV of IMEP of non-linear regression model using MATLAB random number generator and compared with experimental results. (W.Dai *et al*, 2000)
- Based on the literature review and field survey study, the major bottleneck of India's Ethanol Blending Program is the availability of feedstock, i.e. sugarcane molasses.

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