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Investigation of Characteristics of Internal Combustion Engine with Producer Gas as a Fuel Running on Dual Fuel Mode

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Abstract

Biomass is alternative energy source which is nowadays building a huge requirement in the world. The present work focuses on the performance of ICE (Internal Combustion Engine) with producer gas as a fuel running on the dual fuel mode and its performance characteristics investigation. The paper shows the methodology and experimental analysis to run the CI (Compression Ignition) engine. The producer gas which is filtered by using the wet bed packed scrubber is supplied to the diesel engine along with diesel at constant engine speed. By doing some modifications in the existing system, the ICE performance parameters like thermal efficiency, volumetric efficiency, fuel consumption, brake specific fuel consumption and the percentage of diesel fuel replacement are calculated at different load conditions. The opacity of the exhaust smoke is calculated by using AVL smoke meter. However, the thermal efficiency of the engine is observed to be dropped to 15.48% on dual fuel mode while the same for diesel fuel mode was 41.92% at the full load operation (3kW load) .This is maybe because of low heating value of the producer gas. The maximum diesel replacement was achieved up to 74.40% at 1kW load. The smoke density of the dual fuel mode was found to be lesser (14.5 %) as compared to diesel fuel mode (20.05%).

Keywords: *Biomass gasifier; renewable energy; compression ignition engine; producer gas; power generation.*

1. Introduction

Power generation by using the diesel engine is getting very popular because of its huge potential to generate high power output and its reliability. Therefore, the diesel engine technology plays a very essential part in the agricultural, transportation and power generation applications. In the current time, there is lot of research is going on the renewable energy technologies because the conventional fuels are on the verge of extinction and the demand of energy is getting increasing day by day(N. Ingle et al, 2016). To overcome this problem, we need to find out some alternative energy sources to fulfill the energy demand. Power generation through biomass can be a very good alternative energy source.

The producer gas (PG) can be obtained from biomass gasifier which can be used in the diesel engines to use it as a substitute fuel for the diesel fuel either partial or full replacement of the diesel. With the help of biomass gasifier based electricity generation plants, we can generate the electricity where there is scarcity of the conventional fuels. These areas are rural sides, remote places and other areas where finding the conventional fuel is very difficult. A lot of researchers have been working on the renewable energy technologies and this energy source seems much promising for the future energy systems.

Mostly, gaseous fuels are found to be very useful and efficient for the ICE because of their property of mixing nicely with the air. A high self ignition temperature and higher compression ratio provide good thermal efficiency and reduction in emissions (R. Sharma 1996).

1.1 ICE operated on dual fuel mode with PG

Biomass gasification technology has been in the existence for more than 80 years since the World War II. The first trial attempted using the producer gas on the ICE in 1881. In dual fuel mode of operation, around 60-70% of diesel replacement was observed on 5.25 kW of load. The power generation efficiency was obtained as 20% and the diesel replacement of 60% was achieved in the dual fuel mode operation (Gosh S, et al 2004).

In the dual fuel mode of operation, the specific energy consumption decreases with the increase in compression ratio(R.N. Singh, 2007). As we increase the brake power up to 4.5 kW, the brake thermal efficiency increases up to 28% and then starts decreasing with the increase in load (N.R. Banapurmath et al 2008).

However, there are lot of concerns needed to be taken care are listed below which affects the engine and its performance parameters are as follows.

- A good quality of producer gas having the maximum heating value
- Producer gas impurities
- Filtration of producer gas
- Pressure losses of the producer gas while connecting gasifier with the IC engine
- Air-fuel mixture

1.2 Impurities in producer gas

The major difficulty is to obtain a good quality producer gas which can be used with the diesel engine. The producer gas should be appropriately filtered so as to reduce the chances of lesser efficient system. The producer gas should be clean and free of tar and other particulate matters which can damage the ICE should be reduced. The optimum or acceptable level of tar and particulate matter content to run the ICE should be less than 100mg/Nm^3 and less than 200 mg/Nm^3 (A. Gurudasani, et al 2016).

2. Materials and methods

The schematic diagram of experimental setup is shown in fig.1. The current system consists of a biomass gasifier, packed bed oil scrubber and an ICE (Diesel engine).

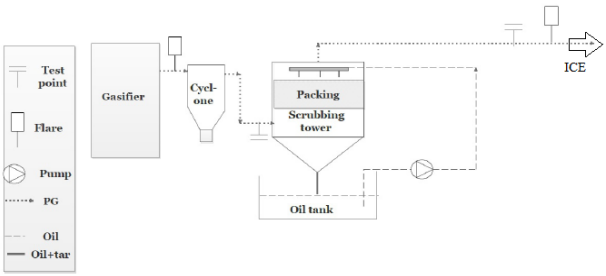

Fig.1 Schematic diagram of the experimental setup

2.1 Producer Gas generation by using biomass gasifier

In the current system, a biomass gasifier used of capacity 20kg/hr. Biomass used was 50mm cubic wooden blocks as feedstock. This gasifier produces 50 Nm^3/hr of producer gas. Gasifier takes 10-15 minutes for starting and it has flaring point to check the quality of gas produced. Tar content of gas was 1g/Nm^3. The outlet temperature of producer gas is 350°C from the gasifier. The calorific value of producer gas obtained is 5.3 MJ/Nm^3 (A. Gurudasani, et al 2016).

Fig.2 The downdraft gasifier and the scrubbing unit

2.1.1 Properties of producer gas

Table 1 Properties of producer gas

Gas	Percentage
CO	17 - 21 %
H_2	12-15 %
CH_4	1-5 %
CO_2	9-12%
N_2	45 - 54 %
Calorific Value	5.3 MJ/Nm^3

2.2 Packed bed oil scrubber to filter the producer gas

The packed bed oil scrubber was made up of mild steel of 3 mm thickness. The diameter of the scrubber used is 750 mm and the height is 800 mm. It has four legs as the stand. In between the four legs, the oil tank is placed. At the lower end of scrubber, a 100 mm pipe is attached to drain of the oil. The raw impure producer gas is fed from the left bottom side of the scrubber after passing through the cyclone. At the height of the 300 mm from tower, supports are provided for the perforated sheet onto which the packing material is placed. With the help of a motor pump, the oil is sprayed from the top of the scrubber. To spray the oil from the top of the system, four showers were used above the perforated sheet where the packing material is placed on to it... The clean gas is obtained from the top side and then sent

to the ICE by controlling the valve attached to it (A. Gurudasani, et al 2016).

2.3 Internal Combustion Engine

Technical Specifications of the diesel engine used for the experimentations are given below.

Table 2 Engine specifications

Rated power output	5 H.P
Rated RPM	1500 RPM
Stroke (L)	110 mm
Bore (D)	80 mm
Burette	50 ml
Orifice diameter	16 mm
Coefficient of discharge	0.64

2.4 Methodology:

The main objective of the present work was to get a good quality producer gas being coupled with the IC engine and investigating the performance parameters of it on the dual fuel mode.

Very limited researches have been done on the producer gas – diesel dual fuel mode engine. The present work focuses on the study of characteristics of the IC engine (CI Engine), which was tested with the producer gas.

2.4.1: Engine modifications:

To get the better efficiency of the engine, we need good combustion of fuel inside the combustion chamber. As we are operating on dual fuel mode, it is very essential to form a lean mixture of producer gas and air. A device which can mix the air and fuel properly was developed. The air and fuel mixing device (venturi) was designed for the maximum load of operation. The diameter of the holes which are drilled on the venturi is designed with the help of mathematical modeling. In the current system, the venturi has three holes of 8mm diameter each connected with the producer gas supply.

Also, a fixture was designed so as to fix the speed of the engine when the load was increased. The fixture fixes the movement of rack of fuel injection system and the speed was fixed (V. Srivastava et al 2012).

Fig. 3 Air –Producer gas mixing device (venturi)

Fig 4. Fuel pump set at idle condition

3. Testing of IC engine

3.1 Trial on ICE with diesel as fuel

The ICE was first tested on diesel fuel mode. The performance of engine was tested on the loads of 0kW, 0.5 kW, 0.5 kW, 1 kW, 2 kW, 2.5 kW, and 3 kW. The various performance parameters such as brake power, fuel consumption, brake specific fuel consumption, volumetric efficiency, and brake thermal efficiency are calculated using mathematical modeling. Also, the exhaust gas analysis is done to check the emission opacity.

3.1 Trial on ICE with producer gas as a fuel (Dual fuel mode)

After connecting every component of the whole system, the gasifier's combustion zone was ignited. After 10-15 minutes, the gasifier started producing the producer gas. Then the gas which is raw and unfiltered was sent to the packed bed oil scrubber. The scrubbing unit need to be started as soon as the gasifier starts burning. After starting the scrubbing unit, the producer gas was filtered. The flow rate of producer gas can be controlled by the valves attached to the system. Then, the clean gas which was coming out from the scrubbing unit was sent to the engine.

Initially the engine started and kept for warm up for 10 minutes. Thereafter the movement of rack is locked in such a way that engine should have speed of 1500 RPM using the fixture as shown in fig. 4.

When the load increases, the engine speed decreases and to maintain the constant speed, the governor increases the supply of fuel. At that time, the producer gas was supplied to the engine. Accordingly, the different loads are applied and the experimental data is recorded.

Also, the exhaust gas analysis is done with the help of AVL smoke meter to measure the emission opacity.

Fig 4. The experimental setup

3.3 Calculations

The following models are used to calculate the performance parameters of the diesel engine (R. Sharma 1996)

3.3.1 Brake Power (kW)

$$B.P. = \frac{Electric\ Load}{Generator\ Transmission\ Efficiency} \quad (1)$$

3.3.2 Fuel Consumption (kg/hr)

$$F.C.\ (\dot{m}_{diesel}) = Volume\ flow\ rate\ of\ fuel * Density \quad (2)$$

$$F.C.(\dot{m}_{diesel}) = \frac{10*10^{-6}}{Time} * 3600 * \rho \quad (3)$$

3.3.3 Brake specific fuel consumption (kg/kWhr)

$$B.S.F.C. = \frac{mf\ (kg/hr)}{B.P.(kw)} \quad (4)$$

For dual fuel mode,

$$B.S.F.C. = \frac{\dot{m}(diesel)+\dot{m}(producer\ gas)}{B.P.} \quad (5)$$

3.3.4 Volumetric efficiency

$$Density\ of\ air\ at\ NTP\ (\rho_a) = \frac{P_\circ}{R*\tau_\circ} \quad (6)$$

$$Air\ head\ causing\ flow\ (H_a) = \frac{Hw*\rho w}{\rho a} \quad (7)$$

Actual volume flow rate of air at NTP

$$(V_m) = \frac{\pi}{4} * d^2 * cd\sqrt{2gHa}\ \ (m^3/sec) \quad (8)$$

$$Piston\ Displacement\ rate = \frac{\pi}{4} * D^2 * L * \frac{N}{2*60} \quad (9)$$

$$\eta_{vol} = \frac{Actual\ volume\ flow\ rate\ of\ air\ sucked\ at\ NTP}{Piston\ Displacement\ rate} \quad (10)$$

3.3.5 Brake thermal efficiency (η_{bther})

$$\eta_{bther} = \frac{Heat\ equivalent\ of\ brake\ power\ (kg/hr)}{Heat\ supply\ by\ the\ fuel\ (kg/hr)} \quad (11)$$

For dual fuel mode,

$$\eta_{bther} = \frac{W}{\dot{m}(diesel)*CV(diesel)+\dot{m}(PG)*CV(PG)} \quad (12)$$

3.3.6 Percentage of diesel replacement by producer gas

$$r = \frac{\dot{m}(diesel)-\dot{m}(dual)}{\dot{m}(diesel)} \quad (13)$$

4. Results and Discussion

The experiments were performed for both diesel fuel mode and dual fuel mode on the loads of 0 kW, 0.5 kW, 0.5 kW, 1 kW, 2 kW, 2.5 kW, and 3 kW. The results of trials are as follows.

4.1 Single Fuel Mode (Diesel)

Table 3 Results: Single fuel mode (Diesel)

Parameter	Load Applied (kW)					
	0	0.5	1	2	2.5	3
Brake power (kW)	-	0.625	1.25	2.5	3.125	3.75
Fuel consumption (kg/hr)	0.4526	0.4594	0.4809	0.57	0.884	0.7328
B.S.F.C. (kg/kWhr)	-	0.735	0.3847	0.228	0.2188	0.195
Volumetric efficiency (%)	79.89	80.04	80.31	81.07	81.66	82.01
Brake thermal efficiency (%)	-	11.45	21.29	35.93	37.42	41.92
Smoke density (%)	1	1.9	2.4	8.5	6	20.5

4.2 Dual Fuel Mode (Diesel +Producer Gas)

Table 4.Results: Dual Fuel Mode (Diesel+Producer Gas)

Parameter	Load Applied (kW)					
	0	0.5	1	2	2.5	3
Brake power (kW)	-	0.625	1.25	2.5	3.125	3.75
Fuel consumption (D+PG) (kg/hr)	0.8986	0.6125	0.604	0.739	0.8763	0.9526
B.S.F.C. (kg/kWhr)	-	9.255	5.296	3.76	3.28	3.12
Volumetric efficiency (%)	82.25	81.34	80.31	80.81	79.83	80.74
Brake thermal efficiency (%)	-	4.8	8.68	12.96	14.18	15.48
Smoke density (%)	2	2.2	2.75	4.98	7.8	14.5
Diesel replacement by P.G. (%)	-	66.67	74.4	70.3	71.88	70

4.3 Graphs

Graphs were plotted for brake thermal efficiency, volumetric efficiency, percentage of diesel replacement for diesel and dual fuel mode are as follows.

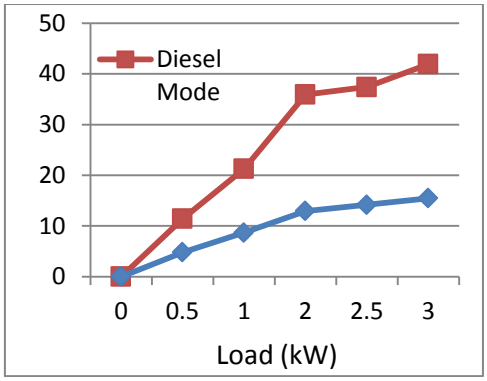

Fig.5 Load Vs Brake thermal efficiency

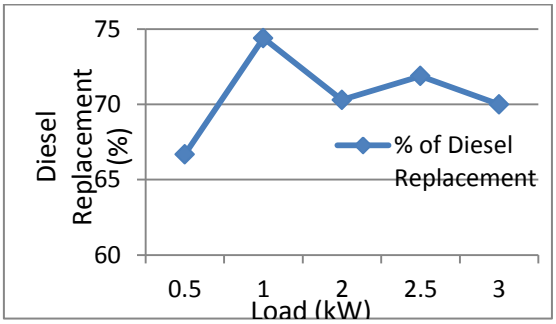

Fig.6 Load Vs Percentage of diesel replacement

4.4 Discussion

From the results, we can observe that the fuel replacement is achieved up to 74.40% on the load of 1kW. Also, it is observed that when we increase the load, the brake thermal efficiency of the engine also increases.

Conclusions

The experimental results show that the emissions in the dual fuel mode are lesser than the diesel mode for all loads. A drop in thermal efficiency was observed as 15.48% (dual fuel mode) compared to diesel mode as 41.92%. This is may be due to the lower calorific value of the producer gas and improper air fuel mixing. The smoke density was obtained as 14.5% on dual fuel mode while on the diesel mode, it was found to be 20.05. This means that, the smoke opacity percentage in the dual fuel mode decreases in the dual fuel mode. Also, the quality of gas might have reduced the thermal efficiency of the system. Higher temperatures were observed at exhaust side when the engine was run on dual fuel mode for all the loads. The BSFC value of the dual fuel mode was obtained as 3.12(kg/kWhr) and 0.195 (kg/kWhr) for diesel mode on 3kW load.
Hence, we can conclude that the producer gas can be a good alternative source of energy and has tremendous scope in future for the energy requirements. However, a lot of research is still need

to be done so as to increase the efficiency of the system and to enhance the performance of the system.

Nomenclature

$\dot{m}_{diesel}$ - Mass flow rate of diesel (kg/hr)
ρ - Density (kg/m^3)
P_0 - Pressure at NTP (kN/m^2)
R - Gas constant
T_0 -Temperature at NTP (°C)
Ha -Manometric pressure difference (m)
η_{bth} -Brake thermal efficiency
R -Percentage of diesel replacement by producer gas

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HP31703317-Experimental Investigation of single cylinder diesel engine operated on eucalyptus oil and cotton seed oil as biodiesel.

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Abstract

Biodiesel has obtained from vegetable oils that have been considered as gifted Non-conventional fuel. The researches regarding blend of diesel and single biodiesel have been done already. Very few works have been done with the combining of two different biodiesel blends with diesel and left a lot of scope in this area. The present study bring out an experiment of two bio-fuel from cotton seed oil and eucalyptus oil and they are blended with diesel at various mixing ratios B9%,B18%,B27%,B36%,B45%. The effect of duel biodiesel work in engine and exhaust emission were examined in a one, cylinder, 4- stroke multi fuel VCR engine at several loads with engine speed of 1500rpm. Result shows that hybrid blends of Cotton Seed Oil Methyl Ester(CSOME) and Eucalyptus Oil Methyl Ester biodiesel an its blends present significance improvement in of CO, unburned HC emission particularly at high loads with corresponding performance to those of bio-fuels, however NOx emission are marginally increases when the biodiesel compositions is improved. The word faces the disasters of energy demand, increasing petroleum prices and reduction of fossil fuel resources

Keywords— *Biodiesel, hybrid, Transterification, CSOME, EOME, performance, emission..*

1. Introduction

Constantly increasing fuel prise, constant increasing of on road vehicles, fast decreasing petroleum resources and continuing gathering greenhouse gases are the main recourses for the development of non-conventional fuels. Many non-conventional fuels are identified and tested successfully in the current engine with and without engine changes. Most of the substitute fuels identified today are bio-fuels and are having one or few objectionable fuel features which are not allowing them to replace the existing petro fuel completely. However, the various admission techniques experimented by the scholars are giving good solution to apply larger section of replacing fuel in the present engine. Biodiesel is may be a uncontaminated burning alternating fuel, made of domestic, renewable resources. It can be used in diesel Engines with little or no changes. Biodiesel is simple to use, decomposable, harmless, and basically free of sulphur and aromatics [1].

At present, India is manufacturing only 35% of the total gasoline fuels required. The residual 65% is being imported, which costs about Rs.80, 000 crores every year. It is an amazing fact that mixing of 10% bio-diesel and 20% fuel to the present diesel fuel is made available in our country, which can save about Rs.500 billion rupees every year. It is predictable that India can produce 390 metric tons of bio-fuel by the completion of 2017.The Government of India has launched a biodiesel project in 200 regions from 18 states in India. It has suggested two plant species, viz. Jatropha and Karanja for bio-diesel production. The current auto fuel strategy document states that bio-fuels are effective, sustainable and 100% natural energy unconventional to gasoline fuels [2]. Remodification technique is used to prepare biodiesel which is shows similar properties as diesel fuels [3]. Blends of Eucalyptus with diesel shows the brake thermal efficiency is improved about 6.2% for blend which has 60% of diesel and 40% of eucalyptus oil as compared to diesel fuel operations and 16.7%, 25% and 15.15% reduction of smoke, carbon Monoxide and Unburned hydrocarbons respectively as compared to Diesel fuel [4]. Kinematics viscosity of Cotton Seed Oil Methyl Ester (COME) is greater than those of diesel fuel the calorific value of COME lesser while its flash point is greater than diesel [5].C20 have closer performance to diesel. Also it is observed that CSOME gives better performance compared to Neem Oil Methyl Ester (NOME) and also the emissions for these diesel blends are fewer as relate to the pure diesel [6].The vegetable oils possesses almost the same heat values as that of diesel fuel. The power output and the fuel depletion of the vegetable oil and its mixture with diesel are almost the similar when the engine is powered with diesel [7]. Vegetable oils can be used as experimental fuel, which removes the use of petroleum diesel fuel [8]. The fuel consumption of rapeseed methyl ester was little higher than diesel fuel operation [9]. Most of the literatures suggested that cotton seed oil is a suitable substitute of diesel and a few research works have also been carried out with Eucalyptus oil. So, the Cotton seed oil and Eucalyptus oil were selected for this current study which is easily and locally available. As a first level of experimentation, the properties of above said fuels

in various combinations were found out in this work. This proved that the calorific value of the dual biodiesels and its combinations with diesel fuel is more than the single biodiesel and its blends with diesel fuel. Hence it is decided to select cotton seed oil and eucalyptus oil and diesel as the fuel for this current analysis. In the second level performance and emission features of a diesel engine with dual biodiesel and its blends and the results are compared with diesel.

2. Biodiesel preparation and its characteristics

A. Biodiesel preparation procedure

Vegetable oils are 3-glycerides of fatty acids and alcohol esters of fatty acids have been prepared by the transesterification of the glycerides, wherever in linear, monohydroxy alcohols reacts with the edible fats in the presence of catalyst to produce alcohol esters of vegetable oil. Transesterification is the process of cutting down heavier molecules into lighter ones in this process of preparing eucalyptus biodiesel ethyl ester was employed ethanol was used as alcohol in this process. In first stage pure eucalyptus oil was taken in a beaker the capacity was 500ml pure eucalyptus oil 100 ml ethanol and 5 grams of NaOH Flakes. The eucalyptus oil in the beaker was heated around 500^0C-600^0C in a burner the oil was stirred thoroughly. While the oil is heated at one side on the other hand simultaneously the NaOH flakes is made to dissolve completely and mixed with ethanol to form sodium ethoxide solution. After heating the oil for an hour the sodium ethoxide is poured into the heated oil and stirred for an another hour maintain the temperature for around 500^0C-600^0C while stirring the colour transformation took place from yellow to light red in colour after that the entire solution from the beaker is now poured into a separation flask for allowing the glycerol to separate from biodiesel it took nearly 24 hrs. for the glycerol to get separated. After separating the glycerol that observed in black colour was isolated from the biodiesel the extracted biodiesel is heated for around one hour to remove any untreated ethanol and the biodiesel was washed with 15% fermentation alcohol to eliminate impurities. The cleaned biodiesel thus obtained was the ethyl ester of Eucalyptus oil, which is known as Eucalyptus biodiesel. Same procedure followed for preparing CSOME.

2.1 Materials and methods

The various properties like viscosity, calorific value, flash point temperature and fire point Temperature of baseline fuel, raw oils and two biodiesel mixed blends were found by using ASTM methods and compared with diesel properties. The experiments were conducted on a 1- cylinder 4- stroke air cooled diesel engine with electrical loading dynamometer and the performance and emission features were matched with reference point data of diesel fuel. experiments were conducted at a constant speed and at varying loads for all dual biodiesel blends. Engine speed was retained at 1500 rpm during all experimentations. Fuel depletion and exhaust gas heats were also measured. The smoke capability of the exhaust gases was measured by the AVL make smoke meter. Gas Analyzer is used to measure exhaust gases. The experimental set up is presented in Fig.4 and the detailed engine specifications are also mention in Table 4.

Table 1 Blending of CSOME and EOME

Sr. No.	Notation	Bio-Diesel Qty (%)		Diesel Qty (%)	Fuel Qty (%)
		CSOME	EOME		
1	B9	4.5	4.5	9	100
2	B18	9.0	9.0	82	100
3	B27	13.5	13.5	73	100
4	B36	18	18	64	100
5	B45	22.5	22.5	55	100
6	B54	27	27	46	100
7	B100	50	50	----	----

3. Result and discussion

Various physical and thermal properties of dual biodiesels of Cotton seed oil and Eucalyptus oil and its blends were evaluated in the laboratory of Indian Biodiesel Corporation.

Table 2 Property of raw Cotton seed and Eucalyptus

Sr. No.	Property	Diesel ASTM	Cotton Seed	Eucalyptus
1	Viscosity Cst	2-4	55.6	1.6-2.1
2	Density Kg/m3	838	912	913
3	Heating value KJ/Kg	42,700	39,500	43,270
4	Flash Point 0C	76	205	54

Table 3 Property of Tested Blends

Sr. No.	Property	B9	B18	B27	B36	B45
1	Viscosity Cst	3.5	3.8	4.1	4.6	4.9
2	Density Kg/m3	935	839	842	844	846
3	Calorific value MJ/Kg	42.3	41.9	41.79	41.5	41.22

3.1 Calorific value of fuel.

The digital bomb calorimeter is used to find out the calorific value of fuels. ASTM D6751 procedure is followed to analyze the calorific value of different test fuels. Fig.1 shows the calorific value of different fuels. The fresh vegetable oil has lower heating value than diesel. After transesterification process, the biodiesels have slightly higher calorific value than raw oil. The EOME has higher calorific value than CSOME by blending the dual biodiesels with diesel; the calorific values of Blend B9 and Blend B18 are close to diesel which is more than single biodiesel blends. The calorific values of Blend B27, Blend B36 and Blend B45 are almost equal to the single biodiesel blends. The heating value of Blend B100 is lesser than the single biodiesel blends due to the presence of pure biodiesel blends without diesel. Hence, dual biofuel and its mixture are used to analyse the performance and emission features of biodiesel.

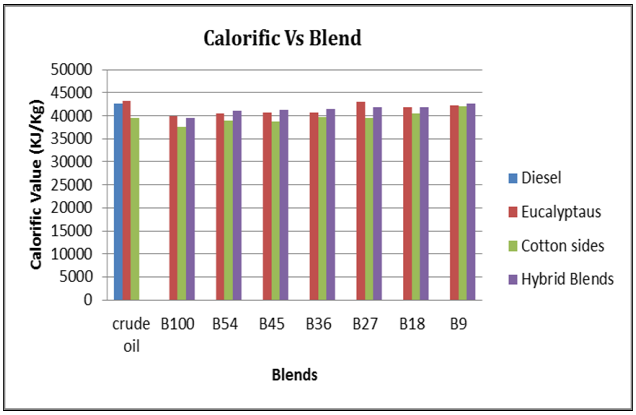

Fig. 1 Comparision of calorific value of various blends with diesel.

3.2 Viscosity of fuel.

Calibrated Redwood viscometer is used for determining the Kinematic viscosity ASTM D 0445 procedure is followed to analyze the viscosity of fuels. Fig.2 shows the viscosity of different fuels. The viscosity of the blends rises with the blend ratio and the viscosities of dual biofuel blends and they are progressive than diesel fuel. As compared to diesel fuel viscosity of cotton seed oil is higher but viscosity of Eucalyptus is low compare to diesel..

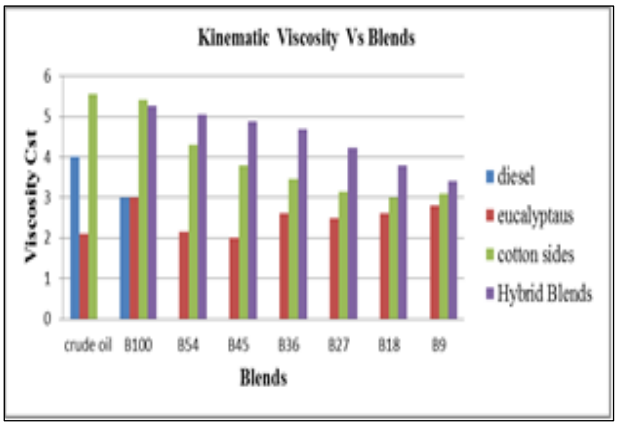

Fig. 2 Comparison of viscosity of various blends with diesel fuel.

Blends B9,B18 shows viscosity closer to diesel fuel. The viscosity of diesel is 4 Cs whereas for the Blend B9 and Blend B18 it is 3.8 Cs and 3.4 Cs respectively.

3.3 Density of fuel.

Density of different blends is measured using a Precision hydrometer. The density of dual biodiesel blends Blend B9, Blend B18 and Blend B27 is 835 Kg/m^3, 839 Kg/m^{3}and 842 Kg/m^{3}respectively whereas for diesel it is 832Kg/m^{3}The other blends have more deviation than diesel. Increase in the fuel density advances the dynamic injection timing by 1° Crank angle thus, the fuel density affect engine combustion and emission. PM emission generally increase with increase in fuel density.

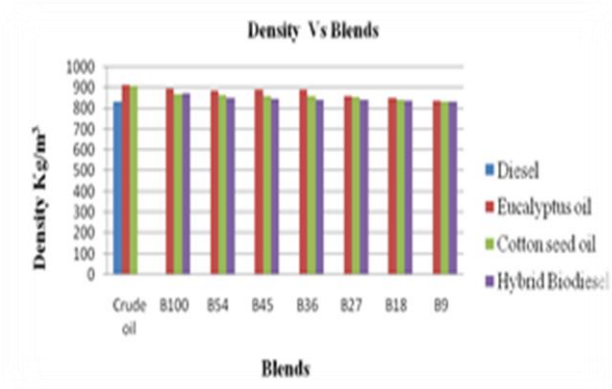

Fig. 3 Comparison of density of various blends with diesel.

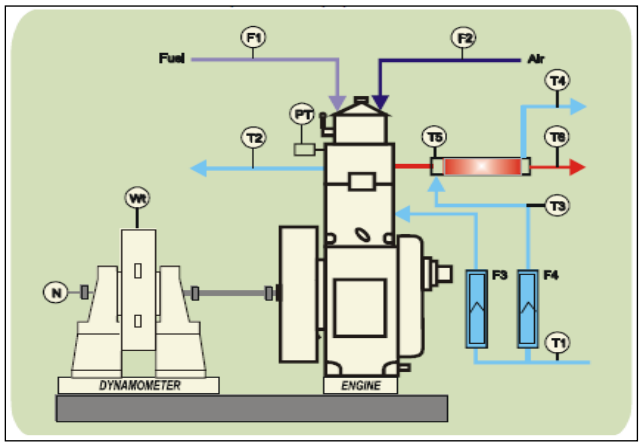

Fig. 4. Single cylinder, four stroke, Multi-fuel VCR engine.

F_1: Fuel in F_2:- Air in

P_T: Petro Diesel Tank T_1:-Temperature in ^{0}C

T_2: Temperature out ^{0}C T_3: Calorimeter water in ^{o}C

T_4: Calorimeter water out^{o}CT_5: Exhaust Temp engine ^{o}C

T_6: Exhaust temp cal. ^{o}C

Table 4 Engine Specifications

Manufacturing	Kirloskar engine Ltd. Pune, India
Engine type	Single cylinder, Four stroke, constant speed, multi-fuel VCR engine.
power	3.5Kw
Bore diameter	87.5mm
Stroke length	110mm
Capacity	661 cc
Compression	12:1-18:1
Injection variation	0-25deg BTDC
Cooling system	Water cooled
Dynamometer	water cooled, Eddy current with loading unit

3.4 Performance analysis.

- Brake Specific Fuel Consumption (BSFC)

The fuel burning characteristics of an engine are usually expressed in terms of specific fuel consumption in Kg/Kwhr. Fig.5 shows the BSFC of the engine with CSOME-EOME (B36,B45) is higher when compared to B9, B18, B27 and diesel at given loads. Due to lower calorific value and higher density and viscosity of cotton seed oil methyl ester the BSFC's reduces from 0.46,0.44,0.42,0.39,0.38 and 0.36 Kg/Kwh for BD45,BD36,BD27,BD18,BD9 biofuels respectively. The major reason is, increase in BSFC with increase in fuel blends is the additional depletion of biodiesel fuel by the test engine in order to retain continuous power output.

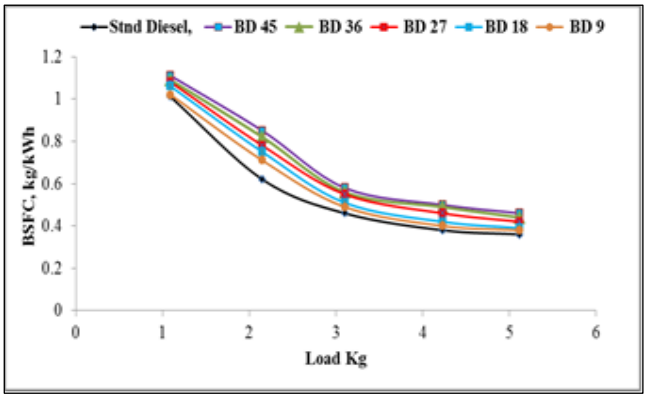

Fig.5 Variations of BSFC Vs Load

- Brake Thermal Efficiency(BTHE)

Brake Thermal Efficiency (BTHE) is the break power of a engine as a function of the heat input from the fuel. It is used to estimate how well an engine changes the heat from a fuel to mechanical energy. As shown in fig.6 BTHE increases with a rising in engine load as the quantity of diesel in the blends rises. The BTHE of the B27 blend was improved than B9, B18, B36, B45 blends, which is near to diesel. Due to a faster burning of biodiesel in the blend, the BTHE enriched. The value is 24.06% as against 29.01% for diesel at 100% load.

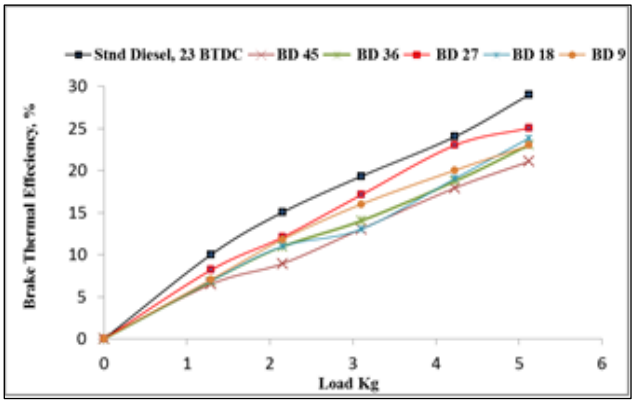

Fig .6 Variations of BTHE Vs Load

- Mechanical Efficiency

The influence of load on mechanical efficiency is revealed in Fig. 7 Indicated power and engine friction is essential for calculating the mechanical effectiveness of the I.C .engine. Efficiency is calculated as a ratio of the measured performance to the performance of an ideal engine. Mechanical

efficiency measures the use fullness of an engine in changing the energy and power that is given as an input to the engine into an output force and movement. Hence, mechanical efficiency indicates how good an engine is, in converting the indicated power to useful power. Blend B36 gives the maximum mechanical efficiency of 42.3% for the maximum load, whereas the diesel gives 40.2% at the same load. For the other blends, mechanical efficiency is lower than diesel fuel.

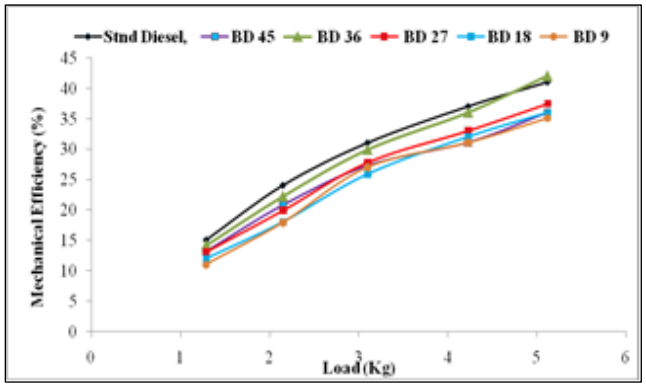

Fig.7 Mechanical efficiency Vs Load

- Effect of Exhaust gas temperature (EGT)

Fig. 8 indications that Exhaust Gas Temperature(EGT) increases with increase in load at wholly injection timing with biodiesel and diesel. It was found that, exhaust gas temperature (EGT) was higher for BD27, and BD45. This may be due poorer cetane number and greater ignition delay of the mixture As a consequence; there is increase in NOx emission with small drop. At lower and higher load exhaust temperature of B18 blend closer to diesel.

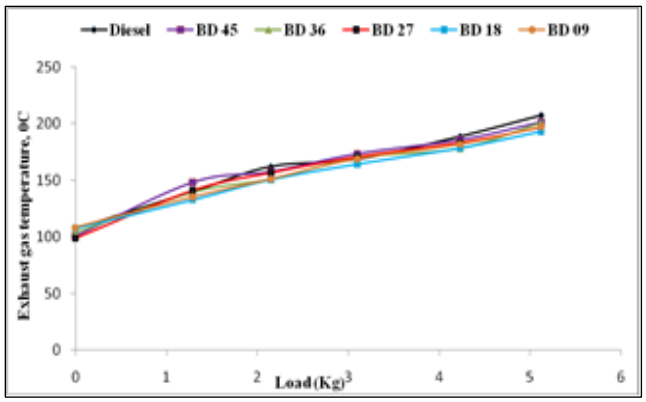

Fig 8 Variation of Exhaust Gas Temperature with Load (EGT)

3.5 Emission Characteristics of biodiesel

- Carbon Monoxides (CO) Emission.

The fig. 9 shows the variations of CO emissions with CSOME-EOME blends. Carbon monoxides not changes for small and medium loads as compare to diesel fuel fuel.CO emission increases as load on engine increases. This may be due to the improvement of O_2 in the EOME. There was a 21% decrease of Carbon Monoxides emission for the B36 blend at maximum load.

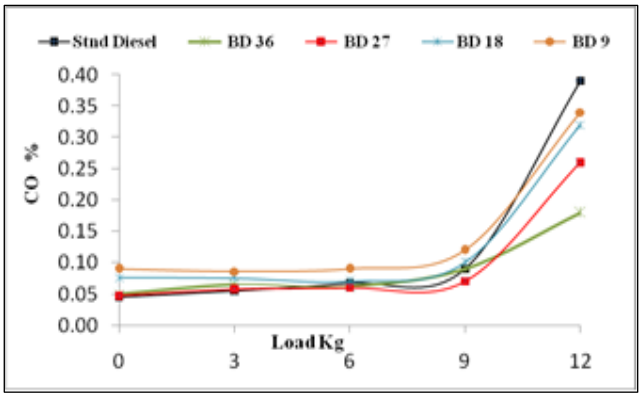

Fig 9 Variation of CO emission with Load

- Unburned Hydrocarbon (HC) Emissions.

Fig. 10 shows variations of hydrocarbon emissions with loads. for diesel fuel HC emission 46ppm at low load and 110ppm at maximum load and for B36 as load increases HC emission increases from 32 ppm to 55 ppm. For CSOME-EOME compositions, the hydrocarbon emissions are lesser than that of diesel, and this may be due to complete burning. The engine operated with methyl ester inside the combustion chamber there are some areas where mixture is too rich and it cannot burn completely. Those un-burnt species are known HC emissions. As the ignition delay period growths, due to a decrease in the fuel CN, flame propagation can reach at each particles of fuel. This may be the reason for the decrease in hydrocarbon emission for blends than the diesel fuel operation.

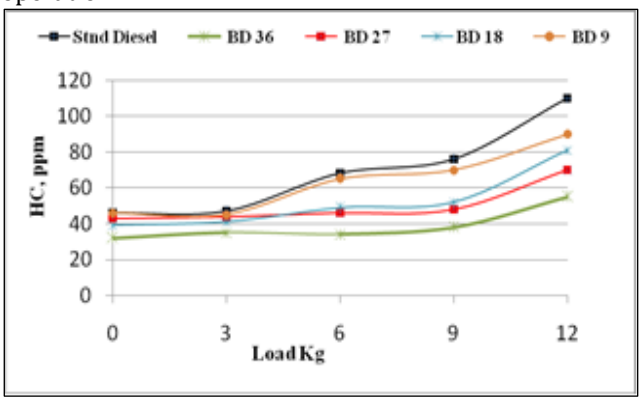

Fig.10 Variations of hydrocarbon emissions with load

- Oxides of Nitrogen (NOx) Emission

Fig. 11 shows that the deviation of NOx emission for CSOME-EOME blends as engine load changes. The increase in NOx may be due to the presence of O_2 in both of CSOME and EOME. Normally NOx formations

occurs due to increase in engine temperature during burning process of fuel. For B36 blend, the NOx emission was 110ppm compare to 775 ppm of diesel fuel.

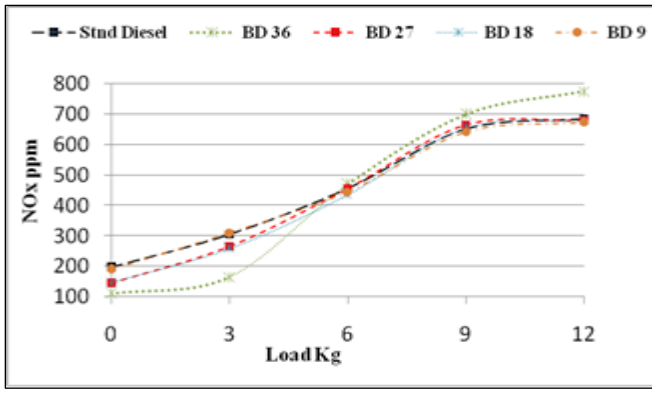

Fig. 11 Load Vs Nox Emissions

Conclusions

Single cylinder high speed diesel engine ran successfully during tests on dual biodiesels and its blends. The blends of diesel and the dual biodiesels of Cotton seed oil and Eucalyptus oil were characterized for their various physical, chemical and thermal properties. The experimental conclusions of this investigation can be summarizing as followed.

- Properties of the 27%, 36% blend of CSOME and EOME are closer to the Diesel.
- The Brake thermal efficiency (BTHE) and mechanical efficiency of Blend B36 were considerably greater than the diesel.
- Blend B18 and Blend B27 were very nearer to the diesel values.
- The specific fuel consumption (BSFC) values of dual biodiesel blends were comparable to diesel.
- Blend B27 and Blend B36 produced slightly lower CO and CO2 than diesel.
- This is a considerable advantage over diesel while using the dual biodiesel blends.
- The diesel fuel gave higher HC emission than that of dual bio-fuel mixture;.Also as proportion of blend increases NOx increases.

Therefore, it may be concluded that dual biodiesel blends of Blend 36 and Blend B27 would be used as an auxiliary fuel for diesel in the CI engines. Various dual biodiesel Blends with diesel can be focused for further recommendations.

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HP31703502-Performance and Emission Analysis of Single Cylinder Diesel Engine by Using Jatropha Methyl Ester and Undi (Calophyllum Methyl Ester)

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Abstract

Now a day's, different nation facing the problem of shortage of energy. And different liquid fuels like kerosene, petrol, diesel is mostly used in industries, transport, agriculture, commercial and domestic sector. Running down of energy resources and environment awareness need to go towards renewable energy resources which is more reasonably priced and environmentally acceptable. Among many fuels, biodiesel considered are more desirable fuel. The world is facing a problem with the twin crises of fuel depletion and environmental degradation. Alternative fuel, conservation of energy and environment protection have become important in recent years.
The properties like flash point, fire point, cetane value, kinematic viscosity, specific gravity, calorific value were found at laboratory. To find the performance characteristic like brake thermal efficiency of single cylinder diesel engine by using diesel, blended Jatropha and blended Undi oil. Also emission of CO, HC, NOx ,smoke opacity is find out. Out of NOx is find 28% higher for Jatropha and 30% higher for Undi than Diesel.

Keywords: *Alternative Fuel, Undi, Jatropha, VCR Diesel Engine.*

1. Introduction

An invention of I.C. The engine has extremely increased energy demand. And this results deflection of petroleum reservoir across the world rapid rate as well as more combustion of fuel, the results rate of increase pollution and it's harmful to human and environment. So it needs to go towards substitute fuel like methane (CH_4), hydrogen (H_2) and vegetable oil. Again, it is found that biodiesel i.e. vegetable oil promise in these regards.

Since it can be produced from plants grounded in rural area and coastal area. How these are many varieties of vegetable oil evaluated in many parts of the world, but only a few vegetables like Karanja, Jatropla and Und ioil can be consid ered is economical specially in India. But the use of direct end vegetable oil is restricted due to some of the Characteristic, density, viscosity, poor fuel atomization, carbon deposited, poor durability and poor thermal efficiency. This problem associated with direct vegetable oil is removed by using transesterification to produce biodiesel and is playing an important role to reduce viscosity of oil. [1]

Biodiesel is referred mono-alkyl ester of long fatty acid. A large number of experiments are carried on a single cylinder diesel engine by unit standard diesel and blend of Undi i.e. calophyllum inophyllum Linn oil 25,50,75,100 as well as the blends of Jatropha 25,50,75,100. This study target to compare Jatropha and Undi oil in diesel engine characteristics is evaluated by using different blend.

2. Literature Review

Yamin et al. have studied biodiesel testing using four stroke, four cylinder diesel engine, which showed that the brake power and brake specific fuel consumption of biodiesel is slightly higher than that of normal diesel for both full and medium throat all over speed range.[1]

Singh et al. have tested a modified four stroke C.I. engine on pure biodiesel made from jatropha curcas oil for engine performance analysis, which may find large application in irrigation sector. By reducing viscosity of jatropha curcas oil, it will make close to that of conventional fuel so that it may find suitable use in C.I. engine for transportation and irrigation for the rural mass.[2]

Raju et al. have carried out an experimental investigation on a single cylinder variable compression ratio C.I. engine using pure Mahua oil as a fuel. Performance characteristics like brake thermal efficiency, exhaust gas temperature, fuel consumption and exhaust analysis were carried out to find the best suited compression ratio.[3]

B.K. Venkanna et al. A direct injection diesel engine typically used in agriculture sector was operated on neat diesel and H100. Injector opening pressure was changed to study the performance, emission and combustion characteristics. It was observed that increasing IOP with H100 from rated injector opening pressure increased the brake thermal efficiency and

reduced CO,HC and smoke opacity emission. However, NOx emission was increased.[4]

3. Material and Method

The Jatropa curcase oil, Undi i.e. calophyllum inophyllum Linn oil is prepared by transesterification process and commercially available diesel.

3.1 Transesterification

Transesterification is a chemical process which converts triglycerides is a converted into mono alkyl ester of fatty acid and it is called as alcoholics. In this reaction fat or oil with alcohol convert biodiesel and glycerin. Catalyst is used in this reaction to faster the rate.

The seeds of Undi i.e. calophyllum inophyllum Linn and seeds of Jatropha were collected, cleaned and dried. The seeds are ground to convent fine powder by heavy duty electric motor. Then seeds powder mixed with a mixture of methanol and ethanol. The transesterificatiom with continuous stirring was carried at 400 RPM oscillation. Heat is given by hot plate to keeping 60^0C for about 60 minutes. The solid cake and mother liquor were separated by vacuum filtration A rotary evaporator was used to separated solvent. The oil fraction separator is about 800C and it is preserved in airtight container for further used and analysis purpose.

Transesterification of Undi and Jatropa was blended with diesel oil in varying proportion like 25, 50,75,100% it also reduces viscosity. Their blends as very stable under normal condition biodiesel oil preparation. [5]

3.2 Material Properties

Fuel properties of diesel and blended of Jatropa curcase L. Oil and Undi i.e. calophyllum inophyllum Linn oil are listed below. It has found out at laboratory of IBDC baramati. [5]

Table. 1*Material Properties*

Sr. No.	Test Description	Ref. STD. ASTM 6751-	Reference		Diesel	Jatropha methyl ester				Calophyllum methyl ester			
			Unit	Limit	B00	B25	B50	B75	B100	B25	B50	B75	B100
1	Density	D 1448-1972	kg/m3	850-900	830	835	846	858	871	832	845	856	868
2	Viscosity	D 445-73	Cst	3.0-6.1	2.9	3.6	4.1	4.6	5.9	3.2	3.8	4.56	5.7
3	Cetane Number	D 613	-	41-55	51	51.3	51.4	52.4	52.91	51.2	51.4	52.3	53
4	Calorific value	D 6751	MJ/kg	34-45	42.5	41.3	41.18	40.1	38.55	41.4	41.16	40	39.17
5	Ash Content	D 482	-	0.1 max	0.10	NA	NA	NA	0.51	NA	NA	NA	0.51
6	Flash point	D 93	-	120-170	65	76	108	130	147	74	105	124	138
7	Fire point	D 93	-	130-185	69	82	114	137	152	78	111	129	145
8	Acid Value	EN 14101	mgKOH	0-0.1	NA	NA	NA	NA	0.75	NA	NA	NA	0.75
9	Moisture	D	% vol.	0.05	NA	NA	NA	NA	NA	NA	NA	NA	NA

		2709		max									

3.3 Experimental Setup and Test Procedure:

A single cylinder, four stroke diesel engines, bore diameter 80 mm, rated power developed 3.7 kW at 1500 RPM, compression ratio 5 to 20, stroke length 110 mm, cylinder diameter of 87.5mm, connecting rod length 150 mm, compression ratio 12:18, water cooled. Indus five gas analyzer was used for measurement of carbon monoxide, CO, Hydrocarbon HC, NOx emission. The engine was operated on baseline diesel first, then on a blend of Jatropa curcase L. Oil and Undi i.e. calophyllum inophyllum Linn oil. The different fuel blend and diesel are subjected to performance and emission test on an engine with a different load range and compression ratio. Then performance data are analyzed from the graphs regarding brake thermal efficiency, brake specific fuel consumption, emission etc. [6]

4. Result and Discussion

A number of engine test is carried out by using standard diesel and different blends of Jatropa curcase L. oil and Undi i.e. calophyllum inophyllum linn oil like BJ25, BJ50, BJ75, BJ100, BU25, BU50, BU75 and BU100. Performance and emission data is carried out by different blend of biodiesel by Jatropa curcase L. Oil and Undi i.e. calophyllum inophyllum Linn oil and is compared with baseline data of standard diesel which shown below. [7]

4.1 Brake Thermal Efficiency

Brake thermal efficiency (BTE) is the most important parameter which denotes how much percentage of energy present in fuel and which can be converted into useful work. The comparative analysis of BTE of various blended of Jatropa curcase L. Oil and Undi i.e. calophyllum inophyllum Linn oil with diesel (BJ25, BJ50, BJ75, BJ100, BU25, BU50, BU75 and BU100) standard baseline diesel is shown in the fig. 1. BTE of Jatropa curcase L. Oil and Undi i.e. calophyllum inophyllum Linn oil are lower than diesel for whole load range. The decreasing trends of efficiency as increase concentration of blends due to lower calorific value of biodiesel. It may also cause by its poor atomization because of high viscosity.[7]

4.2 Brake Specific Fuel Consumption

The break specific fuel consumption is not considerable parameter due to different blends are having calorific value and densities are different.

3.2 Unburned Hydrocarbon HC Emission

Unburned HC emission is one of the important parameter to determine the behavior of the engine. The biodiesel blend gives relatively Low HC as compared to the diesel. This is because better combustion of biodiesel inside the combustion chamber due to availability of excess content of oxygen in biodiesel as compare to diesel as shown in fig. 2. Also observed that HC emission of the various blends is low at partial load and increased at high load. This is due to the availability of less oxygen for the reaction when more fuel is injected into the engine at higher load.[7]

4.3 Carbon-monoxide, CO Emission

Carbon monoxide emission occurs due to the incomplete combustion of fuel. Biodiesel blends give less CO due to complete combustion. When the percentage of blends of Jatropa curcase L. Oil and Undi i.e. calophyllum inophyllum Linn oil increases, emission of CO decreases as shown in fig. 3. Co emission is toxic. The more amount of oxygen present in biodiesel results in complete combustion of fuel and it will supply necessary oxygen convert CO to CO2.[7]

4.4 Emission of NOx

NOx is one of the important emission factors because it creates problems such as cough, asthma, etc. NOxemission is more for biodiesel as compare with diesel because of more oxygen is present and it will faster the rate of reaction and hence the temperature of combustion is reached high. NOx emission is high at high output conditions in both of Jatropa curcase L. Oil, Undi i.e. calophyllum inophyllum Linn oil and diesel is shown in fig. 4. [8]

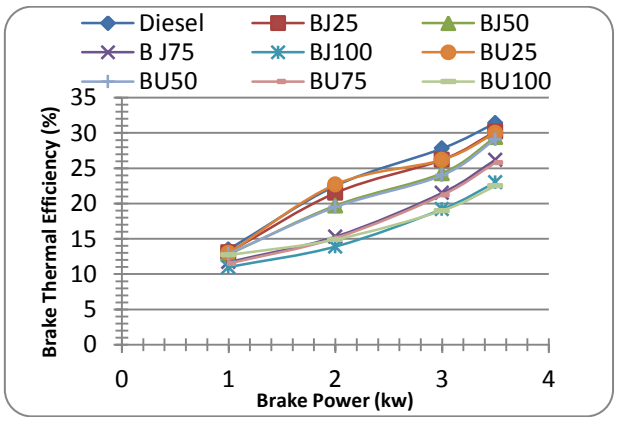

Fig. 1 *Variations of BTE with brake power*

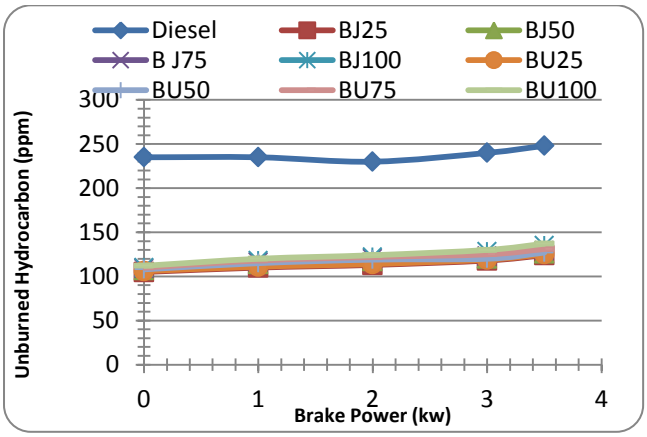

Fig. 2.*Variation of Unburned Hydrocarbon with brake power*

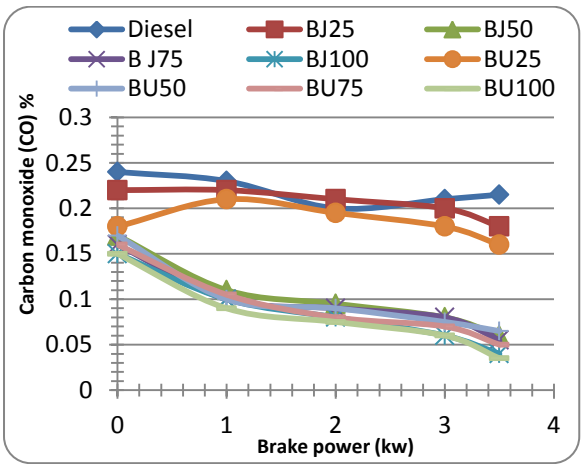

Fig. 3*Variability of Carbon monoxide with brake power*

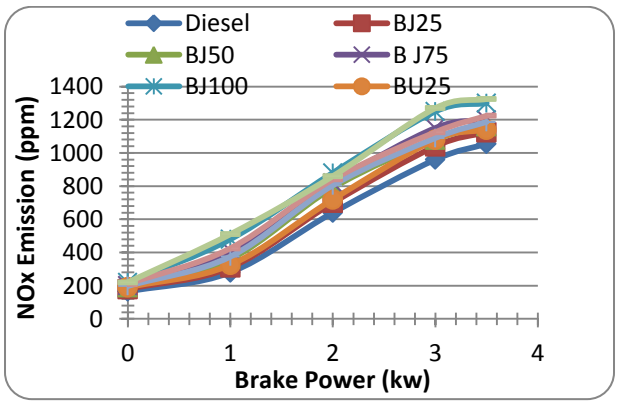

Fig. 4*Variation of Oxide of Nitrogen (NOx) with brake power.*

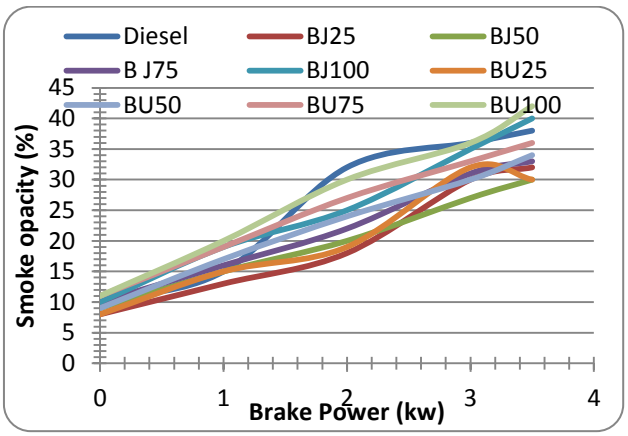

Fig. 5*Variation of Smoke opacity with brake power*

3.6 Emission of Opacity

Emission opacity means smoke is produced due to incomplete combustion of fuel. In case of Jatropa curcase L. Oil and Undi i.e. calophyllum inophyllum Linn oil smoke produced less as compared to diesel due to complete combustion because more oxygen is present. Now, as thepercentage of blends increase from B15 to B100 for both of Jatropa curcase L. Oil and Undi i.e. calophyllum inophyllum Linn oil, smoke density are reduced. Variation of smoke emission with brake power is shown in fig. 5.

4. Conclusion

The performance of the engine and emission characteristic of Jatropa curcase L. Oil and Undi i.e. calophyllum inophyllum Linn oil are analyzing with standard baseline diesel. It is found that, brake thermal efficiency is 22% lower for BJ50 and 24% lower for KU50 of Jatropa curcase L.Oil and Undi i.e. calophyllum inophyllum Linn oil as compared with diesel. Co, HC, and smoke emission are also lower for biodiesel as compare with diesel. An emission of NOx is 28% higher for Jatropa curcase L. Oil and 30% for Undi i.e. calophyllum inophyllum Linn oil than standard baseline diesel.

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HP31703702-Performance & emission analysis of VCR engine at different compression ratios by using diesel &Babassu Biodiesel blends.

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Abstract

Abstract- In today's scenario, we are facing huge problem of pollution due to diesel cars, transport vehicles and industries. Petroleum resources are finite and almost 90% of energy needs of the world are provided by fossil fuels which are depleting at an alarming. Apart from the depleting resources, of petroleum, another important aspect of their use is in the alarming rise of pollutants like CO, HC, NOX, CO2 etc. We need to find out alternative sources of diesel which should be affordable, having good performance and less polluting. Biodiesel is typically made by chemically reacting lipids (e.g. vegetable oil, soybean oil, animal fats with an alcohol producing fatty acid esters. Biodiesel can be used alone, or blended with petro diesel in any proportions. In this context Babassu oil is used as an alternative and potential feed stock to overcome the environmental problems.

Keywords:*Babassu oil biodiesel, Transesterification, VCR engine, Performance and emission testing.*

1. Introduction

Due to very high pollution from conventional energy sources, it's time to think about new sources which give us comparative performance as well as less pollution. With the intensifying search for a sufficient alternative to oil-based energy, the development of energy sources becomes increasingly relevant. Most of the pollution is happened due to diesel engines so we need to do analysis on diesel engines with blends of biodiesel. Additive treatment of biodiesel experienced the controversy of its suitability in terms of performance, combustion and emissions. Hence the aim of proposed project is to check the biodiesel and diesel mixture with different combinations for performance, combustion and emission. Although several oil bearing trees like Karanja, polang, Kusum, Neem, Simarauba, Sal, Linseed, Castor, Jatropa, etc are native to India. Babassu oil is chosen for the present work of experimental investigation of performance and emission characteristics on VCR diesel engine. Babassu oil is extracted from Orbinyamartiana, a tree whose coconuts contains in average 7wt. % of almost 62 wt. % of oil. Babassu is a palm native to the Amazon Rainforest region in south-America, north-eastern Brazil and South India. The kernels of its hard shelled nuts are the source of Babassu oil. The main objective of this work is to analyze the engine performance and emission characteristics of diesel engines fuelled with biodiesel produced from Babassu oil and /or its blends with diesel and blends with additives which will helps in both the direction of controlling emission problems and search of alternative fuel for

diesel engine. However there is major disadvantage in the use of biodiesel as it has lower heating value, higher density and higher viscosity, higher fuel consumption and higher NOX emission, which limits its application. Here fuel additives become essential and indispensable tool not only to minimize these drawbacks but also generate specified products to meet the regional and international standards. A variety of additives are used for ex-metal based additives, oxygenated additives, antioxidants, cetane number improvers, lubricity improvers and cold flow improvers.

2. Materials and Methodology:

2.1 Materials:-

VCR engine, Babassu oil from Aromex industry (Mumbai), Biodiesel with antioxidant (Propyl-galate)

Methodology-

Available oil has following properties:

Oil Name- Babassu Oil

Table1. Properties of Babassu oil

Sr. No.	Property	Value
1	Density	930 Kg/m3
2	Flash Point	260 deg C
3	Fire Point	265 Deg C
4	Viscosity	36 cst
5	Pour Point	20 deg C
6	Water content	0.02%
7	Acid number	1.34 mg KOH/g

2.2 Preparation of Biodiesel:

The Transesterification is also known as alcoholysis. It is the reaction of oil with an alcohol to form ester and glycerin. Transesterification of Babassu oils can be done in a simple process. However, the process conditions must be carefully controlled to achieve optimal yield at the optimal temperature and reaction time

1. Take oil in beaker and stirred & heat it on magnetic stirrer and maintain it to boiling point of methanol (64.7 ^{0}C)
2. Add methanol in amount of 3 to 15 equivalents of oil
3. Then add KOH as a catalyst in amount of 0.1 to 1 wt%
4. The reaction mixture is then heated to the boiling temperature of methanol and refluxed for a certain length of time under agitation.
5. After stoppage of agitation the reaction mixture will separate into an upper layer of methyl esters and a lower layer of glycerol diluted with un-reacted methanol.
6. Upper layer fatty acid is then neutralized and vacuum distilled for removal of excess methanol before use as fuel.

Table 2. Experimental Set up-VCR Engine

Sr No.	Description	Specification
1	Make	Kirloskar,1 cylinder
2	Bore	87.5mm
3	Stroke	110mm
4	RPM	1500
5	Brake Power	3.5 kW
6	Compression ratio	16,17,18
7	Fuel oil	Diesel
8	Fuel flow Transmitter	DP transmitter, Range 0-500 mm WC
9	Air flow transmitter	Pressure transmitter, Range (-) 250 mm WC
10	Piezo sensor	Range 5000 PSI, with low noise cable
11	Software	Engine soft LV" Engine performance analysis software

One of the major drawbacks of biodiesel is that it can be more susceptible to oxidation at room temperature .It is due to chemical structure of fatty acid methyl ester .There is change in composition of biodiesel and a change in Physico-chemical properties of biodiesel such as increase in acid value ,viscosity, and flash point due to oxidation. To inhibit fuel degradation and increase storage life antioxidant used, such as Propyl-galate (PrG) id added to biodiesel and stored at room temperature for 15 days.

Experiment is carried out initially using neat diesel fuel to generate the base line data denoted by B00. After recording the base line data, tests are carried out using 20% and 40 % Babassu biodiesel blends which are denoted by B20 and B40 respectively. The engine tests are conducted at various loads and the parameters related to performance and emission characteristics are recorded. After taking readings of biodiesel blends we also test the biodiesel with additive Propyl-galate in proportion of 1ppm.

3. Result and Discussion:

Table 3. Babassu oil Biodiesel properties

Sr. No.	Property	Value
1	Density	986 Kg/m3
2	Flash Point	132 deg C
3	Fire Point	135 Deg C
4	Viscosity	4.46 cst
5	Pour Point	12 deg C
6	Water content	0.0015%
7	Calorific value	9124 cal/gm

Engine performance –The performance parameters such as BP, BSFC, BTHE, Volumetric Efficiency Obtained with B00, B20, B40 are found to be affected by fuel blends and engine loading at different Cr i.e. Cr16, Cr17& cr18 are discussed in the following section-

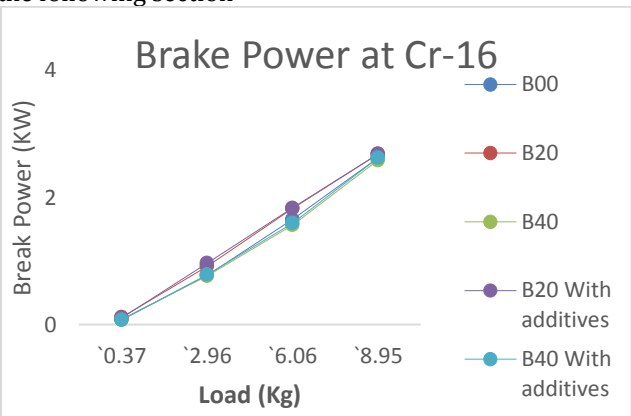

Fig.1.1.Brake power Cr16 values at various load

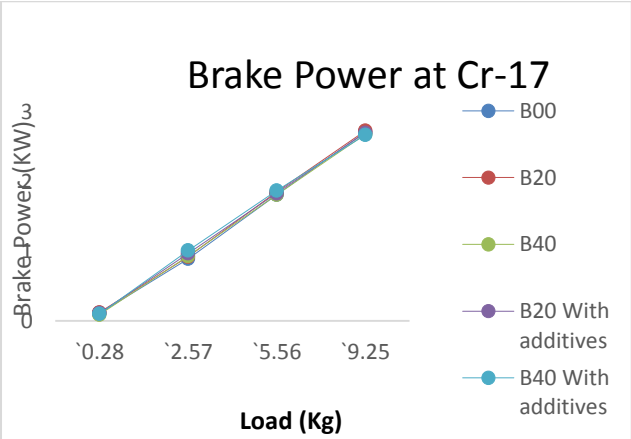

Fig.1.2. Brake power Cr 17 values at various loads

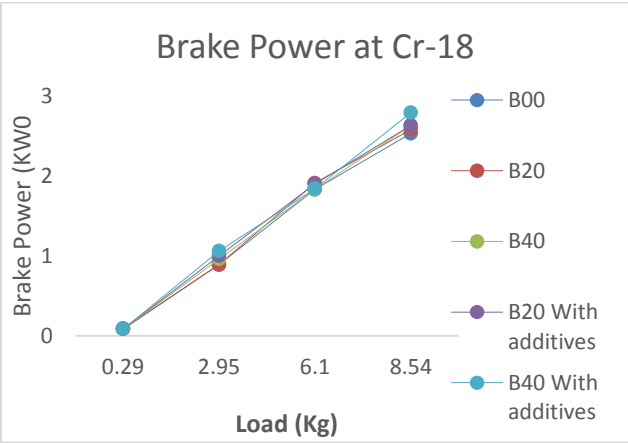

Fig.1.3.Brake power Cr 18 values at various loads

As the graph Load vs. Brake power at different compression ratio shows that as we increase the load at any particular ratio then brake power increases, from above graphs it can be said that brake power of B20 is improved as compared to pure diesel (B00) .It is also observed that optimal value of brake power occurs at compression ratio 17 for blend B20.

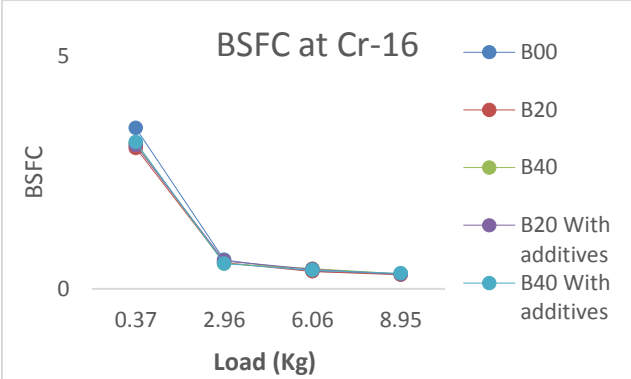

Fig.2.1.BSFC Cr 16 values at various load

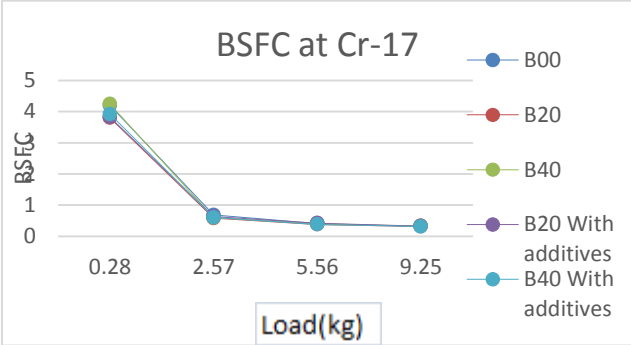

Fig.2.2.BSFC Cr 17 values at various load

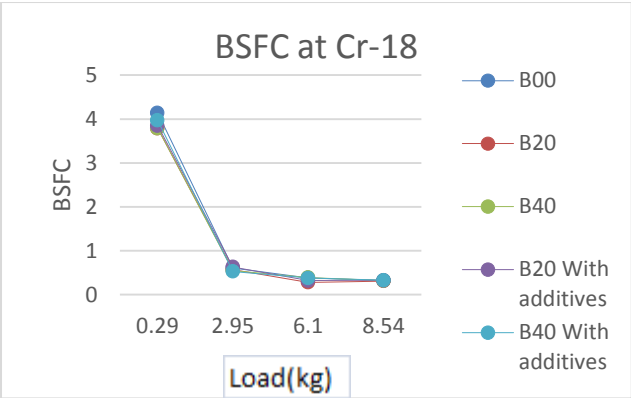

Fig.2.3.BSFC Cr 18 values at various load

Above graphs shows that variation in BSFC with respect to load. In the graph Load vs. BSFC at different compression ratio shows that as we increase the load at any particular ratio then BSFC decreases. From the graph it can be said that BSFC of B20 is improved as compared to pure diesel (B00).It also observed that optimal value of BSFC occurs at Compression ratio 18 for blend B20.

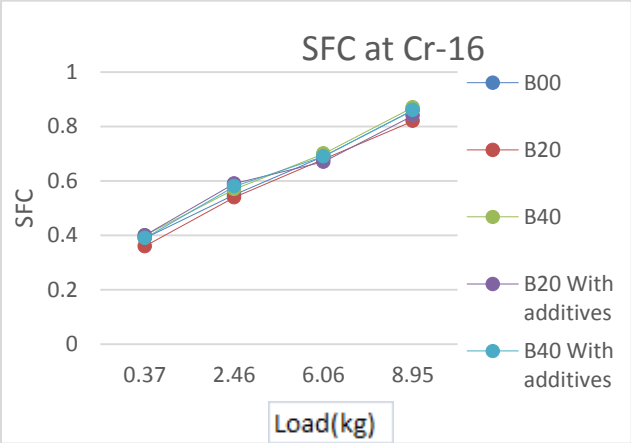

Fig.3.1.SFC Cr 16 values at various load

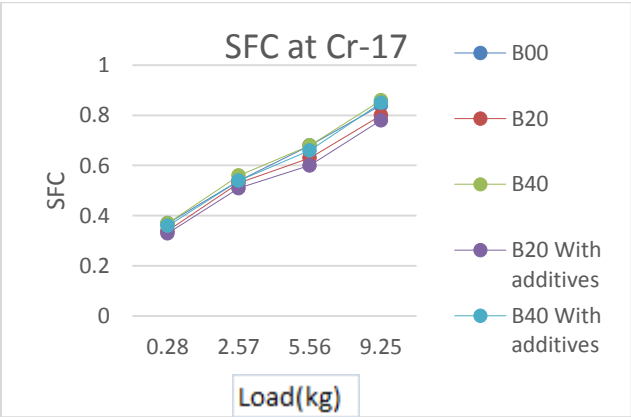

Fig3.2.SFC Cr 17 values at various load

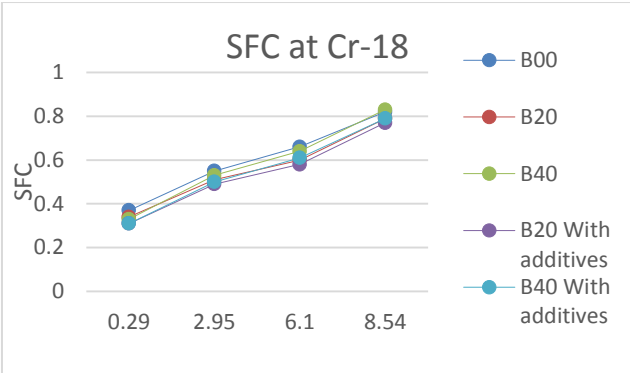

Fig.3.3.SFC Cr 18 values at various load

Above graphs shows that variation in SFC with respect to load. In the graph load vs. SFC at different compression ratio shows that as we increase the load at any particular ratio then SFC also increases. From the graph it can be said that SFC of B20 is improved as compared to pure diesel (B00). It also observed that optimal value of SFC occurs at Compression ratio 18 for blend B20.

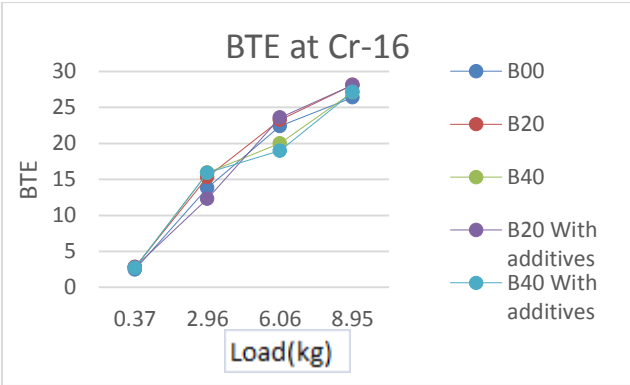

Fig.4.1.BTE Cr 16 values at various load

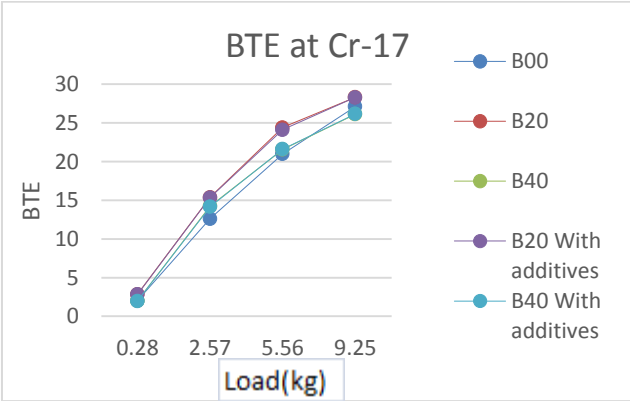

Fig.4.2.BTE Cr 17 values at various load

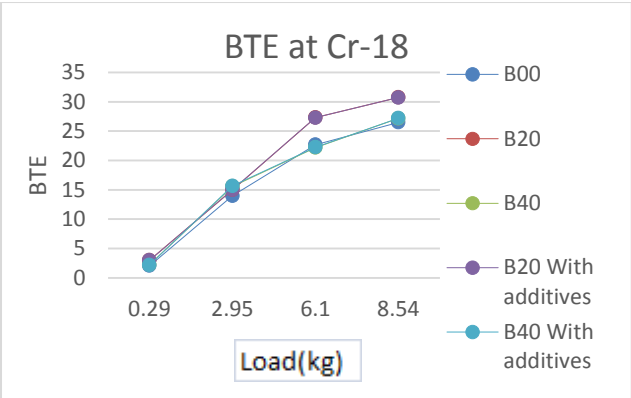

Fig.4.3.BTE Cr 18 values at various load

As the graph Load vs. BTE at different compression ratio shows that as we increase the load at any particular ratio then BTE increases, from above graphs it can be said that BTE of B20 is improved as compared to pure diesel (B00).It is also observed that optimal value of BTE occurs at compression ratio 18 for blend B20.

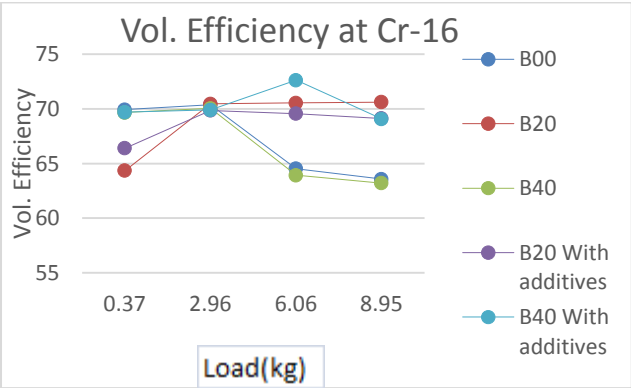

Fig.5.1.Volumetric efficiency Cr 16 values at various load.

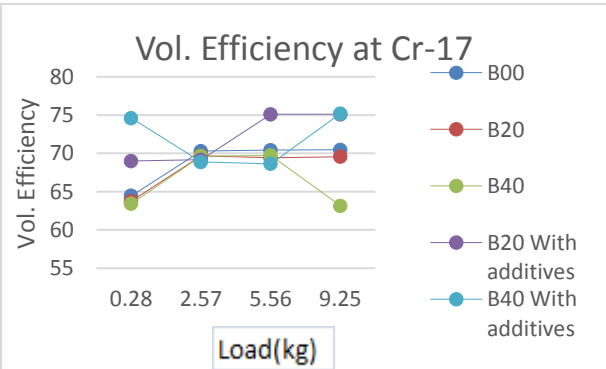

Fig.5.2.Volumetric efficiency Cr 17 values at various load.

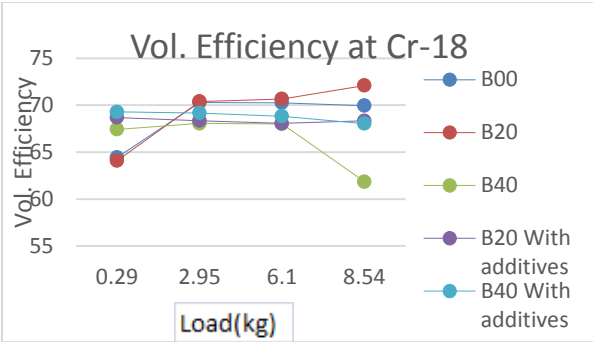

Fig.5.3.Volumetric efficiency Cr 18 values at various load.

As the graph Load Vs volumetric efficiency at different compression ratio shows that as we increase the load at any particular ratio then Volumetric efficiency increases, from above graphs it can be said that Volumetric efficiency of B20 is improved as compared to pure diesel (B00).It is also observed that optimal value of Volumetric efficiency occurs at compression ratio 18 for blend B20.

Engine Emission–The emission parameters such as CO,HC,CO2 & NOX Obtained with B00, B20, B40 are found to be affected by fuel blends and engine loading at different Cr i.e. Cr16, Cr17& cr18 are discussed in the following section-

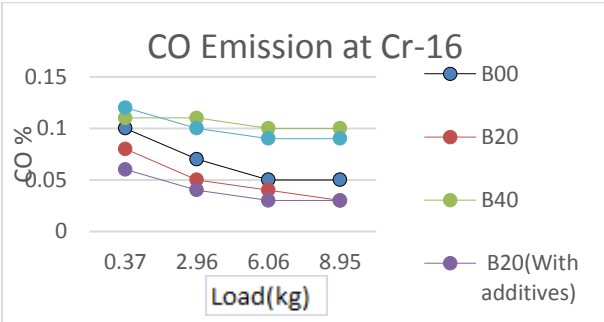

Fig.6.1.CO emission at Cr 16 values at various load.

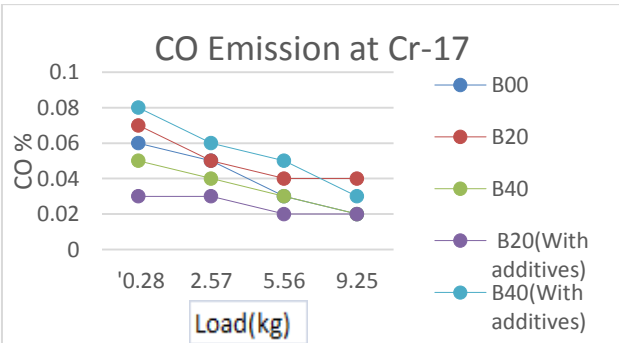

Fig.6.2.CO emission at Cr 17 values at various load.

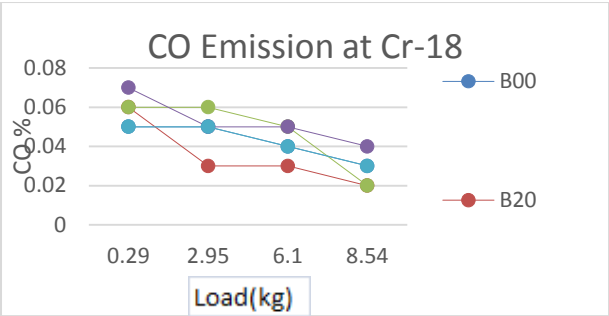

Fig.6.3.CO emission at Cr 18 values at various load

As the graph Load Vs CO emission at different compression ratio shows that as we increase the load at any particular ratio then CO emission decreases, from above graphs it can be said that CO emission of B20 is reduced as compared to pure diesel (B00).It is also observed that optimal value of CO emission occurs at compression ratio 18 for blend B20.

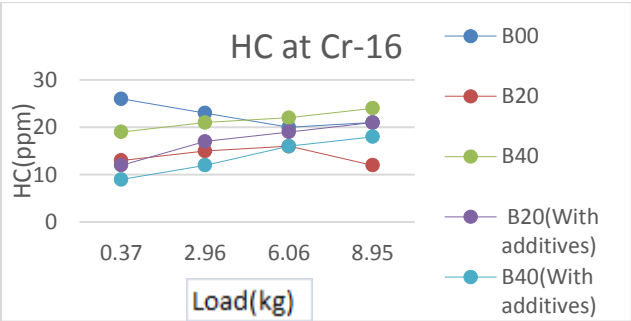

Fig.7.1.HC emission at Cr 16 values at various load.

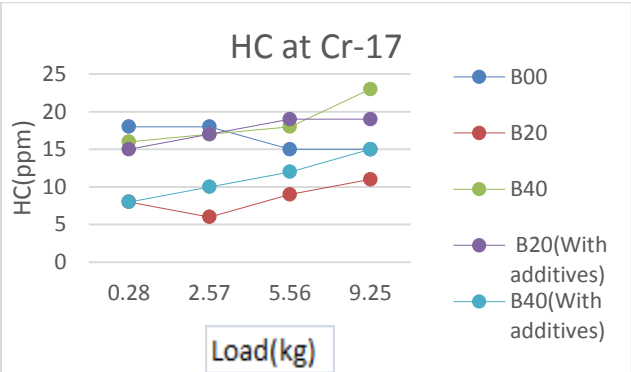

Fig.7.2.HC emission at Cr 17 values at various load.

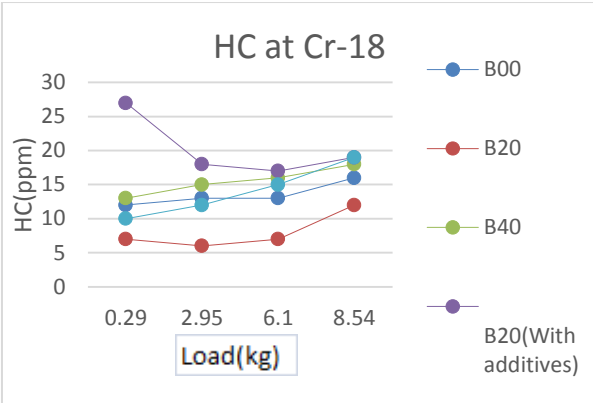

Fig.7.3.HC emission at Cr 18 values at various load.

As the graph Load Vs HC emission at different compression ratio shows that as we increase the load at any particular ratio then HC emission increases, from above graphs it can be said that HC emission of B20 is reduced as compared to pure diesel (B00).It is also observed that optimal value of HC emission occurs at compression ratio 18 for blend B20.

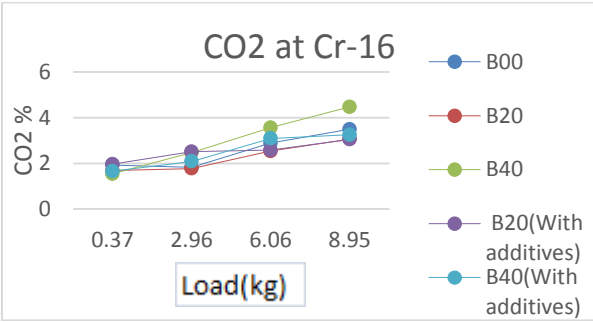

Fig.8.1.CO2 emission at Cr 16 values at various load.

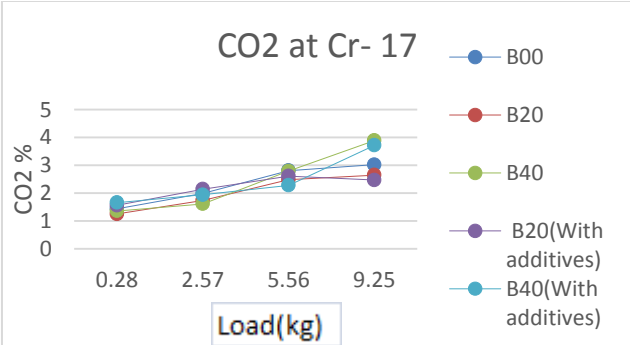

Fig.8.2.CO2 emission at Cr 17 values at various load.

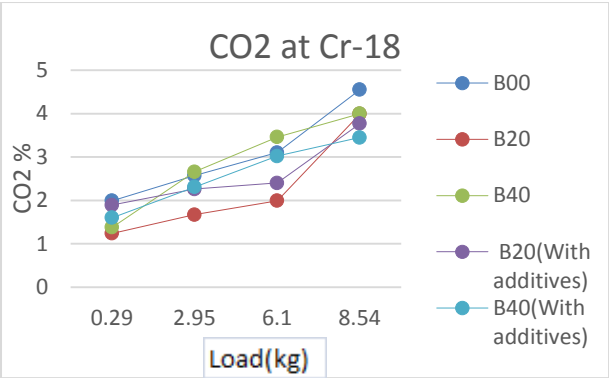

Fig.8.2.CO2 emission at Cr 18 values at various load.

As the graph Load Vs CO2 emission at different compression ratio shows that as we increase the load at any particular ratio then CO2 emission increases, from above graphs it can be said that CO2 emission of B20 is reduced as compared to pure diesel (B00).It is also observed that optimal value of CO2 emission occurs at compression ratio 17 for blend B20.

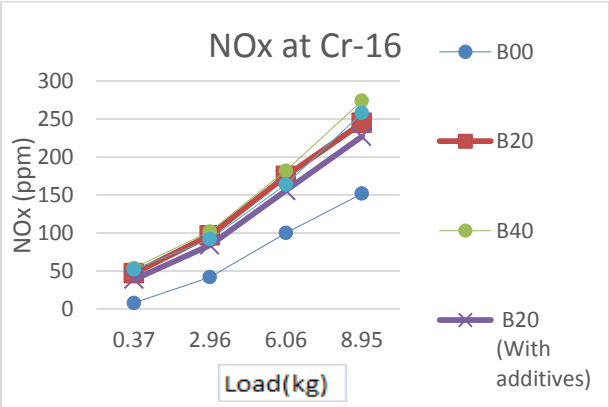

Fig.9.1.NOX emission at Cr 16 values at various load.

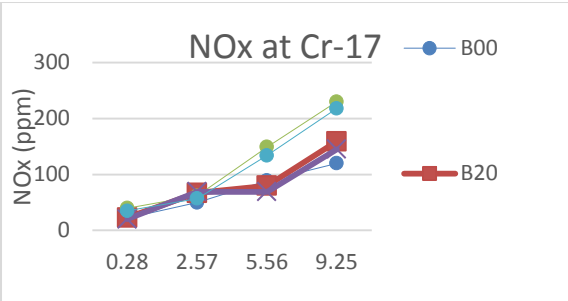

Fig.9.2.NOX emission at Cr 17 values at various load.

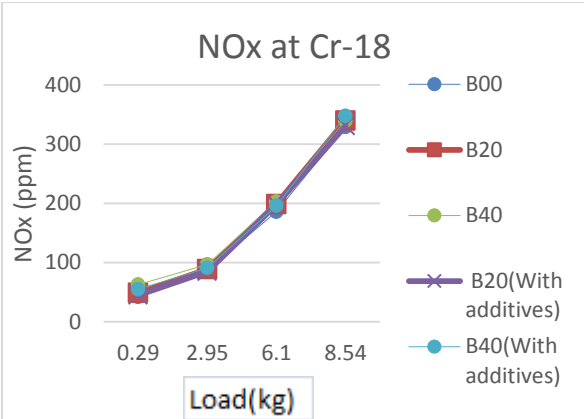

Fig.9.3.NOX emission at Cr 18 values at various load.

As the graph Load VsNOx emission at different compression ratio shows that as we increase the load at any particular ratio then NOx emission increases, from above graphs it can be said that as we increases biodiesel percentage in blend NOx value increases. It is also observed that after addition of antioxidant in blend NOX value decreases.

3. Acknowledgment

I wish to express my warm and sincere thanks to Dr. M. S. Deshmukh, for his invaluable guidance along with the care extended throughout the project work. I thank RSCOE Pune for providing facilities and resources for my project work.

4.Conclusion:

After studying the above results it can be said that B20 blend has improved results than pure diesel for both performance and emission of diesel engine. Also addition of antioxidant Propyl-galate increase the stability of Biodiesel which shows reduction in NOx emission. So, B20 blend of Babassu oil biodiesel can be the alternative source of fuel.

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HP31703703-Performance and evaluation of VCR diesel engine using deodorizer distillate oil based and additised biodiesel

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Abstract

The present study of oxidation stability and NO_X emission of biodiesel is a challenge to adopt biodiesel. Experimental investigation of deodorizer distillate oil biodiesel with antioxidant (pyrogallol) has been carried out to analyze performance, combustion and emission, characteristics in diesel engine with blends in the diesel (0%, 10%, 20%) by changing compression ratio and engine load. An addition of antioxidant in the biodiesel proved improvement in the oxidation stability. The oxidation stability increases with increasing in antioxidant proportional.The deodorizer distillate oil has to be converted to biodiesel in two step process because deodorizer distillate oil containing high FFA. The emissions from the diesel engine calculated with the help of smoke meter & gas analyzer. NO_X emission from the engine reduced at all load conditions and different compression ratio.

Keywords: *Biodiesel, Antioxidant, Oxidation stability, performance, Emission*

1. Introduction

In recent years, the consumption of petroleum products in India has been increased significantly. As far as India is concerned the need to search an alternative fuels argent to meet the demand for transportation, agricultural sector. All the investigations carried out Biodiesel, an alternative diesel fuel, comprising of alkyl monoesters of fatty acids obtained from contemporary feedstocks such as vegetable oils, animal fat and waste cooking oil, etc. In the recent past, fatty acid methyl esters, produced from different feedstocks have been used as an alternative fuel for conventional diesel in compression ignition engines. Due to its biodegradability and nontoxic nature, biodiesel attracted the attention of global researchers. However, the first generation biodiesel produced from the edible oil encountered the issues of food versus fuel along with its higher feedstock cost and energy policies. On the other hand, the production of biodiesel from the non-edible feedstocks, such as Jatropha, Pongamia Mahua, Neem was found to be more expensive compared to petrodiesel. The biodiesel production cost includes about 85% feedstock price, which results in higher cost of biodiesel (Cunshan et al.,2011). However, the production of biodiesel from deodorizer distillate feedstock oil was found to be lower compared to petrodiesel (Sorate and Bhale,2014) .

In addition to oxidation stability, the effects of antioxidant on engine performance and emission have been presented. In this investigation, oxidation stability of biodiesel derived from non-edible feedstocks such as Neem, Karanja, and Jatropha, stabilized with antioxidant pyrogallol (PY) was studied by Rancimat test. It was found that stability increases with increasing the dosage of antioxidants (Khurana and Agarwal,2011).

Effect of antioxidant additives on the performance and emission characteristics of a DICI engine using neat lemongrass oil–diesel blend the effect of antioxidant additives on the performance and the exhaust emissions has been studied. The test fuel used in this study was neat lemongrass oil (LGO)–diesel blend. The emission results showed a significant reduction of NOx (Sathiyamoorthi et al.,2016).

Emission like HC,CO, and smoke reduced while using biodiesel and its blends due to presence of oxygen in biodiesel but NOx emission increases and increase in peak temperature in combustion (Selvan and Nagarajan, 2012).

The main objective of this work is to analyze the oxidation stability, engine performance, combustion and emission characteristics of diesel engines fuelled with biodiesel produced from deodorizer distillate oil blends with antioxidant with diesel which will help in the direction of controlling stability, emission problems of biodiesel and search of alternative fuel for diesel engine.

2. Materials and Methods

2.1 Deodorizer distillate oil

Deodorizer Distillate oil is by-product of vegetable oil. Oil containing high viscosity (31.46mm2). The higher viscosity of oil leads to the poor atomization of fuel and mixing it with air when used directly in a diesel engine. Deodorizer distillate oil containing high FFA i.e 51.6% FFA. To overcome this limitation, the deodorizer distillate oil has to be converted to biodiesel in two step i.e. esterification followed by transesterification. After the conversion, the viscosity value of deodorizer distillate oil biodiesel is reduced as per standard.

2.2 Materials

Deodorizer distillate oil used in the present study was obtained from M/s Ghodhawat Foods International Private Limited, Kolhapur, a nearby medium scale vegetable oil refining industry. The unit processes about 20,000 tones of soybean oil and produces about 1000 tones of deodorizer distillate oil per annum as a byproduct. All chemicals used in the experiments such as 99.8% methanol/ethanol, sodium hydroxide, and 98% sulfuric acid were of analytical reagent grade are available at chemistry lab RSCOE, Pune.

2.3 Methods

2.3.1 Preheating

The deodorizer distillate oil was preheated to a 120oC temperature in open cauldron to remove moisture. When the temperature falls to 600C, it is poured into the glass reactor for esterification.

2.3.2 Acid-catalyzed esterification

The acid-catalyzed esterification reaction was conducted in a laboratory-scale experiment. The apparatus used for the experiment contained of round bottom reaction flask and hotplate with a magnetic stirrer. The volume of the reaction flask capacity was 1 liter and contained of two necks, one for a condenser, and the others for a thermometer. A known amount of preheated deodorizer distillate oil feedstock was poured into the reaction flask. The 0.35 v/v of methanol/ethanol was added to the preheated deodorizer distillate oil and stirred for a few minutes. The sulphuric acid (0.7 % v/v of oil) was then added as a catalyst to the mixture and the reaction was carried out at 60oC for 60 minutes (Nakpong and Wootthikanokkhan,2010, Sorate and Bhale,2014). After this reaction, the mixture was allowed to settle for 4 hours in the separating funnel and the methanol-water fraction at the top layer was removed. The lower layer consisted of deodorizer distillate oil having a lower content of FFA and impurities were purified by washing gently with hot distilled water at 55oC until the washing water had a pH value that was similar to that of distilled water. The deodorizer distillate oil layer was then dried at 110oC by the hot air temperature oven.

2.3.3 Alkali-catalyzed transesterification

The alkali-catalyzed transesterification reaction was carried out by using the same experimental setup of acid-catalyzed esterification step. The oil product from the pretreatment step was preheated to the 60 oC temperature in the reaction flask. Methanol to oil ratio of 0.4 v/v, catalyst concentration of 1.5 % w/v was used at the reaction temperature of 60oC and reaction time of 60 minutes (Sorate and Bhale,2014, Nakpong and Wootthikanokkhan,2010).The solution of sodium hydroxide in methanol/ethanol was prepared freshly in order to avoid the moisture absorbance and maintain the catalytic activity. The methanolic/ethanolic solution was then added to the heated oil in the

reaction flask. After the reaction, the mixture was allowed to separate into two layers by settling overnight in the separating funnel. The upper layer consisted of methyl/ethyl esters, whereas the lower layer contained a mixture of glycerol and impurities. The methyl/ethyl ester layer was purified by washing gently with hot distilled water at 55oC until the washing water had a pH value that was similar to that of distilled water. The methyl/ethyl ester layer was then dried at 110oC by the hot air temperature oven. The biodiesel properties were determined by using standard test methods.

3. Test procedure and test setup

Test fuel blend was prepared by blending deodorizer distillate oil biodiesel with antioxidant pyrogallol (ppm). Experiments are conducted in kirloskar engine by using biodiesel blended with diesel and with an antioxidant by volume as B10, B20 by changing compression ratio and engine load.

Table 1Engine specification

Make & model TYPE	Kirloskar SV1 single cylinder, 4stroke, DI, water cooled, Diesel Engine
Power (hp)	5.7
Speed (rpm)	1500
Bore (mm)	87.5
Stroke length (mm)	110
Injection bar (bar)	210
load type	Eddy current dynamometer
Compression ratio	16,17,18

After start engine is run on diesel for 15 min to remove out carbon particle present inside cylinder block. Here test is taken at a constant speed of 1600 rpm with varying compression ratio from 16 to 18 in step of one. At the same time, different loads are applied on the engine using dynamometer. The range selected here is 0 to 9 in a step of 3. Software used is “EnginesoftLV” for Engine performance analysis. Test engine coupled with an electrical dynamometer to apply load to the engine. Electrical Dynamometer consists of electrical power bank which applies loads in the range of 0 to 50 kg loads on an engine and it is controlled with the aid of ammeter and voltmeter. The engine is connected to the computer to record and analyze the output data. The performance analysis combustion parameters such as cylinder pressure, instant heat release rate, mean gas temperature and rate of pressure rise are evaluated. Exhaust gas analyzer is used to measure engine emissions such as NOx, unburnt hydrocarbon (HC), carbon monoxide(CO) and Carbon dioxide(CO2).

Fuel properties

Table 2 Physical properties of test fuels.

Properties	Limit	B100	Diesel
Density at 15 0C, kg/m3	860-900	869	820
Kinematic Viscosity At 40 0C, mm2/s	1.9-6.0	4.63	2.5
Acid value, mg KOH/g	Max.0.5	0.35	-
Calorific value, MJ/kg	-	37.5	42.5
Flash point , 0C	Min 130	174	55
Cloud point, 0C	Report	6	-18
FAME content, %	Min 96.5	98.3	-
Pour point, 0C	-25	2	-23
Cetane number	Min. 51	49	-
Oxidation stability at 110 0C,h	Min. 6	2.6	-

Nikhil analytical & research Pvt. Ltd, laboratories Test Report No: Ch7/L1708 from Sangli.

4. Results and discussions

4.1 Oxidation stability

Oxidation stability of biodiesel was lower than the standard limit because of deodorizer distillate oil containing high FFA. Pyrogallol a suitable antioxidant was selected based on the previous experimental work reported by the authors (Dwivedi and Sharma, 2016; Tang et al 2010; Obadiah et al., 2012; Chen et al., 2011; Kivevele et al.,2011; Chakraborty and Baruah,2012; Jain and Sharma,2010). The biodiesel (B100) samples was dosed with an antioxidant (pyrogallol) in the concentration of 1000, 2000 ppm and stored at room temperature. For reference, the biodiesel (B100) sample without an antioxidant was stored for the same period under similar conditions. Samples were stored in bottles. During the storage period, the room temperature was noted to be within the range of 18°C to 44°C. Check the oxidation stability for different ppm.

4.1.1 Rancimat test

The oxidative stability (EN 14112) was determined by the Rancimat method. As per standard biodiesel, manufacture to have at least 6 h of induction period at 110°C Oxidation stability was found to be only 2.6 h at 110°C as determined by Rancimat apparatus. The presence of polyunsaturated and unsaturated fatty acid derivatives are the important factor for the biodiesel. To improve oxidation stability of biodiesel, it is dosed with pyrogallol in concentration, i.e. 1000, 2000 ppm. This study reveals the best improvement in oxidation stability of deodorizer distillate oil biodiesel (B100) is 8 h at a concentration of 2000 ppm of pyrogallol. The addition of antioxidant increases the oxidation stability.

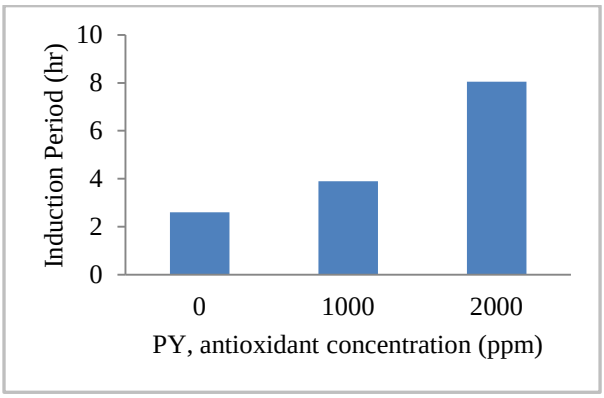
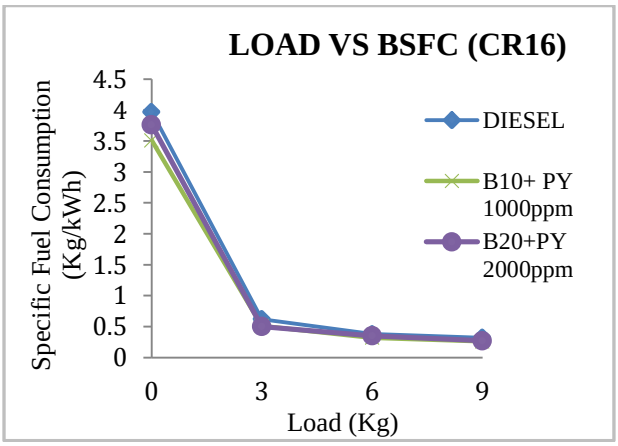


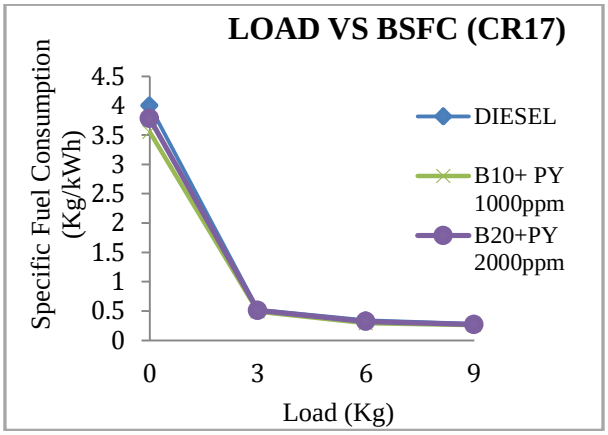
Fig.1.Effect of pyrogallol on the stability of biodiesel.

4.2 Performance characteristics

4.2.1. Brake specific fuel consumption

The variation of brake specific fuel consumption (BSFC) with respect to load at various compression ratio & various load is shown in Fig .2. The BSFC was found higher at low loads and lower at higher loads. It was found that specific fuel consumption decreases with an increase in loads. Also, it was observed that BSFC for biodiesel with antioxidant is decreased than diesel fuel.





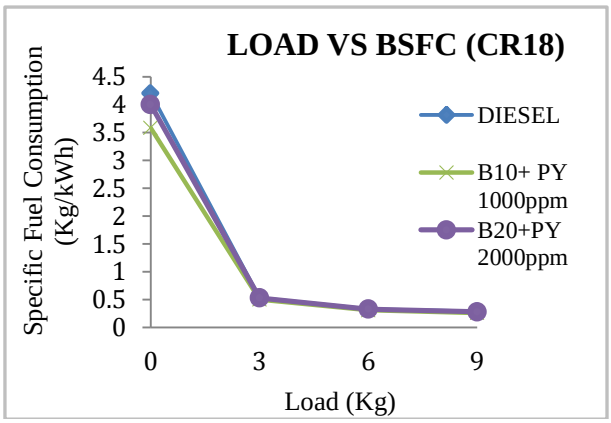
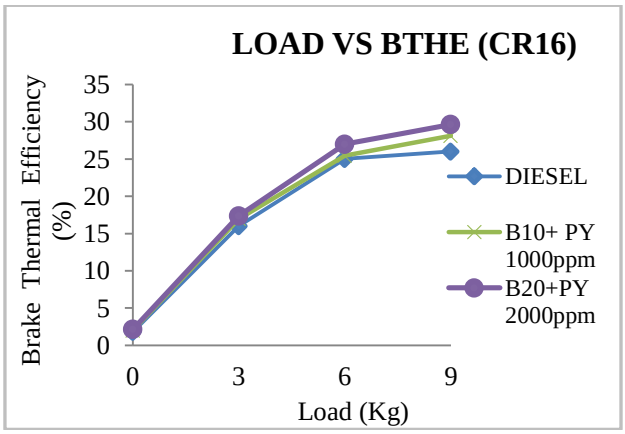
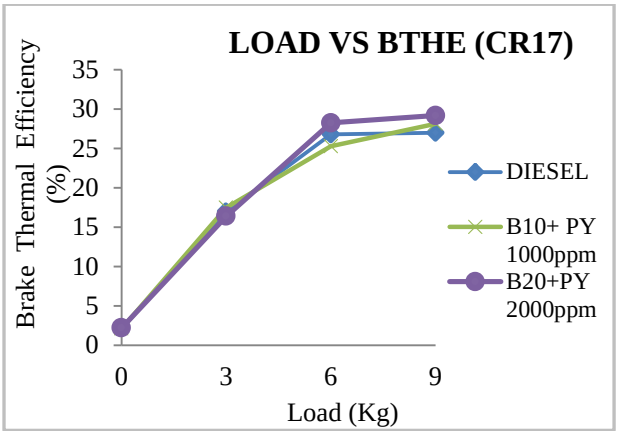


Fig.2. Variation of BSFC with respect to Load

4.2.2. Brake thermal efficiency

The variation of brake thermal efficiency with load is shown in Figure 3. From the test results, it was observed that the brake thermal efficiency of biodiesel with antioxidant increases gradually. Also, it was observed that BTE for biodiesel with antioxidant is increased than diesel fuel.





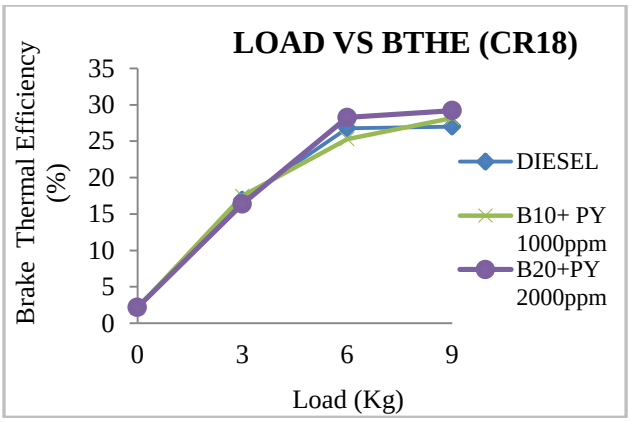


Fig.3. Variation of BTE with respect to Load

4.3 Combustion characteristics

4.3.1 Cylinder pressure and peak pressure

The pressure generated for diesel and biodiesel with antioxidant shown in figure 4. It was clear that maximum cylinder pressure is lower for biodiesel with antioxidant at all engine loads. In CI engine, the peak pressure depends on combustion rate in initial stages.

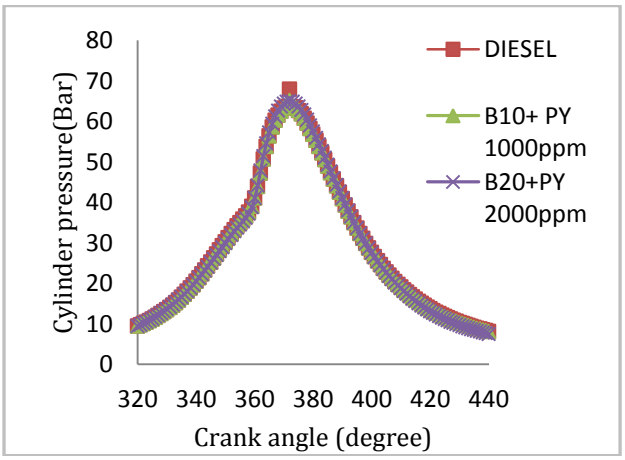


Fig. 4. Variation of cylinder pressure with crank angle

4.3.2 Net heat release rate (NHRR)

The net heat release rate is shown for diesel and biodiesel with an antioxidant in figure 5. The NHRR curves show the potential availability of heat energy which can be converted into useful work. It can be observed from figure 5 that the NHRR for biodiesel with an antioxidant is lower than diesel.

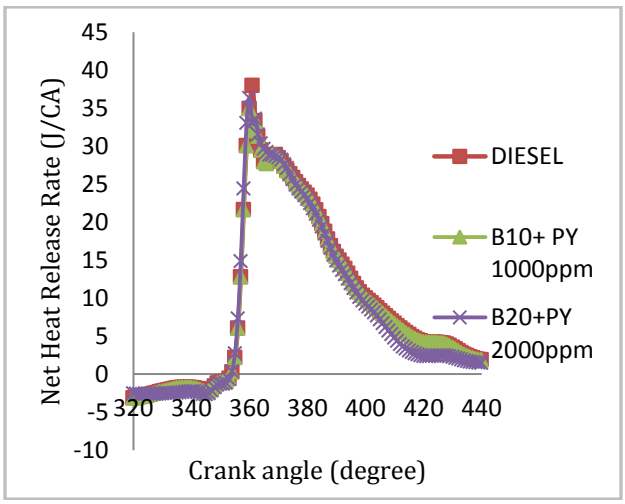
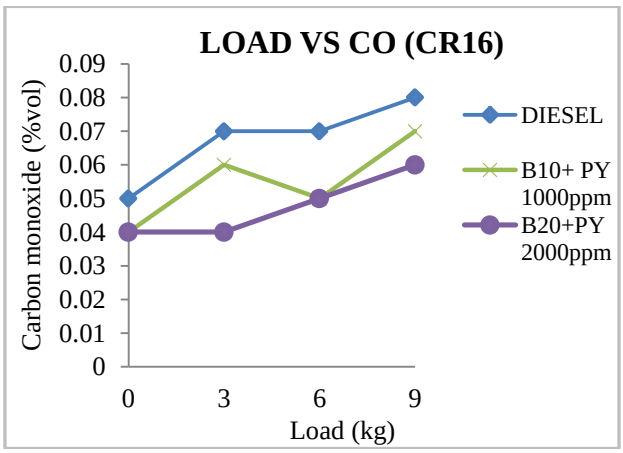


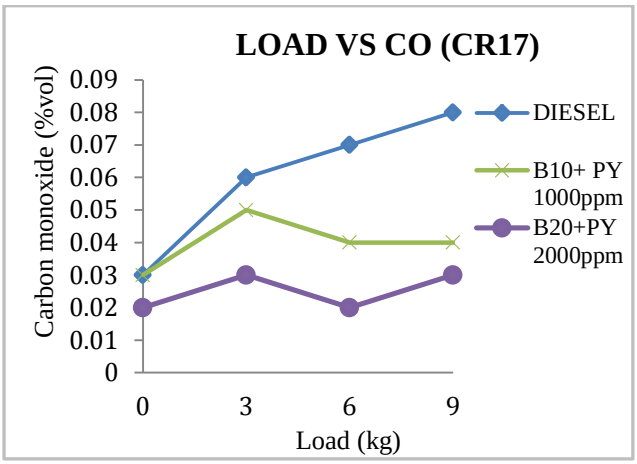
Fig.5. Variation of net heat release rate with crank angle

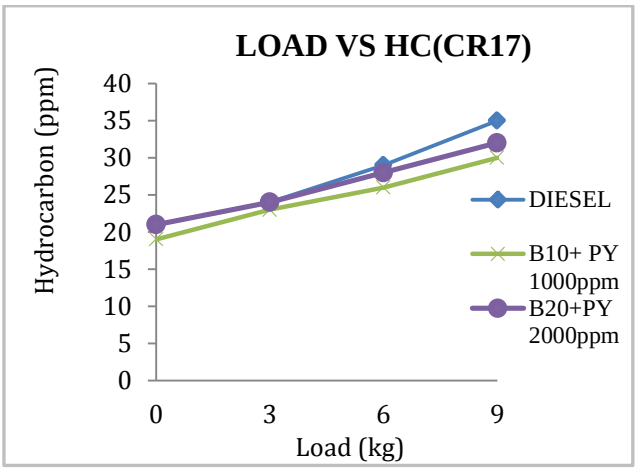
4.4 Emissions characteristics

4.4.1 Carbon monoxide emission (CO)

The variation of CO emission with load is shown in Fig. 7. It can be observed from Fig.7 that the CO emission for biodiesel with antioxidant is lower than diesel fuel at various compression ratio and engine load due to the presence of an antioxidant.







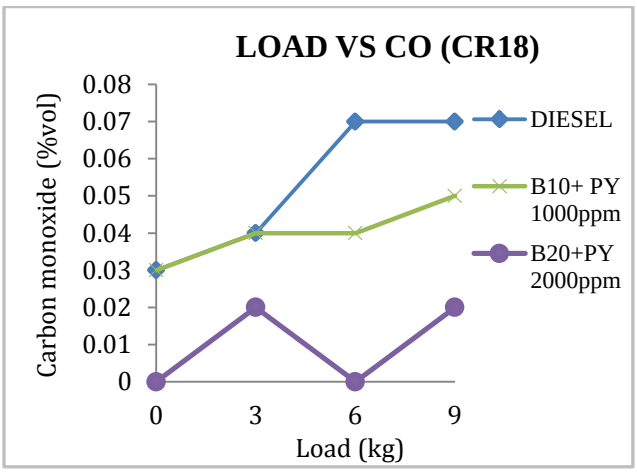


Fig.7. Variation of carbon monoxide with load

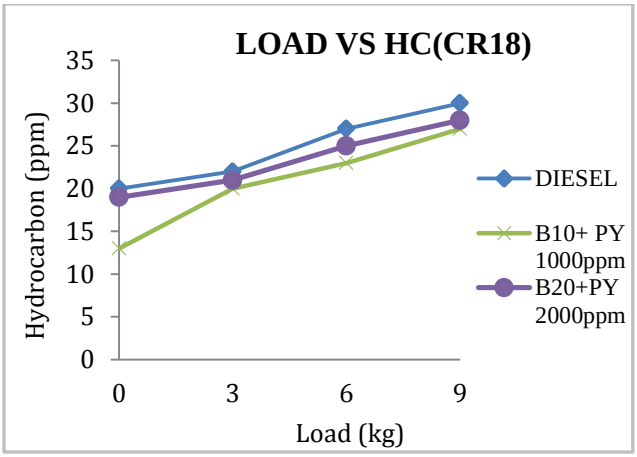


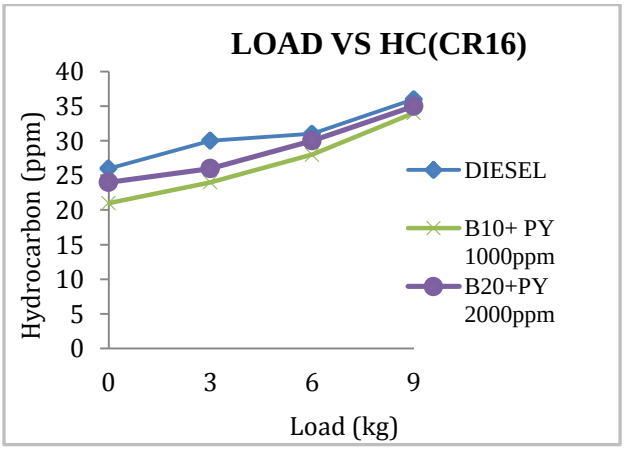
Fig. 8. Variation of hydrocarbon with load

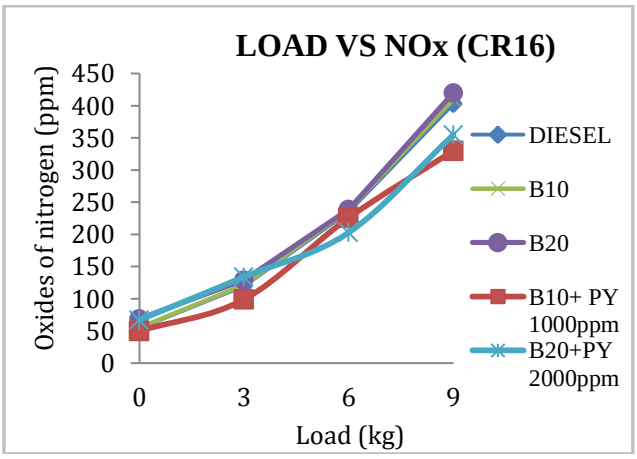
4.4.2 Hydrocarbon emission (HC)

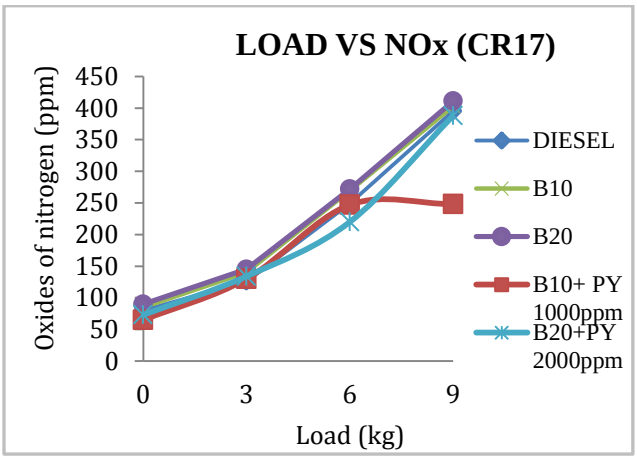
The variation of HC emission with load is shown in Fig. 8. It can be observed from Fig.8 that the HC emission for biodiesel with an antioxidant is lower than diesel fuel at various compression ratio and engine load due to the presence of oxygen in biodiesel and higher cetane number.

4.4.3 Oxides of nitrogen emission (NOX)

Temperature plays main role in NOx formation. When combustion temperature exceeds 1500°c in the combustion chamber will lead to NOx formation. The variation of NOx emission with load is shown in Fig. 9. It can be observed from Fig. 9 that the NOx emission for biodiesel with an antioxidant is lower than diesel fuel at various compression ratio and engine load due to addition of an antioxidant.







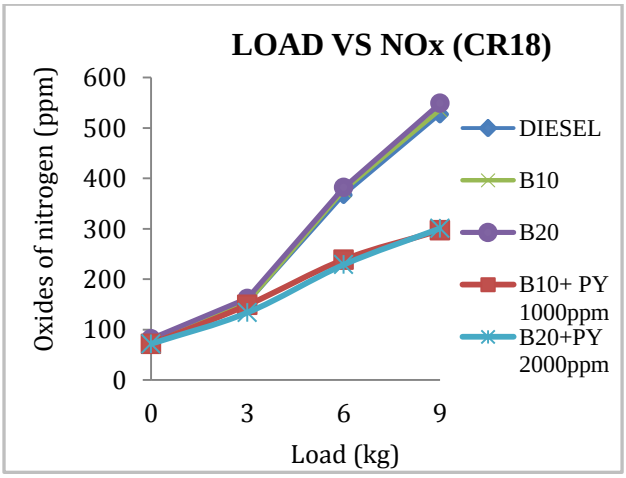


Fig. 9. Variation of oxides of nitrogen with load

5. Conclusion

Test were conducted in diesel engine, with diesel, deodorizer distillate oil with antioxidant and the following conclusion were arrived.

1. On the basis of experiment work, it is observed that viscosity of biodiesel is close to diesel.
2. In the present study, of deodorizer distillate biodiesel has poor oxidation stability. It has been found that using antioxidant (pyrogallol) improve the oxidation stability.
3. NOX emission from the engine reduced at all load conditions and different compression ratio when compared to that of pure diesel.
4. Hence it is concluded that, pyrogallol can be used as a renewable replacement for synthetic fuel additive while using biodiesel blend in the diesel.

Acknowledgement

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HP31703704-EXPERIMENTAL ANALYSIS OF I.C ENGINE RADIATOR WITH Al2O3 NANO FLUID

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Abstract

In cooling system of automobile IC engine the water plays important role for cooling also evaporate at high temperature, so we need to add water and also water is low capacity of absorb heat. By using Al2O3 nano fluids in radiator instead of water, we can improve the thermal efficiency of radiator. So cooling effect of the radiator is improved and the overall efficiency of engine will increased.

Keywords*: Al2O3 Nano Fluid, Radiator, Cooling, Engine, Base fluid (Water)*

1. INTRODUCTION

A Nano fluid is a fluid containing nanometer-sized particles, called nano particles. The main objective of the project is to improve the efficiency of the radiator cooling system in IC engine. Efficiency of the cooling system can increase by mixing the Al2O3 nano fluids with the base fluids Water. Nano fluid is mixed with the base fluid. The base fluid used in radiator is Water and other coolants. Conventional heat transfer fluids such as Water, mineral oil, and ethylene glycol play an important role in many industries including power generation, chemical production, air conditioning, transportation, and micro electronics. However, their inherently low thermal conductivities have hampered the development of energy-efficient heat transfer fluids that are required in a plethora of heat transfer applications. This new type of heat transfer suspension is referred to herein as a nano fluid. In particular, aluminum oxide tube-containing nano fluids provide several advantages over conventional fluids, including thermal conductivities far above those of traditional solid/liquid suspensions, a nonlinear relationship between thermal conductivity and concentration, fully temperature-dependent thermal conductivity, and a significant increase in critical heat flux. An emerging and new class of coolants is nano fluids which consist of a carrier liquid, such as water, dispersed with tiny nano-scale particles known as nano particles. Purpose-designed nano particles of e.g. alumina, ti oxide, carbon nano tubes, silica, or metals e.g. copper, or silver nano rods dispersed into the carrier liquid the enhances the heat transfer capabilities of the resulting coolant compared to the carrier liquid alone.

The experiments however do not prove so high thermal conductivity improvements and also efficiency Of IC engine, but found significant increase of the critical heat flux of the coolants.

Therefore, this study attempts to investigate the heat transfer characteristics of an automobile radiator using mixture of water and based Al2O3 nano fluids as coolants. Thermal performance of an automobile radiator operated with nano fluids is compared with a radiator using water based coolants. The effect of volume fraction of the Al2O3 nano particles with base fluids on the thermal performance and potential size reduction of a radiator were also carried out. Al2O3 nano particles were chosen in this study.

2. SETUP FOR EXPERIMENT

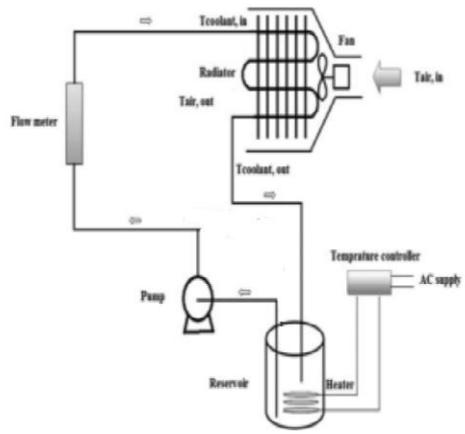

Fig 1.Experimental model [1]

System is shown in fig. it contains.

1. Radiator
3. Fan
4. Pump
5. Heating Element
6. Rota meter
7. Battery
8. Inlet and Outlet tubes.
9. Water + Concentration of nano fluid.

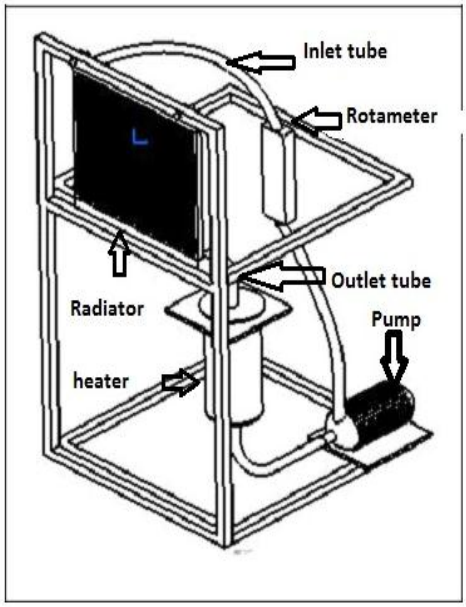

Fig2. Schematic diagram for experimental model

3. COOLING PROCESS [2]

The radiator is part of the cooling system of the IC engine in Automobile radiators mostly a cross flow heat exchanger. The two working fluids are generally air and coolant nano fluid + water. As the air flows through the radiator, the heat is transferred from the coolant to the air. The flow of the air on radiator is to remove heat from the coolant, which causes the coolant to exit the radiator at a lower temperature than it entered at. Coolant is passed through engine, where it is absorb heat. The hot coolant is then feed into tank of the radiator. From tank of radiator, it is distributed to the radiator upper storage tank through tubes to another tank on opposite bottom side of the radiator. As the coolant passes through the radiator tubes on its way to the opposite tank, it transfers much of its heat to the tubes which, in turn, transfer the heat to the fins that are lodged between each row of tubes. The radiator acts as a heat exchanger, transferring excess heat from the engine's coolant fluid into the air.

The below figure 2 shows schematic diagram of experimental set up which consists of closed loop circuit. The experimental test rig includes and heating element, drive pump, Rota meter, radiator fan(speed control DC motor) and Automobile radiator. drive pump gives the flows 1-10 LPM, the flow rate of the test section is regulated by valve which is appropriate adjustable to the recycle line as shown in fig 1. The total volume of the circulating liquid is constant in all the experiments. The circuit include 0.030m diameter pipeline which is made of the rubber pipe. A Rota meter is used to measure the flow through the test section. The specification of the Rot meter is 1-10LPH . For heating the working fluid an electric heater of capacity 2000 watt. Two K type thermocouples were implemented on the flow line to record the radiator inlet and outlet temperature

The results obtained are based on the following assumptions:

A) Velocity and temperature at the entrance of the radiator core on both air and coolant sides are uniform.
B) There is no phase change (condensation or boiling) in all fluid streams during experiment.

C) Fluid flow rate is uniformly both side inlet and outlet of radiator. No stratification, flow bypassing, or flow leakages occur in any stream.

D) The flow condition is characterized by the more speed at any cross section.

E) The temperature of each fluid is uniform over every flow cross section, so that a single bulk temperature applies to each stream at a given cross section. Heat transfer area is distributed uniformly on each side Both the inner dimension and the outer dimension of the tube are assumed constant.

F) The thermal conductivity of the tube material is constant in the axial direction. No internal source exists for thermal-energy generation.

G) There is no heat loss or gain external side or body of radiator and no axial heat conduction in the radiator.

4. PREPARATION OF NANO FLUID [3]

4.1 Single step methods

The single-step method is a process combining the preparation of nano particles with the synthesis of nano fluids, for which the nano particles are directly prepared by physical vapour deposition PVD technique or liquid

chemical method. In this method the processes of drying, storage, transportation, and dispersion of nano particles are avoided, so the agglomeration of nano particles is minimized and the stability of fluids is increased. But a disadvantage of this method is that only low vapour pressure fluids are compatible with the process. This limits the application of the method . In the previous researches, used a one-step physical method to prepare nano fluids, in which Cu vapour was directly condensed into nano particles by contact with a flowing low vapour pressure liquid ethylene glycol. copper dioxide nano fluids by a single-step method called SANSS. The established SANSS demonstrated to be effective in avoid particle aggregation and producing uniformly distributed and well-controlled size of CuO nano particles dispersed in a deionised water suspension. Presented a novel one-step chemical method for preparing copper nano fluids by reducing $CuSO_4 \cdot 5H_2O$ with $NaH_2PO_2 \cdot H_2O$ in ethylene glycol under microwave irradiation. Non-agglomerated and stably suspended Cu nano fluids were obtained.

4.2 Two-step methods

The two-step method for preparing nanofluids is a process by dispersing nano particles into base liquids. Nanoparticles, nano fibers or nanotubes used in this method are first produced as a dry powder by inert gas condensation, chemical vapour deposition, mechanical alloying or other suitable techniques, and the nanosized powder is then dissolve into a fluid in a second processing step. This step-by-step method isolates the preparation of the nanofluids from the preparation of nanoparticles. As a result, agglomeration of nanoparticles may take place in both steps, especially in the process of drying, storage, and transportation of nanoparticles. The collection of particle will not only result in the settlement and clogging of micro channels, but also decrease the thermal conductivity. Simple techniques such as ultrasonic agitation or the addition of surfactants to the fluids are often used to minimize particle aggregation and improve dispersion behaviour. Since nano powder synthesis techniques have already been scaled up to industrial production levels by surfactants and ultrasonic agitation were employed. 2002 prepared Al_2O_3/H_2O, Al_2O_3/EG, Al_2O_3/PO nanofluids by two-step method, and intensive ultrasonication and magnetic force agitation were employed to avoid nanoparticle aggregation. Most few companies there are potential economic advantages in using two-step synthesis methods that depends on the use of such powders. But an important problem that needs to be solved is the stabilization of the suspension prepared.

Base Fluid

1. Water
2. Mineral oil
3. Vegetable Oil, natural oil
4. Synthetic oils

Following methodology is followed during the process of Experiment:

1. Development of model heat exchanger.
2. Development of experimental set up.
3. Testing or Experimentation.
4. Software analysis using ANSYS software.
5. Comparison of results for water coolant and Al_2O_3 nano fluid for checking Performance and improvement.

Experimental Results for temperature

We take fluid flow rate 7 litre/minute and time duration 5 minute to take each reading.

We take initial temperature 60 deg.c for experiment for all four readings the results shown as follows:

GRAPH SHOWING TEMPERATURE DIFFERANCE BETWEEN INLET TEMPERATURE AND OUTLET TEMPERATURE:

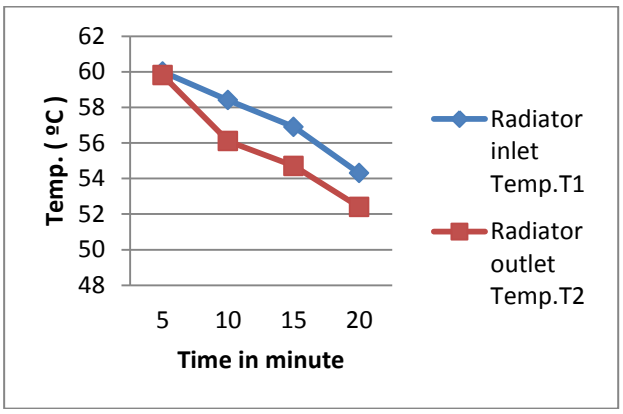

Fig 3. Temperature difference of water through the system

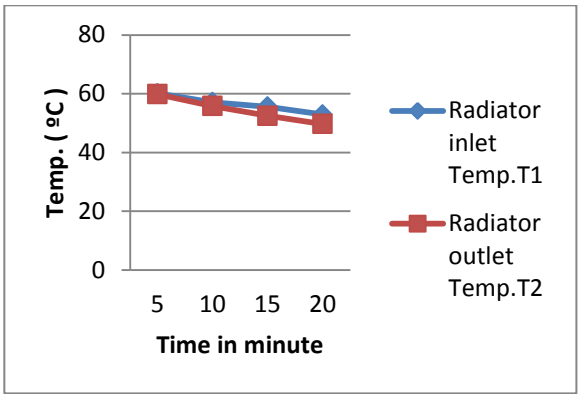

Fig 4. Temperature difference Concentration 0.1% nano fluid through the system**.**

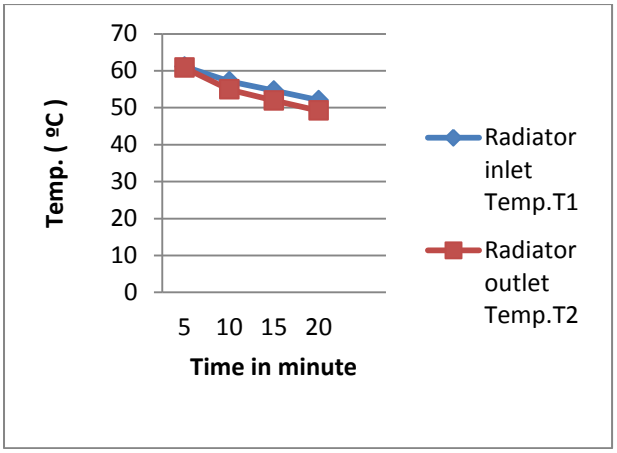

Fig5. Temperature difference Concentration 0.3% nano fluid through the system

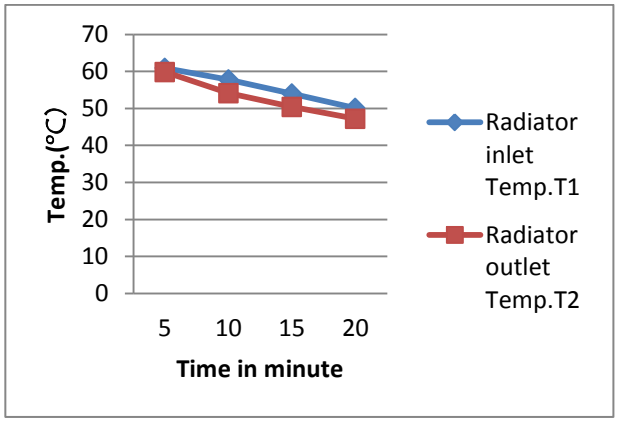

Fig 6. Temperature difference Concentration 0.5% nano fluid through the system.

ANSYS Results For Temperature difference:

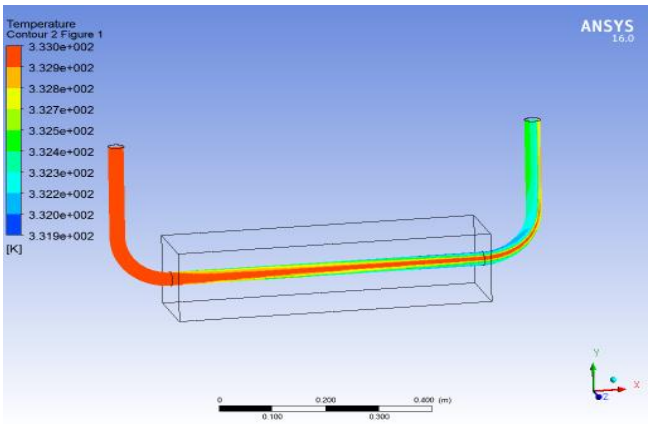

Fig 7. Temperature reading of water through the system

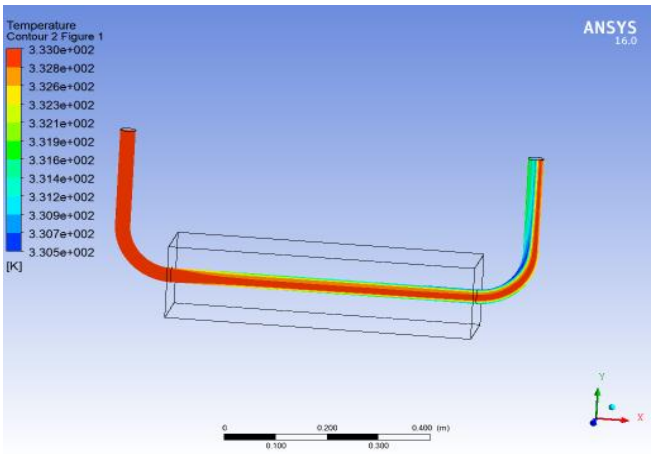

Fig 8. Temperature reading of Concentration 0.1% nano fluid through the system

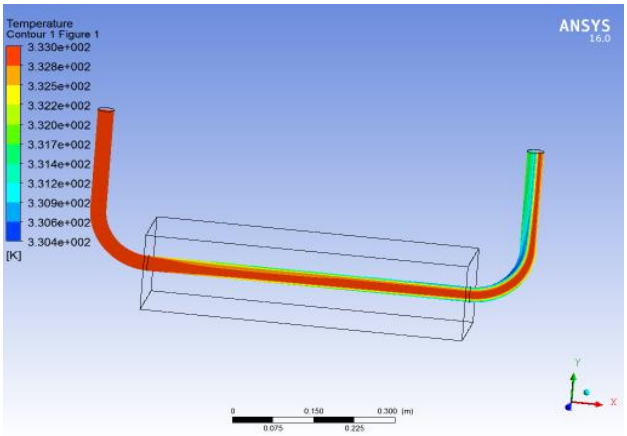

Fig 9 Temperature reading of Concentration 0.3% nano fluid through the system

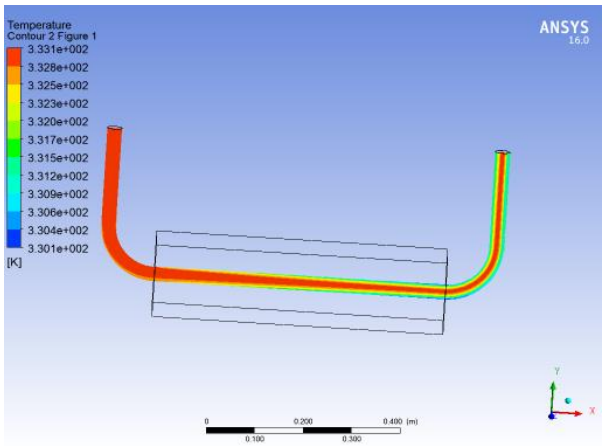

Fig 10. Temperature reading of Concentration 0.5% nano fluid through the system

5. CALCULATION PARAMETER

Factor Considered in Design of Radiator

Radiator core dimension = 0.365×0.348×0.016 m
Inner diameter of tube = 0.005 m
Outer diameter of tube = 0.007 m
Number of tubes=37
Tank height = 0.2 m
Tank diameter =0.12 m
Tank capacity = 2 liter

Thermo physical properties of base fluid and nano fluid:

Table 1: Thermo physical properties of base fluid and nano fluid

Sr.No	Properties	Al2O3	Water
1	Density kg/m^3	3950	1000
2	Specific heat(J/Kg)	873.336	4187
3	Thermal conductivity	31.922	0.561
4	Viscosity(N/m^3	-	4.068* 10^3

The air-side and coolant side heat transfer rates can be calculated as:

$$Q_a = m_a * C_{pa} * (\Delta ta) \qquad (1)$$

$$Q_c = m_c * C_{pc} * (\Delta tc) \qquad (2)$$

The mass flow rates are calculated as:

$$m_c = \rho_c * V_c * A_{tubes} \qquad (4)$$

$$m_a = \rho_a * V_a * A \qquad (5)$$

The Effectiveness of the radiator is given below

Effectiveness of fin = $\frac{Actual\ heat\ transfer}{maximum\ heat\ transfer}$

$$\varepsilon = \frac{mc\ Cpc(\Delta t)}{maCpa(\Delta t)} \qquad (6)$$

$$C_{min} = m_a * C_{pa} \qquad (7)$$

Total Heat Transfer in radiator is:

$$Q_T = \varepsilon C_{min} * (Tc\text{-Ta}) \qquad (8)$$

Overall Heat Transfer coefficient based on the air side can be express below:

$$U=\frac{Qt}{A(Tc-Ta)} \qquad (9)$$

Air Heat transfer coefficient can be expressed as follows:

$$h_a = \frac{J_a * G_a * C_{pa}}{Pr_a^{\frac{2}{3}}} \qquad (10)$$

Where,

$$J_a = \frac{0.774}{R^{0.333}{}_{ea}} \quad , \quad G_{a=\frac{R_{ea}\mu_a}{D_{ha}}}$$

Pressure drop modelling:

$$\Delta P_{nf} = \frac{2XG_{nf}{}^2 Xf_{nf}XH}{\rho_{nf}XD_{hnf}X(\frac{\mu_{nf}}{\mu_{bf}})^{0.25}} \qquad (11)$$

$$G_{nf=\frac{R_{efX}\mu_{nf}}{D_{nf}}} \qquad (12)$$

Pumping Power is given by:

$$P=V_{nf}X \triangle P_{nf} \qquad (13)$$

6. CONCLUSION

1. It has been seen that nanofluids can be considered as a potential candidate for Automobile application.

2. For base fluid water, radiator temperature decreases by 7.4ºC for trial of 5 minute to 20 minute.

3. For concentration of 0.1% nano fluid the temperature decreases by 10.1 ºC for trial of 5 minute to 20 minute.

4. For concentration of 0.3% nano fluid the temperature decreases by 11.6 ºC for trial of 5 minute to 20 minute.

5. For concentration of 0.5% nano fluid the temperature decreases by 12.6 ºC for trial of 5 minute to 20 minute.

6. Heat transfer rate is increased with increase in volume concentration of nanoparticles (ranging from 0.1% to 0.5%).heat transfer enhancement was achieved with addition of 0.1% to 0.5% Al2O3 particles at 84391 air Reynolds number and constant mass flow rate (1 Kg/s) .

7. Overall heat transfer based on air side increased up with addition of 0.1% to 0.5% Volume Al2O3 particles than the base fluid at constant air Reynolds number and constant mass flow rate.

8. Effectiveness of the radiator increased with addition of Al2O3 particles than the base fluid at constant air Reynolds number and constant mass flow rate.

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HP31703705-Experimental Investigations on Biodiesel-Alcohol-Diesel Low Proportion Blends with a Single Cylinder Diesel Engine

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Abstract
The use of fossil diesel in transportation is through CI engine increases. In this investigation additives are used to improve the performance, combustion and emission characteristics of Caster biodiesel (B) and Diesel (D) blends. Methanol (M) and Ethanol (E) used as additives in existing CI engine. The effective additives improve performance in diesel engine is methanol. The reduction of NOx, CO, CO_2, HC emission in diesel engine and also increase performance in addition of 10% to 15% methanol. From this conclude the result methanol is alternative to improve efficiency of diesel engine by using the blended biodiesel.
Keywords: Castor-Biodiesel, ethanol, methanol, performance, combustion, emission.

1. Introduction

The rapid industrialization has increase the demand of fossil fuels. But depletion fossil fuel day by day the major issues. From world energy scenario the fossil fuel available only to complete demand up to 2035. The need of alternative fuel is developed and the fulfillment of that Biodiesel is best option. The many research is done on biodiesel as a fuel in diesel engine at various load and compression ratio but the major issues is lower performance, incomplete combustion and increase rate of emission. The additives are added in the blend of Diesel and Biodiesel is effective to increase performance, complete combustion and reduce emission. Additives used alcohol first two order Methyl alcohol called as Methanol and Ethyl alcohol called as Ethanol due to more oxygen content.

2. Literature Survey

2.1 Summery

The use methanol and ethanol are very practical in the biodiesel blends due to its miscibility with the pure biodiesel [1]. Alcohol additives are very helpful to reduce the viscosity and density of the biodiesel which is higher compared to standard mineral diesel. The alcohol additives improve the combustion efficiency and produce lower exhaust emission when fuelled the diesel engines. Ethanol and methanol has approximately 35% and 30% higher oxygen in basis as compared to mineral diesel that help diesel engine to achieve higher complete combustion [2]. More oxygen in fuels means more complete combustion to be achieved. Qi et al [3] conducted the study on biodiesel-methanol-diesel blends with the volume ratios of 5% and 10% of methanol as an additive in the compression ignition engine. The investigated survey results showed that B20D55M25 and B20D50M30 produced a significant decrease of smoke emission and CO emissions. As for combustion results, B20D55M25 and B20D50M30 have a combustion delay compared to B50D50 at low engine load. However at the high engine load, the engine start for B20D60M20 and B20D65M15 were comparable to B40D60. Najafi and Yusaf[4]investigated the performance of methanol-diesel blends on a diesel engine. Methanol was added to diesel as additives with volume concentrations of 10%, 20% and 30% (methanol (10%)-diesel (90%), methanol (20%)-diesel (80%) and methanol (30%)–diesel (70%). The lowest exhaust temperature produced for the mixing ratio fuels was methanol (15%)-diesel (85%). While mineral diesel produced the highest exhaust temperature as compared to methanol-diesel blends. The comparison of brake specific fuel consumption (bsfc), the diesel produced lower bsfc as compared to other methanol-diesel blends while the bsfc for methanol (30%)-diesel (70%) was the highest. Anand et al. [5] the concude influence of the methanol as an additive on the combustion, performance and exhaust emission when blended with neat jathropa biodiesel in a single-cylinder direct injection. The test fuel was blended according to the volume ratio (90% biodiesel and 10% methanol). The combustion results indicated that the peak cylinder pressure and peak energy release rate decreases for biodiesel diesel methanol blend. However, the unburned hydrocarbon and emissions of CO were slightly higher for the methanol blend compared to neat biodiesel at low load conditions [6]. The experimental study, biodiesel-methanol-diesel blends were tested in the same diesel engine under the same operating conditions. Those finding results were compared to B20 and mineral diesel as for the baseline. Biodiesel-methanol-diesel blends were prepared with 20:15:65 and 20:20:60 ratios (B20 M15 and B20 M20). Brake specific fuel consumption (bsfc), exhaust temperatures, CO and NO emissions were compared based on the fuel type and mixing ratio. Biodiesel is mainly methyl ester of triglycerides prepared from animal fat and virgin or

used vegetable oils. It can be used in diesel engines as a single fuel or as a diesel–biodiesel blend. These require little or no engine modifications [5,6]. Ethanol is also an attractive renewable fuel. But it cannot be used as a single fuel in diesel engines thus it is blended with diesel which results in an oxygenated fuel. This blend of ethanol and diesel is also known as diesohol diesel. Diesohol has a number of advantages [7,8]. It is already known that adding ethanol to the fossil diesel fuel increases the ignition delay, increases the rate of premixed combustion, increases the thermal efficiency and reduces the smoke exhaust. The solubility of ethanol in the diesel fuel is mainly affected by hydrocarbon composition of diesel, temperature and water content of the blend [9]. However, there are some technical barriers in the direct use of diesel–ethanol blends in the CI engine. Many researchers have tested these blends with different additives (emulsifiers) but all of the blends contained small quantity of ethanol as the additives can only improve the solubility but other properties of the blend are not affected [10]. The low flash point of this blend without biodiesel, is another critical problem, which hinders the application of this blend in the CI engine and studies have shown no effect of emulsifiers on this property [11]. This blend is stable well below under sub-zero temperature and have equal or superior properties to fossil diesel fuel [12]. Studies haves that the diesel–biodiesel–ethanol blend has improved physicochemical properties compare to diesel–biodiesel or diesel–ethanol blends separately. This blend has better water tolerance and stability than the diesel–ethanol blend. Some researchers have studied this blend with hydrous ethanol [13] while some of the used anhydrous ethanol [14]. From previous studies it is obvious that for better physic chemical properties, anhydrous ethanol must be used in ternary blends [8] but the quantity of ethanol in ternary blend to demonstrate best performance needs to be determined. while some of them used maximum 80% biodiesel in a single ternary blend with 10% ethanol and 10% diesel [16]. Their results showed very good performance of this ternary blend. Although many researchers have reported good performance of this blend there are also many of them who reported very high BSFC and emissions from this blend. So there is need to evaluate research works done on this blend to conclude about its performance. The present study reviews the literature one valuating power, torque, fuel consumption, efficiency and emissions (soot, smoke, NOx, CO, CO2, HC, PM, unregulated emission, sulfur dioxide and exhaust gas temperature) of this ternary blend found by many researchers around the globe. In this review the data from research studies conducted for evaluating diesel–biodiesel–ethanol blends are collected, summarized and compared to high light potential of this blend as an alternative to diesel fuel. Performance, Power and torque Diesel–biodiesel–ethanol blends reduces engine power and torque output as the portion of oxygenated compounds (biodiesel and ethanol) in the blends increases [17]. This is due to the low cetane number and calorific value and higher ignition delay of the blends, compared to diesel fuel. This found approximately 4.4–8.7% reduction in maximum power output by using diesel–biodiesel–ethanol blends compared to fossil diesel fuel [18].

2.2 Remark on literature

Overall review we conclude that mixing of Biodiesel 20% in diesel is giving optimum result of Performance, combustion and emission. Availability of Caster oil in India in large amount and result for above proportion is good, so we use Caster Biodiesel as fuel in Blends.

3. Experimental setup and Methodology

3.1 Experimental setup

Experiments were carried out using a single cylinder, four stroke, direct injected, water cooled kirloskar TV1 VCR engine. Table 1 shows the technical specification of the test engine. Engine performance characteristics were determined using eddy current dynamometer with loading unit. The performance characteristics measure using sensors inserted at various location in combustion chamber. And the emission characteristics measure by exhaust gas analyzer in which non dispersive infrared sensor for CO, CO_2, HC and electromechanical sensor for O_2, NO. The data acquisition technique connected computer and by using engine performance and combustion software result are calculated. Figure 1 shows the block diagram of all engine setup.

TABLE 1 Technical specification of engine [19]

Specification	Values
Bore x stroke	87.5 x 110 mm
Cubic capacity	0.661 liter
Compression ratio	18:1 (vary 12 to 18)
Peak cylinder pressure	77.5 kg/cm^2
Maximum speed	2000 rpm
Sump capacity	2.70 liter
Connecting rod length	234 mm
Overall dimension	617 x 504 x 877
Weight	160 kg

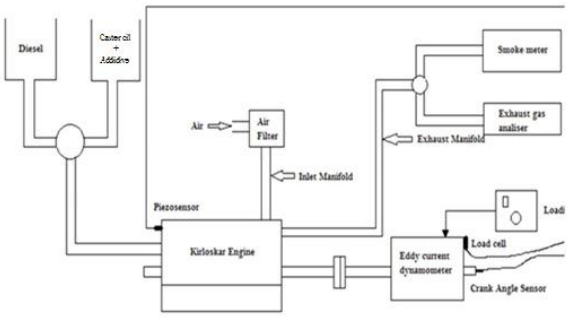

Fig 1 Block Diagram of Experimental setup

3.2 Methodology

The various blends are tested as per volume basis proportion listed in table II and the properties of diesel, biodiesel, Methanol and Ethanol are tabulated in table III. The engine load varies as in kilogram are 0, 4, 8, 12 and 16. The compression ratio of engine is constant 18:1.

Compare the result on engine performance, combustion and emission of pure Diesel, Diesel Biodiesel and Diesel Alcohol biodiesel.

TABLE 2 Percentage of Blend to Tested

Blends	Diesel	Biodiesel	Methanol	Ethanol
Diesel	100%	-	-	-
Blend I	80%	20%	-	-
Blend II	75%	20%	5%	-
Blend III	70%	20%	10%	-
Blend IV	65%	20%	15%	-
Blend V	75%	20%	-	5%
Blend VI	70%	20%	-	10%
Blend VII	65%	20%	-	15%

TABLE 3 Properties of Diesel, biodiesel, ethanol and methanol [20]

Properties	diesel	biodiesel	ethanol	Methanol
Viscosity (cSt)	18.75	21.91	15.25	12.24
Flash point (°C)	126	155	13	10
Calorific value (Kcal/kg)	10867.49	10440.34	3057.23	5493.45
Heating value (Kcal/kg)	10091.25	10439.34	6830.99	5421.80
Cetane number	52	52.87	6	5
Density (g/cm^3)	0.835	0.935	0.789	0.792
Moisture (g/100g)	0.07	0.09	-	-

4. Result and Discussion

4.1 *Performance parameter*

4.1.1 Brake power

Figure 2 shows that there is little variation in result of brake power at various loading condition. At low loading condition only variation in result of B20M15D65 blend as compare to other blends. But as compare to diesel brake power increases by adding additives.

4.1.2 Brake mean effective pressure

Figure 3 shows the variation of brake mean effective pressure at various loading condition. At low load the brake mean effective pressure increases and at high load the result are not vary by blend B20M15D65 as compare to other blends. The effect additives at low proportion nothing so much variation as compare to diesel and biodiesel.

4.1.3 Indicated mean effective pressure

Figure 4 shows the IMEP increases as increases a loading condition. The variation of result shows that the IMEP of diesel is low as compare to other blends. At low load IMEP of B20M5D75 blend is small value and increases maximum at high load.

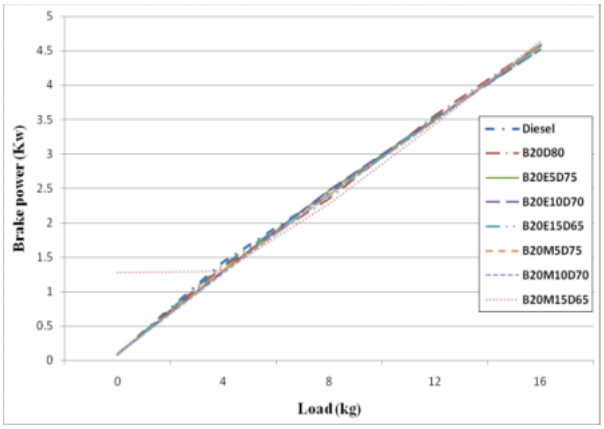

Fig 2 Variation of brake power at different load

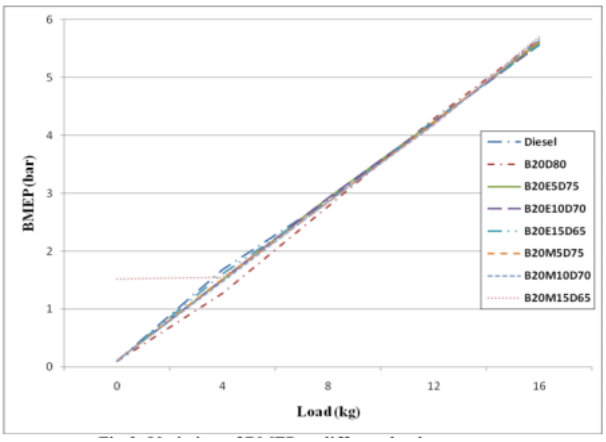

Fig 3 Variation of BMEP at different load

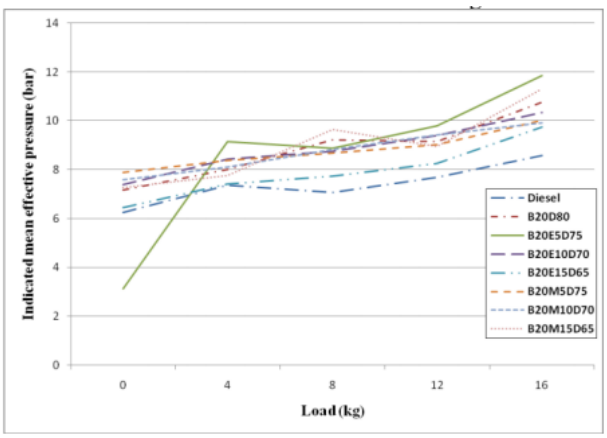

Fig 4 Variation of IMEP at different load

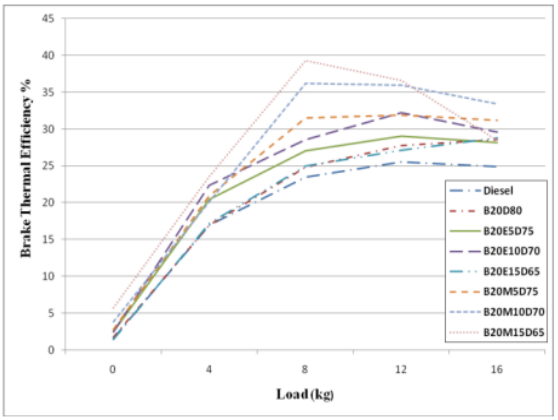

Fig 5 Variation of Brake thermal efficiency at different load

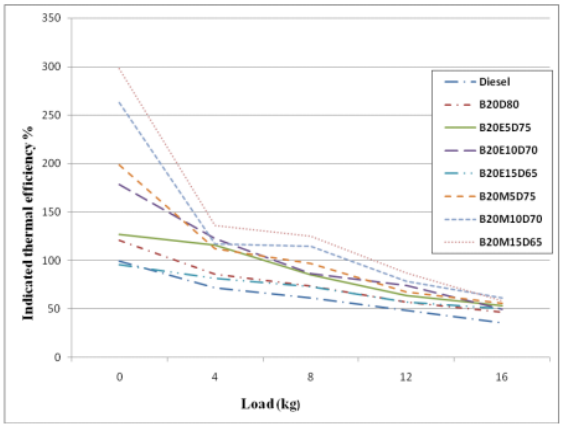

Fig 6 Variation of Indicated thermal efficiency at different load

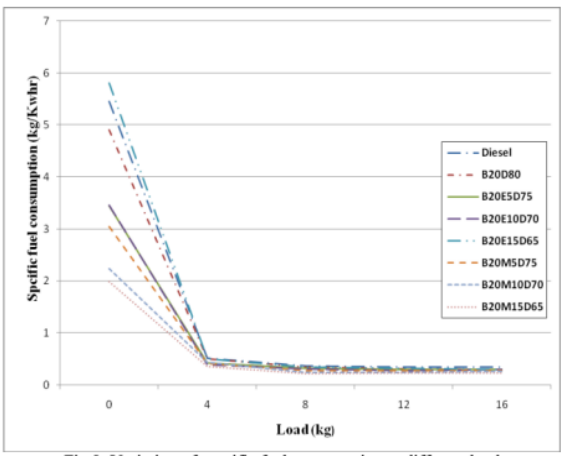

Fig 7 Variation of Specific fuel consumption at different load

4.1.4 Brake thermal efficiency

Brake thermal efficiency is the ratio of brake power to the input fuel energy. Figure 5 shows that the variation of brake thermal efficiency of various blends at various loading condition. The brake thermal efficiency of diesel is very low as compare to other blends. The maximum brake thermal efficiency at various loading condition is B20M15D65 blend below that B20M10D70, B20M5D75, B20E10D70, B20E5D75, B20D80, B20E15D65 and last pure biodiesel.

4.1.5 Indicated thermal efficiency

Figure 6 shows the variation in result of indicated thermal efficiency according to engine load. The indicated thermal efficiency decreases as increase in engine load.. The result shows that the as the concentration of additives of ethanol increases then increase in indicated thermal efficiency as compare to diesel. Also the methanol concentration increases the indicated thermal efficiency increases as compare to diesel as well as ethanol additives blends.

4.1.6 Specific fuel consumption

Figure 7 shows the variation of specific fuel consumption with respect to various loading condition. Specific fuel consumption is the ratio of fuel consumption per unit time to power consumption. The specific fuel consumption of diesel is higher than ethanol and methanol concentration blends. Methanol higher concentration blends less specific fuel consumption as compare to ethanol concentration blends. The B20D80 blend specific fuel consumption is less than diesel and B20E15D65 as compare to other blends.

4.2 Combution parameter

4.2.1 Pressure verses crank angle

The variation of pressure verses crank angle as shown in figure 8. The variation of crank angle is 320° to 390°. The variation of pressure in diesel as a fuel is lower than the B20M15D65, B20M5D75, B20D80, B20M10D70, B20E10D70 respectively and higher than the B20E5D70, and B20E15D65 respectively.

4.3 Emission parameter

4.3.1 CO

Figure 9 shows the variation of carbon monoxide emission with respect to various loading condition. The CO emits diesel is maximum as compare to other blends. The ethanol concentration blends emits less CO as compare to diesel but high as compare to methanol concentration blends and B20D80.

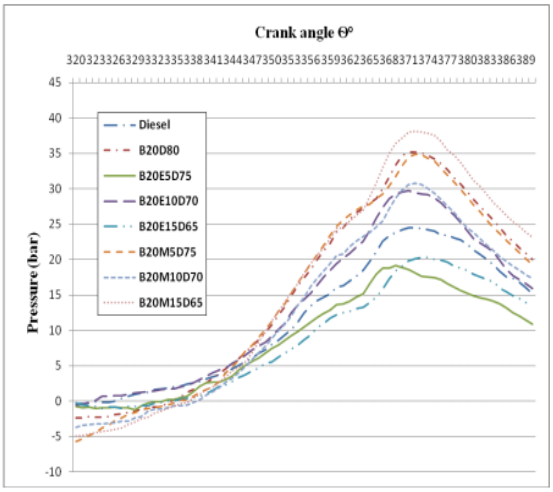

Fig 8 Variation of pressure verses crank angle

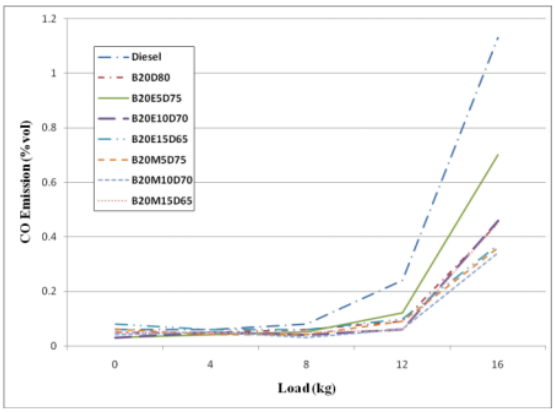

Fig 9 Variation of CO emission at different load

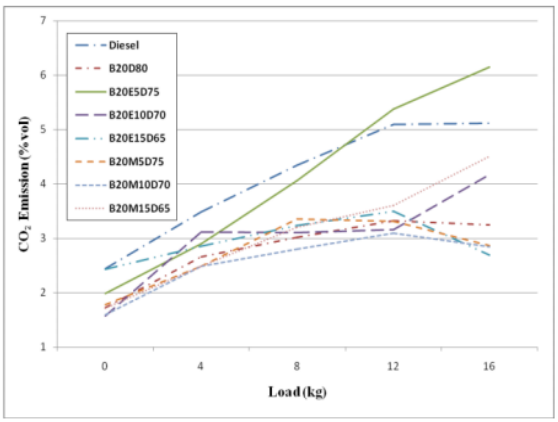

Fig 10 Variation of CO_2 emission at different load

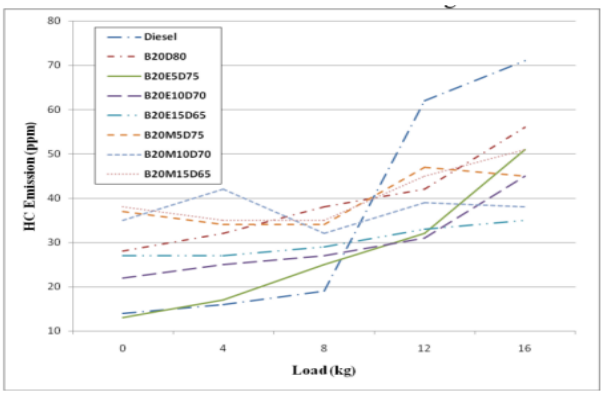

Fig 11 Variation of HC emission at different load

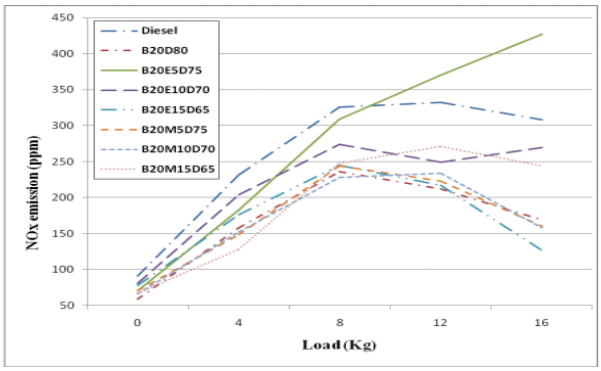

Fig 12 Variation of NOx emission at different load

4.3.2 CO_2

Figure 10 shows the variation of carbon dioxide emission with respect to various loading conditions. The diesel is emits maximum CO_2 at low load condition at higher loading emits less as compare to B20E5D75 blend. Other than B20E5D75 other are emits lesser CO_2 as compare diesel. The methanol concentration blends giving best result for CO_2 emission at all loading condition.

4.3.3 HC

Figure 11 shows the variation of hydrocarbon emission with respect to various loading condition. The HC emission at low load diesel is minimum as compare to other blends but as load increases the emission also increases. At higher loading ethanol concentration blends minimum emission of hydrocarbon. The methanol concentration blend B20M10D70 emits minimum amount of HC at high loading condition.

4.3.4 NOx

Figure 12 shows the variation of nitrous oxide emission with respect to various loading condition. The use of low concentration of ethanol in blends gives more NOx emission and methanol higher concentration blends obtained same result. At low loading condition methanol concentration giving optimum result but at high load B20M10D70 gives

variation in result. The blend without additives also getting accurate result as compare to low concentration of ethanol and high concentration of methanol blends.

Conclusions

The experimental investigation concludes as given below

1) In performance the brake thermal efficiency is increase as compare to diesel. The maximum for the methanol additives blend.
2) The combustion of biodiesel is incomplete when used in diesel engine. The addition of additives gives the complete combustion of blends in methanol concentration as compare to ethanol concentration.
3) The emission of CO, CO_2, HC and NOx is reducing by addition of methanol in diesel and biodiesel.
4) The experimental investigation we conclude that comparisons of ethanol and methanol as additives in diesel- biodiesel blend methanol is best additives.

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HP31703708-Optimization of performance parameters and emission characteristics of a diesel engine using Carbon Nanotubes blended Cottonseed Methyl Ester Emulsions

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Abstract

The impacts of nano additives in the diesel and biodiesel fuel is latest scope of research. It is recognized that at higher blend ratio emission of NOx and particulate matter is of great concern. An experimental investigation was conducted in a single cylinder diesel engine to study the effects of Carbon Nanotubes(CNT) with the Cottonseed Methyl Esters (CSOME) Emulsion fuel. The cottonseed biodiesel was produced from the Cottonseed oil by transesterification process and emulsion fuel was prepared in the proportion of 83% of CSOME, 15% of water and 2% of surfactants (by volume) with a hydrophilic–lipophilic balance of 10.72. The Carbon Nanotubes are blended with the different blends of CSOME-Diesel emulsion fuel in the various dosages. The complete investigation was conducted in the diesel engine using the following fuels: neat Diesel, different blends of CSOME and CNT blended CSOME emulsion fuels accordingly. The experimental results revealed an considerable enhancement in the brake thermal efficiency for the CNT blended CSOME emulsion fuels compared to that of neat CSOME. Additionally due to the combined effects of micro-explosion and secondary atomization phenomena occurring due to emulsified fuels, the level of harmful pollutants in the exhaust gases (such as NOx and smoke) was drastically reduced when compared to that of neat CSOME.
Keywords: *Carbon nanotubes, Water- Diesel emulsion, CI engine.*

1. Introduction

A diesel engine (CI Engine) is an internal combustion engine which utilizes the heat of compression to initiate ignition to burn the fuel, which is injected into the combustion chamber. This is in contrast to spark-ignition engines such as a petrol engine, which uses a spark plug to ignite an air-fuel mixture. The diesel engine has the highest thermal efficiency a compared to any regular internal or external combustion engine due to its very high compression ratio.

The trend towards cleaner fuels for reduce emissions and improved compatibility with after treatment devices has led to renewed interest in Diesel and biodiesel blend in fuel in recants year. In addition, these biodiesel is completely miscible with conventional diesel making it an ideal candidate as both as blending agent with and eventual replacement for conventional petroleum based diesel fuel. Furthermore, the current states of global politics and rising oil prices make these fuels and increasingly alternative to petroleum based fuel with the potential to dependence on foreign oil imports as well. Due to the depletion of fossil fuel worldwide and increasing demand of the fuel has made up to go with the alternative resources to cope up the demand of the fuel. The world survey of the fossil fuel indicates that the traditional fuels are likely to deplete within 130 to 150 years. Due to plodding depletion of world petroleum reserves and the influence of environmental pollution of increasing exhaust emissions, there is a crucial necessity for suitable alternative fuels for use in diesel engines. Therefore, it is very important to search the alternative fuels for diesel. As per the environmental and economic concern, the biodiesel seems to be one of the best emerging alternative fuels for the diesel. In view of this, vegetable oil is a promising alternative because it has several advantages. The problem of high viscosity of vegetable oils has been approached in several ways such as preheating the oils blending or dilution with other fuels, transesterification and thermal cracking / pyrolysis.

Biodiesel chemically called as 'mono methyl ester' can be extracted from a wide variety of plant oils, both edible and non-edible. Most of the developed countries produce biodiesel from sunflower, peanut, palm and several other feed stocks which are principally edible in Indian context. Hence, in the developing countries such as India, it is desirable to produce biodiesel from the non-edible oils which can be extensively grown in the waste lands of the country. The biodiesel serves the better performance characteristics in the sense of the BSFC, brake thermal efficiency, brake power, mean effective pressure, volumetric efficiency etc. Also the properties of the biodiesel such as cetane number, viscosity, calorific value are closer to the petroleum diesel. The economic feasibility of biodiesel depends on the price of crude petroleum and the cost of transporting diesel over long distances to remote areas. It is a fact that the cost of diesel will increase in future owing to increase in its demand and limited supply.

Strict emission norms and environmental concerns are prime motives for developing interest in water diesel emulsion. Induction of water has a convincing effect on various component of exhaust escaping to environment, such as nitrogen oxides (including NO and NO2, which are collectively termed as NOx), particulate matter as well as soot formation. Water diesel emulsions are of more interest due to micro size dispersion of water molecule, which is desirable for better combustion of fuel. Due to endothermic reaction of water in W/D emulsion fuel decreases the temperature of combustion which leads to the decrease in NOx Formation. The presence water also reduces the formation of soot particles.[1]

Nanoparticles are potential additives which decrease the emission parameter and can improve combustion efficiency by improving the ignition delay and fuel properties. Idea behind using nanoparticle is that as they have high surface to volume ratio provides more reactive surface, allowing them to act as more efficient chemical catalyst, thus increasing fuel combustion. The presence of these nanoparticles also increases fuel-air mixing in the fuel, which leads to more complex burning.

2. Literature Review

Many researchers observed that compared to conventional diesel fuel, use of biodiesel is generally reduces emissions of hydrocarbons (HC), carbon monoxide (CO), and particulate matter (PM) but NOx emission increases slightly. [2]

For higher blend ratios of cottonseed oil methyl esters increase in NOx was reported because of higher peak and Exhaust temperature. Emission of Nitrogen oxide might have increased due to presence of extra Oxygen in biodiesel blends.[3]

To meet the emission legislation researchers adopted different techniques like Exhaust gas recirculation, additive treatment. However these techniques failed to obtain the method which decreases reduction in Particulate matter and NOx simultaneously.[4],[5]

The introduction of water diesel emulsion in diesel engine is effective method which reduces NOx emission by decreasing the temperature of combustion products. The presence of water enhances the burnout characteristics and reduces formation of soot particles.[5]

In addition water diesel emulsion is cost effective method and doesn't require engine modification.[6]

However prolonged ignition delay takes place due to endothermic reaction of water particles during the premixed combustion phase.[7]

In order to achieve shorter ignition delay with W/D emulsion fuel various metal based nano additives such as alumina, cerium oxide, Carbon nano tubes are mixed and their improvements have been recorded by various researchers.[8],[9]

Studies showed that carbon nanotube acted as a catalyst which accelerated the burning rate which resulted in decreased ignition delay and smoke reduction.[10]

From the brief literature survey, it may be observed that by using W/D emulsion fuel with nano additives is the challenging field of research in recent years and no work has been recorded for additive based W/D emulsion fuel for Cottonseed oil methyl esters.

3. Experimental setup and procedure

Experimental Setup (Fig.1) consists of a single cylinder diesel Engine manufactured by Kirloskar oil Engine Ltd. with power rating of 3.7 KW and compression ratio of 17.5. Engine was coupled to an Eddy Current Dynamometer through universal coupling. The engine and the dynamometer were mounted on a common bed made from Iron C-Channel which was bolted to the cement foundation.

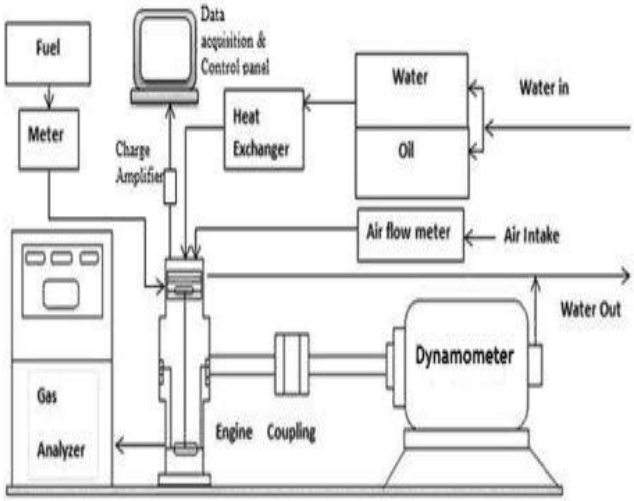

Fig.1. Details of the experimental set up

The performance characteristics were measured placing sensors at various locations in combustion chamber. And the emission characteristics were measured by exhaust gas analyzer which uses non dispersive infrared sensor for CO, CO_2, HC and electromechanical sensor for O_2, NO. The data acquisition technique was connected computer and by using engine performance and combustion software results are calculated.

Table.1 Specification of the engine

Specification	Values
Bore x stroke	87.5 x 110 mm
Cubic capacity	0.661 liter
Compression ratio	18:1 (vary 12 to 18)
Peak cylinder pressure	77.5 kg/cm^2
Maximum speed	2000 rpm
Sump capacity	2.70 liter
Connecting rod length	234 mm
Overall dimension	617 x 504 x 877
Weight	160 kg

All the experiments were carried out at constant speed of 1500 rpm by varying the loads.

2.2. Preparation of Jatropha Methyl Esters and JME emulsion fuel

Alkaline transesterification method was used to prepare the CSOME fuel process in the laboratory. The experiment was adopted for the preparation of vegetable methyl esters of Cottonseed oil (Cottonseed Biodiesel). The prepared CSOME was subjected for the preparation of CSOME emulsion fuel

For preparation of the CSOME emulsion fuel, initially the importance of the surfactant was investigated for the preparation of the CSOME emulsion fuel. A mechanical homogenizer set at an agitation speed of 2000 rpm was used to mix the water and CSOME without surfactants for 30 min. The prepared fuel was kept in the test tubes for the stability investigation under static conditions. It was observed that within fifteen minutes there was a considerable separation of water and CSOME in the test tube. Henceforth, it was inferred that a surfactant was needed to prepare the stable Cottonseed Methyl Ester emulsion fuel. The details of surfactants with a mixture of Span80 and Tween80 and their proportions are tabulated in the Tables 2. A metering pump, reactor vessel and mechanical homogenizer are utilized to prepare the stable CSOME emulsion fuel in two phases. In the first phase, the surfactants Span80 (HLB = 4.3) and Tween80 (HLB = 15) are measured in order to obtain HLB of 10.72 and filled in the reactor vessel with 2% volume and subjected for agitation at a constant speed of 1000 rpm. After the agitation, both the surfactants mixed thoroughly and kept in a separate container. The magnitude of Hydrophilic–Lipophilic Balance (HLB)

Levels	**BR**	**Additive amount (PPM)**	**LOAD (kg)**
1.	20	30	3
2.	30	60	6
3.	40	90	9

Table. 2 Process parameters and their levels

between the two surfactants indicates the relative strength of the hydrophilic and lipophilic and the emulsion stability.

In the second phase, CSOME (83% by volume) was filled in the reactor vessel and it is mixed at various agitation speeds (1000–3000 rpm) with the surfactant mixture which was prepared in the first phase. The CNT blended CSOME emulsion fuels were prepared in three stages with the aid of a mechanical homogenizer , ultrasonicator and a reactor vessel. In the first stage, the CNTs were weighed separately by means of a digital weighing machine to a predefined dosage of 30,60,90 ppm and dispersed in the distilled water (15% by volume) for 45 min with the help of an ultrasonicator. In the second stage, the surfactant mixture (Span80 and Tween80) was prepared as explained. In the third stage, CSOME (83% by volume) was filled in the reactor vessel and it is mixed with the surfactant mixture at various agitation speeds (1000–3000 rpm). In the same time, CNT dispersed in distilled water (15% by volume) was dropped by means of a metering pump at a rate of 30mL/min in the reactor vessel. Thus, the resulting solution obtained from the reactor vessel is the CSOME emulsion fuel (B20E15CNT30) and had a creamy black color and the same method was carried out for the other dosages (60 and 90 ppm) with 30,40 blend ratios to prepare 9 blends of varying proportion

Table 3 Details of Tween80 and Span80 surfactants

Type	HLB
Tween80 (polyoxy ethylene sorbitan monooleate)	15
Span80 (sorbitan monooleate)	4.3

Utility can be defined as the usefulness of a product or a process in reference to the levels of expectations to the Measurements of responses consumers. The performance evaluation of any machining process depends on number of output characteristic. Therefore, a combined measure is necessary to gauge its overall performance, which must take into account the relative contribution of all the quality characteristics. Such a composite index represents the overall utility of a process. It provides a methodological framework for the evaluation of alternative attributes made by individuals, firms and organizations. Utility refers to the satisfaction that each attributes provides to the decision maker. Thus, utility theory assumes that any decision is made on the basis of the utility maximization principle, according to which the best choice is the one that provides the highest satisfaction to the decision maker.

4. Measurements of responses

The proper blend and amount of additive emulsion is prepared before experimentation. Brake thermal efficiency, brake specific fuel consumption and exhaust gas temperature are recorded in computer. The emission parameters are measured with the help of five gas analyzer along with smoke opacity attachment. The data is recorded by analyzer with the help of auxiliary storage.

5. Analysis of results

Using Taguchi's analysis and the ANOVA, the optimal settings Engine performance parameters and emission characteristics for NOx, CO, CO2, HC, Smoke, BTE, BSFC, EGT and BP were obtained separately and the optimal values of the selected characteristics were predicted. The average values of the Engine performance parameters and emission characteristics at each level

and against each parameter were calculated

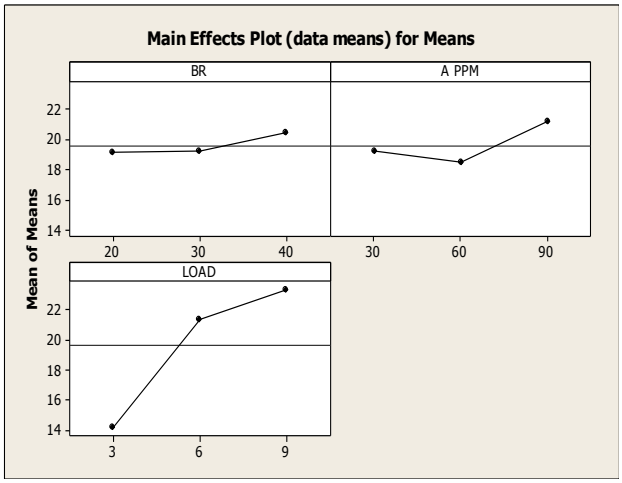

Fig. 2 Main Effects Plot (data means) for Means for BTE

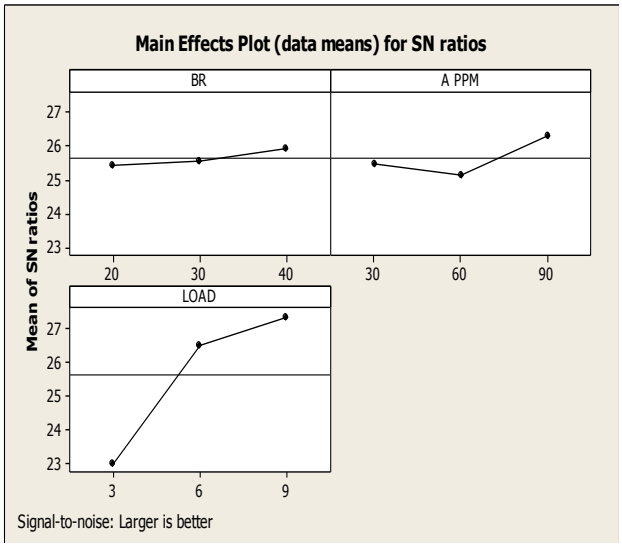

Fig. 3 Main Effects Plot (data means) for SN ratios for BTE

Fig 2 and Fig 3 show main effect plot of parameters on mean and S/N ratio for brake thermal efficiency. It should be maximum. The optimal values found from S/N ratio are to at blend ratio (40), amount of additive (90 ppm) and load (9 kg) for Cottonseed biodiesel.

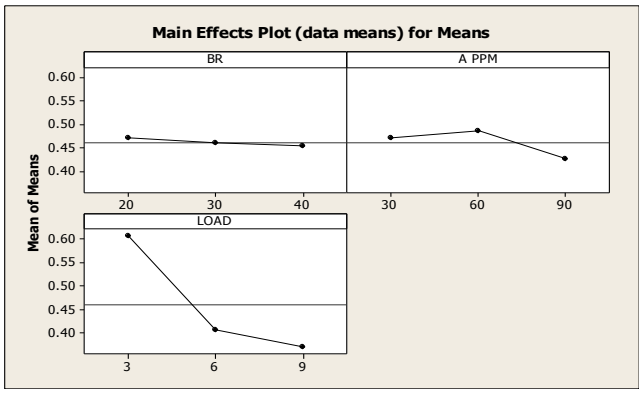

Fig. 4 Main Effects Plot (data means) for Means BSFC

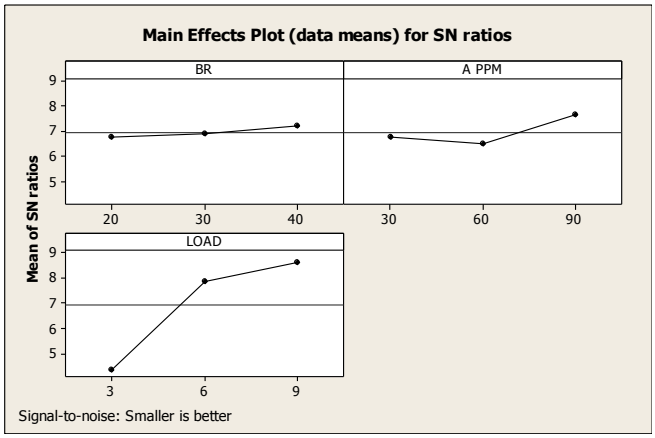

Fig. 5 Main Effects Plot (data means) for SN ratios for BSFC

Fig 4 and Fig 5 show main effect plot of parameters on mean and S/N ratio for exhaust gas temperature. It should be minimum. The optimal values found from S/N ratio are to at blend ratio (60), amount of additive (90 ppm) and load (9 kg) for Cottonseed biodiesel.

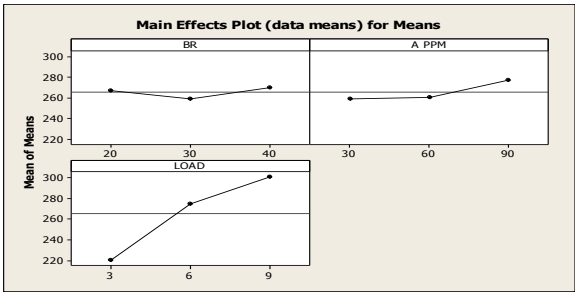

Fig. 6 Main Effects Plot (data means) for Means for EGT

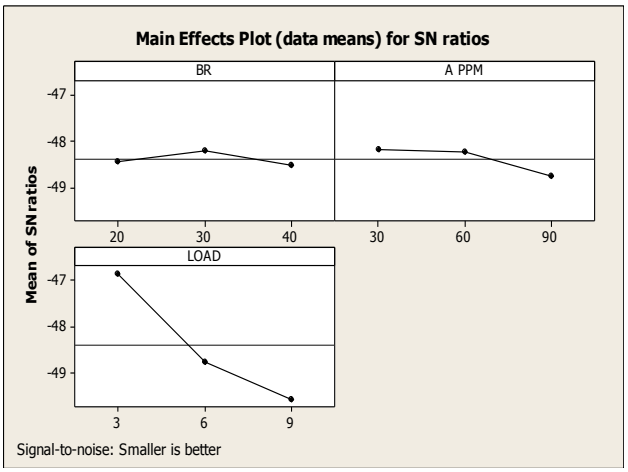

Fig. 7 Main Effects Plot (data means) for SN ratios for EGT

Fig 6 and Fig 7 show main effect plot of parameters on mean and S/N ratio for brake thermal efficiency. It should be minimum. The optimal values found from S/N ratio are to at blend ratio (30), amount of additive (30 ppm) and load (3 kg) for Cottonseed biodiesel.

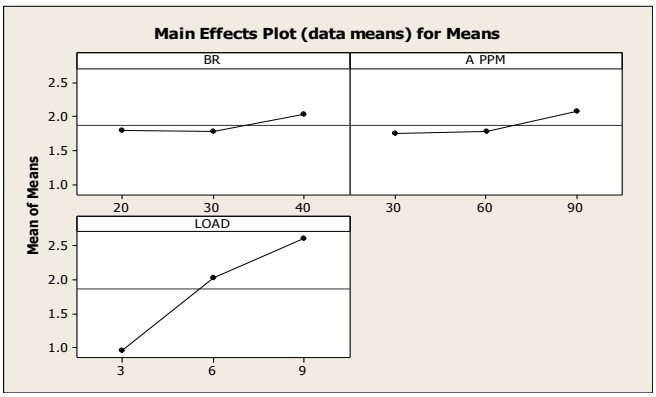

Fig. 8 Main Effects Plot (data means) for Means for BP

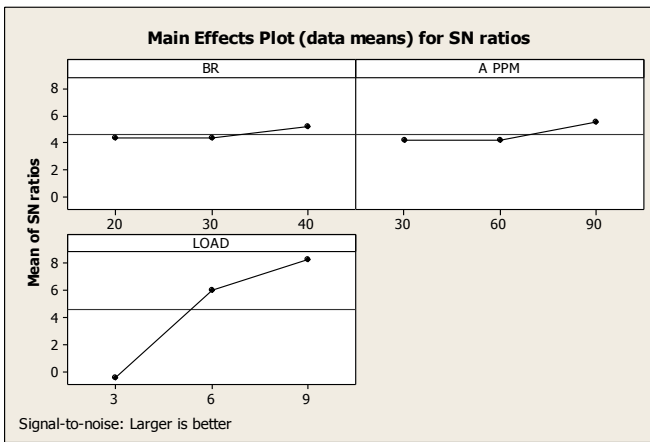

Fig. 9 Main Effects Plot (data means) for SN ratios for BP

Fig 8 and Fig 9 show main effect plot of parameters on mean and S/N ratio for brake power. It should be maximum. The optimal values found from S/N ratio are to at blend ratio (40), amount of additive (90 ppm) and load (9 kg) for Cottonseed biodiesel.

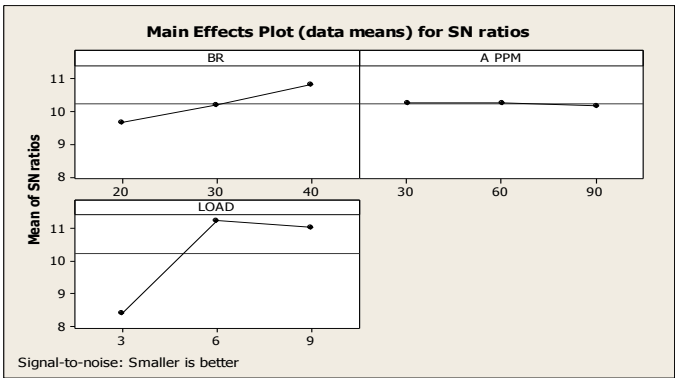

Fig. 10 Main Effects Plot (data means) for Means for NOx

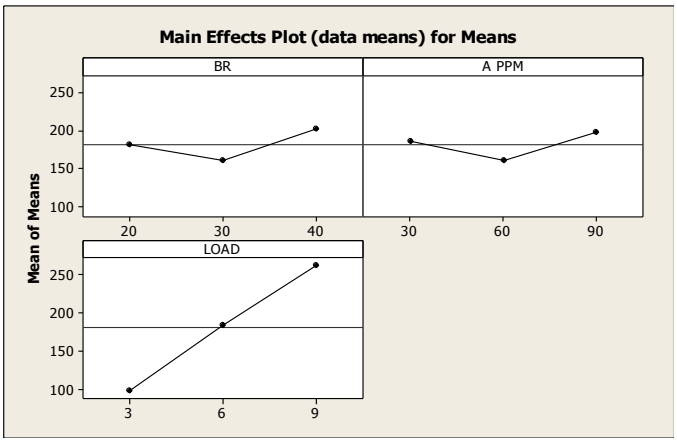

Fig 11 Main Effects Plot (data means) for SN ratios for NOx

Fig 10 and Fig 11 show main effect plot of parameters on mean and S/N ratio for NOx emission. It should be minimum. The optimal values found from S/N ratio are to at blend ratio (30), amount of additive (30 ppm) and load (3 kg) for Cottonseed biodiesel.

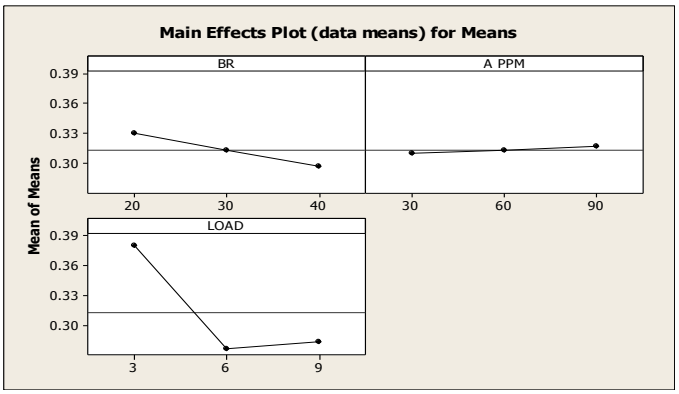

Fig. 12 Main Effects Plot (data means) for Means for CO

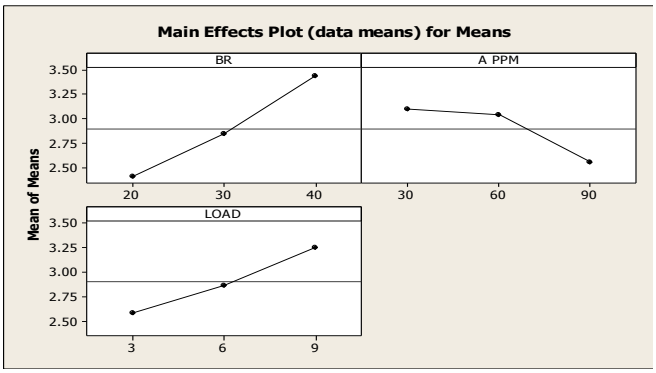

Fig. 13 Main Effects Plot (data means) for SN ratios for CO

Fig 12 and Fig 13 show main effect plot of parameters on mean and S/N ratio for CO emission. It should be minimum. The optimal values found from S/N ratio are to at blend ratio (40), amount of additive (30 ppm) and load (6 kg) for Cottonseed biodiesel.

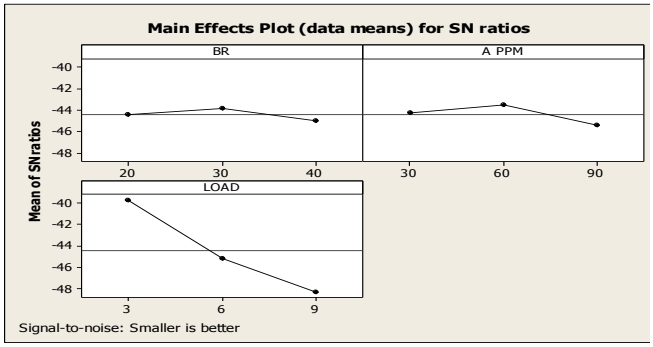

Fig. 14 Main Effects Plot (data means) for Means CO_2

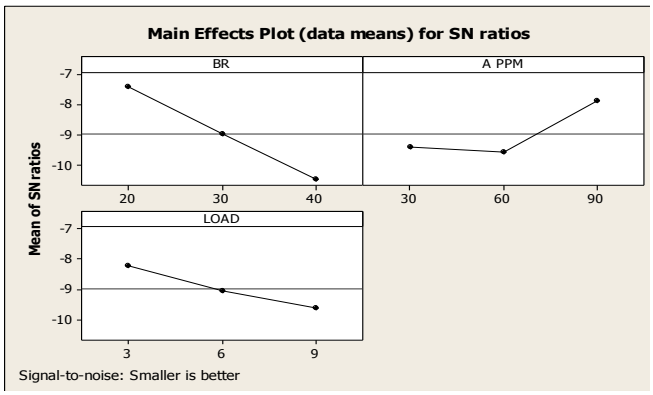

Fig. 15 Main Effects Plot (data means) for SN ratios CO_2

Fig 14 and Fig 15 show main effect plot of parameters on mean and S/N ratio for CO_2 emission. It should be minimum. The optimal values found from S/N ratio are to at blend ratio (20), amount of additive (90 ppm) and load (3 kg) for Cottonseed biodiesel.

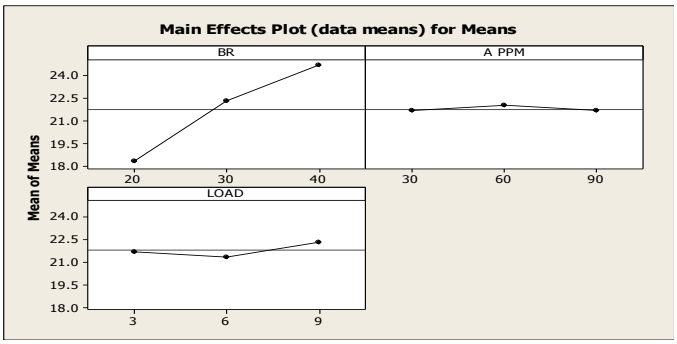

Fig. 16 Main Effects Plot (data means) for Means for HC

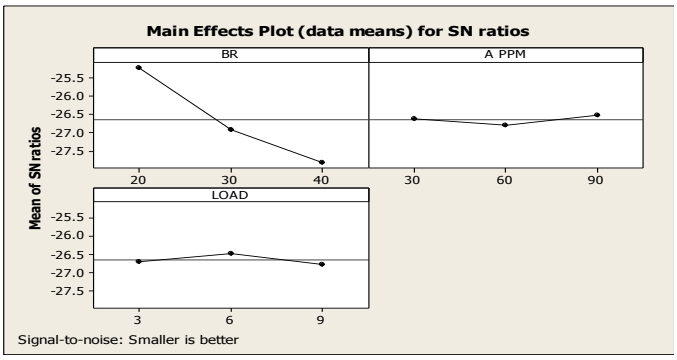

Fig. 17 Main Effects Plot (data means) for SN ratios HC

Fig 16 and Fig 17 show main effect plot of parameters on mean and S/N ratio for HC emission. It should be minimum. The optimal values found from S/N ratio are to at blend ratio (20), amount of additive (90 ppm) and load (6 kg) for Cottonseed biodiesel.

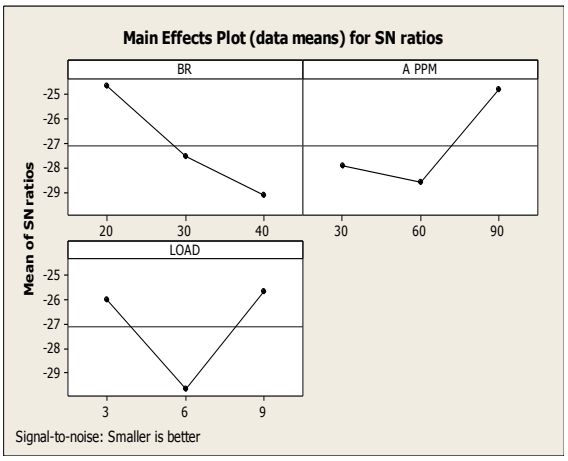

Fig. 18 Main Effects Plot (data means) for Means for smoke opacity

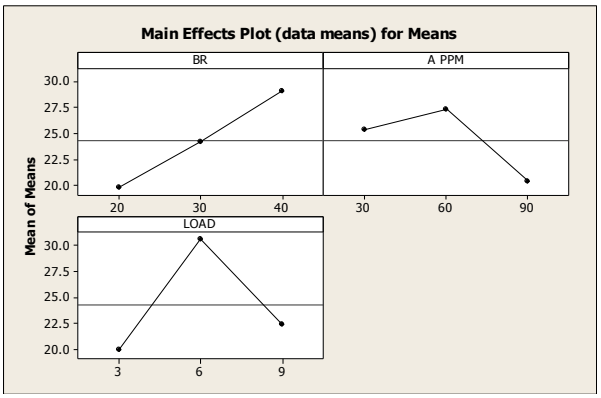

Fig. 19 Main Effects Plot (data means) for SN ratios for smoke opacity

Fig 18 and Fig 19 show main effect plot of parameters on mean and S/N ratio for smoke opacity emission. emission. It should be minimum. The optimal values found from S/N ratio are to at blend ratio (40), amount of additive (60 ppm) and load (6 kg) for Cottonseed biodiesel.

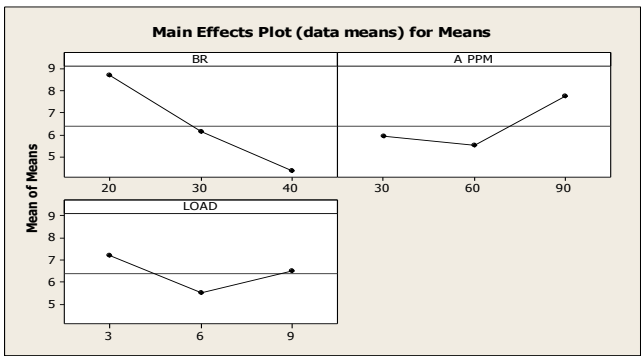

Fig. 20 Main Effects Plot (data means) for Means for utility values

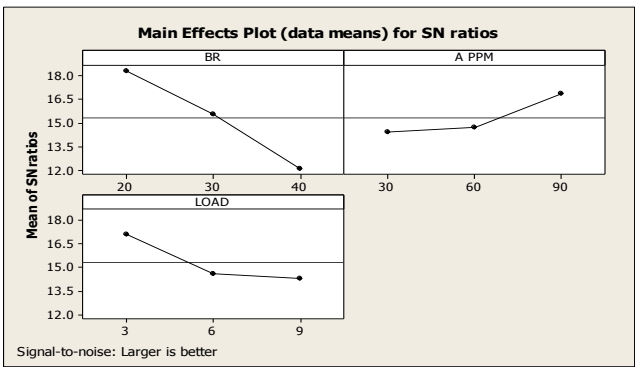

Fig. 21 Main Effects Plot (data means) for SN ratios for utility values

It is clear from the Fig 21 that the first level of blend ratio (20), the third level of additive amount (90 ppm) and first level of load (3 kg) would yield best performance in terms of utility value and S/N ratio within the selected range of parameters. Blend ratio (A) are showing about 45% contribution on responses followed by load (about 27%) and additive amount (about 23%).

The confirmation test for the optimal parameter setting with its selected levels was conducted to evaluate the response characteristics for performance parameters and emission characteristics.

6. Conclusions

1) It can be seen from the graph for CO to be minimum, BR has to be at level 3 (30), Additive amount has to be at level 2 (60), Load has to be at level 2 (6 kg).
2) It can be seen from the graph for CO2 to be minimum, BR has to be at level 1 (20), Additive amount has to be at level3 (90), Load has to be at level 1 (3 kg).
3) It can be seen from the graph for HC to be minimum, BR has to be at level 1 (20), Additive amount has to be at level 3 (90), Load has to be at level 2 (6 kg).
4) It can be seen from the graph for Smoke opacity to be minimum, BR has to be at level 1 (20), Additive amount has to be at level 3 (90), Load has to be at level 3 (9 kg).
5) It can be seen from the graph for brake thermal efficiency to be maximum, BR has to be at level 2 (30), Additive amount has to be at level 3 (90), Load has to be at level 3 (9 kg).
6) It can be seen from the graph for brake specific fuel consumption to be minimum, BR has to be at level 2 (30), Additive amount has to be at level 3 (90), Load has to be at level 3 (9 kg).
7) It can be seen from the graph for exhaust gas temperature to be minimum, BR has to be at level 2 (30), Additive amount has to be at level 1 (30), Load has to be at level 1 (3 kg).
8) It can be seen from the graph for brake thermal efficiency to be maximum, BR has to be at level 3 (40), Additive amount has to be at level 3 (90), Load has to be at level 3 (9 kg).
9) According to the utility concept with taguchi technique we found that, the optimal values for NOx, CO, CO2, HC, Smoke, BTE, BSFC, EGT, BPis at blend ratio 20, additive amount 90 ppm and load 3 kg.
10) From ANOVA table for means and S/N ratios for utility overall vales, it is showing that blend ratio is showing about 45% contribution on responses followed by load (about 27%) and additive amount (about 23%).

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HP31703803-Testing and Calibration of ISLe8.9 Diesel Engine for Power Upgradation

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Abstract

The present work deals with the power upgradation of a multi-cylinderCummins' diesel engine i.e. ISLe8.9L with a capacity of 8.9 liters by changing its calibration with the least possible hardware change (target was No-change), and still be within the thermal and mechanical limits of the engine. The outputswere achieved by changing the turbo pressure (engine boost), main injection timing/Start of Injection (SOI), rail pressure, pilot fueling, pilot separation, and Air-fuel ratio. The end application of this engine was for various defense vehicles.

Keywords:*Design of experiment (DOE), Air-fuel ratio (AFR), Peak cylinder pressure (PCP), Start of Injection (SoI), Heat Rejection (HR)*

1. Introduction

Initially all the I.C. engines were fueled mechanically which is a robust technique but also a polluting one, with the ever tightening emission norms across the world and the effect it has caused on our environment the world is rapidly moving toward electronic engines (electronically fueled) which has provided the freedom of controlling various parameters in the engines which were not possible earlier with mechanically fueled engines.

This paper outlines the process by which a current production commercial diesel engine was modified for high performance military use. The constraint faced while working towards achieving this goal was that the engine hardware could not be changed.

The engine's outputs were achieved by changing the turbo pressure (engine boost), main injection timing/Start of Injection (SOI), rail pressure, pilot fueling, pilot separation, and Air-fuel ratio. All these quantities were changed and optimized such that the required power output from the engine can be extracted without the risk of crossing the thermal and mechanical limits of the base engine and its allied peripherals.

A similar approach for the increment of power density of the engine was used by [1] James W. Brogdon *et.al*, but the DoE was designed using CFD analysis and Ricardo DIPEN (Diesel SprayPenetration).

The parameter of SOI was selected as the effect of varying it has been studied quiet extensively [2]-[4] and [6]. The SOI though increases the value of engine out NOx, it increases the peak power of the engine to a very high extent [2]. The balance of increase in power and that in NOx has to be balanced critically to obtain the desired result.

Apart from injection timing, airpath and fuelpath dynamics are also important for controlling the combustion. [6]M. Hillion *et al.* have shown the variation of performance with the classic fuel and air path of and engine and with modified paths. The fuelpath can be varied by changing the pilot injection, fueling quantity and also the pressure at which the fuel is being injected. The airpath can be modified by changing the boost pressures and controlling the amount of EGR entering back into the system.

All the changes made onto the base engine were validated at the engine dynamometer and by performing test on 3 engines for consistency for the optimized calibration

2. Existing System Architecture and Test Set-up

The base engine used for this application was "*ISLe 8.9L"* engine which is a Cummins 8.9 liter turbocharged, intercooled diesel engine with the following architecture as shown in Table 1.:

Table 1. *Base engine architecture*

Engine Model	ISLe 8.9L
Number of cylinders	6
Displacement	8.9 liters
Fuel system	BOSCH Common rail
Bore × Stroke (mm × mm)	114 × 144.5
Rated power (HP)	435 @ 2100
Peak Torque (Nm)	1627 @ 1300

The base engine was tested at Power Train Engineering's New Engine Design Lab cell #1 ARAI, Pune with the engine set-up shown in Fig 1, for determining the baseline values for all the parameters. Test cell specifications are shown in Table 2:

Table 2. *Test cell specifications at ARAI*

Sr No.	Equipment	Make	Model name
1.	Dynamometer	APICOM	FR 800 Eddy Current
2.	Test cell automation system	iASYS	Orbit-e
3.	Fuel conditioning system	APICOM	CMF-025
4.	Emission system	AVL	AMA i60 R1 EGR
5.	Condition Air system	KS engineering	IACU3000
6.	Air flowmeter	ABB	Sensyflow (SFI-12)
7.	Opacimeter	AVL	S439-03
8.	Smoke meter	AVL	415
9.	Particulate system	AVL	SPC 472
10.	Weighing Micro balance	Sartorious	CP2P-F

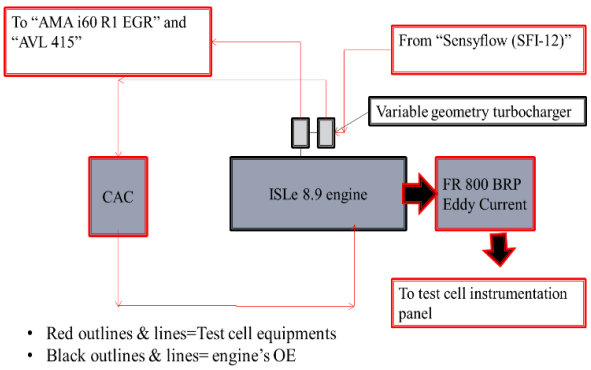

Fig 1. *Engine setup at ARAI*

3. Design of Experiment (DoE)

DoE allows to manipulate the independent variables and see the effect on the dependent variables. In other words it is used to find cause and effect relationship and optimize the output.

3.1 Procedure description

The experiment was designed in the followingsteps: define the test plan, development of statistical model, generate suitable calibration, experimental measurement, its optimization and finally validation of the optimized calibration.

3.2 Problem specification

After the baseline determination of the existing calibration the thermal and mechanical limits' margins for the enginewas found to be well within the acceptable limit. Thus the input parameters viz, SoI, rail pressure, AFR, fuel flow.

3.3DoE solution

Half factorial method was used for DoE model. ($2^n/2$) optimized combinations were formed where n is number of independent variables. Here the number of independent variables are 6, therefore 32 combinations for the rated speed and peak torque modes were the optimized output given by MINITABsoftware which is summarized in Table 3. The optimized DoE trends thus obtained are shown in Fig2, these were the trends which were expected when the actual test was carried out.

Table 3. *Concept of DoE*

Operation Points	2
DoE Method	Half Factorial
DoE Points	32 per mode
Optimization Object	Torque, Smoke, Turbo inlet Temperature, Turbo speed, Peak Cylinder Pressure
Optimization Constraints	Rail Pressure, SOI, Pilot fueling, Pilot Separation, AFR, Total Fueling

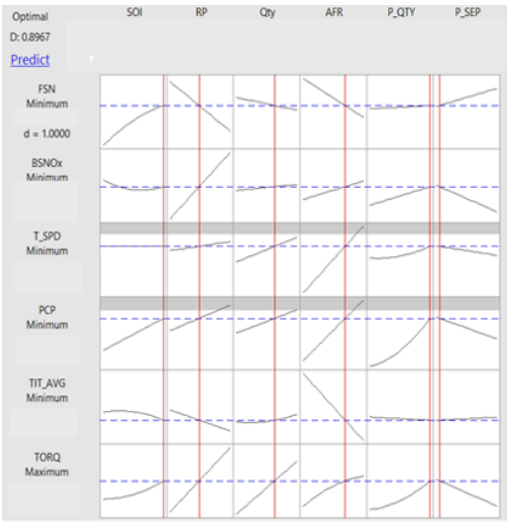

Fig 2. *DoE's result at rated speed*

The trends shown in Fig 2 are obtained by running the DoE at the rated speed viz., 2200 rpm.

3.4 DoE Validation

The different values of the engine parameters that were found out during the DoE needed to be validated experimentally, thus these values were actually fed to the ECM and the engine was run at these points within the engine's duty cycle. The trends obtained experimentally were in agreement with the one's obtained during the simulated DoE, thus validating the simulated results of MINITAB.

4. Development Test

The engine was coupled with the test cell equipment mentioned in Table 2, and the tests were performed. These boundary conditions were kept the same for all the engines that were used for the testing activity viz. the base engine (which was used to determine the baseline values of all the parameters), the development engine and the engines used for consistency

1. First the engine is warmed up gradually by increasing the load in steps of 10% of the full load, then after the coolant out temperature has been attained the remaining boundary conditions are set as shown in table 4:

Table 4. *Boundary conditions*

Parameter	Value
Intake manifold temperature (°C)	48 ± 2
Charged air cooler ΔP (kPa)	11.5 ± 0.5
Exhaust back pressure (kPa)	9.5 ± 0.5
Air intake restriction (kPa)	3.74 ± 0.1
Air inlet temperature (°C)	25 ± 2
Relative humidity (%)	40 ± 2
Fuel inlet temperature (°C)	39 ± 1
Coolant outlet temperature (°C)	88 ± 2

The above conditions are maintained at the rated speed i.e. 2200 rpm.

2. The boundary conditions once maintained are not changed manually and are allowed to drop in free-fall.
3. The input/independent variables for the preparation of the DoE are captured.
4. The optimized points of the DoE are run and validated.
5. DoE points with repetitive values of independent variables were omitted for reducing the number of trials.
6. The changes in the input values for the new calibration were made in the SoI, rail pressure, AFR and fueling quantity, these trends are shown in Fig 3, Fig4, Fig 5 and Fig 6.

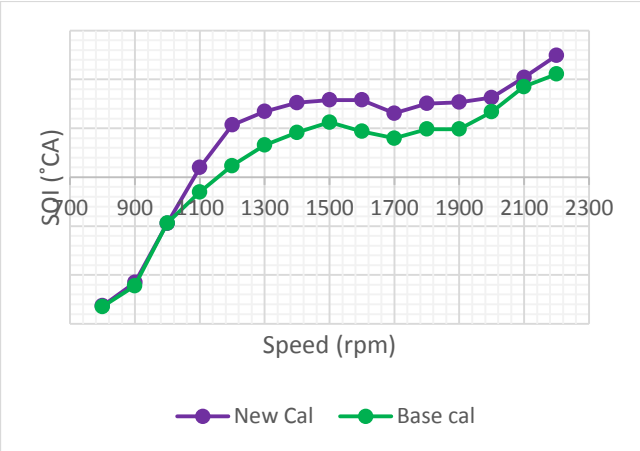

Fig 3. SoI trend in new calibration v/s base calibration

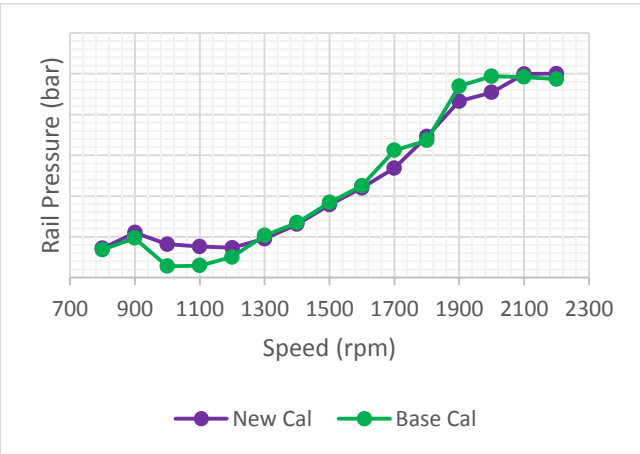

Fig 4. Rail pressure trend in new calibration v/s base calibration

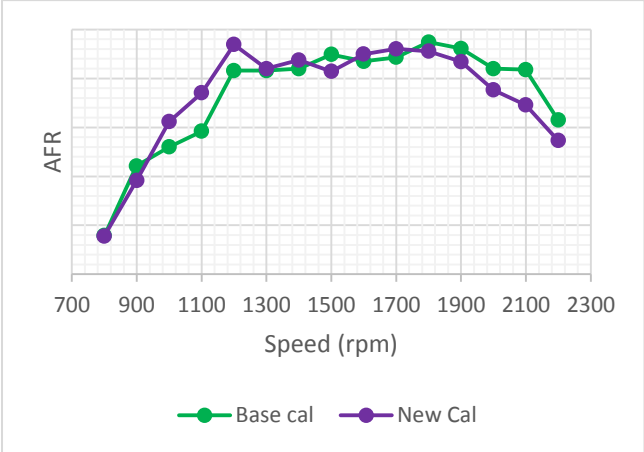

Fig5. AFR trend in new calibration v/s base calibration

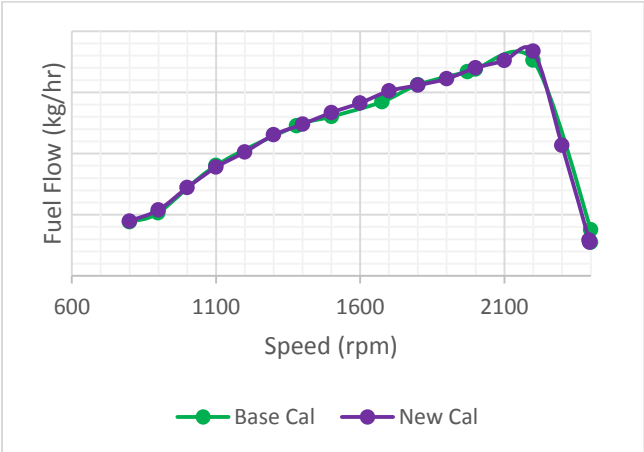

Fig 6. Fuel flow trend in new calibration v/s base calibration

5. Result and Discussion

The optimized DoE gave the targeted results and the target increase is achieved in torque and power (3.12%)over the base calibration and is shown in Fig 7.

The new calibration thus formed was supposed to be a robust one and the consistency was checked by performing test on 3 engines. These values were validated and the graphs shown in Fig7 and Fig 8 are the average of the performance of 3 engines' trial.

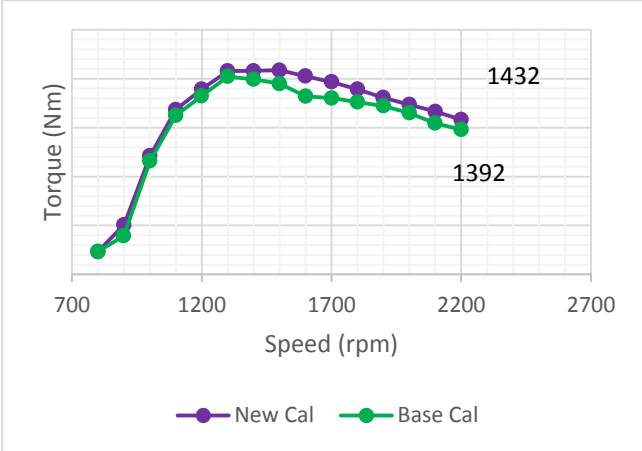

Fig7. Torque curve new calibration v/s base calibration

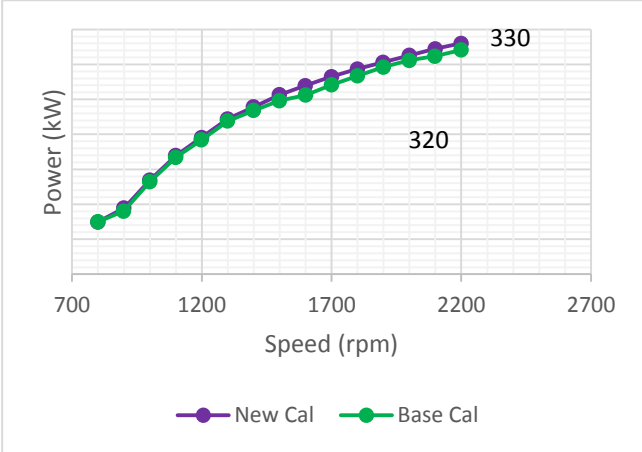

Fig8. Power curve new calibration v/s base calibration

With the change in the fueling values at different points in the duty cycle, the total fueling for the system was changed, this changed the value of the total heat added to the system. The new calibration was required to achieve the target without causing any outliners which might cause the hardware to be changed. Even with the increase in the total value of heat added to the system, the percentage of heat rejected to each of the constituents remained the same, as no physical changes were made to the engine or its peripherals, thus the Heat Rejected (HR) with the old calibration shown by Fig 9 and the Heat Rejection (HR) with the new calibration shown by Fig 10 were also supposed the same or within the limits defined for the auxiliary components.

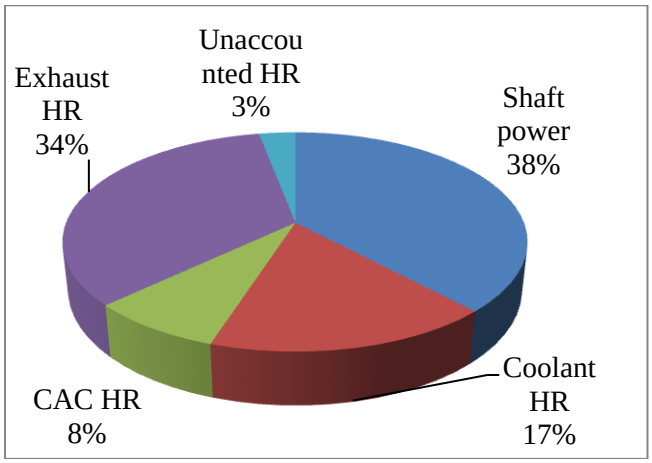

Fig 9.HR @ rated power with base calibration

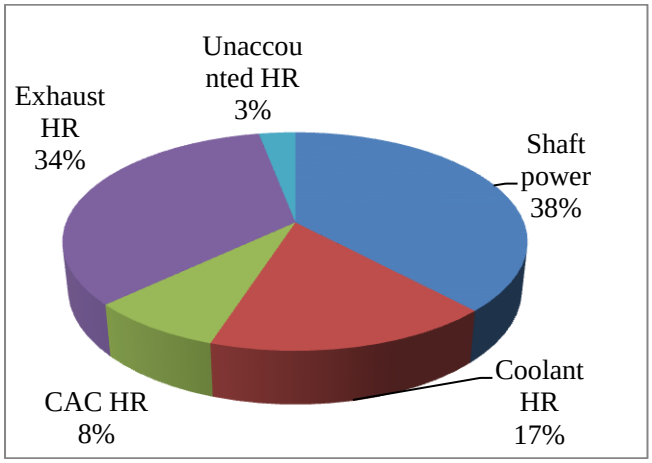

Fig 10. HR @ rated power with new calibration

Thus we can conclude the new calibration is a robust one as it was consistent and within the engineering margin. As the target values of power and torque were achieved, the other major constraint was that of no hardware change and that was validated by the heat rejection trials which shows that no new heat exchanger was required for dissipating the engine's heat.

The other hardware which were at a risk of reaching their limits were the cylinder block and head assembly and the turbocharger. The block and head assembly were found to be within the safe operating limits as shown in the Fig 11 at all the points except the rated point with full load, but that can be accepted as the engine never runs at this point in its entire duty cycle.

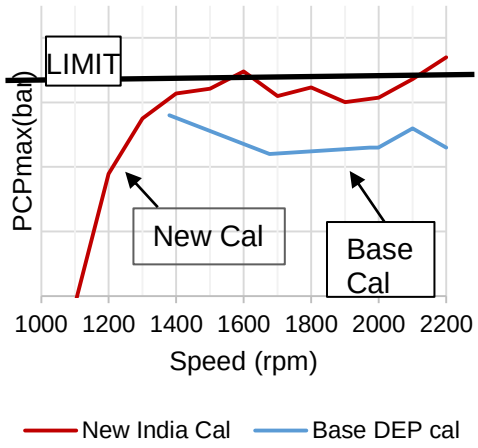

Fig 11. PCP trend new calibration v/s base calibration

The other component, the turbocharger too was at a risk only by the extreme Turbo inlet temperature (TIT), but was controlled by the increased AFR. As shown by Fig 12, the TIT was also found to be within its thermal limit.

Even with the change in the calibration, the AFR increased compensated for the increase of TIT. The increase in the TIT at the worst point in the duty cycle is still only 1.8% over that with base calibration

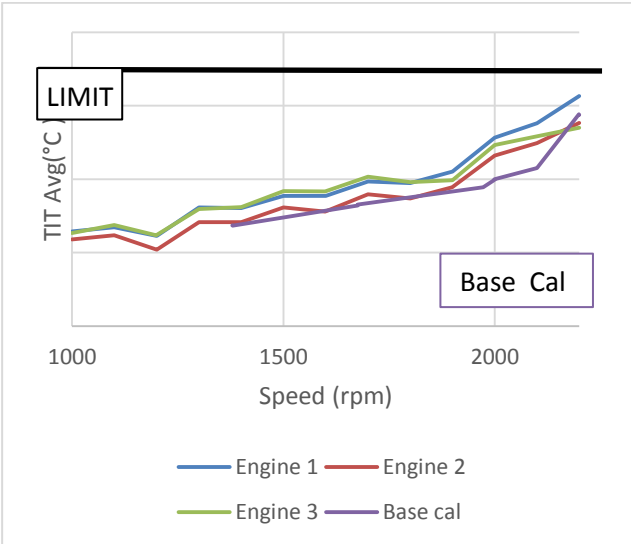

Fig 12. 3 engine TIT data with new calibration v/s base calibration

6.Conclusions

- As we advance the start of injection (SOI), the combustion (higher combustion efficiency)in terms of crank angle increases which in turn increases thepower from 320kW to 330 kW.
- The rail pressure is also increased in the new calibration which in turn causes better atomization of the fuel mist thus contributing in increasing the efficiency of combustion.
- The increase in Air-fuel ratio (AFR) although increases the Peak Cylinder Pressure (PCP) but it contributes in lowering the Turbo inlet temperature (TIT) which helps the component (turbocharger) to be within the thermal limit.
- Lastly, increasing the fueling quantity will increase the amount of energy fed into the engine, which helps to achieve the desired output.

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HP31703806-Testing and calibration development of Generic Cummins 9 liter engine to meet BS IV emission norms and customer demands

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Abstract

With the enhancement in transportation luxuries for human beings, the amount of exhaust gases emitted from various vehicles is increasing rapidly. Emission control on vehicles is necessary to limit the amount of pollutants and environmentally damaging substances released from these vehicles. Every vehicle must comply with the respective judicial emission norms with minimum tolerance to be passed for production. This work aims to develop a Cummins 9 liter engine to meet the Bharat Stage IV emission norms. Also, customer defined performance parameters i.e. duty cycle torque and Brake Specific Fuel Consumption (BSFC) are to be met. This can be achieved by testing the engine and calibrating the engine's Electronic Control Unit (ECU) with reference to a base calibration in the engines. ECU combustion parameters like fuel injection timing, rail pressure, and fuel injection quantity and oxygen fuel consumption are optimized to achieve the desired results. Upon optimizing the above parameters, emission exhausts like unburned hydrocarbons, carbon monoxide and particulate matter are found to pass the Bharat Stage IV emission norms along with the customer defined demands. The optimization of NOx emissions is beyond the scope of this project since its emission development has to be carried out by a different entity in Cummins India Limited.

Keywords:*Cummins ISLe8.9, Bharat Stage IV emission norms, Electronic Control unit (ECU), fuel injection timing, rail pressure, fuel injection quantity, oxygen fuel consumption.*

1. Introduction

With improved modes of transport and their affordability, the world nowadays relies a lot on them for quick and rapid functioning. Automotive vehicles take the premier slot when it comes to vehicle consumption by today's society. With the ever increase in number of vehicles, the amount of pollutants and other undesirable products released by these vehicles is increasing rapidly. Strict norms have been set to regulate the vehicular environmental impact in every region. Any vehicle aimed at a particular market must legally comply with the emission norms of that judicial system with minimum tolerance. For India, Bharat Stage emission norms have been defined for every category of vehicle i.e. petrol and diesel vehicles in light duty, medium duty and heavy duty sector. Many amendments have been made in Bharat Stage norms to reduce the amount of hazardous emission gases. Automobile companies are facing a lot of challenges to limit the emission of such products while maintaining the affordability of vehicle.

Combustion of fuel inside the cylinder releases the chemical energy of fuel in the form of thermal energy. Since the combustion takes place at extremely high temperatures, the incomplete combustion of fuel releases exhaust gases like unburned hydrocarbons, carbon monoxide, oxides of nitrogen, particulate matter etc. The chemical energy of the fuel is distributed to engine shaft power, heat to engine coolant, heat to intercooler, heat to exhaust gases and to some unaccounted losses. The consequences of exhaust gases have been proven with the increase in number of respiratory disorders among public along with the ever increasing global warming.

Engines with electronic fuel injection technology (common rail diesel injection) have the Electronic Control Unit (ECU) to govern the engine functioning. The ECU enables precise control over a common rail fuel injection system. The main combustion parameters to be manipulated for reduced engine emissions are rail pressure, fuel injection quantity and fuel injection timing. Depending on the application, an engine can have pilot or post injections.

Walke N.H. et al. (2009) developed a 3.9 liter diesel engine to meet Euro IV emission norms. A base 3.9 liter TCIC engine was upgraded to meet Euro IV norms. The mechanical inline fuel injection pump was replaced by high pressure common rail fuel injection system. By developing suitable combustion mapping the ECU was calibrated to meet the targeted emission levels.

P.R. Ghodke et al. (2014) reviewed multi-cylinder diesel engines to meet future Indian emission norms. Vehicle simulation model was developed to meet Euro

V emission norms. Base engine was also developed with a new piston bowl of lower compression ratio, high capacity EGR cooler and turbocharger to attain desired emission levels.

Mayur S. Sawade et al. (2015) carried out performance and emission optimization of single cylinder diesel engine to meet BS IV norms. The base engine's (BS III) capacity was increased along with certain modifications like changes in fuel injection system of vehicle with exhaust after treatment devices.

This project aims to develop a generic Cummins 9 liter engine to meet Bharat Stage IV emission norms along with the customer specific demands with most of the engine hardware being localized. The base calibration used for development is from Darlington, UK. Testing is carried out on the engines to verify the base calibration. Further testing and development of base calibration is done on the engines to achieve the desired target. Some engine parts like ECU, turbocharger etc. is imported but the rest of engine is built in India. The reduction of NOx emissions is associated with a different entity in Cummins and hence is beyond the scope of this project.

The Bharat Stage IV emission norms which are to be fulfilled are:

Table 1 Bharat Stage IV norms for heavy duty diesel vehicles

Test	CO (g/kWhr)	HC (g/kWhr)	NOx (g/kWhr)	PM (g/kWhr)
ESC	1.5	0.46	3.5	0.02
ETC	4.0	0.55	3.5	0.03

ELR smoke (m^{-1}): 0.5

2. Objectives of experiment

1) To carry out testing of ISLe8.9 engine with base calibration.
2) To carry out analysis of baseline data.
3) To develop and verify an optimized calibration which meets customer defined energy targets and also comply with Bharat Stage IV emission norms.

3. Existing system

The existing system used is a Cummins Interactive Series ISLe engine with a light weight, compact size 9 liter configuration.

The specifications of test engine used are as follows

Table 2 Engine specifications

Rated power	400 hp (294 kW)
Peak Torque	1700 Nm at 1400 rpm
Number of cylinders	6
Displacement	8.9 liters
Fuel system	Cummins common rail
Bore × Stroke	114 mm × 144.5 mm
Oil system capacity	27.6 liters
Dry weight	706 kg
Injector make	BOSCH
Turbocharger	Wastegated

4. Engine Test cell Layout

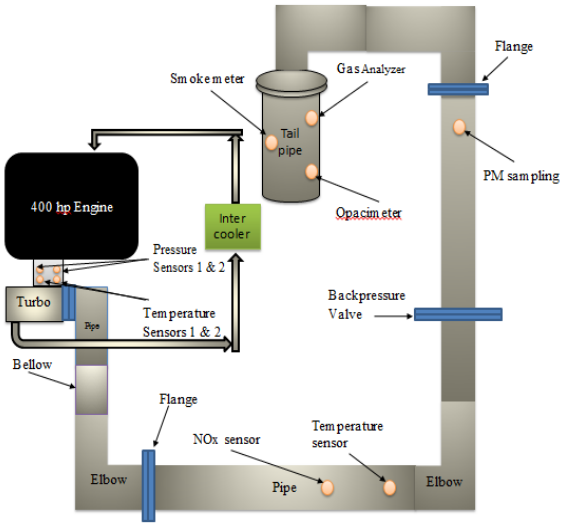
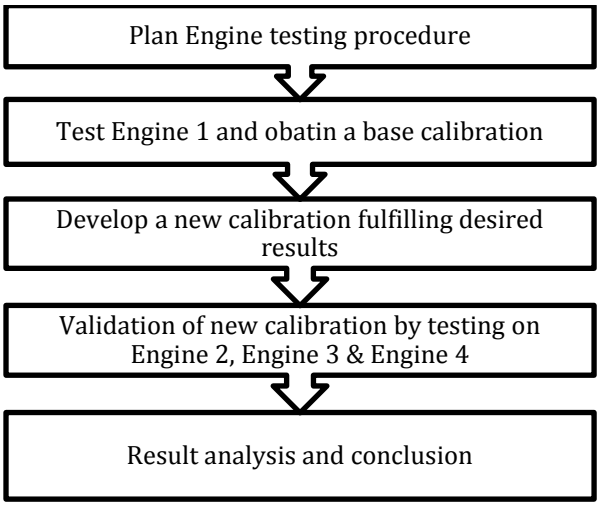


Fig.1 Engine Test cell layout

The complete testing of ISLe8.9 is done in Test Cell 14 of Powertrain Engineering department of Automotive Research Institute of India, Pune. The dynamometer installed in the test cell is a HORIBA 460kW water cooled type dynamometer. The total length of exhaust pipeline is set as per customer specifications. Suitable adjustments have been done considering the layout of Test Cell. Flexible bellow is used to dampen the engine vibrations on the exhaust pipeline. A backpressure valve is used to maintain the exhaust backpressure boundary condition suitable for emission control. AVL415 smoke meter is installed near the tail pipe. HORIBA emission analyzers are installed for emission measurement. AVL SPC is connected for sampling PM particles on a filter paper using gravity sampling effect.

5. Methodology



6. Emission Regulation Cycles

For emission measurement of heavy duty diesel engines some regulation cycles are used. These are European Stationary Cycle (ESC), European Transient Cycle (ETC) and European Load Response (ELR) test.

6.1 European Stationary Cycle

This is a 13 mode steady state procedure for measurement of emissions. The engine is operated for the prescribed time in each mode, completing engine speed and load changes in first 20 seconds. Emissions measurement is done during each mode and averaged over the cycle using a set of weighing factors. For measurement of particulate matter, filter paper sampling is done over the 13 modes. The final emission results are expressed in g/kWh. The engine modes during the cycle are defined by using the high speed and low speed. The high speed (n_{hi}) is the highest speed where 70% of declared maximum net power occurs. The low speed (n_{lo}) is the lowest speed where 50% of the declared maximum net power occurs.

$$A = n_{lo} + 0.25(n_{hi} - n_{lo}) \quad (1)$$
$$B = n_{lo} + 0.50(n_{hi} - n_{lo}) \quad (2)$$
$$C = n_{lo} + 0.75(n_{hi} - n_{lo}) \quad (3)$$

6.2 European Transient Cycle

This cycle is used to measure real road emissions of heavy duty vehicles. The different driving conditions are divided in three parts namely urban, rural and motorway driving. The entire cycle is of 1800 seconds. The duration of each part is 600 seconds.Urban part represents city driving with a maximum speed of 50km/h, frequent starts, stops and idling. Rural driving starts with steep accelerationwhere average speed is about 72 km/h. while in motorway driving the average speed is about 88km/h.

6.3 European Load Response:

This test is used for measurement of smoke opacity from heavy duty diesel engines. The test consist of a sequence of three load steps at each of the three engine speeds A (cycle 1), B (cycle 2) and C (cycle 3), followed by cycle 4 at a speed between speed A and speed C and a load between 10% and 100%, selected by the certification personnel.

7. Modifications in the calibration:

7.1 Calibration for fuel injection quantity

Fuel injection quantity is a critical parameter to achieve the desired torque in any ECU calibration. The base calibration had no pilot or post injection. The effect of fuel injection quantity on main injection in the base calibration was studied. Some changes in fuel injection quantities (mg/stroke) were made taking the base calibration as reference for the new calibration at suitable points to achieve desired flat torque curve. These changes were reflected in the ECU 3D map containing speed (rpm), torque (Nm) and fuel quantity (mg/stroke).

7.2 Calibration for oxygen fuel consumption

The air fuel ratio has a great influence on proper combustion in a diesel engine. During the testing of local engines, the particulate matter emission results were crossing the desired limits in transient operation. To bring down the particulate matter emissions, the oxygen consumption of engine at high smoke points in the ETC cycle was optimized. This is done by the optimizing the equivalence ratio of the engine.

$$\text{Equivalence ratio} = \frac{\text{Actual fuel/air ratio}}{\text{Stoichiometric fuel/air ratio}} \quad (4)$$

Increasing the equivalence ratio increases the oxygen-fuel ratio in the combustion. This results in sufficient amount of oxygen for the liquid fuel to burn off prior to the end of combustion. Also enrichment of oxygen improves the fuel oxidation and suppresses the formation of soot.

The equivalence ratio was changed at points of high smoke spikes in the ETC cycle to achieve the desired particulate emission results.

7.3 Calibration for fuel injection timing

Start of injection is the time at which injection of fuel in the combustion chamber begins. It is expressed in crank angle degrees relative to TDC of the compression stroke. In order to reduce the particulate matter emissions, the start of injection was advanced. This resulted in more time for fuel to burn resulting in more complete combustion during the progress of compression stroke. This achieved a high peak temperature resulting in lower formation of particulate matter.

7.4 Calibration for rail pressure

In a common rail system, fuel is distributed to the injectors from a high pressure accumulator called rail. The rail is fed by high pressure fuel pump. High rail pressure enables better spray penetration and mixing leading to better atomization of fuel. Small droplets of fuel have better chance to vaporize and participate in combustion.

The rail pressure at suitable points was increased leading to low HC, CO and soot emissions but increasing the NOx emissions slightly.

Table 3 Summary of modifications in base calibration

Parameter	Action	Effect
Fuel injection quantity	Fuel injected quantity manipulated (mg/stroke) at duty cycle points	To achieve uniform torque curve at duty cycle speeds as per customer demands
Oxygen fuel consumption	Equivalence ratio increased up to 15% at points of higher smoke spikes	To reduce the amount of particulate emissions in transient conditions
Injection Timing	Start of injection advanced	To achieve high peak temperatures leading to lower PM emissions
Rail Pressure	Rail pressure increased up to 10% at certain points	To achieve better atomization of fuel to reduce HC, CO and soot emissions

8. Results

Various results involved in this development are shown with performance parameters like torque, power, brake specific fuel consumption (bsfc), air fuel ratio along with the emission results for both ESC and ETC cycle for all the engines. The results are based on comparison of base engine calibration and final revised developed calibration.

8.1 Torque

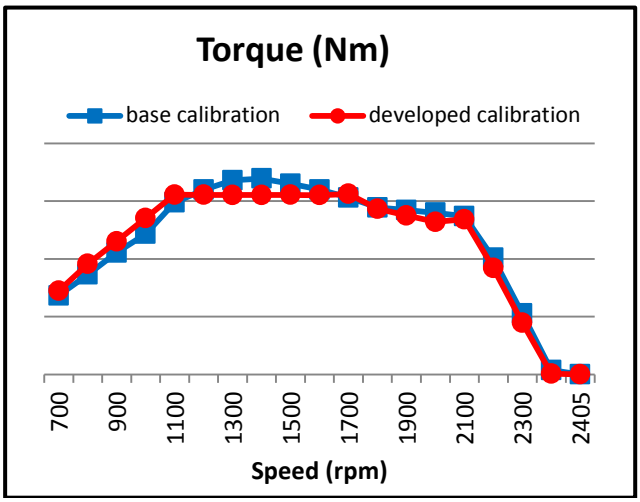


Fig. 2 Torque (Nm) vs. Speed (rpm)

Appropriate changes in fuel injection quantity were made to achieve a flat torque curve in the duty cycle region.

8.2 Power

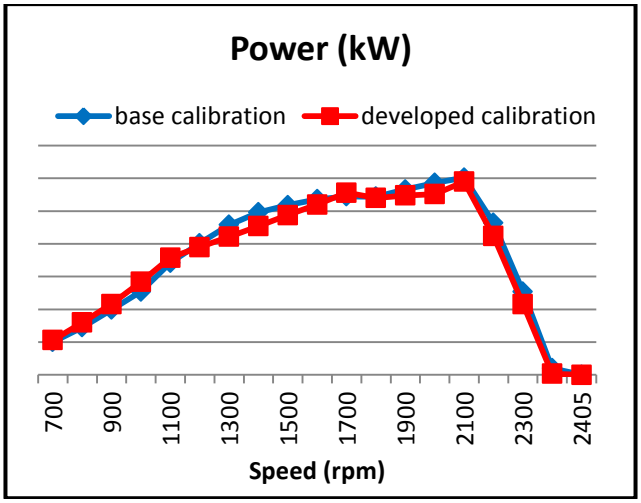


Fig. 3 Power (kW) vs. Speed (rpm)

The power curve corresponds to changes in the torque curve. Due to reduction in fuel injection quantities at certain points, some amount of power is dropped in the final revised calibration which is within the acceptable band of the base calibration.

8.3 Brake specific fuel consumption (BSFC)

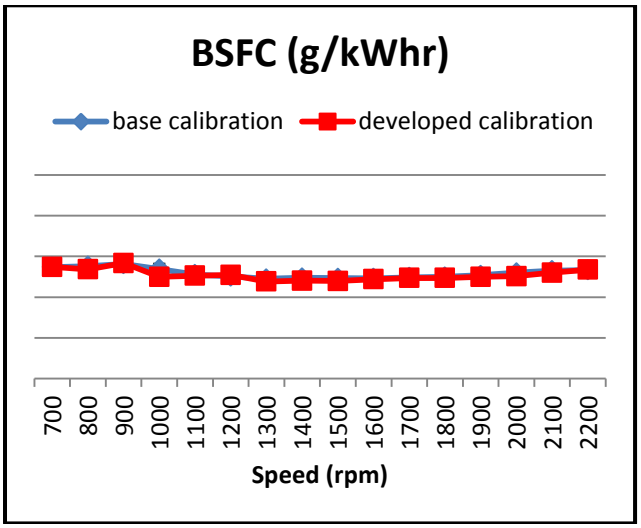


Fig. 4 BSFC (g/kWhr) vs. Speed (rpm)

The optimal developed calibration resulted in reduction in brake specific fuel consumption by changing the fuel injection quantities with respect to the base calibration. The reduced fuel injection quantity at same power produced lower brake specific fuel consumption than the base calibration.

8.4 Air fuel ratio

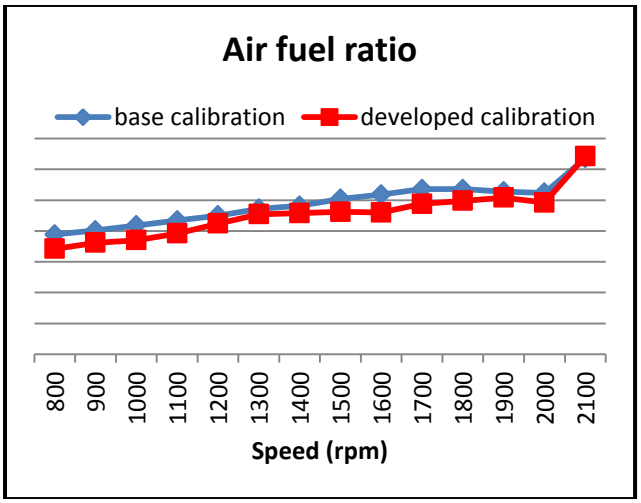


Fig. 5 Air fuel ratio vs. Speed (rpm)

Fig. 4 shows the air fuel ratio vs. the speed for the base and final revised developed calibration. The final revised calibration bears a lower but much more suitable air fuel ratio than the base calibration. This was achieved with a lower air intake along with lower fuel intake quantity. The new air fuel ratio is closer to stoichiometric air fuel ratio.

8.5 Regulation Cycle results

The whole engine development took place by testing four engines with both steady state and transient operation cycles. Results from Engine 1 signify the base calibration testing and verification. The base calibration was then revised and developed over testing of three other engines. Of these 4 engines, Engine 3 (local made) showed high blowby readings. So Engine 3 was given for tear down analysis. The number of tests and thus their results on Engine 3 are lower in number as compared to the other engines.

Blowby is the amount of fuel, air and moisture leaked into the crankcase leading to pressure buildup inside the combustion chamber. As blowby increases significantly, it can allow oil to leak into combustion chamber leading to severe destruction of piston and chamber.

The tear down analysis concluded that the cylinder liners of Engine 3 had initiated scuffing due to intense contact between the top piston ring and the cylinder liner bore on the anti-thrust side of the cylinder liner. The same was reflected on the piston ring in the form of barrel drop. The emission results for both the regulation cycles are further shown.

8.5.1 ETC cycle results

For on highway application of any engine it is important that it provides suitable transient response while simultaneously fulfilling the transient emission norms for heavy duty applications. The results for four European Transient Cycle tests are shown for Cummins 9 liter engines.

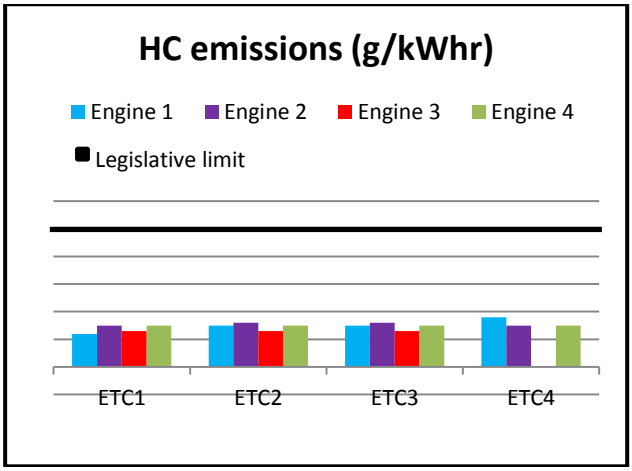


Fig. 6 HC emissions for ETC tests

Fig. 6 shows the HC emissions for the ETC tests for all the four engines. Engine 1 with the base calibration gave favorable unburned HC emissions. Engine 3 gave lower HC emissions as the high blowby effect caused much of the HC to leave via crankcase ventilation to atmosphere rather than through turbine.

Engine 4 with the final revised calibration gave the best HC result; this is because of increase in oxygen content in the combustion giving the liquid fuel

sufficient oxygen to burn off before the end of combustion.

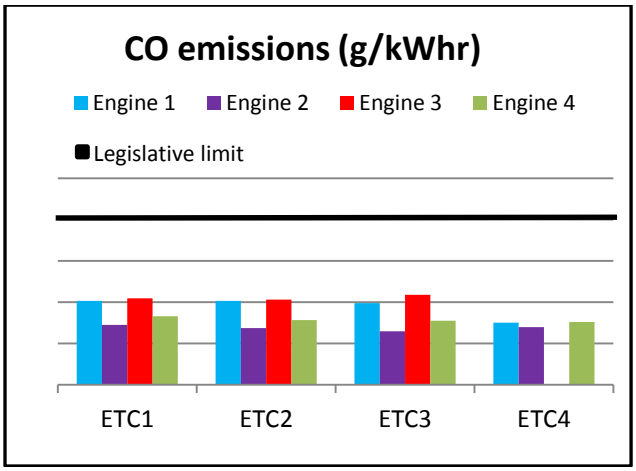


Fig. 7 CO emissions for ETC tests

The carbon monoxide emissions for all four engines are shown in Fig. 7. The base calibration gave maximum CO emission with Engine 1. Engine 2 achieved 30% reduction in CO emissions from the base calibration. CO formation takes place mainly because of insufficient oxygen for complete oxidation. Advancing the injection timing allows a more time and thus more complete combustion reducing the CO emissions.

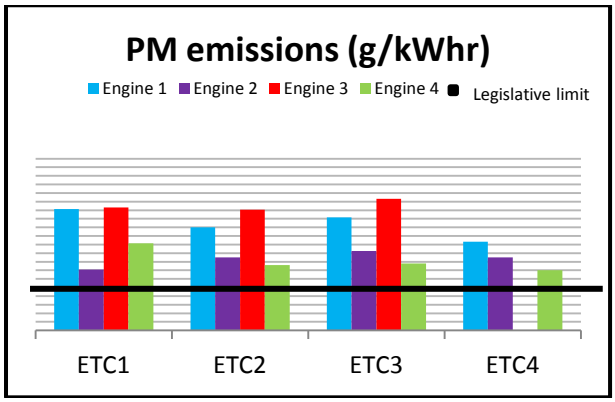


Fig. 8 PM emissions for ETC cycle

Particulate matter is generated due to formation of soot in the combustion leading to increased smoke density. Oxygen enrichment also reduces ignition delay leading to higher burning rate and shorter combustion duration resulting in decrease in soot formation.

The base engine calibration gave high PM emissions with Engine 1. With revision of the base calibration, 17% reduction in PM emissions was achieved for Engine 2.

Engine 3 with high blowby gave the highest PM emissions because of high soot formation. The final revised calibration with enhanced oxygen consumption, advanced injection timing and high rail pressure achieved 30% drop in PM emissions with Engine 4.

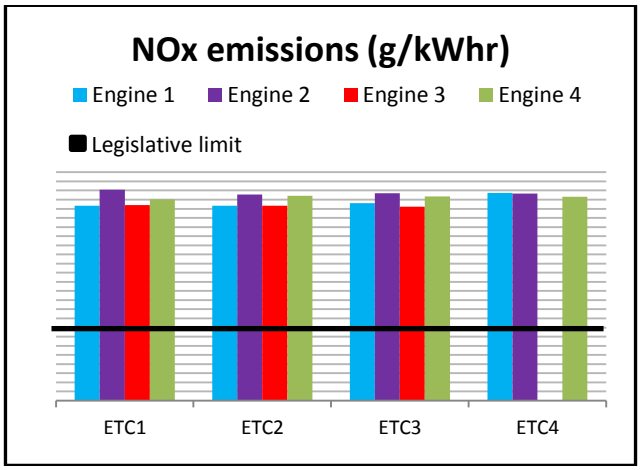


Fig. 9 NOx emissions for ETC tests

NOx emissions are a reaction product of nitrogen at high temperature and oxygen enrichment. Higher flame temperatures and oxygen concentrations during the combustion leads to formation of high NOx emissions.

Engine 2 emitted highest amount of NOx among all. Due to the NOx-PM emissions trade off factor, revised calibration emitted slightly higher NOx than the base calibration. This is because increasing oxygen content leads to a much better combustion at higher temperatures reducing the PM emissions and increasing NOx in the exhaust. Also, advancing the injection timing only increases peak temperature favoring higher NOx. These NOx emissions can be controlled with the use of aftertreatment devices.

8.5.2 ESC cycle results

Engine 1 with the base calibration underwent only two ESC tests for verification since the emission results were very much under the legislative norms.

Although a revised calibration was used on Engine 3, but due to its high blowby readings it had to be tear down to know the root cause of high blowby and high smoke values. Hence only one ESC test was done on Engine 3.

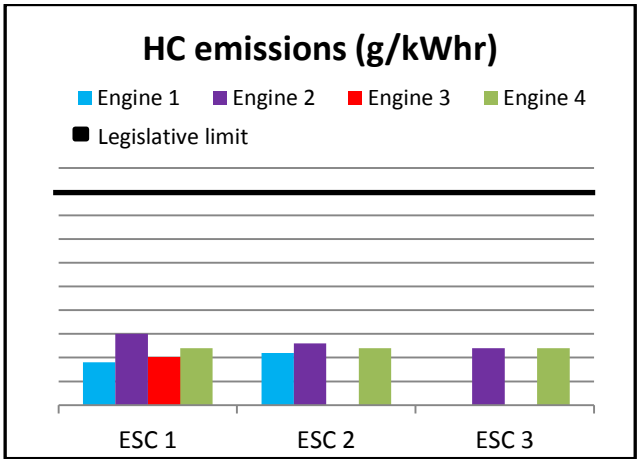


Fig. 10 HC emissions for ESC tests

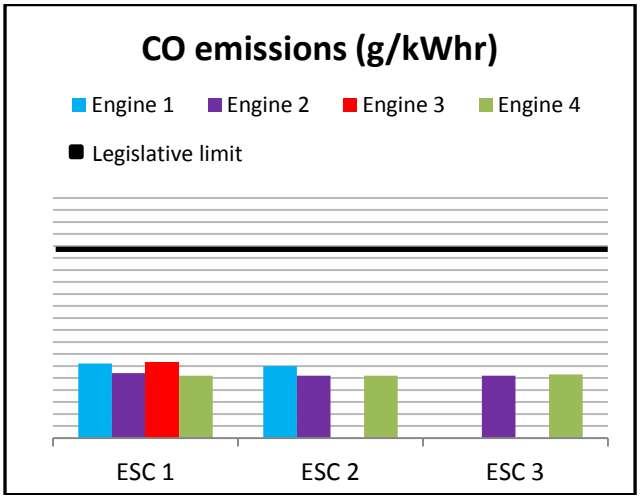


Fig. 11 CO emissions for ESC tests

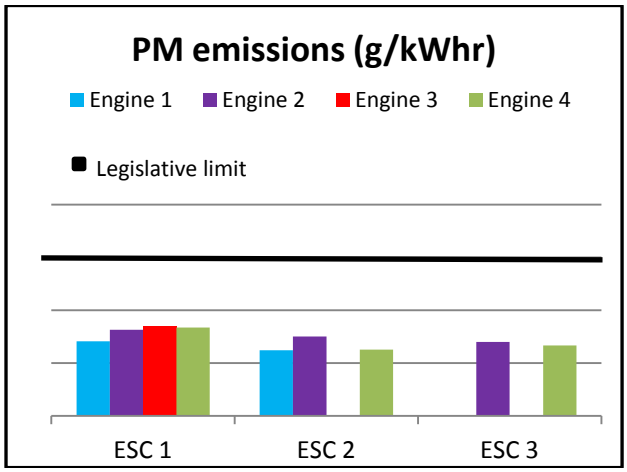


Fig. 12 PM emissions for ESC tests

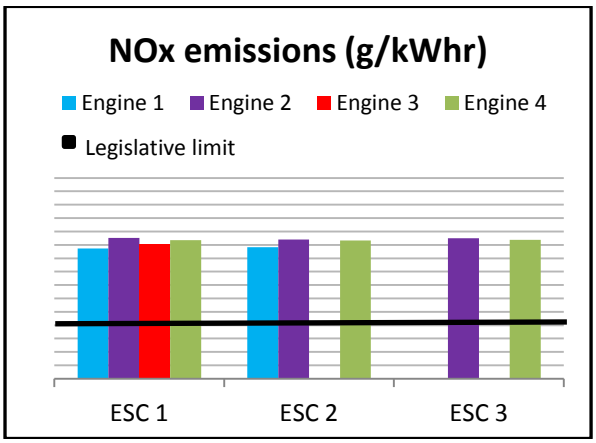


Fig. 13 NOx emissions for ETC tests

The base calibration gave best results for HC emissions for steady state operations. The consistency of these results is also proved by Fig. 10 and all are passing the legislative limit.

Fig. 11 shows the CO emissions for the 4 engines. A 14% drop in CO emissions was achieved for developed calibration with respect to base calibration.

The particulate matter emissions for steady state operations are shown in Fig. 12. For steady state operations, 5% reduction in PM emissions was achieved for developed calibration from base calibration.

Steady state operations showed higher NOx concentrations than transient operations. Engine 1 released minimum NOx among all the engines. Engine 2 and Engine 4 with advanced injection timing emitted higher NOx.

8.5.3 ELR cycle results

The smoke opacity instruments measure optical properties of diesel smoke, providing an indirect way of measuring diesel particulate emissions.

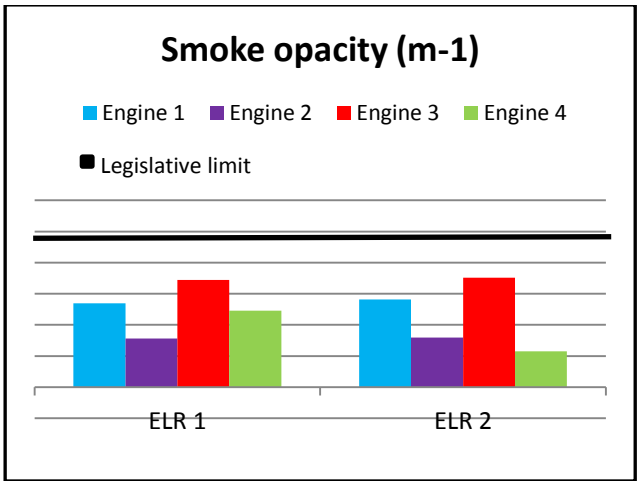


Fig. 14 Smoke opacity results for ELR tests

Engine 2 achieved 40% reduction in smoke opacity values as compared to base calibration. After revising the calibration Engine 3 with high blowby correspondingly showed increase of 25% in smoke opacity. Engine 4 ran at a calibration involving high oxygen consumption at high smoke spikes, reducing the smoke opacity to 60% compared to the base calibration.

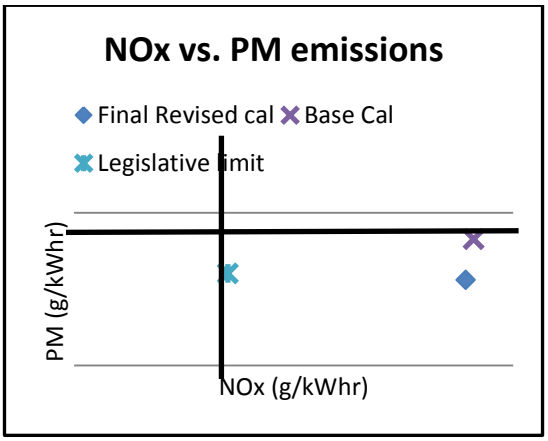


Fig. 15 NOx vs. PM emissions

The final revised calibration achieved engine out Particulate Matter emissions under the legislative limit. NOx emissions can be brought down under the limit by implementing the aftertreatment devices e.g. Selective Catalytic Reduction (SCR) which has great NOx conversion efficiency.

SCR uses ammonia as a reducing agent which converts the exhaust NOx into inert N2 gases thus reducing the amount of NOx emissions from the engines. SCR have been found to be 80% efficient in NOx conversion.

9. Conclusions

The development of Cummins ISLe8.9 engine keeping all parts localized (except turbocharger, ECU) to meet Bharat Stage IV emission norms and simultaneously satisfying customer specified demands is achieved.

1) A flat torque curve in duty cycle region is achieved by tuning the fuel injection quantity at suitable regions.
2) In the process a 5% reduction in brake specific fuel consumption is achieved thus meeting customer requirements.
3) Engine 3 faced high blowby issues and hence a tear down analysis was done which reflected initiation of scuffing on the cylinder liner.
4) Advancing the start of injection along with increased rail pressure lead to reduced PM emissions but increased NOx emissions.
5) By increasing the oxygen consumption, better oxidation of fuel is achieved.
6) Increasing the oxygen consumption also result in lower soot formation leading to reduced particulate emissions in transient operations.

10. Future Scope

1) The optimal calibration for engine emissions always has to give the best trade-off between NOx and PM emissions.
2) The high NOx emissions for the calibration can be brought down by the use of aftertreatment devices.

Nomenclature

n_{hi}– highest speed at which 70% of the declared maximum net power occurs

n_{lo}– lowest speed at which 50% of the declared maximum net power occurs

A, B, C speeds – speeds used during 13 mode ESC cycle

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HP31703814-Analysis of Four Cylinder Diesel Engine using Biodiesel and Blends

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Abstract

A good amount of research is being done the world over to search for alternative fuels to be used in engines. Biofuel,such as alcohol and biodiesel,could partly replace petroleum fuel, reduce toxic emissions and restrain the life cycle emission of CO. After many trails single cylinder diesel engines have shown positive results and biodiesel such asmoringa, cotton seed etc as a fuel can be used in diesel engines. The next stage is the analysis and research of such biodiesel fuels in four cylinder diesel engines. In view of this an attempt has been made to investigate the diesel engine performance and emission characteristics of jatropha ,karanja and cotton seed oil and its blends and compare its performance with diesel engine. Three types of blends with three different types of biodiesel were tested and results obtained showed positive trends.

1. INTRODUCTION

The drastic use of fossil fuels the world over has made man to search for alternative fuels. The fuels which can be used in transport sector for effective utilization are preferred. Biodiesel over the period of time has emerged as an alternative for fossil fuels. It is renewable and non toxic.

Biodiesel is produced from triglycerides. Triglycerides are substances which have high viscosity and are composed of 3 long chain fatty acids. They can be converted to biodiesel by transesterification reaction. This conversion is done by reacting them with alcohols in presence of a catalyst. Transesterification results in reduction in viscosity and increasing the cetane number. Biodiesels of various types have been used for research by various authors.They have been applied in diesel engines directly with no or a small modification. Experiments and analysis have concluded that there is a substantial reduction in hydro carbons, carbon monoxide, and particle-mass emissions .Common biofuels such as n-butanol,diethyl ether etc. blended with diesel have also been tested on engines.Ethanol has been tested and havegiven positive results.It has got high heat of evaporation which helps in reducing NOx.Its high oxygen content reduces PM.

Biodiesel helps to reduce smoke, PM, CO and HC emissions.The NOx emissions are high and the engine efficiency is not changed or sometimes increased.

Many fluid properties such as viscosity,heating value and density are important parameters that effect the engine performance as a whole.The efficient use of biodiesel helpsto enhance the efficiency of engines and make a pollution free environment.

2. LITERATURE REVIEW

In paper [1], author analyzed 20 %.moringa biodiesel which showed positive results.It was compared with palm, jathropha and diesel fuel.Average brake power is lower and brake specific fuel consumption values are higher.

The reduction in CO and HC emissions were seen but NOx emissions increased a little.Palm oil showed better performance than moringa and jathropa.

In paper [2] it was seen that after esterification process acid value reducedsignificantly and after transesterification process the percentageof yield was found more than 90%.Over the entire speed range, B10 and B20 yielded an averagereduction of BP compared with diesel fuel. Meanwhile, the averageBSFC of B10 and B20 was slightly higher than that of diesel fuel.B10 and B20 reduced CO emission and slightly increased NO emissionthan diesel fuel. After the above mentioned findings the author found that M. oleifera is a potential feedstockfor biodiesel production, and B10 and B20 blends can be used as directdiesel fuel substitute.

The paper [3] studied properties of the J. curcas and M. oleifera methylesters and their blends and they agreed with the ASTM D6751 and EN14214 standards.

Over the entire range of engine speeds, the JB10 and MB10biodiesels gave average brake powers of 27.32 and 27.51 kWthat were 5 and 4% lower than B0 fuel, respectively. Theaverage brake specific fuel consumptions were 399 and 406 g/kWh for the JB10 and MB10 respectively, which were slightlyhigher (3 and 5%) than B0 fuel. These results were attributed tothe higher viscosity and density and the lower

energy contentof these biodiesel blends.-Hence it was concluded that the J. curcas and M. oleifera oils are potentialfeedstocks for biodiesel production, and the JB10 and MB10 biodieselscan replace diesel fuel in unmodified engines to reduceexhaust emissions into the environment.

Ignition delay was a parameter which was studied by the author[4] for DBE and ULSD blends.For DBE blends pressure and peak heat release rate are higher. ULSD and biodiesel showed lower values.Higher values of ethanol fraction in blended fuel showed shorter diffusion combustion duration. DBE blends showed lower BSPM and BSPN emissions.

M. M. Rashed[5]used moringa biodiesel blends on amineantioxidant additives (DPPD and NPPD).Oxidation stability was increased when antioxidants in MB20 blends were used. DPPD had a higheroxidation stability with MB20 than NPPD. Thecalorific value was reduced andkinematic viscosity increased due to addition of amine antioxidant.The density,flash point and oxidation stability was also high. The HC emission as comparedwith diesel, was still lower.Blends could be stored safely due to this process.
The author Ahmed Sanjid [6] in his work used kapok and moringa combineddiesel–diesel blends. Combustion performance and emissions were investigated. BSFC values were higher as compared to that of B0. It was due to their lower heating value and higher density.Decrease in engine brake power was also seen. It was compared to diesel fuel . This was due to their higher density, viscosity and lower heating value. NO and CO2 emissions were high due to their higher oxygen contents, saturated fatty acids, in cylinder temperature and pressure etc.
The available literature focuses on the performance study of biodiesel on multicylinder diesel engine using different fluid blends. There is little information available on the performance study of multicylinder engine using different blends simultaneously.
The present study attempts the performance evaluation of multicylinder engine using 3 different oil types with multiple blends with emission studies. The study can be helpful to decide the appropriate biodiesel for commercial engines for marine and generator applications.

3. EXPERIMENTAL SETUP

The engine used was MUL make BSIV CRDI Diesel with microprocessor based engine management system 4 cyclinder ,4 stroke water cooled power 55 kW at 4000 rpm,torque 190 Nm.The capacity was 1248cc,bore 69.6 ,stroke 82mm.The dynamometer is eddy current type water cooled.It has a load sensor, temperature indicator ,fuel flow transmitter ,air flow transmitter,piezeosensor,crank angle sensor.The experimental setup is shown in fig 1.
Specificiations of test engine

Make and model MUL Swift BS IV CRDI	
No. of cylinder	4
Cycle	4 stroke
Bore _ stroke	69.6 mm _ 82 mm
Compression ratio	17.5:1
Rated power	55 kW @ 4000 rpm

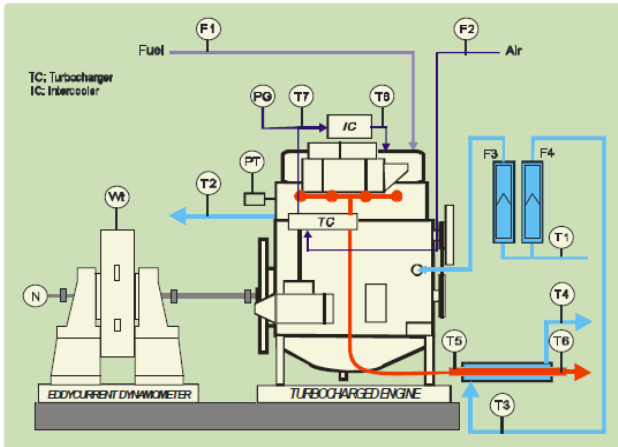

Fig 1. Engine Setup

where, F1 fuel consumption kg/hr, F2 air consumption kg/hr, F4 calorimeter water flow kg/hr,T1 jacket water inlet temperature K,T2 jacket water outlet temperature K,T3 calorimeter water inlet temperature K, T4 calorimeter water outlet temperature K, T5 exhaust gas to calorimeter inlet temp K, T6 exhaust gas from calorimeter outlet temp K.

Test Procedure
In the present work, experiments were carried out with various blends of diesel with Karanja oil jathropa and cotton seed oil (B5, B10, B15, B20etc) were tested on various load conditions.The performance characteristics of the engine with these different working fluids at constant speed were analysed. For optimization of blend, performance characteristics blends were then compared with those of diesel.

4. RESULTS AND DISCUSSION

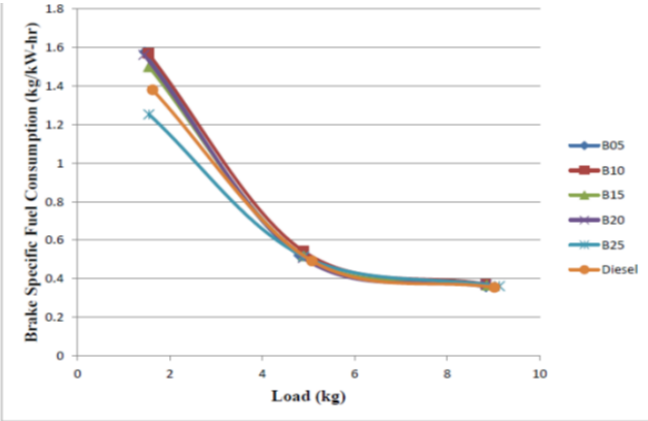

Fig 2. Plot of BSFC Vs Load for Karanja Oil and its blends

Refer figure 2 in which Load and BSFC are inversely related. BSFC increases at high loads owing to increased fuel waste (smoke) associated with high fuel-air ratios. At lower loads, BSFC increases due to decrease in mechanical efficiency.

Under the trial conditions of constant speed and variable load, and at a constant air-fuel ratio, BSFC will rise consistently and rapidly as load is increased. The reason for rapid increase in BSFC with reduction in load is that the frictional power (FP) remains essentially constant while the indicated power (IP) is being reduced. The BP drops rapidly than fuel Consumption and thereby the BSFC rises.

However, we observed that BSFC at low compression ratios and low loads is less than that for higher compression ratios and low loads. Hence, at low loads lower compression ratios can be used.

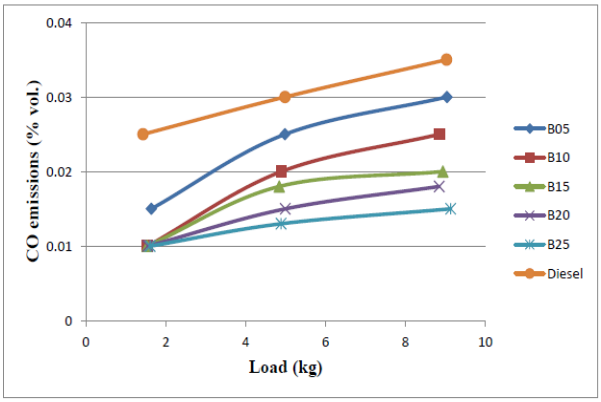

Fig 3.Plot of CO emissions and load for Karanja Oil and its blends.

The variation of CO produced by running the engine when using various blends with compared to diesel is shown in figure 3 at various compression ratios. Higher viscosity leads to difficulties in atomization and also at low speeds. As the resulting local mixture is rich hence more CO is produced and also at low speeds, engine needs more air for the combustion to complete.

Biodiesel has more oxygen content than diesel fuel. This led to more complete combustion. Therefore the CO emissions decreased for biodiesel for as biodiesel percentage increase due to more amount of oxygen available in fuel.

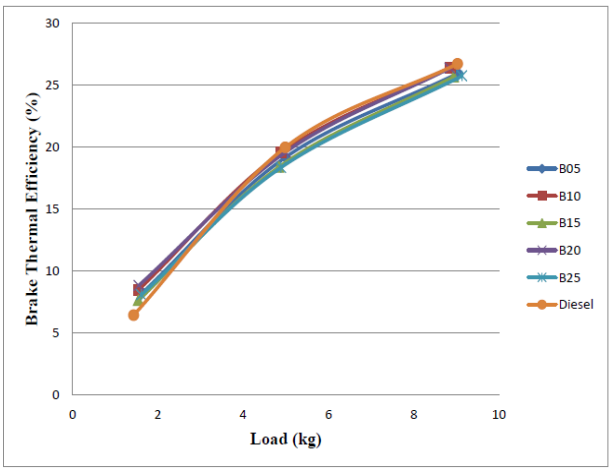

Fig 4.Plot of brake thermal efficiency and load for Karanja Oil and its blends.

The brake thermal efficiency is based on BP and fuel consumption of the engine. The expression for indicated thermal efficiency is given by:

Brake thermal η = Brake Power / (mass flow rate of fuel * Calorific value of fuel)

Brake thermal efficiency gives an idea of the output generated by the engine with respect to heat supplied in the form of fuel. For the current Kirloskar TVS VCR single cylinder diesel engine, brake thermal efficiency varies from 2% to 25% with variation in load for fish oil biodiesel. The corresponding range for karanja oil biodiesel is 5% to 25%.

The brake thermal efficiency increases with increase in load. The expression for brakeFrictional Power remains more or less unaffected by changes in load, while indicated power rises with load. The net effect is increase in brake power proportionate to increase in load.The same trend is also portrayed in variation of brake thermal efficiency with load.

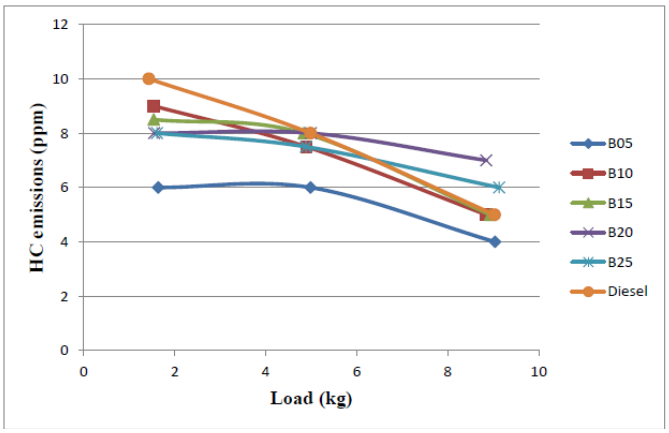

Fig 5.Plot of HC emissions and load for Karanja Oil and its blends.

Karanja oil and fish oil blends generally exhibit lower HC emission at lower engine loads and higher HC emissions at higher engine load compared to mineral diesel. This is because of relatively less

oxygen available for the reaction when more fuel is injected into the engine cylinder at highengine load.

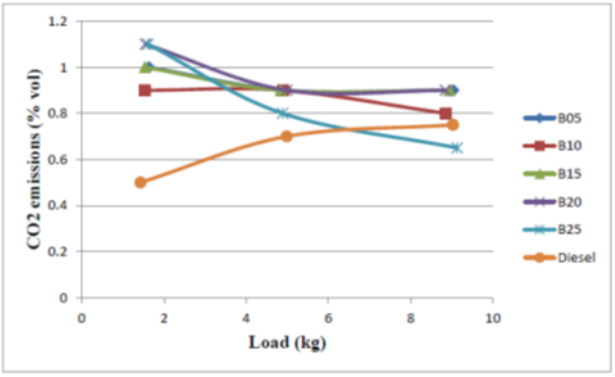

Fig. 6. Plot of CO2 emission and load for Karanja Oil and its blends

A CO2 emission represents complete combustion of fuel this is observed more than diesel due to more oxygen available in fuel. For to reduce the CO emissions EGR method can be utilized which will reduce the emissions so that emissions will be under the specified limits of BS IV.

5. CONCLUSION

- In the current study after analyzing the graphs, we observed that break thermal efficiency of B10 is better than that of other Karanja oil blends, but lower than that of disel.
- From the plots of brake thermal efficiency versus load, we observed that break thermal efficiency of any blend is maximum for maximum compression ratio.
- Thus we can conclude that adding 10% of Karanja oil with diesel improves engine performance but for emissions we must incorporate suitable method.

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HP31704003-Effect of condenser subcooling on performance of VCR system

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Abstract

Heat is the energy and energy saving is the important for the protection of global environment. so it is necessary to find possible ways to save the energy through the different medium in order to increase the performance of system to make it as compatible. Themain objectiveofthispaperistostudy "WasteHeatrecoverysystemfordomesticrefrigerator".An attempt has been made toutilize wasteheat from condenser ofrefrigerator. This heatcan be used fornumberofdomesticandindustrialpurposes.In minimumconstructional,maintenanceand running cost,thissystem ismuch usefulfor domesticpurpose.Itisvaluable alternative approach to improve overall efficiency and reuse the waste heat. The study has shown that such asystem is technically feasible and economically viable.

Keywords:*COP, Refrigeration effect, subcooling, Waste heatrecovery .*

1. Introduction

The increase in surrounding temperature due to climatic condition affect the performance of refrigeration and AC system and results in reducing the cooling capacity, increase in power consumption leads to detoriate the ability of comfort. Before we discussing the subccoling of refrigerant, it is necessary to understand what refrigeration is. Refrigeration is the removal of heat or transmission of heat from one part of system to another. Subcooling in refrigeration implies that cooling the refrigerant in liquid state at uniform pressure, to temperature that is less than the saturation temperature which corresponds to condenser pressure. Refrigeration is improved when a liquid refrigerant is subcooled by circulation of cold water in heat exchanger i.e condenser. Due to this characteristics, design of condenser needs to change for getting better liquid subcooling.

Therefore the present study has one of the alternative approach for the performance improvement of refrigerator and AC system, with minimal cost.

2. Literature Review

Useofwasteheatrecoveryfromthermalsystemisnotanew techniquealtogether.Thefocusis placed onaneedtodevelopeffective,lesscostlyandmaintenance-freeauxiliaryintegratedwith mainsystemtoachievewasteheatrecovery.If thisideaisimplementedatsystemdesignlevel, then therewould beconsiderable saving ofenergy.

Stinson et al.[1] conducted research in dairy refrigeration by recovering the heat fromcondenser. They found out that by using the water cooled condenser COP of the system is enhanced by 10% to 18%. They also found that increase in condenser pressure reduces COP, and inclusion of heat recovery heat exchanger reduces head loss.

Clark et al. [2] carried out experimentation on 18 ft3 domestic refrigerator. They used watercooled condenser and regular air cooled condenser in parallel. Following are the findings of this research: (i) rise in temperature of cooling water is 350C in 100 hours of continuing operation, (ii) 18% - 20% energy savings for hot water, and (iii) no deterioration of the refrigerant performance.

Milind V. Rane et al. [3] developed sensible heat recovery unit and carried out experiments.Waste heat recovered is utilized for water heating. Their findings are: (i) chiller cooling capacity enhanced by 30% and COP by 20%, (ii) fuel saving reported 81liters HSD/day, annual savings of Rs. 10 Lakh/year, (iii) Reduction in CO2 Emissions 450 ton in 4 years, and (iv) simple payback of 3 to 6 months.

Gustavo POTTKER et. al.[4] a theoretical and experimental analysis of the effect of condensersubcooling on the performance of vapor-compression systems. It is shown that, as condenser subcooling increases, the COP reaches a maximum as a result of a trade-off between increasing refrigerating effect and specific compression work. The thermodynamic properties associated with the relative increase in refrigerating effect, i.e. liquid specific heat and latent heat of vaporization, are dominant to determine the maximum COP improvement with condenser subcooling. Refrigerants with large latent heat of vaporization tend to benefit less from condenser subcooling. For a typical AC system, numerical results indicate that the R1234yf would benefit the most from condenser subcooling in comparison to R410A, R134a and R717 due to its smaller latent heat of vaporization. On the other hand, the value of COP maximizing subcooling does not seem

to be a strong function of thermodynamic properties. Experimental results comparing R1234yf and R134a confirmed the trends observed during the numerical study. For a given operating condition, the system COP increased up to 18% for R1234yf and only 9% for R134a.

AmitPrakash [5] To improve the coefficient of performance, it is to require that compressorwork should decrease and refrigerating effect should increase. Modifications in condenser are meant to increase degree of sub-cooling of refrigerant which increased refrigerating effect or more cooling water is required in condenser. The purpose of a compressor in vapor compression system is to elevate the pressure of the refrigerant, but refrigerant leaves the compressor with comparatively high velocity which may cause splashing of liquid refrigerant in the condenser tube, liquid hump and damage to condenser by erosion. It is needed to convert this kinetic energy to pressure energy by using diffuser. By using diffuser power consumption is less for same refrigerating effect so performance is improved.

3.Problem Description and Objectives

Vaporcompression refrigerationsystemisusedindomesticrefrigeration,food processingand coldstorage, industrialrefrigerationsystem, transportrefrigeration andelectroniccooling.So improvement of performance of system is too important for higher refrigerating effect or reduced power consumptionforsamerefrigeratingeffect.Manyeffortshavetobedonetoimprovethe performanceofVaporCompressionrefrigerationsystem. Toimprovethecoefficient of performance,itistorequirethatcompressorworkshoulddecreaseandrefrigeratingeffect shouldincrease. It is only possible by subcooling refrigerant by either water or air. Modificationsincondenserarenecessary. Thedilemmathatindustry isfacing regardingCFCphase-outandtheproblemsassociated withCFCalternativespresentlyunder development. Thechallenge**of**replacingR-11andR-12presents**a**greatdilemmato manufacturersofhomerefrigerator/freezerssinceR-12isavery popularworkingfluidforthis application.Toxicity isanother issuewithusing nonCFCrefrigerant.

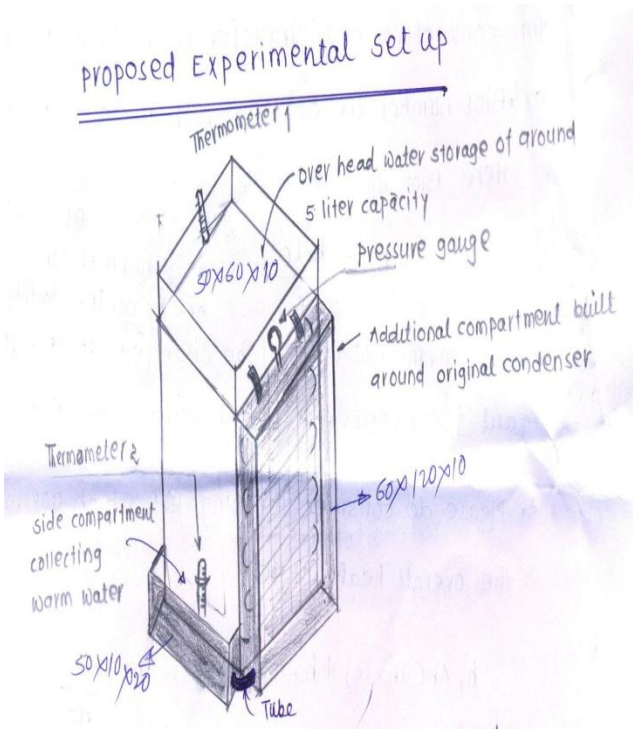
4.Methodology and Experimental Setup

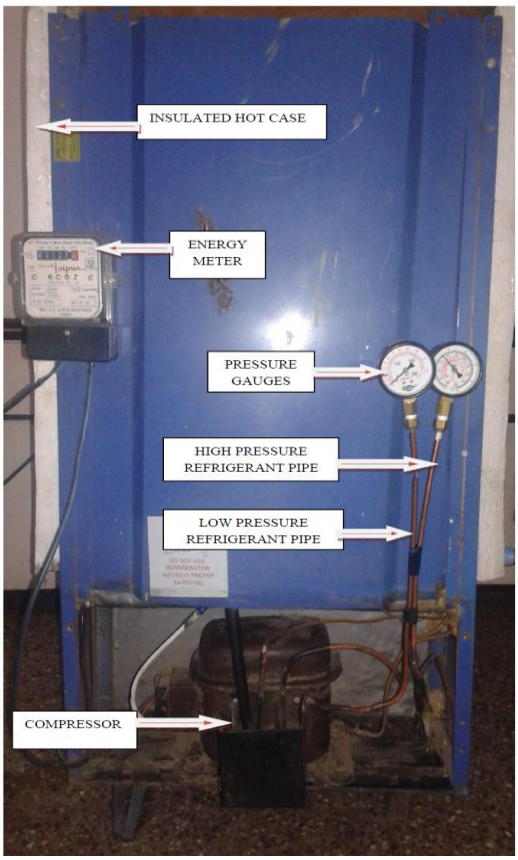
4.1.Methodology

- Optimize the use ,distribution and operation of refrigeration system.
- Examine waste heat streams in refrigeration system.
- Quantify potential value of this heat.
- Examine uses for the Waste heat.
- Quantify useable heat and its value.
- Design heat recovery system.
- Develop and Implement project.
- Result and Analysis

4.2.Experimental Setup

Photograph shows the assembly of hot case, energy meter, temperature sensor and load kept in the evaporator of the refrigerator





5.Experimental Method (Apparatus and Procedure)

The present research used 165 liters household refrigerator working on vapor compression cycle includes four major equipment as Compressor, condense, capillary tube, and evaporator, with a cabinet made over the original condenser
In which cold water is store and dummy load of water is kept in evaporator and evaporator main cabin. With the variation of load in evaporator the system performance is analyzed.

5.1Idea Behind The Project

Due to the presence of subcooled liquid, the two phase heat transfer area would decrease relative to the condition without subcooling. As result saturation temperature would be rise in condenser which would increase the specific compression work, on the other hand the temperature at outlet of condenser decreases with the increase in enthalpy as results of this COP of the system increases.

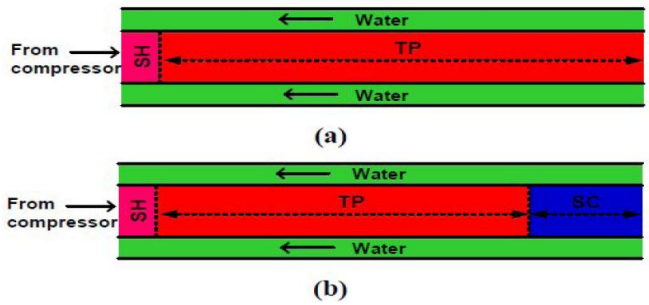


Fig.1 Schematic of water cooled condenser with and without subcooling

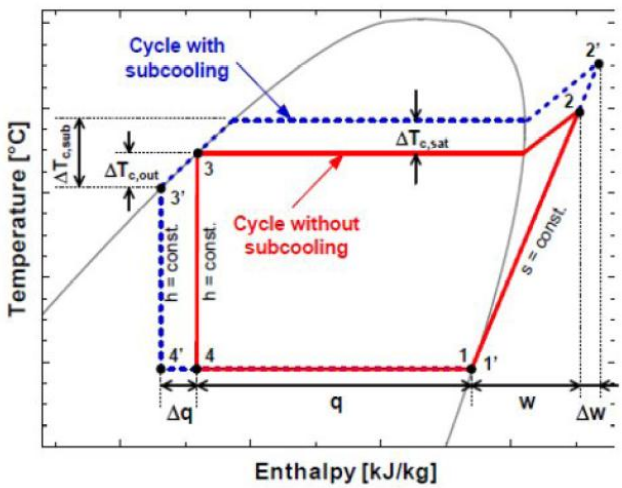


Fig.2 T-s diagram with and without subcooling

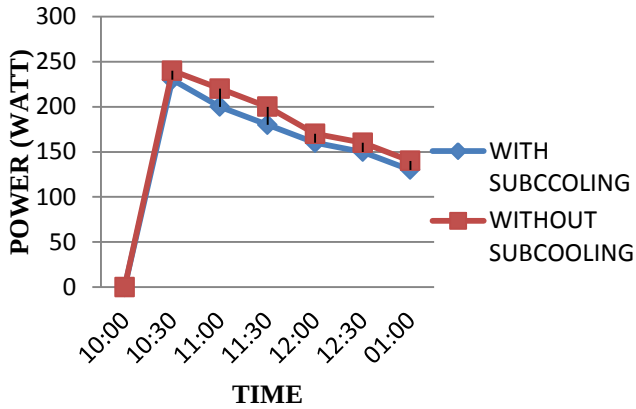
6.Experimental Results

6.1. COP of the system calculated on the theoretical data provided by the refrigerator manufacturing company (GODREJ).

For Refrigerator of 165 liters capacity, given data from Kirloskar Ltd manual follows-
Refrigerator cooling capacity:(PROVIDED BY GODREJ)

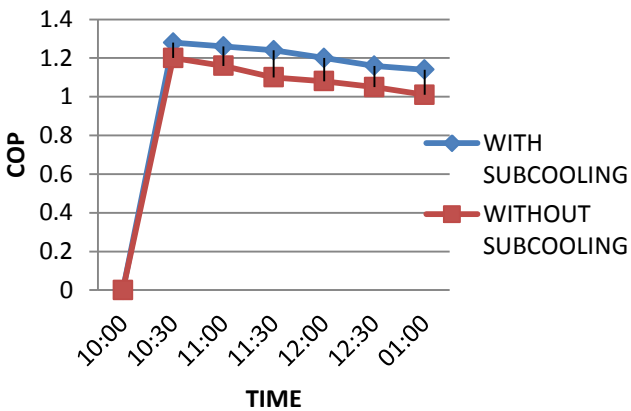
=76 kcal/hr
= 76×4.187×1000/3600
= 88.392 W
Power required running the compressor
= 1/8 HP
= 1/8×746
= 93.25 W

The coefficient of performance (COP)
$=\frac{\text{REFRIGERATION EFFECT}}{\text{WORKSUPPLIED}}$
$=\frac{88.5}{93.2}$
=0.948



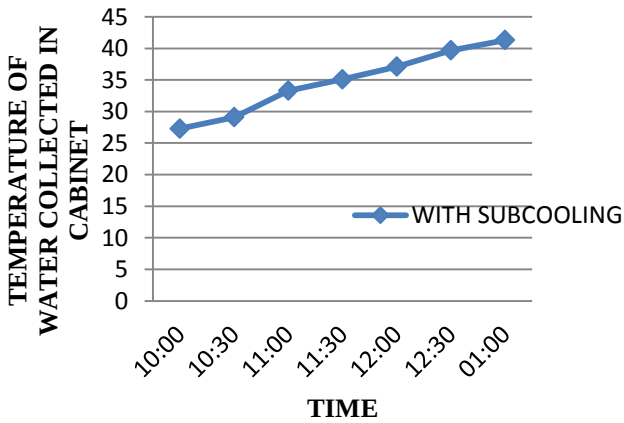
6.2 Experimental COP of the system

OPERATING CONDITION:
Water kept in evaporator = 5 kg
Water kept in evaporator main cabin = 15 kg



As shown in above fig, the refrigerating effect keeps decreasing as the temperature difference between the refrigerant and article placed is decreased. The COP of system remains almost constant though, it decrease little bit. Average COP is in case with subcooling is 1.04.Somewhat higher thanWithout hot case.

As shown in above fig, Power consumption of refrigerator is somewhat more than that of refrigerator with hot case similar to the case by changing load in other cases without hot case.



As shown in above fig .Same trend of graph showing increasing the temperature of water in outside compartment. It is shown that as the load on refrigerator increases, heat rejected by condenser also increases and as result of this temperature of water in outside compartment also increase. The maximum temperature achieved in this case is 41.3

6.3 Calculations for increased rate of waste heat

Heat Recovery Achieved, Q = Heat Absorbed By Water
Given data-
Mass of water in the outside compartment, M = 10 kg

Specific heat of water, Cp = 4.187 KJ/Kg K

Initial temperature of water = 27.3 °C

Final temperature of water = 41.3 °C

Time required for reading Δt = 180 min
Q = 10 x 4.187 x (41.3−27.3)/180 X60

= 3.74 J/s

Heat recovery achieved Q = Heat Absorbed by Water

= 3.74 W

6.4 Result and discussion

From experimental results it can be concluded that with time the energy consumption of the refrigerator decreases for certain time and then it remain constant. The refrigerating effect keeps decreasing as the temperature difference between the refrigerant and article placed is decreased. The C.O.P. remains almost constant though it decreases a little bit. With hot case, as if we add up heating effect in desired effect, then the COP is increased also otherwise it is almost little bit more than the unit with the hot case. Thus the hot case has not bad effect on the refrigerator. Here with use of hot case, we can keep some food stuff, is hot condition, also temperature of food/milk, etc can be increased without change in taste, so amount of electrical energy used for hot case, as in case of conventional system, can be saved.
The same procedure is followed to show different behavior of the system by changing the different loading condition as mentioned below and results obtTained
CASE 1:
Water kept in evaporator = 3 kg
Water kept in evaporator main cabin = 15 kg
CASE 2:
Water kept in evaporator = 3 kg
Water kept in evaporator main cabin = 20 kg
CASE 3:
Water kept in evaporator = 5 kg
Water kept in evaporator main cabin = 15 kg

The following result shows the comparison between COP of modified setup and waste heat recovery at various operating conditions.
CASE 1: Average COP with and without hot case is obtained as 1.211 and 0.9625 respectively. Waste heat recovery achieved is 3.48 watt
CASE 2: Average COP with and without hot case is obtained as 1.132 and 0.9542 respectively. Waste heat recovery achieved is 4.16 watt
CASE 3: Average COP with and without hot case is obtained as 1.041 and 0.9548 respectively waste heat recovery achieved is 3.74 watt

The result show that modified setup gives better instantaneous efficiency as compared to normal setup. It is shown from results obtain in three different cases as load on refrigerator increases, the power consumption of refrigerator increases and COP decrease a little bit but the temperature of water in outside compartment increases.

Conclusion

A theoretical and experimental study about the effect of condenser subcooling on the performance of vapor compression system has been presented. This study showed that, as condenser subcooling increases, the COP undergoes a maximum as a result of a trade-off between increasing refrigerating effect, due to the reduction of the condenser exit temperature, and increasing specific compression work, due to the increase in the condensing pressure. "Waste heat recovery system" is an excellent tool to conserve available energy. An attempt is made to recover the waste heat from 165 L refrigerator used for domestic purpose. As indicated in this paper, recovered heat can be utilized as food and snacks warmer, water heater, grain dryer. So one can save lot of time and energy also

Future Scope

Now a day saving & regeneration of energy is become a very needful issue due to energy crisis. Cost of fuel is increasing day by day so anybody is providing any equipment for energy saving or regeneration then he will get a lot of future scope, help & research facilities to work for it by any government of developing countries. Worldwide attempts are being made to phase out the production and consumption of chlorofluorocarbons, as these chemicals are responsible for
Depletion of stratospheric ozone layer. Refrigeration, A/C and heat pumps sectors are one of the principal users of these chemicals. Due to the environmental concerns ozone depletion potential (ODP) and global warming potential (GWP) of the existing refrigerants, industry and researchers in this field are investigating long-term solutions.

According to that, following are few recommendations for the future work.
1. A design optimization of refrigeration system such as condenser coil modification and arrangement of hot case around the refrigerator can be made.
2. Application of this type of techniques in air conditioning system can be studied.
3. Change of different refrigerant and optimizing the performance of system can be studied.

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HP31704004-Effect of Ethanol as an Additive on Properties as well as Performance, Combustion and Emissions of Palm Oil Biodiesel in Diesel Engine

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Abstract

As we need an alternating fuel to replace diesel fuel in order to decrease the harmful emissions of diesel fuel coming out of engine as exhaust by products, which makes it necessary to improve the engine performance and combustion characteristics of fuels in the combustion chamber. As biodiesels are having high viscosity and flash point as compared to diesel, it will be impractical to use biodiesel as a fuel alone in present diesel engines. Hence it will be beneficial to blend biodiesel with diesel to get required properties of blend that will work with the present diesel engines. Higher kinematic viscosity of biodiesel/diesel blends as compared to diesel affects the atomization of fuel in the combustion chamber reducing the combustion pressure and temperature and thus the power output of the engine. This makes it necessary to add ethanol as an additive in the blends which further improves the hot flow, cold flow and thermo-physical properties of the biodiesel/diesel blends. Therefore in this study, feedstock of palm oil biodiesel is used as fuel with 5% ethanol by volume as an additive in the blends of palm oil biodiesel/diesel blends. Hot flow and cold flow properties of blends of palm oil biodiesel/ethanol/diesel blends are experimentally investigated as per IS 1448 standards. Investigation outcome shows, an ethanol as an additive improves the kinematic viscosity of blends of palm oil biodiesel/diesel by 4% lesser than biodiesel/diesel blend without ethanol and 12% higher than diesel. But on the other hand calorific value of the blends with ethanol as additive decreases calorific value by 6.27% than biodiesel/diesel blend without ethanol and 6.87 % than diesel. The cold flow properties are enhanced by the addition of ethanol in the blend such as cloud point increases by 13 % than biodiesel/diesel blend without ethanol and pour point increases by 14% than biodiesel/diesel blend without ethanol. The combustion and performance analysis are improved with the addition of ethanol and decreases the harmful emissions from the exhaust manifold of the engine. The effect of ethanol as an additive in the blends of palm oil biodiesel/diesel blend on emissions, performance, and combustion and on properties are studied in this paper.

Keywords:*Ethanol, Combustion, Palm oil biodiesel, Performance, Diesel engine.*

1. Introduction

Generally biodiesel (triglycerides) have properties like high viscosity, high pour point high density and low calorific value than a diesel fuel. Thus it becomes difficult to use biodiesel oil as a fuel in conventional diesel engines. Biodiesel is derived from vegetable oils or animal fats (triglycerides). Vegetable oils possess high viscosity, pour point and density. Hence vegetable oils only are not used as a fuel in conventional diesel engines. The transterification process brings properties of vegetable oils to certain required level [JainS. and SharmaM.P.2010]. The transestrified vegetable oil is treated as biodiesel.

Transterification process is also called as alcoholysis, where exchange of alcohol from an ester by another alcohol in a process similar to hydrolysis, except that an alcohol is used instead of water [JainS. and SharmaM.P.2010.]. In order to carry reaction in shorter time the time, pressure ratio reaction temperature of alcohol to oil, concentration and type of catalyst, mixing intensity, and kind of feedstock, are relevant variables which affect the transterification process. Addition of methanol to fat or oil gives the biodiesel .Biodiesel is a mixture of mono-alkyl ester of saturated and unsaturated long chain fatty acids. Diesel is nothing but the mixture of paraffinic, naphthenic and aromatic hydrocarbons [WanNorMaawaWanGhazali 2015]. As biodiesel fuel has technical, environmental and strategic advantages it can be accepted as an alternative fuel for diesel in diesel engines [Sahoo P.K. and Das L.M. 2009]. Biodiesel has biodegradability, less toxicity and good lubricity. Biodiesel is also miscible

with petroleum diesel thus blending of biodiesel becomes possible in any proportion. Biodiesel/diesel fuel blend does not require any significant changes in present diesel engines. There is difference in chemical nature of biodiesel and diesel, which altered basic properties of blend and thus affects performance and emissions of diesel engine. Biodiesel generally has high viscosity, high density, high pour point, high cloud point and high cetane number and lower heating value and volatility. In order to find out the specifications of blending mixture whether it fits with diesel engine, properties of a blend composition must be investigated. Higher kinematic viscosity of biodiesel/diesel blends as compared to diesel affects the atomization of fuel in the combustion chamber which further reduces the combustion pressure and temperature and reduces the power output of the engine [Schumacher L.G., Van Gerpen J.H.2004]. This makes it necessary to add ethanol as an additive in the blends which further enhances the hot flow, cold flow and thermo-physical properties of the biodiesel/diesel blends.

Ethanol [CH_3CH_2OH] is a colourless liquid and is also known as ethyl alcohol. Ethanol has higher octane number. Ethanol is oxygenated low cost fuel containing 34% higher oxygen by weight [6]. Ethanol is made by fermentation process biologically from variety of biomass resources like Sugarcane, Corn, Sugar beet etc. It is produced by catalytic hydration of ethylene using sulphuric acid as catalyst [Azad A. K.,AmeerUddinS. M.2012]. Table 1 shows the properties of 99.99% pure ethanol investigated as per IS 1448 standards in the NABL accredited laboratory.

Table1. Properties of 99.99% pure ethanol [16, 17, 18]

Properties	Ethanol
Density (kg/m^3)	790
Viscosity (Cst)	1.4
Cetane number	8
Calorific value (MJ/Kg)	26.95
Flash point °C	13
Cloud point°C	<-26

2. Palm oil biodiesel

Since last 100 years palm oil became world's major aThe palm oil biodiesel /diesel blends were prepared with blending concentration of 20%, 40%, 60% and 80% palm oil biodiesel with diesel fuel but without ethanol and 25%, 45%, 65% and 85% palm oil biodiesel with 5% ethanol by volume in diesel fuel. All the blends with and without ethanol were prepared on the volumetric analysis. The thermo physical properties or hot and cold flow properties such as pour point, cloud point, fire point calorific value, density, viscosity, and flash point were experimentally investigated as per IS: 1448 standards for all blends of palm oil biodiesel/diesel with and without ethanol.

gricultural commodity from minor crop in west and central Africa. Whereas Palm oil has been cultivated in Africa for centuries, but it has dramatically increased in Southeast Asia, Africa and Latin America and Malaysia. Palm oil production is driven by producers responding to consumer demand much of which is from India and China. Palm oil is an industrial crop and it is also a smallholder crop too increasing rural development in humid tropics [Jeffrey S. and Jaboury G. 2012]. Three important things about palm oils are as follows

1. Demand for palm oil will keep increasing in response to growing population as it is edible one.
2. Palm oil plantation possesses more carbon than other alternatives.
3. Oxidation stability of palm oil is because of presence of higher concentration of saturated fatty acids i.e. for more time palm oil is stable when stored as compared to other feedstock of vegetable oil.

The feedstock of palm oil biodiesel was collected from SVM Agro Industries, Nagpur. Table 2 shows the properties of palm oil biodiesel (B100-100% biodiesel), which are investigated as per IS 1448 standards in the NABL accredited laboratory.

Table 2: Properties of Palm oil biodiesel (B100)

Test parameter		Units	Value	IS 1448 Standards
1	Gross calorific value	kJ/kg	41173	Bomb Calorimeter
2	Kinematic viscosity at 40°C	cSt	5.6	IS 1448 Part I (P-25)
3	Cloud point	°C	18	IS 1448 Part I (P-10)
4	Pour Point	°C	15	IS 1448 Part I (P-10)
5	Density @ 15 °C	kg/m^3	896.3	IS 1448 Part I (P-16)
6	Flash point	°C	70	IS 1448 (P20)
7	Fire point	°C	130	IS 1448 (P-69)

3. Fuel blend preparation

The palm oil biodiesel /diesel blends were prepared with blending concentration of 20%, 40%, 60% and 80% palm oil biodiesel with diesel fuel but without ethanol and 25%, 45%, 65% and 85% palm oil biodiesel with 5% ethanol by volume in diesel fuel. All the blends with and without ethanol were prepared on the volumetric analysis. The thermo physical properties or hot and cold flow properties such as pour point, cloud point, fire point calorific value, density, viscosity, and flash point were experimentally investigated as per IS: 1448 standards for all blends of palm oil biodiesel/diesel with and without ethanol.

4. Properties investigation and discussions

Fuel properties plays very important role in complete combustion of the fuel in compression ignition engines. Addition of ethanol as an additive changed the fuel blends properties [SenthilkumarS., SivakumarG. and Siddarth M.2015]. Table 3 shows properties of blends of palm oil biodiesel with and

without ethanol and these properties are compared with conventional diesel fuel.

Table: 3 Properties of Palm oil biodiesel/diesel blends with and without ethanol.

Fuel	CV kJ/kg	Kinematic viscosity cSt	Cloud point °C	Pour point °C	Density Kg/m^3	Flash point °C	Fire point °C
Diesel	43851	2.5	-23	-21	817.4	40	42
P20	43593	2.9	-5	-7	855.2	28	44
P20 E5	40840	2.8	-8	-10	846.4	18	24
P40	43483	3.4	1	-5	857.3	30	48
P40 E5	40229	3.2	-3	-6	850.2	20	32
P60	42590	3.9	6	1	860.2	32	52
P60 E5	39229	3.6	5	1	854	22	40
P80	41706	4.8	12	10	887.3	42	98
P80 E5	37025	4.4	8	3	876.2	26	40

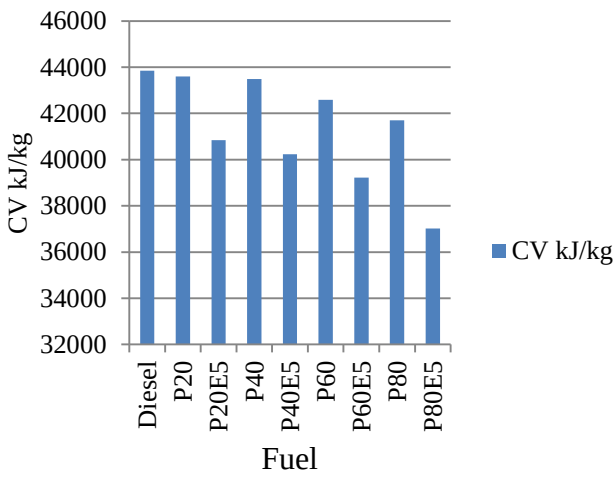

Fig.1 Variation of Calorific value of Palm oil biodiesel with and without ethanol for different Blend ratio

Fig 1 shows Calorific value variation of various blends of palm oil biodiesel with and without ethanol for different blend ratio. Calorific value is the important property of fuel which indicates the heat release capacity of the fuel. The properties as lower heating value or calorific value of biodiesel and ethanol, decreases the overall calorific value of the all the blends. For the blend of P20E5 the calorific value is 7% less than diesel, for P80E5 it is 15% less than diesel. So increase in percentage of biodiesel in the blend, the calorific value of the overall blend decreases.

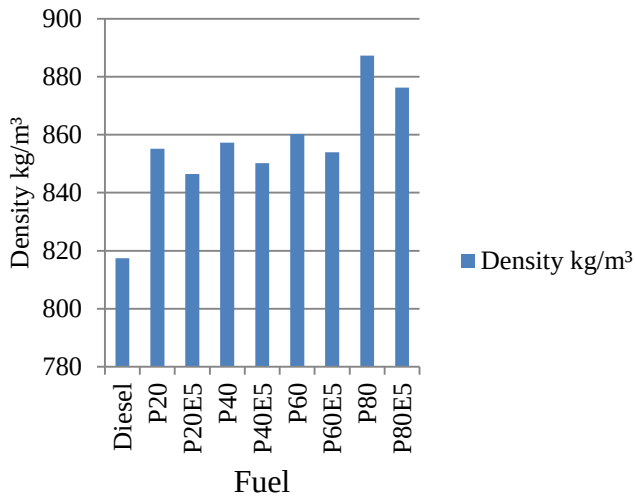

Fig.2 Variation of density of Palm oil biodiesel with and without ethanol for different blend ratio

The variation in density of blends of palm oil biodiesel with and without ethanol for different blend ratio is shown in fig 2. To reduce the overall density of palm oil biodiesel/diesel blend for all blend ratios low density of ethanol is responsible. Density of P20E5 is 3% greater than diesel but it is 1% less than P20. So the density of P20 blend gets reduced by addition of ethanol. For all other blends the similar trend was noticed.

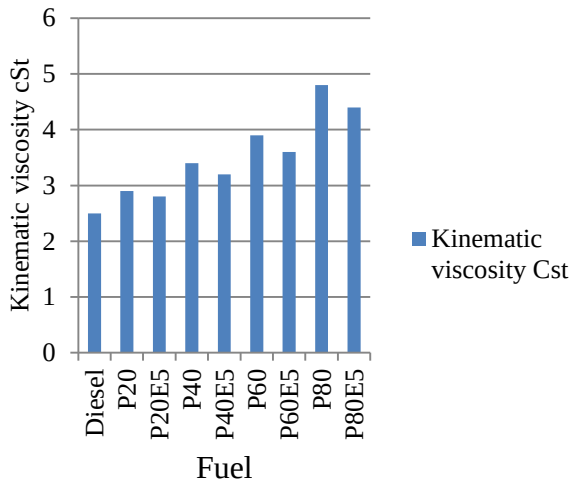

Fig.3 Variation of kinematic viscosity of Palm oil biodiesel with and without ethanol for different blend ratio

Variation of kinematic viscosity of Palm oil biodiesel with and without ethanol for different blend ratio is shown in fig 3. For the complete combustion Low viscosity of fuel is always preferable. Viscosity of the blend increases with addition of the palm oil biodiesel with diesel. Ethanol is added in the blend in order to address this problem and addition of ethanol reduces the overall viscosity of the blends. This is done as ethanol has low viscosity. Kinematic viscosity of P20E5 becomes less by 3.44% than P20. For blend

ratios of P40E5 up to P80E5 the reduction in the viscosity takes place from 6 to 9 % as compared with P40 to P80 blend ratio of palm oil biodiesel.

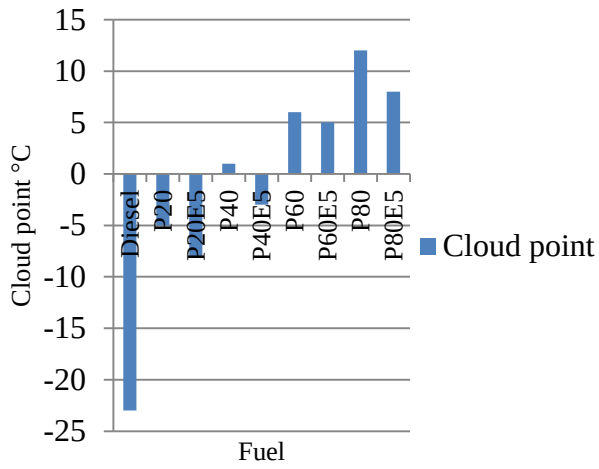

Fig.4 Variation of cloud point of Palm oil biodiesel with and without ethanol for different blend ratio

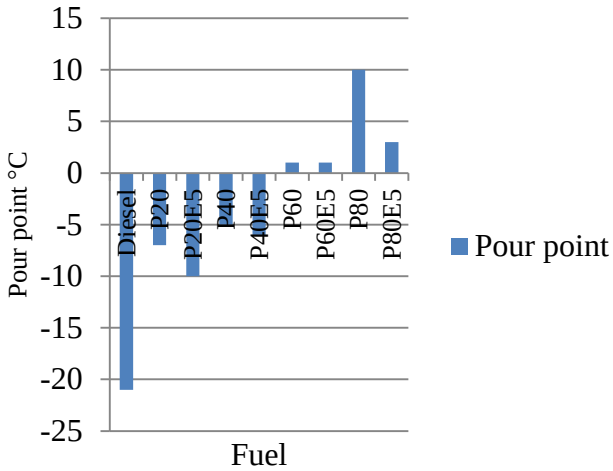

Fig.5 Variation of pour point of Palm oil biodiesel with and without ethanol for different blend ratio

Variation of cloud point and pour point of palm oil biodiesel with and without ethanol for different blend ratios is shown in Fig 4 and fig 5. Application and usability of biodiesel in very cold climatic conditions becomes limited, as the cloud point and pour point of palm oil biodiesel is higher than the diesel fuel [Verma P., Sharma M.P. and Dwivedi G. 2016]. But addition of ethanol in the blends of biodiesel reduces the cloud point and pour point of the biodiesel/diesel blends as the cloud point and pour point of ethanol is lower than palm oil biodiesel/diesel blends. Pour point of P20E5 is 14% lower than P20 and Cloud point of P20E5 is 13% lower than the P20.

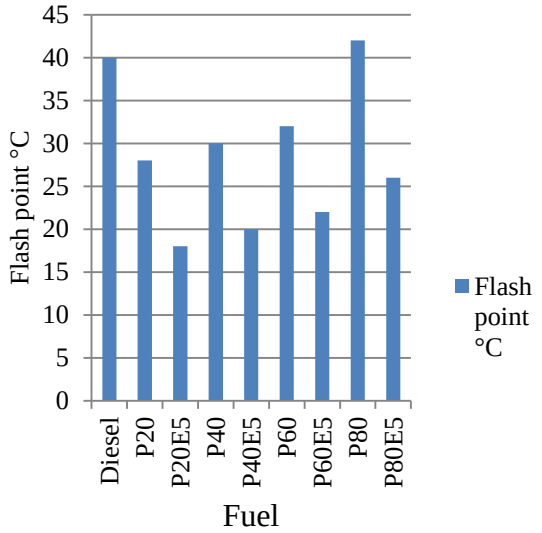

Fig .6 Variation of flash point of Palm oil biodiesel with and without ethanol for different blend ratio

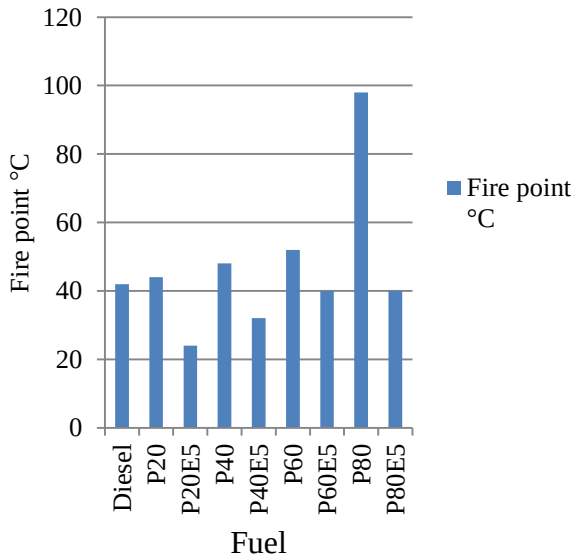

Fig .7 Variation of fire point of Palm oil biodiesel with and without ethanol for different blend ratio

Increase in the flash and fire point of the blends is caused by addition of palm oil biodiesel in the diesel, which is not at all desirable for the combustion. The ignition delay increases with higher flash point and fire point and affect the combustion temperature and pressure, which will further reduce the power output of the engine. Addition of ethanol in the blend reduces the flash point and fire point of the blend as ethanol is having very low kinematic viscosity than the biodiesel. Flash point is reduces by 25% than P20 and fire point reduces by 47% than P20 respectively for P20E5. The variation of flash point and fire point of palm oil biodiesel with and without ethanol for different blend ratios is shown in Fig 6 and 7.

5. Experimental fuel set up and fuel

The 75% diesel is mixed with 20% palm oil biodiesel and the 5% ethanol was added as an additive i.e P20E5. The test was carried for engine performance, combustion and for emission in single cylinder variable compression diesel engine. Same test was carried on same engine for 20 % palm oil biodiesel and 80% diesel blend without ethanol i.e P20. The results of engine performance combustion and emission of P20E5 and P20 are compared with diesel.

Single cylinder, four stroke, and variable compression ratio diesel engine was used for test. Engine was equipped with eddy current dynamometer for load adjustment. The compression ratio of the engine was maintained at 18. Pressure transducer and crank angle measurement sensor were installed on engine test rig. The signals were interface to computer for various P-θ and P-V diagrams with the help of data logger. Air box, U-tube manometer, and fuel measuring system, air flow and fuel flow measuring instruments were also used in engine. Tests were carried out on the engine for the load of 3 kg, 6kg, 9kg and 12 kg running at 1500 rpm for varying load conditions. Engine soft; lab view based software was interface with engine to computer in order to test and record real time data.

6. Results and discussions

6.1 Performance Analysis

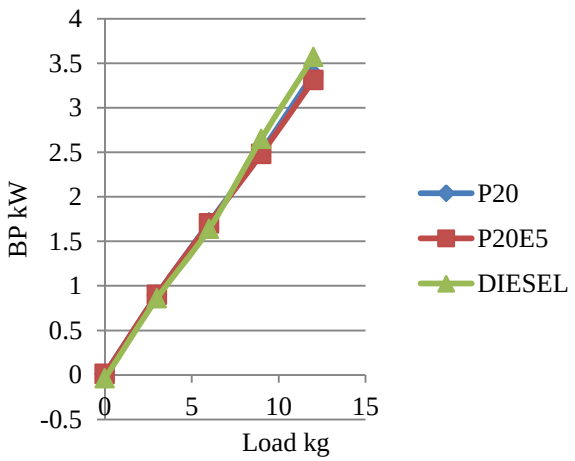

Fig .8 Variation of Brake Power with Load

Variation of brake power with the different load conditions are shown in Fig 8. It is observed that the brake power of P20E5 gets reduced in the range of 7 to 8% as compared with diesel, because of lower calorific value of P20E5 as compared with P20 and diesel fuel. The variation of brake mean effective pressure with load on engine is shown in Fig 9. Because of the lower kinematic viscosity of the blend, the BMEP of P20E5 is 3% more than the conventional diesel which ensures the maximum combustion in cylinder and builds the maximum BMEP.

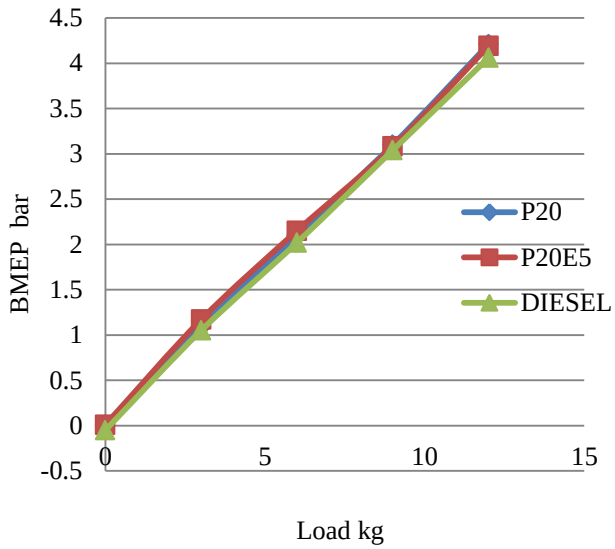

Fig.9 Variation of Brake Mean Effective Pressure with Load

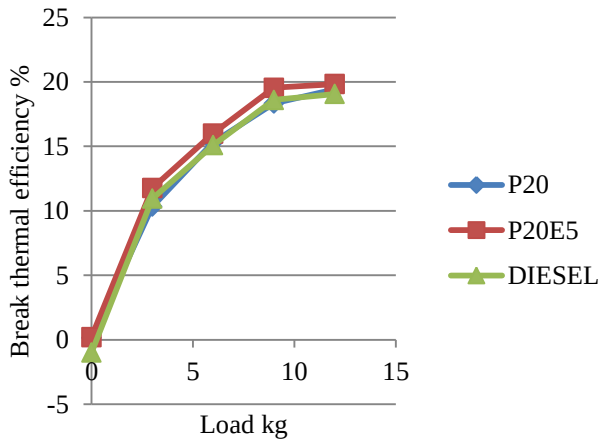

Fig.10 Variation of Brake thermal efficiency with Load

The variation of brake thermal efficiency with Load is shown in Fig 10. The brake thermal efficiency increases from 4% to 7 % than conventional diesel fuel and 3% to 6% more than P20 for P20E5. The variation of Specific fuel consumption with load is shown in Fig 11. The fuel consumption is increased from 1 to 7% as compared to diesel for P20E5, due to the lower heating value of the biodiesel and ethanol.

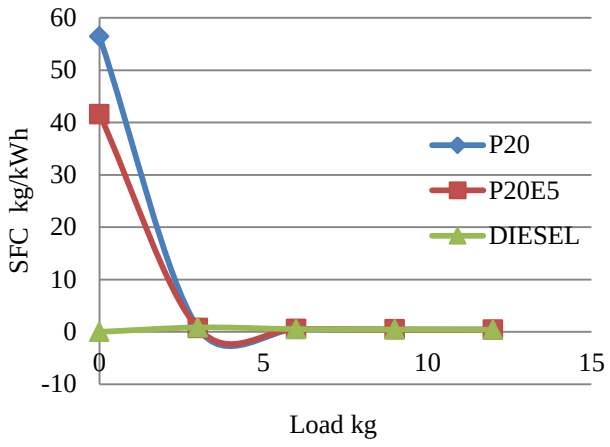

Fig.11 Variation of Specific fuel consumption with Load

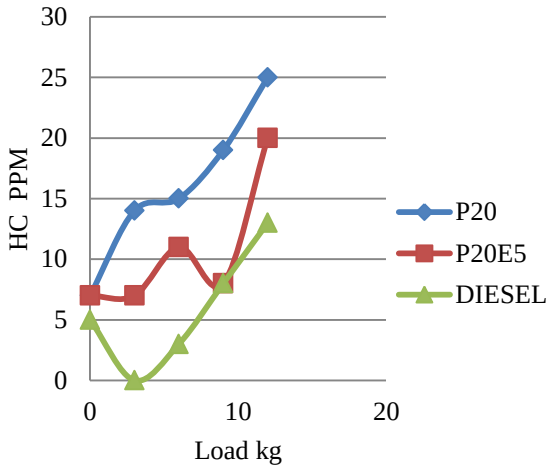

Fig.13 Variation of Hydrocarbon in PPM with load

6.2 Emission analysis

The variation of CO, HC, CO_2 and NOx emissions with varying load respectively is shown in Fig 12, Fig 13, Fig 14 and Fig 15. Investigation shows that CO emissions of P20E5 are more than diesel but less than P20 by 30% to 40%. Hydro Carban emissions of P20E5 blend are more than diesel by 30% but less than P20 by near about 40% to 50%. CO_2 emissions of P20E5 are more only at 12 kg load and are less than diesel by 5% to 10% but on the other hand CO_2 emissionsare less than P20 blend by 5% to 15%. Nitrogen oxide emissions of P20E5 reduces as compared with P20, this is because of more availability of nitrogen for the combustion as we all know that 70% nitrogen is available in air. As biodiesel is an oxygenated fuel and liberates more amount of energy and increases combustion temperature which dissociates nitrogen in to nitrogen oxide.

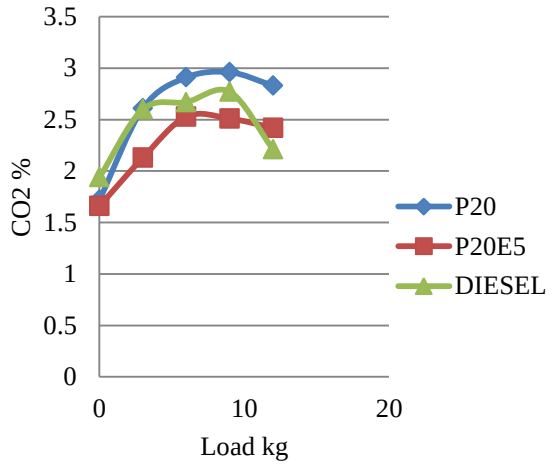

Fig.14 Variation of Carbon Dioxide in (%) with load

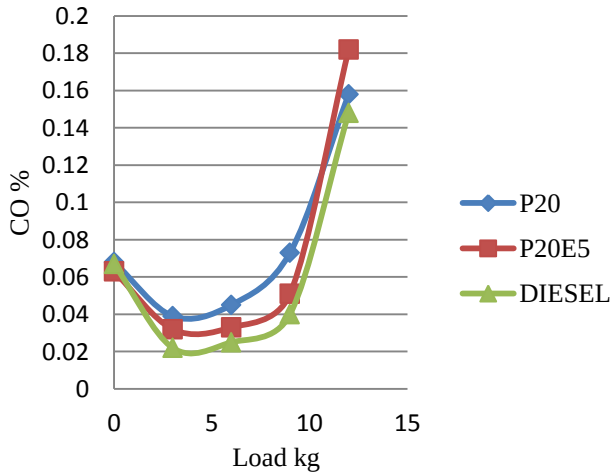

Fig.12 Variation of Carbon Monoxide in (%) with load

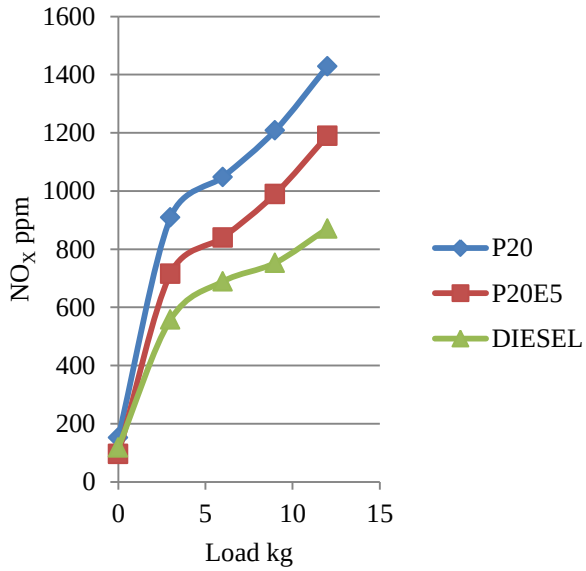

Fig.15 Variation of Nitrogen Oxides in PPM with load

6.3 Combustion Analysis

The variation of pressure, build near TDC with crank angle as shown in Fig 16. It is clear from the figure that the maximum pressure is built after the TDC in case of diesel fuel as compared with P20E5 and P20 because of the higher heating value of the diesel who possesses higher heating value than diesel P20E5 and P20.

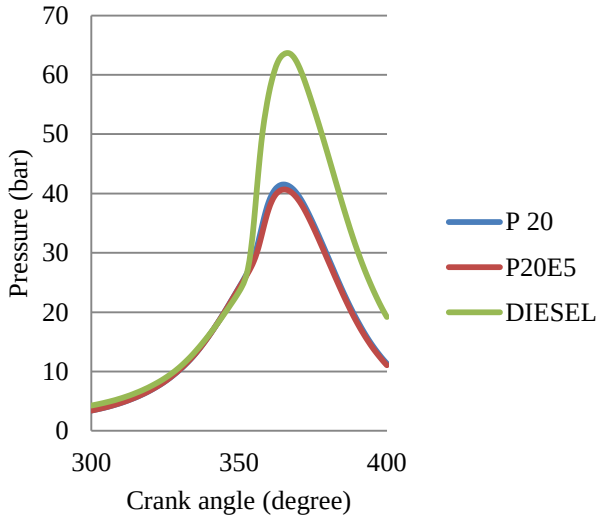

Fig.16 Variation of Pressure with Crank Angle

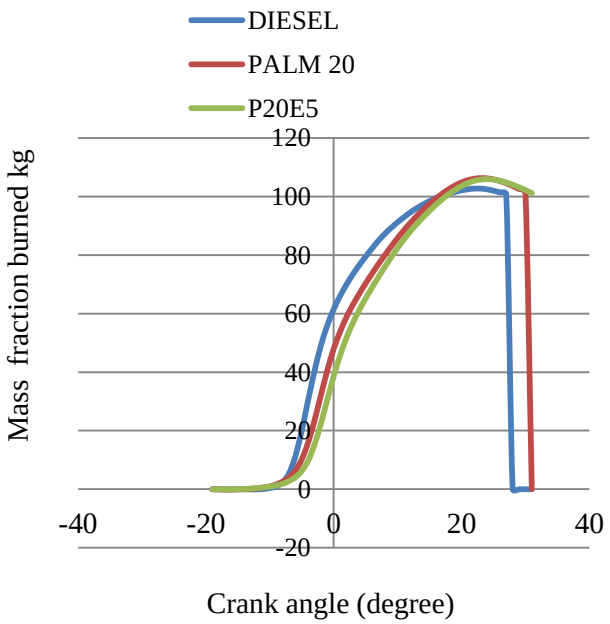

Fig.17 Variation of Mass fraction burned with Crank Angle

Fig 17 shows the quantity of fuel consumed for the combustion. Mass fraction of P20E5 is greater than the P20 because of the lower kinematic viscosity, which ensures good atomization and maximum combustion. But mass fraction of P20E5 and P20 is less than diesel fuel.

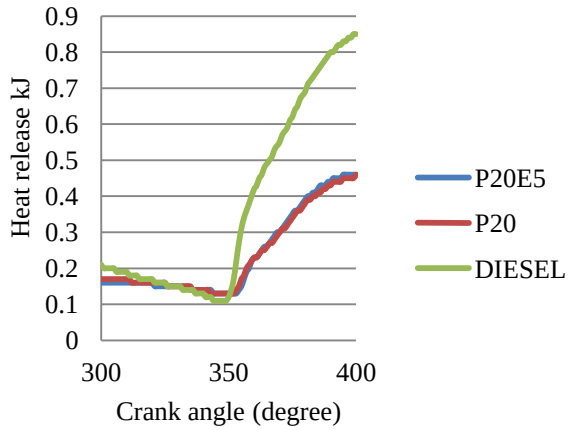

Fig.18 Variation of heat release with Crank Angle

The variation of heat release with crank angle is shown in Fig 18. Near the TDC at the time of combustion the heat release in case of diesel fuel is more as compared with P20E5 and P20.

7. Conclusion

The blend of Palm oil biodiesel/diesel should be prepared with the addition of 5% ethanol as an additive only because itself ethanol has low calorific value and its more percentage will reduce calorific value of overall mixture. Hot flow, cold flow and thermo-physical properties of P20E5 are showing improvement by addition of 5% ethanol by volume as seen in graphs of cloud point and pour point. Kinematic viscosity of P20E5 reduces by 4% than P20 which becomes beneficial to the process of atomization and combustion.Cloud point and pour point of P20E5 reduces by 13 % and by 14 % than P20 respectively this gives more application and usability of biodiesel/diesel blend in cold atmospheric conditions. Combustion quality of the fuel is defined well by flash point and fire point, the flash point of P20E5 enhances by 25 % than P20 and fire point by 47 % than P20 by addition of ethanol.

Addition of ethanol as an additive in the fuel i.e P20E5 improves the performance of the engine than the fuel without ethanol i.e. P20. P20E5 shows improvement in brake thermal efficiency than P20 by 6%. CO emissions of P20E5 are higher than diesel but less than P20 by 30% to 40% from Emission analysis. HC emissions of P20E5 blend are more than diesel by 30% but less than P20 by near about 40% to 50%. Carbon dioxide emissions of P20E5 are only more at load of 12 kg and it is less than diesel by 5% to 10% but it is less than P20 blend by 5% to 15%. NO_X emissions of P20E5 reduces as compared with P20. Improvement in the properties of the blends and performance is observed and also reduction in the harmful exhaust emission from diesel engine because

of addition of ethanol as an additive in palm oil biodiesel.

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HP31704012-Effect of injection timing, nozzle orifice geometry on performance of direct injection Diesel engine

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Abstract

Fuel injection parameters and nozzle orifice geometry plays an important role for obtaining better combustion in direct injection diesel engine. The performance and emission characteristics are mainly depends on various parameters such as injection pressure, injection timing, nozzle hole size and spray characteristics. An experimental investigation is carried on single cylinder, four stroke diesel engine for nozzle configuration of 3, 5 holes having diameters 0.2mm,0.3mm at 19°, 23° and 27° bTDC respectively. Important parameters governing are fuel droplet size, distribution & concentration, injection velocity etc. From the experimental investigation it is found that 0.2-5 holes nozzle configuration at 23° bTDC injection timing is better than 0.3-3 holes and 0.3-5 holes respectively.

Keywords:*Diesel engine, emission, injection timing, Performance characteristics, Nozzle geometry*

1. Introduction

Diesel engines are becoming more important for transport and power generation applications as they have high thermal efficiency and lower HC and CO emissions compared to gasoline engines. Also this can handle wide variety of fuel types; hence they can be predominantly used for transportation and power generation applications. Diesel engines are widely employed for the above mentioned applications due to their simple and sturdy mechanical structure, low fuel cost, high reliability and durability, and high power to weight ratio. The advantages of diesel engines are high fuel efficiency, reliability and durability. In present diesel engines, fuel injection systems have designed to obtain higher injection pressure. So, it is aimed to decrease the exhaust emissions by increasing efficiency of diesel engines.

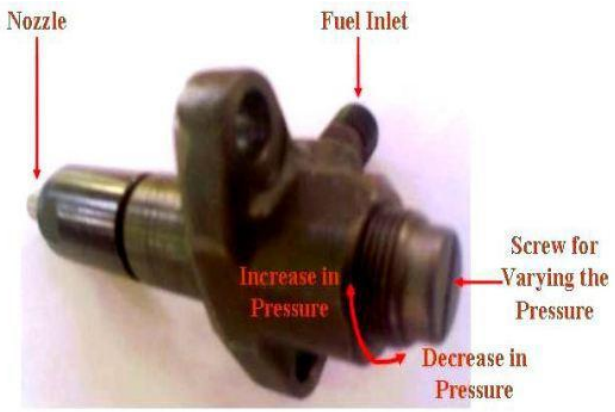

Fig.1 Fuel injector

A. Injection timing:

In diesel engine 23° bTDC injection timing is commonly used as it gives better performance characteristics and low emission level. If injection timing is retarded, there may be possibilities that fuel is not properly atomized and hence HC, CO emission increases due to un-burnt part of fuel. If injection timing is advanced from 23°bTDC to 27° bTDC, fuel gets more time for mixing but ignition delay period increases and hence emission level increases.

B. Nozzle geometry

The geometry of the nozzle in an injector plays a vital role in controlling diesel spray atomization and combustion. In order to bring fuel droplet size small, the nozzle-hole size is required to be reduced to produce smaller droplets. By decreasing the nozzle hole size, the spray tip penetration is reduced due to the low spray momentum. High injection pressures with small nozzles are common in the modern diesel engine as they reduce injection duration and improve combustion efficiency.

Fig.2 Injector nozzle model S639

2. Literature Review

1 Swamy L., Nashipudi in their study on "Effect of injection timing, combustion chamber shapes and nozzle geometry on Diesel engine performance" find the variation of performance and emission characteristics such as BTE, HC, CO, NOx emission for different injection timing such as 19, 23, 25, 27 deg. bTDC, for various combustion chamber shapes and nozzle orifice diameter of 0.2, 0.25, 0.3mm of 3,4,5 holes respectively by conducting an experiment on single cylinder kirloskar diesel engine.

2 IsmetCelikten in their work on "Experimental investigation of effect of injection pressure on engine performance and emission in diesel engines" found that the variation of NOx, HC, CO emission with injection pressure and throttle position at 50%, 75% and 100% etc.

3 ARYA PIROOZ in their work on "Effect of injector nozzle geometry on spray characteristics , An analysis"investigate effect of various nozzle geometry parameters such as number of nozzle holes, nozzle hole diameter, needle seat angle and diameter of nozzle sac on pressure variation inside cylinder.

4 Sibendosom, Anita I. Ramirez and suresh K. Aggarwal in their work on "Effect of nozzle orifice geometry on spray, combustion and emission characteristics under diesel engine conditions" obtain the effect of conicity and hydrogrinding on spray cone angle, spray distribution and other performance parameters computationally.

5 Deva kumar, K. Vijayakumar Reddy in their on "Effect of fuel injection pressure on performance of single cylinder diesel engine at different manifold inclination " found the effect of injection pressure at various load on performance characteristics such as BTE, ITE, mechanical and volumetric efficiencies as well as on HC, CO, NOX emissions experimentally.

6 Rosli Abu.Bakar, Semin and Ismail in their study on "Fuel injection pressure effect on performance of DI diesel engine" conducted an experiment on two-stroke, four cylinder direct injection diesel engine with an eddy current dynamometer to find out effect of injection pressure ranging from 180 to 220 bar on BTE, ITE and mechanical efficiency under fixed speed – variable load and variable speed- fixed load conditions.

3. Experimental Setup

The setup consists of single cylinder, four stroke, VCR (Variable Compression Ratio) diesel engine connected to eddy current dynamometer. It is provided with necessary instruments for Combustion pressure, crank-angle, airflow, fuel flow, temperatures and load measurements. These signals are interfaced to computer through high speed data acquisition device. The setup has standalone panel box consisting of air box, twin fuel tank, manometer, fuel measuring unit, transmitters for air and fuel flow measurements, process indicator and piezo powering unit. Rotametersare provided for cooling water and calorimeter water flow measurement.

Fig.3 Experimental set up

Table 1: Engine specification

Sr. No.	Engine Parameters	Specification
1	Engine Type	Kirloskar TV1, single cylinder, four stroke, water cooled diesel engine
2	Rated power	3.5 KW (5 HP)@ 1500 rpm
3	Bore	87.5 mm
4	Stroke	110 mm
5	Injection pressure	205 bar
6	Compression ratio	18:1
7	Temperature sensor	Digital, PT-100 Type
8	Speed measurement	Rotary encoder
9	Load measurement	Strain gauge load cell, Range 0-50 Kg
10	Water flow measurement	Rotameter

Table 2: Dynamometer specification

Sr. No.	Parameters	Specification
1	Type	Eddy current dynamometer, water cooled
2	Maximum KW	7.5KW, Model TME-10
3	Speed	1500-6000 rpm
4	Arm length	185 mm

Table 3: Exhaust gas analyzer specification

Sr. No.	Measuring Item	Range	Resolution
1	CO	0-10 %	0.001%
2	HC	1-15000 ppm	1 ppm
3	CO2	0-20 %	0.01 %
4	O2	0-25 %	0.01 %
5	NOx	0-5000 ppm	1 ppm

Fig.4 Exhaust gas analyzer

Problem statement:

The aim of project is to study and analyze performance and emission characteristics of DI diesel engine for various fuel injector nozzle configuration using Kirloskar made TV 1 engine.

Objectives:

i) Development of injector prototype model of specific configuration

ii) Investigate performance of injector model for various geometric parameters experimentally.

Aim and scope for future study:

The performance and emission characteristics of direct injection diesel engine for different nozzle configuration have been analyzed to know which nozzle configuration is better. It helps us in reducing emission and hence pollution level for protecting environment. It also reduces fuel consumption to get fuel economy.

4. Results and discussion:

Effects of injection timing and nozzle orifice geometry on performance and emission characteristics are discussed below:

1) Indicated power

From fig.5 it is observed that indicated power increases with increase in load but it is minimum at injection timing of 23° bTDC. It is seen that indicated power at full load decreases with increase in number of nozzle holes. At 50% load, there is significant effect of injection timing on indicated power for all nozzle configurations.

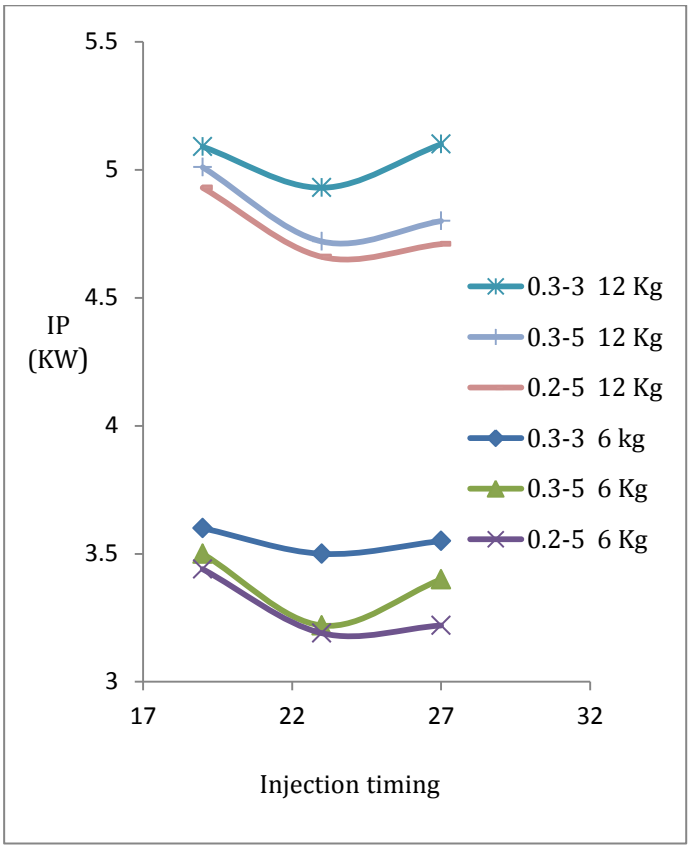

Fig.5 Effect of injection timing and nozzle orifice geometry on indicated power

2) Friction power

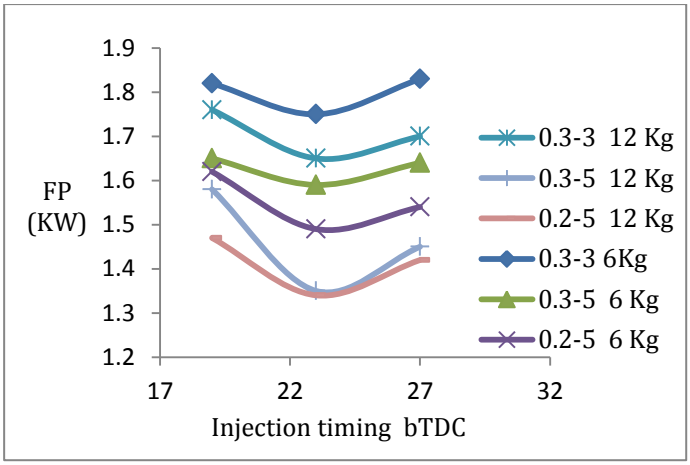

Fig.6 Effect of injection timing and nozzle orifice geometry on Friction power

Fig.6 shows effect of injection timing, number of nozzle holes and nozzle orifice size on friction power Friction power is minimum at 23° is for full load condition. It also reduces with increase in number of nozzle holes. It is found that friction power for 0.3-3 holes, 0.3-5 holes and 0.2-5 holes are 1.65 KW, 1.35 KW and 1.34 KW respectively.

3) Brake mean effective pressure

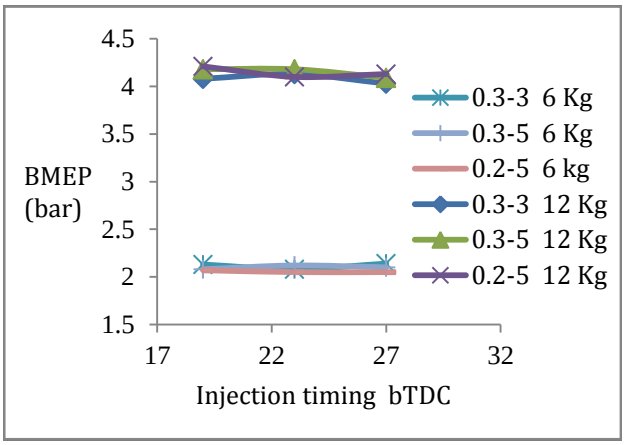

Fig.7 Effect of injection timing and nozzle orifice geometry on Brake mean effective pressure

Fig.7 shows variation of brake mean effective pressure with injection timing for different nozzle configurations at 6 Kg and 12 Kg load respectively. It is found that injection timing and nozzle configuration has no significant effect on brake mean effective pressure at 6 Kg and 12 Kg respectively.

4) Indicated mean effective pressure

Fig.8 shows variation of Indicated mean effective pressure with injection timing for different nozzle configurations at 6 Kg and 12 Kg load respectively. It is mainly associated with indicated power and friction power. It follows same trend as that of indicated power. At 23° bTDC injection timing, Indicated mean effective pressure falls down marginally with increase in number of nozzle holes and decrease in nozzle orifice size. At 19° and 27° bTDC with decrease in nozzle orifice size from 0.3 to 0.2 mm, indicated mean effective pressure decreases from 6.15 bar to 6.01 bar and 5.85 to 5.76 bar respectively.

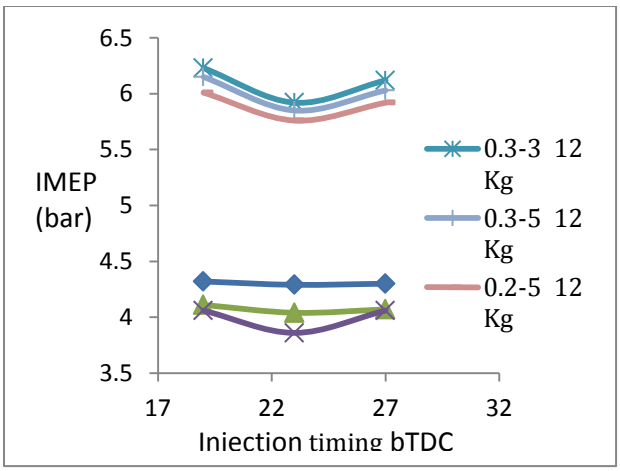

Fig.8 Effect of injection timing and nozzle orifice geometry on Indicated mean effective pressure

5) Brake thermal efficiency

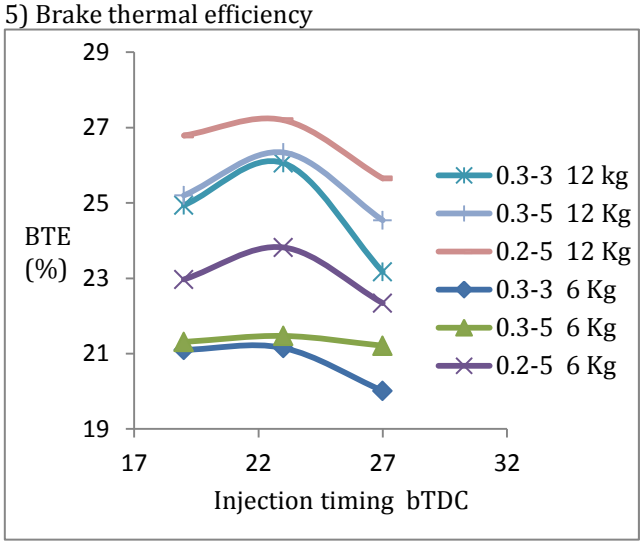

Fig.9 Variation of Brake thermal efficiency with injection timing at 6 Kg and 12 Kg Loads

Fig.9 shows variation of brake thermal efficiency with injection timing ranging from 19° bTDC to 27° bTDC at 50% and 100% load. It is found that brake thermal efficiency increases with increase in load irrespective of injection timing. Brake thermal efficiency increases with increase in number of nozzle holes and decrease in nozzle orifice size. This is due to better air fuel mixing and improved combustion at 23° bTDC.

6) Indicated thermal efficiency

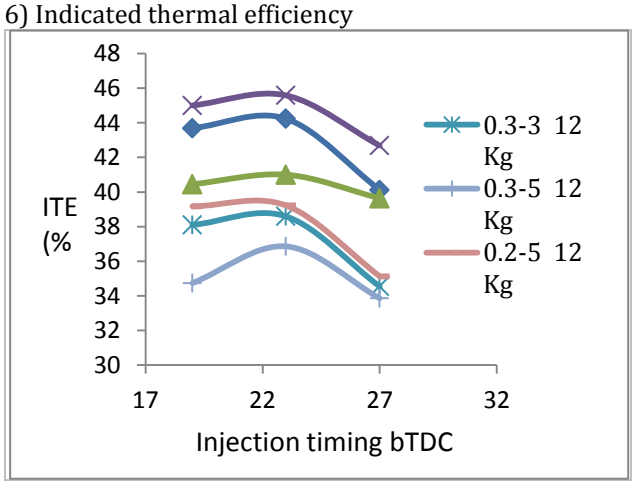

Fig.10 Variation of indicated thermal efficiency with injection timing at 6 Kg and 12 Kg Loads

Fig.10 shows effect of injection timing 19°, 23°, 27° bTDC and nozzle orifice geometry 0.3-3 holes, 0.3-5 holes, 0.2-5 holes on indicated thermal efficiency at variable load condition. If load is increased from 6 Kg to 12 Kg, indicated thermal efficiency decreases due to reduction in friction power with increase in load. Indicated thermal efficiency decreases by advancing or retarding by 4° crank angle due to increase in fuel consumption.

7) Mechanical efficiency

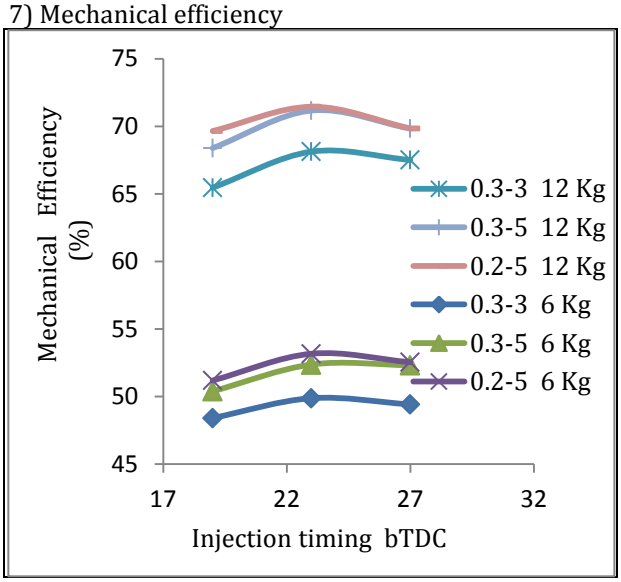

Fig.11 Variation of Mechanical efficiency with injection timing at 6 Kg and 12 Kg Loads

Fig.11 shows behavior of mechanical efficiency at 50% and 100% load for injection timing 19°, 23°, 27° respectively. At 23° bTDC injection timing, mechanical efficiency increases with increase in number of nozzle holes from 3 to 5 and decrease in orifice diameter from 0.3 to 0.2 mm. mechanical efficiency found to be maximum for 0.2-5 holes at 23° bTDC is 71.45% for full load condition. This is due to reduction in friction power and better air fuel mixing at 23° bTDC.

8) Volumetric efficiency

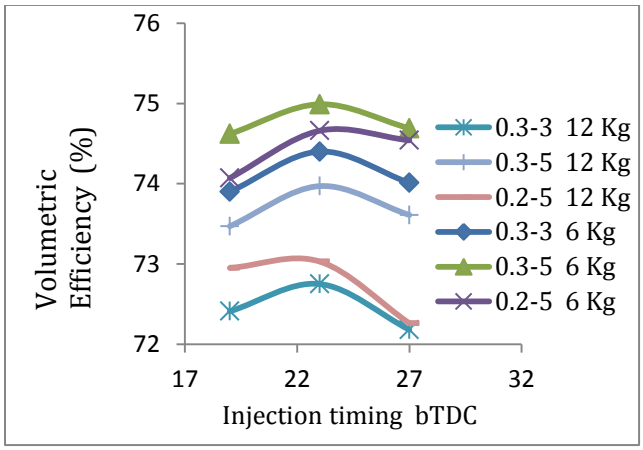

Fig.12 Variation of Volumetric efficiency with injection timing at 6 Kg and 12 Kg Loads

Effect of nozzle orifice geometry on volumetric efficiency at 19°, 23°, 27° bTDC injection timing for 50% and 100 % load is shown in fig.12. It is seen that volumetric efficiency increases with increase in number of nozzle holes and size. As load is increased, fuel consumption is increased for obtaining rich mixture. Therefore, volumetric efficiency falls down. Injection timing has no such significant effect on volumetric efficiency.

9) Brake specific fuel consumption

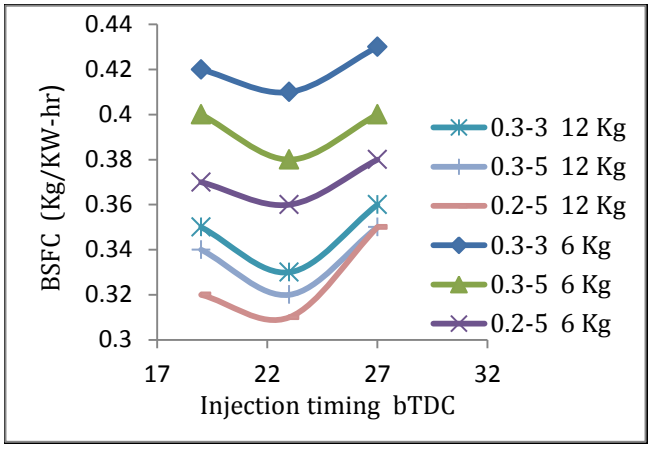

Fig.13 Effect of injection timing and nozzle orifice geometry on Brake specific fuel consumption

Fig.13 shows variation of brake specific fuel consumption with injection timing for different nozzle configuration such as 0.3-3 holes, 0.3-5 holes, 0.2-5 holes etc. At 50 % load, it is found that brake specific fuel consumption increases slightly with injection timing but at 100 % load brake specific fuel consumption is low at 23° bTDC. As number of nozzle hole increases, brake specific fuel consumption decreases due to better combustion, improvement in spray characteristics.

10) Hydrocarbon (HC)

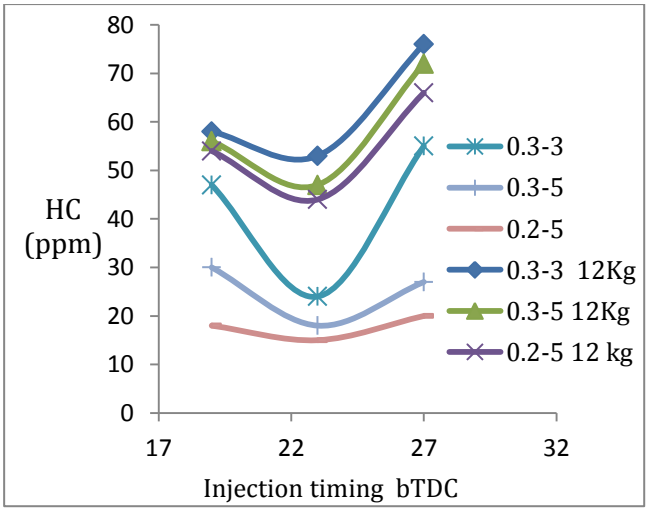

Fig.14 Effect of injection timing and nozzle orifice geometry on HC emissions

Effect of nozzle orifice geometry on HC emission at injection timing of 19°, 23°, 27° bTDC is shown in fig.14. It is observed that HC emission increases with load but decreases with increase in number of nozzle holes and reduction in nozzle hole size. For 0.2-5 holes,

HC emission is found to be 54, 44, 66 ppm respectively. This is due to proper fuel droplet distribution and reduction in droplet size which enhances combustion.

11) Carbon monoxide (%)

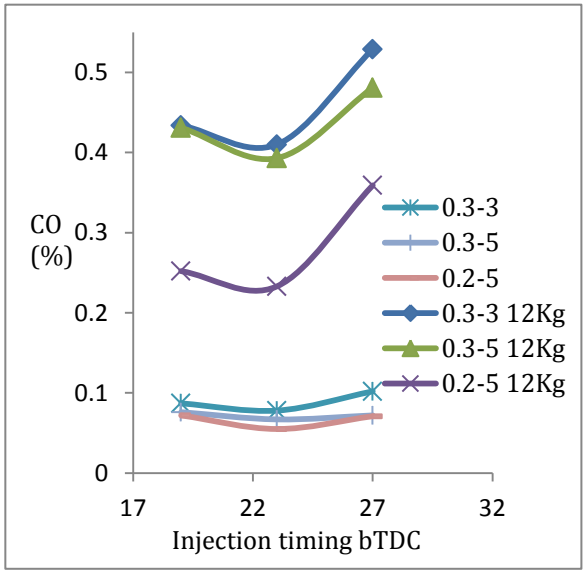

Fig.15 Effect of injection timing and nozzle orifice geometry on CO emissions

Fig.15 shows variation of CO with change in load from 6 Kg to 12 Kg at injection timing of 19°, 23° and 27° bTDC respectively. For 0.3-3 holes nozzle configuration, CO emission is increased from 0.078 % to 0.410 % at 23° bTDC. This is due to fact that fuel consumption increases with load. CO emission decreases with increase in number of nozzle holes and found to be mimimum at 23° bTDC because better distribution of fuel droplet leads to proper air fuel mixing and improved combustion.

12) Nitrogen oxide

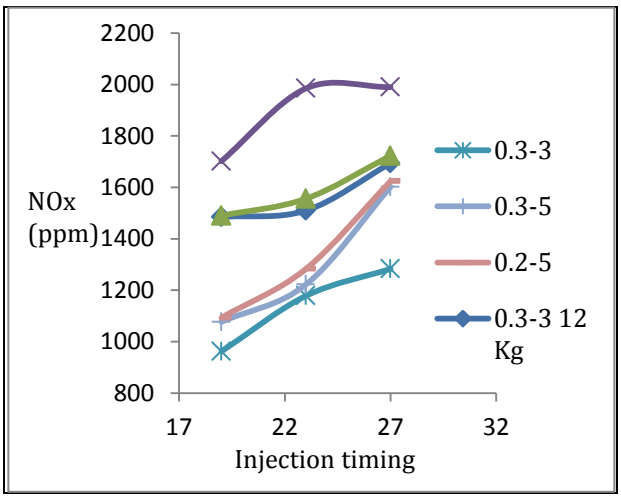

Fig.16 Effect of injection timing and nozzle orifice geometry on NOx emissions

Fig.16 shows variation of NOx with change in load at 19°, 23°, 27° bTDC injection timing. NOx emission increases with increase in number of nozzle holes and decrease in nozzle hole size. For 0.2-5 holes NOx found to be maximum i.e. 1990 ppm at 27° bTDC. NOx emission also increases with injection timing as improved combustion leads to peak combustion temperature.

Conclusions

From the experimental investigation it is found that 0.2-5 holes nozzle configuration at 23° bTDC injection timing is better than 0.3-3 holes and 0.3-5 holesrespectively due to following reasons:

i) Friction power is lowered by 23.13% and 17.91% compared to 0.3-3 holes and 0.3-5 holes configurations respectively.

ii) Brake thermal efficiency and indicated thermal efficiency are improved by 8.30% and 2.85% respectively due to proper fuel atomization and combustion.

iii) Mechanical efficiency is improved by 4.64% and 0.5% when compared with 0.3-3 holes, 0.3-5 holes configurations at same injection timing whereas specific fuel consumption is reduced by 6.45% and 3.22% respectively.

iv) HC and CO emission are lowered due to reduction in fuel consumption and improvement in spray characteristics by 37.5% and 46.1%.

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HP31704306-Emission Development on Six Cylinder Diesel Engine to Achieve CPCB-II Emission norms with Mechanical FIE System

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Abstract

Diesel engines are commonly being used for power generation due to its higher thermal efficiency and it is better-quality fuel consumption compared to gasoline engines. CPCB-II emission norms, which are the most stringent limits in the world for this category of diesel engines up to above 75 kW. An engine manufacturer has a big challenge to achieve stringent emission norms with least modification. In engine design to reduce manufacturing and developments cost. Research work carried out to achieve the norms for the power ratings (1500 rpm and 115 kW).Optimization such as of EGR flow rate and injector nozzles (which includes nozzle type, no. of holes, hydraulic through flow, spray cone angle, nozzle tip protrusion etc.) and turbocharger, appropriate injection timing with mechanical fuel injection pump etc.To have a better trade-off between NOx-PM and NOx-BSFC, carrying out tests on 5.7-liter six cylinder turbocharged diesel engine.

Keywords: *CPCB-II, Turbocharged, EGR, engine optimization, Injector Nozzle, Mechanical FIE.*

1. Introduction

Indian Ministry of Environment and Forests-Central Pollution Control Board (CPCB) has proposed new emission limits for diesel genset engine. Diesel engines are the most powerful and efficient thermal engine, the diesel engines are comprehensively making use of in almost all on road and off road applications.

Indian emission norms for stationary gensets engine are upgraded from CPCB I to CPCB II. These revised emission norms call for a significant change in emission limits. CPCB II emission norms call for 62% reduction in NOx+HC and 33% reduction particulates for engines above 75 kW up to 800 kW power range compared to existing CPCB I emission norms. CPCB II norms are more stringent as compared to European Stage IIIA and CEV BS III.

This paper present experimental work carryout to achieve stringent emission norms and fuel economy, The major challenge in meeting the proposed norms is with minimum cost impact to the customer. This paper deals with the strategies to meet CPCB-II emission norms by appropriate selection of engine hardware and optimization of injection parameters, EGR flow rate, turbocharger, mechanical fuel injection pump, high pressure pipe, static injection timing, intercooler to achieve emission performance without major changes in the engine configuration and cost impact for above 75 kW power rating engines. All above different combinations tested during engine optimization trials and configurations with optimum result selected as the final configurations.

Table1. Emission norms comparison between CPCB-Stage I and CPCB-II

Power range (kW)	**CPCB-I Stage I - Existing**				
	Emission Limit (g/kWh)				**Smoke (1/m)**
	NOx	**HC**	**CO**	**PM**	
Up to 800 kW	**9.2**	**1.3**	**3.5**	**0.3**	**0.7**
	10.5				
Fuel Spec: Less than 500 ppm Sulfur diesel					
Power range (kW)	**CPCB-I Stage II - Proposed**				
	Emission Limit (g/kWh)				**Smoke (1/m)**
	NOx +HC		**CO**	**PM**	
Up to 19 kW	**7.5**		**3.5**	**0.3**	**0.7**
More than 19 up to 75	**4.7**		**3.5**	**0.3**	
More than 75 up to 800	**4.0**		**3.5**	**0.3**	
Fuel Spec: Less than 350 ppm Sulfur diesel					

This engine emission has developed for genset applications and used test cycle (5 mode cycle

Emission test (as per ISO 8178 Type D2).The main challenging task to reduce Nox as well as reduce particulate matter both results are trade-off, and The direct injection diesel engine is one of the most efficient thermal engines known. The use of diesel engines for road applications has been widely extended during the last decade due to their relatively lower fuel consumption when compared to spark ignition engines. For this reason, DI diesel engines are widely used for heavy-duty applications and especially for the propulsion of Generators & Tractor. Even though the efficiency of these engines is currently at a high level, there still exist possibilities for further improvement. On the other hand, there are problems associated with its use that result from the relatively high values of particulate emissions and NOx values. Furthermore, that NOx also contributes to the greenhouse effect. Environmental concerns have led to progressively more stringent emission regulation for diesel engine. Keeping this in view, the government of India keeps regulations on the exhaust emission level of engines from time to time. Currently applicable emission norms for GENSET in India are Central Pollution Control Board (CPCB)- 1 and proposed CPCB - 2 emission norms will be applicable from July 2014, for Generator up to 19kw, >19kw.CPCB-II engine are smoother as compare to CPCB-I due to performance related hardware has changed and optimized the better combustion. The motive of the present work is to present an experimental investigation aiming towards meet the desired engine performance and emission levels for development of 125kva diesel genset engines to this problem in cost effective way so that the minimum design modification is required in the existing engine and to reduce the lead-time. Tests were conducted under various operating conditions like Full / part throttle performance test, 5 mode cycle Emission test (as per ISO 8178 Type D2).

2. EXPERIMENTAL SETUP

Experiments were carried out on six-cylinder diesel engine (Rating 115 kW @ 1500 rpm), engine tested on test cell with all standard condition with instrumentations and the Engine setup with Instrumentation required for measuring & data acquisition during the emission test on engine dynamometer. The experimental measurements for the present investigation were performed to the achieve emission target.

The major task in the present experimental work is to evaluate the performance and emissions on diesel engine for genset application, used software to execute the emission test results. Hence, this engine selected for the present research. All test data captured during test and emission.

This paper deals with the strategies applied and experimentation details to meet the proposed CPCB Stage-II emission limits. The criticality increases exponentially for new versions. The experimental investigations carried out on various capacities turbocharger diesel genset engines.

Table.2. Engine Specification

Configuration	Base Engine (CPCB-I)	Upgraded Engine (CPCB-II)
Power	75 KW	114.9 KW
Type of Aspiration	Turbocharged	Turbocharged
No. of Cylinder	6	6
Swept Volume	5. 67 Ltr.	5. 67 Ltr.
Bore X Stroke	97X128	97X128
FIP	Bosch, A2000	Bosch, A4000
SIT	9 deg. BTDC	9 deg. BTDC
Injector	P-Type	P-Type With K-Factor
EGR	Without EGR	With EGR

Eddy Current Dynamometer: Eddy current dynamometer is a device, which is used to measure moment of force (torque) and power of the engine.
Emission Gas Analyzer: This analyzer used to measure the engine exhaust emission raw gases in ppm like HC, NOx, CO, CO2, and O2 etc.
Smoke Meter: Smoke meter is a device which used to measure smoke (FSN) of engine exhaust, smoke instruments measure optical properties of diesel smoke and used AVL 415S Smoke meter.

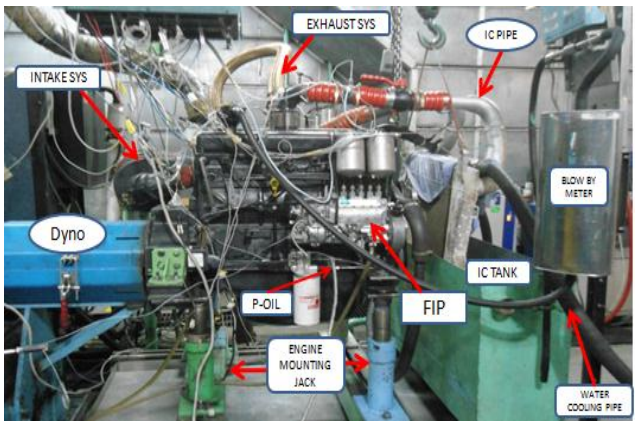


Fig.1. Experimental setup and instrumentations.

Fuel meter: Used AVL 735S/753C fuel meter to measured fuel flow rate, AVL Fuel mass flow meter is a high precise and continuous fuel consumption measurements system, which is worldwide.
Data Acquisition System: For studying the processes inside the cylinder, a data acquisition system used. This used for analyzing the measured cylinder pressure data and combustion parameters with variations in the crank angle other parameters.

3. STRATEGY TO MEET CPCB II EMISSION NORMS

Choosing the most significant hardware from a whole scope of options is difficult to meet the

requirements in an efficient and effective manner. The following combinations tried out during engine optimization trials to achieve optimum results.

Mechanical FIE –Use Mechanical fuel injection system to achieve stringent emission norms, mechanical A4000 FIP used to required fuel injection pressure and fuel quantity. The fuel-injection system supplies the diesel into engine. The fuel-injection pump generates the required fuel pressure for injection and delivers the fuel at the required rate. The fuel pumped through a high-pressure fuel line to the nozzle, which injects it into the engine's combustion chamber. The combustion processes in a diesel engine are primarily dependent on the quantity and manner in which the fuel introduced into the combustion chamber.The in-line fuel-injection pump used all over the world in medium-sized and heavy- duty trucks as well as on marine and fixed- installation engines. It is controlled either by a mechanical governor, which may be combined with a timing device, or by an electronic actuator mechanism. In contrast with all other fuel-injection systems, the

inline fuel-injection pump lubricated by the engine's lubrication system. For that reason, it is capable of handling poorer fuel qualities. The fuel injection equipment must be able to achieve precise control of fuel metering. Bosch, A4000 FIP used instead of existing A2000 FIP in order to increase the pump end pressure & indirectly increasing the peak combustion pressure, which increase the power as compare existing engine rating 75 kW, and optimizing the appropriate fuel pressure, which helps in reducing the NOx emission from the engine exhaust. NOx can also be decrease by changing other parameter like after treatment devices but these devices increases the cost and lead-time of the system.

Standard in-line fuel injection pumps - the range of standard in-line fuel injection pumps currently produced encompasses a large number of pump types. They are used on diesel engines with anything from 2 to 12 cylinders and ranging in power output from 10-to200 kW per cylinder. They are equally suitable for use on direct-injection (DI) or indirect-injection (IDI) engines.

EGR System-EGR is the most extensively used techniques for reducing NOx formation during combustion. NOx reduction in combustion chamber of engine, exhaust gas is presence in gaseous form and It is having latent heat and carbon contains to which are reacts with fresh air and fuel in combustion chamber. Transform the chemical reaction, which formation the emission, exhaust gas recirculation (EGR) used to reduce formation nitrogen oxide (NOx), external or internal EGR system works by recalculating a portion of an engine's exhaust gas back to the engine cylinders. In a diesel engine, the exhaust gas replaces some of the excess oxygen (from air flow) in the pre-combustion mixture. The NOx formation mainly when the mixture of nitrogen and oxygen is subjected to high temperature, NOx formation due to high combustion temperature, When EGR supply into engine combustion chamber, it dilute the air fuel ratio and decrease the combustion pressure and temperature caused by reduce the NOx formation in combustion chamber, EGR decrease combustion pressure which result decrease the engine power and efficiency, Most modern engines now require exhaust gas recirculation to meet NOx. In modern diesel engines, the EGR gas is cooled with a heat exchanger ie.EGR cooler. Since diesels always operate with excess air, they advantage from EGR rates as high as 40% (at idle, when there is otherwise a large excess of air) in controlling NOx emissions. Exhaust recalculated back into the cylinder can increase engine wear as carbon particulate wash past the rings and into the oil. Exhaust gas-largely carbon dioxide and water vapor-has a higher specific heat than air, so it still serves to lower peak combustion temperatures. However, adding EGR to a diesel reduces the specific heat ratio of the combustion gases in the power stroke. This reduces the amount of power that can be extracted by the piston. EGR also tends to reduce the amount of fuel burned in the power stroke. This is evident by the increase in particulate emissions that corresponds to an increase in EGR Particulate matter (mainly carbon) that are not burned in the power stroke is wasted energy. The most familiar is a diesel particulate filter in the exhaust system, which cleans the exhaust but reduces fuel efficiency. Since EGR increases the amount of PM that must be dealt with and reduces the exhaust gas temperatures and available oxygen, these filters need to function properly to burn off soot. Automakers inject fuel and air directly into the exhaust system to keep these PM filters from becoming blocked up.

EGR reduces the NOx emission by four ways:

A) Dilution Effect: The dilution of the intake charge with EGR reduces the mass fraction with oxygen. This lower oxygen mass fraction is the dilution effect. Adding together EGR to the intake air flow charge also affects average properties of the intake charge such as the specific heat capacity and molecular mass introducing other effects.

B) Thermal Effect: - EGR contains water and CO2, both of which having higher specific heat capacities than fresh air mass. The effect of amplified heat capacity is the thermal effect. The nitrogen in the air is replaced with inert gas helium to study the effect in separation. Intake air mass dilution with EGR simultaneously introduces the dilution and thermal effect. The oxygen mass fraction in the intake air

needs to be held constant to avoid interference from dilution effect.

C) Chemical Effect: - A quantity of the diluents gases may disconnect or actively participate in chemical reactions during the combustion progression, this is the chemical effect. One way to isolate the chemical effects is to replace nitrogen in the air with argon while the diluents is present this maintains a constant average charge heat capacity and oxygen concentration in the intake charge relative to the undiluted. This avoids interference from the thermal and dilution effect. However, it is not used in this project.

D) Add mass effect: - If adding diluents into the intake charge results in an increased mass flow rate, an additional effect is introduced. This added flow has an additional heat capacity due to its mass. The EGR used is 18 mm diameter EGR orifice tube in which the exhaust gases flows from the exhaust to intake due to pressure difference between Exhaust and Inlet system.

This EGR has a disadvantage that at full load EGR is not required but due to pressure difference the exhaust gases flows to the inlet and reduces the power and fuel efficiency at full load.

EGR Cooler – To reduce EGR gas temperature (From 550 °C to 125 °C).The heat absorbed from the combustion process is proportional to EGR rate, its specific heat, and the difference between combustion and EGR temperatures. Therefore, cooling the EGR stream allows for greater heat absorption from the combustion process, which leads to a lower rate of NOx formation. In addition, EGR cooler occupies less volume in the inlet system. Lower EGR volume displaces a smaller fraction of fresh filtered intake air, thus displacing less O2, which helps maintain combustion efficiency.

High Pressure Pipe –High-pressure pipe designed to sustain high fuel injection pressures, high-pressure pipe is used between fuel injection pump to injector to maintain fuel pressure, and high-pressure pipe internal diameter reduces from 2 mm to1.8 mm to increase the fuel pressure. Diesel fuel injection pressures in diesel engine plays a vital role for engine performance and emission treatment of combustion. Manufactured from special low-carbon steel as international standards, Special coating avoided corrosion and rust, High quality basic seamless steel tube, standardized inner bore diameter and fine finish ensuring smooth fuel flow and fuel pressure conforms to the OEM specification.

NTP (Nozzle Tip Protrusion) – Nozzle Tip Protrusion (NTP) plays a vital role in air fuel interaction. NTP adjusted such that fuel spray hits the piston bowl at the lip or the entry of the bowl.

Too low NTP may result in spray hitting the cylinder wall, the results show that a lower NTP led to slightly faster heat release, and too high value may result in insufficient charge mixing.
It may be noted that the NTP should be closely paired with injection timing. Effect of NTP on emissions is shown in Figure 2 and 3. [4].
Nozzle tip protrusion (NTP) is having wide scope for optimization for the combustion system. The influence of the variation in NTP on NOx and PM emissions, as well as BSFC

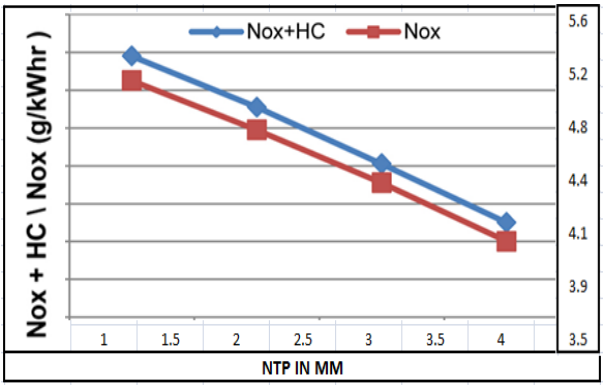


Fig.2. Effect of NTP on NOx+ HC and NOx emissions.

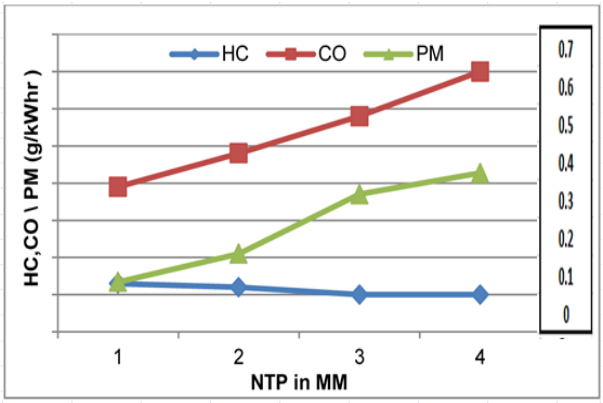


Fig.3. Effect of NTP on HC, CO and PM emissions

Injector Nozzle Hydraulic Through Flow-Through flow of injection quantity and numbers of hole of fuel injection for better atomization of fuel, fuel injection spray pattern. The nozzle injects the fuel into the combustion chamber of the diesel engine. It is a determining factor in the efficiency of mixture formation and combustion and therefore has a fundamental effect on engine performance, exhaust-gas behavior and noise. In order that nozzles can perform their function as effectively as possible, they have to be designed to match the fuel-injection system and engine in which they are used. The nozzle is a central component of any fuel-injection system. It requires highly specialized technical

knowledge on the part of its designers. The nozzle plays a major role in shaping the rate-of-discharge curve (precise progression of pressure and fuel distribution relative to crankshaft rotation).Optimum atomization and distribution of fuel in the combustion chamber, and Sealing off the fuel-injection system from the combustion chamber. Because of its exposed position in the combustion chamber, the nozzle is subjected to constant pulsating mechanical and thermal stresses from the engine and the fuel-injection system. The fuel flowing through the nozzle must also cool it. When the engine is overrunning, when no fuel is being injected, the nozzle temperature increases steeply. Therefore, it must have sufficient high-temperature resistance to cope with these conditions.

Injection timing –Optimization of fuel injection parameters static injection timing (SOI) and injection duration. Ignition delay is of two parts, physical delay and chemical delay. Physical delay is the atomization, vaporization, mixing with air and the chemical delay is the pre-combustion reactions which lead to the auto ignition. The physical component is affected by the injection pressure, air density and air flow characteristics. Injection timing will largely affect the chemical delay as it is dependent on the temperature and pressure of the charge when the fuel is injected. Advancing the injection timing will increase the chemical delay as the temperature and pressure are lower. This means that the ratio of premixed combustion is increased thus resulting in sharp increase in pressure and temperature in the cylinder. This results in higher NOx formation and soot gets reduced as the ratio of diffused combustion is lower. Here we observe reduction in HC and CO as the time available for combustion is longer. NOx emissions increase dramatically, when the in cylinder temperature and the Air-fuel Ratio (AFR) increase, whereas PM emissions increase at a rich AFR and lower temperatures. Generation of NOx and PM is a local phenomenon, so the local AFR and temperature at each point of the cylinder are of relevance to the generation of emissions.

Turbocharger Selection -The selection of a proper turbo charger enables to minimize the engine out smoke and to reduce the exhaust gas temperature and better fuel economy as well as better torque curve and improving engine performance. Required airflow to complete combustion of fuel and turbocharger increase the volumetric efficiency of engine. Matching an engine to a turbocharger to achieve well efficient turbocharger operation on a various range of engine speed is a complicated and compromising procedure. Matching large airflow turbine to improve the performance on higher engine speed, the exhaust turbine energy will be inadequate to operate the turbine at high sufficient speed to provide the charge to combustion fuel. This can result lower smoke and fuel consumption, matching a low air flow turbine to an engine develop the performance at low engine speed and output in excessively high boost pressure at high engine speed.

The optimization of turbocharger in work, the required air flow, boost pressure, boost pressure ratios, intake manifold temperatures & turbine inlet temperatures are estimated, accordingly three turbochargers were selected jointly in consultation with the turbocharger supplier. Sufficient database was scrutinized to arrive at the estimations.

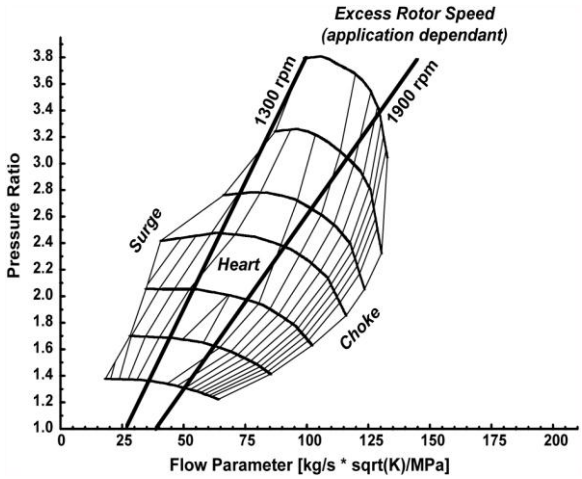


Fig.4. Optimized compressor map.

In turbocharger measured the turbine speed by speed sensor which is increase the turbine speed then increase the high mass air flow supply to the engine and decrease turbine speed decrease the mass air flow supply to engine, turbine speed depends on the exhaust gas pressure or exhaust gas energy. Non waste gate turbocharger is used in this application, Selection of turbocharger based on desire torque curve as well as achieves targeted BSFC and smoke limit, necessary adequate amount of EGR flow from exhaust to required boost pressure and exhaust manifold pressure which allow the EGR drive.

Intercooler-Intercooler is simply a heat exchanger mechanical device, which reduces the temperature of the compressed intake air mass, as a result of this volumetric efficiency of the engine is increased, intercooler makes density of air more oxygen rich air to the engine allowing more fuel to be burn thus improving combustion and giving more power. Due to Retard in fuel injection timing & EGR the power is reduced and specific fuel consumption is increased. Intercooler reduced the temperature of compressed air from 145 deg. C to 45 deg. C, Higher temperature of air decreased the performance and power of the engine due to incomplete combustion, and pressure drop depends on the intercooler size.

Load Sensing Principle- Estimating load on an internal combustion engine using only engine crank pickup sensor as an input.This technology has wide applications in control of internal combustion engines. One such application is Exhaust Gas Recirculation (EGR) control. It is well known that if amount of recalculated exhaust gas in an IC engine is varied as a function of both engine speed and load on engine, better engine performance (as measured in terms of emissions and specific fuel consumption) can be obtained, as compared to varying the amount of recalculated exhaust gas based only on engine speed. Using load estimation technology, such better engine performance to be obtained using only engine crank pickup sensor as an input to EGR control Electronic Control Unit (ECU).Test Setup Description. Overall objective of the tests was to demonstrate ability of EGR ECU to distinguish between different loads on the engine using only crank sensor as an input. The setup used during the tests on the engine dynamometer.

A magnetic pickup unit (MPU) was used as a crank pickup sensor. Output of this sensor was interfaced with EGR ECU. It should be noted that EGR ECU also serves as the ECU for Electronic Governor. The ECU implements digital signal processing methods on signal obtained from crank pickup sensor, and forms an estimate of load on engine (hereafter called Load Index).It was possible to operate the dynamometer in torque control mode (where a specified torque is applied to engine) or in speed control mode (where engine speed is regulated by dynamometer at a specified value).Following two sets of tests were conducted as a part of demonstration exercise. Torque controlled by dynamometer, speed controlled by mechanical governor. In this mode, torque on engine was controlled at various points between part load and rated load conditions for the engine. Variation of Load Index computed by EGR ECU as a function of engine load torque was studied. Speed controlled by dyno, torque controlled by manual fueling adjustment. In this mode, engine speed was regulated at a fixed value (1800RPM) using the dynamometer. Engine torque was varied manually by changing effective fuel rack position. Variation of Load Index computed by ECU as a function of engine load torque studied. Observations torque controlled by dynamometer,, speed controlled by mechanical governor: As mentioned earlier, in this test, load was imposed on engine using the dynamometer. The fueling, and hence engine speed, was controlled by mechanical governor. Thus, for a given load, engine speed was determined by mechanical governors droop characteristics.

4. RESULTS AND DISCUSSION

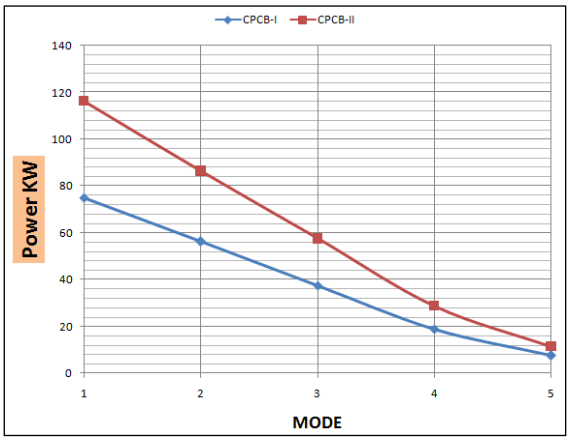


Fig.5. Power comparison between CPCB-I & CPCB- II Rating

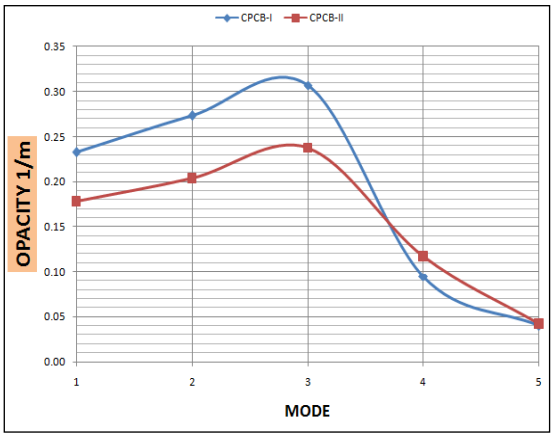


Fig.6. The Opacity comparison between CPCB-I &CPCB-II.

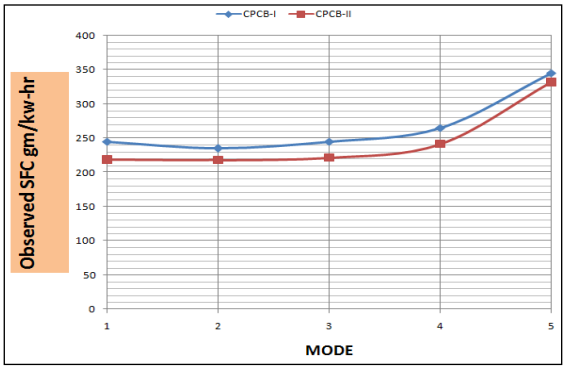


Fig.7. The BSFC comparison between CPCB-I & CPCB-II

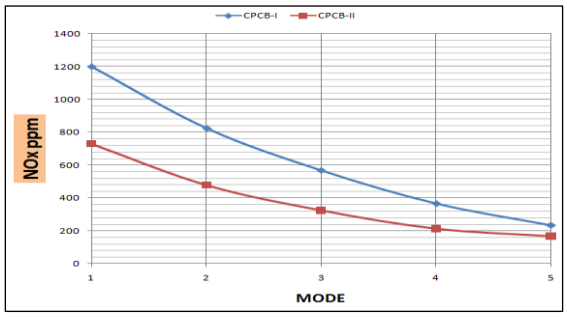


Fig.8. The NOx comparison between CPCB-I & CPCBII.

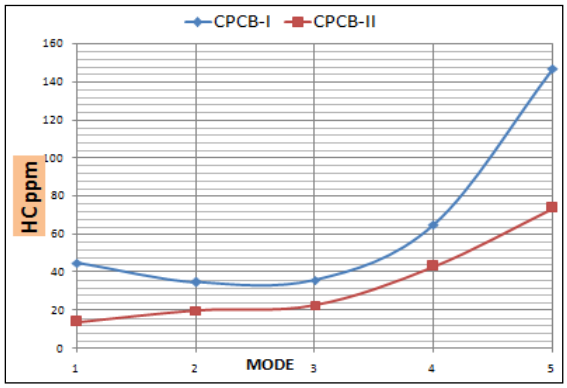
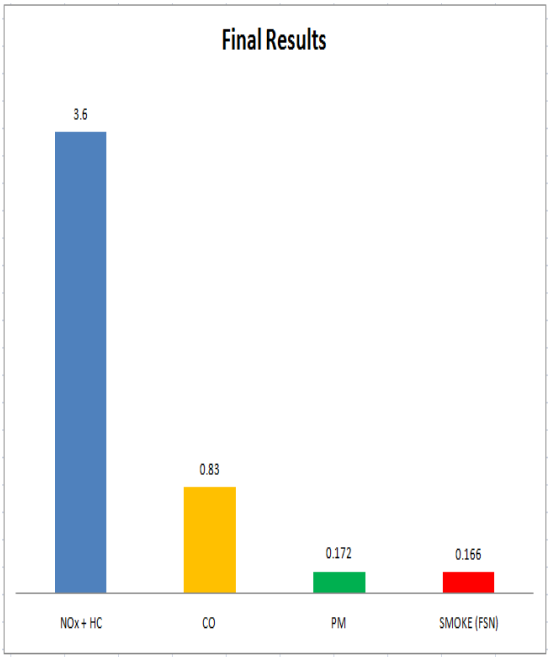


Fig.9. The HC comparison between CPCB-I & CPCBII.

Table.3. Actual results V/s Engg. Targets

Emission Parameter	CPCB-II Limits	Engg. Target	CPCB-I Results	Iteration -I (A4000 FIP & K-Fac Inj.)
Nox + HC (g/KWh)	4.0	3.5	7.23	3.60
CO (g/KWh)	3.5	2.5	1.1	0.83
PM (g/KWh)	0.2	0.15	0.251	0.172
Smoke (m-1)	0.7	0.56	0.541	0.166

From above figures 5. It can be seen that the Power requirement in CPCB-II is high because during this rating calculation customer has demanded 10% overload Power at rating speed i.e.1500 rpm. Hence there is 10 % power rise in CPCB-II engine rating as compared to CPCB-I rating.
Also from figures 6, 7, 9 it can be seen that there is drastic improvement in Smoke, BSFC and HC this is because of use of K- Factor Injector & Fig. 8 shows improvement in Nox & this is observed because of use of A4000 FIP.



5. CONCLUSION

Stringent CPCB-II Emission limits have been achieved with mechanical fuel injection system. Emission limits for NOX + HC are decreased by 50.20 % as compared to baseline limits. Also there is very large margin for Carbon Monoxide (CO) – 24.54%, Particulate Matter (PM) - 31.47% & Smoke-69.31%.CO External cooled EGR has play a significant role t o decrease NOX, and the external EGR system is low cost due simple design on and off type EGR valve are used and A4000 pump i.e. high fuel pressure as compare to A2000 FIP which lead to decrease PM, smoke and fuel consumption. Results Smoke and CO emission is decreased due to use of K-Factor injector and PM emission are increased due to retarded injection timing. Some of the Particulate matter is oxidized during exhaust stroke but the amount of oxidation of fuel is very less, Turbocharger optimization for better torque and better fuel efficiency with low smoke and PM as well as targeted EGR drive.

6. ACKNOWLEDGEMENTS

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HP31705103-Performance Evaluation of Single Cylinder Diesel Engine in Dual Fuel Mode with Biogas as Primary Fuel and Diesel and Biodiesel as Pilot Fuel

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Abstract- *In whole world energy demand is increasing day by day. It is very difficult to meet this high energy demand with coal and petroleum based fuels. The environmental pollution and global warming caused by fossil fuels consuming at faster rate, the energy focus now shifting toward nuclear energy and renewable source of energy but nuclear energy required careful handling raw and waste material, so only option is left that is renewable source of energy. Biogas and Biodiesel are best alternative fuel. The fuel can be directly used in diesel engines with/without any modification in the existing engine. In the proposed work experiments will be performed on Kirloskar (AV1model) 4 stroke single cylinder diesel engine. The primary fuel, biogas was mixed with diesel and Biodiesel pilot fuel. The experiment were performed to measure various parameters like brake power, energy conversion efficiency, and emission such as percentage of CO, CO_2, HC particulate matter and NO_x in exhaust gas. The effect of diesel replacement on emissions was also studied*

Keywords*: Dual fuel*, *biogas*, *biodiesel*, *diesel.*

1. Introduction

In India 75% of population is engaged in agriculture related activities. This population is living in rural area, some at very remote areas. Over past many years there has been phenomenal increase in the use of diesel engines in rural areas. Diesel engines are commonly used in power generation in rural applications, viz. tractors, irrigation pumps, generate sets chaff-cutters, crushers, mills etc., the demand for conventional fuel- diesel is increasing even at rural areas. On the other hand, the global huge pressure on energy issues and environmental pollution and global warming caused by burning fossil fuels has driven researchers as well as industrialists and governments to look for the other energy sources. This made essential that the electricity be introduced to rural areas in sustainable and environmentally sound way. Local production of fuel and generation of electricity is a sustainable option for rural electrification, which only can contribute to economic development and poverty reduction of rural areas. The diesel engines are most reliable and efficient combustion device, there is a growing need to adopt/modify such diesel engines for efficient and trouble free operation with biomass based gaseous fuels. Since biomass is available in huge amount in rural areas, it could be helpful to generate these fuels at the local sites; and using them to operate diesel engines. These fuels are also environment friendly because it increases renewable fuel consumption. These fuels can be used directly in the diesel engine with no/minimal modification in the engine, which is most advantages part of system. In the Present scenario, diesel engines are being extensively used for variety of applications in villages, while petrol engines or petrol operated generator sets are almost nil.

The invention and thermal efficiency in the second case was increased by 8% in second case. In second stage the by keeping the biogas flow rate fixed and reducing the pilot flow rate, it was noticed that the pilot quantity can be reduced to as low as 10%. This also resulted in lower exhaust gas temperature with turbocharger system. Better thermal efficiency in the experiment with turbocharger was probably due to better mixing of biogas with air. HC emissions were reduced notably with the system having longer mixing length. It showed a good mixing of air and biogas results in lower emissions and also with increased thermal efficiency (Pisarn Sombatwong et al. 2013) The Study of a diesel engine performance with exhaust gas recirculation (EGR) system fuelled with palm biodiesel. Experimental works using a multi-cylinder diesel engine with EGR and using Diesel-RK were performed at a constant engine speed of 2500 rpm in full load condition. On the basis of results the experimental works, palm biodiesel significantly increased fuel consumption increased NOx and slightly decreases in other emissions including CO2, CO, and unburned hydrocarbon (UHC). However, the use of EGR showed a significant reduction in the NOx emission and exhaust temperature with increases in fuel

economy, CO, CO2, and UHC emissions (MohdHafizil Mat Yasin et al. 2017) The effects of varying composition of biogas on performance and emission characteristics of compression ignition engine using exergy analysis carried out by (SaketVerma et al. 2017). It was found that the peak cylinder pressures with biogas dual fuel modes were always higher than that for diesel fuel mode. Peak in-cylinder pressure for diesel fuel mode was found 64.73 bar; and that for dual fuel modes: BG93, BG84 and BG75, it was found to be 71.56 bar, 70.15 bar and 68.84 bar respectively. It was concluded that the high methane fraction in biogas equitable performance can be obtained in dual fuel operation without any major engine modifications. However, for improved performance with high CO2 concentration in biogas, significant changes in operating parameters are suggested. By increasing compression ratio in improvement in BTE was observed by (Bhaskor J. Bora et al. 2014) This, in turn, increases the probability of more amount of biogas to undergo complete combustion. The effects of CO_2 content was studied in biogas on the fuel consumption and NO_x emission. Thermal efficiency improved and NO_x reduced with using a lean burn strategy. He concluded that a significant reduction in NOX emissions is expected using biogases containing CO_2; however, an increase in fuel consumption appears unavoidable. A lean burn strategy was effective for reducing both fuel consumption and NOX emissions; however, the use of biogas with the stoichiometric air–fuel ratio (which can better handle transient operating conditions) appears effective in reducing NOX emissions and can improve the fuel economy at higher loads (Yungjin Kim et al.2016)

2. Objectives

The objective of this work is to study engine performance

a. Using diesel fuel,
b. Using biodiesel fuel,
c. Using Bio gas in dual fuel mode with diesel and biodiesel

The parameters in this work are under,

1. Variation in break power
2. Brake specific fuel consumption
3. Thermal efficiency
4. Mechanical efficiency
5. Volumetric efficiency

The engine performance was compared with on Diesel and Biodiesel fuel performance against Bio Gas Diesel or Biogas Biodiesel fuel performance.

3. Experimental Setup

Table -1: Technical specifications for Kirloskar TV1 engine

Make	**Kirloskar Oil Engines**
Type	**Compression Ignition, Constant speed, Four Stroke ,Water cooled,**
No. of cylinder	**One**
Bore X stroke	**87.5 mm X 110 mm**
Cubic capacity	**0.661 litres**
Compression ratio	**17.5:1**
Peak pressure	**77.5 kg/cm^2**
Maximum speed	**2000 rpm**
Min. idle speed	**750 rpm**
Operating speed	**1500 rpm**

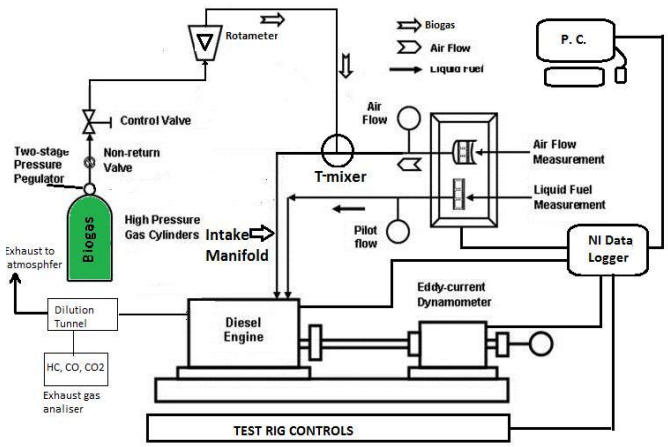

Fig.1 Block diagram of test setup

4 Results and Discussion

From figure 2 and 3 it is observed that, pressure rise is maximum for Biodiesel and diesel mode in both cases. The pressure rise is decreasing with increasing portion of biogas when engine is run in dual fuel mode.

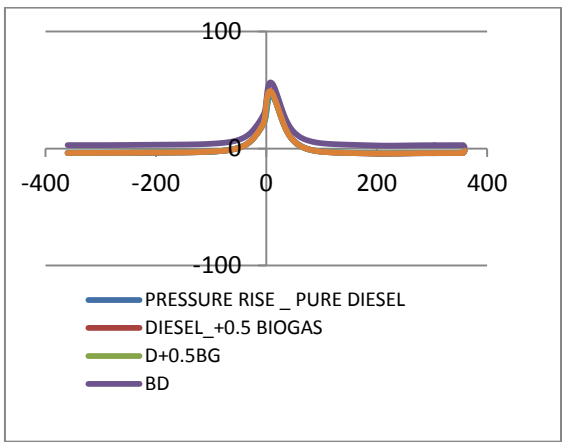

Fig. 2comparison of ICP vs CA (1.7 k W, 1500 rpm)

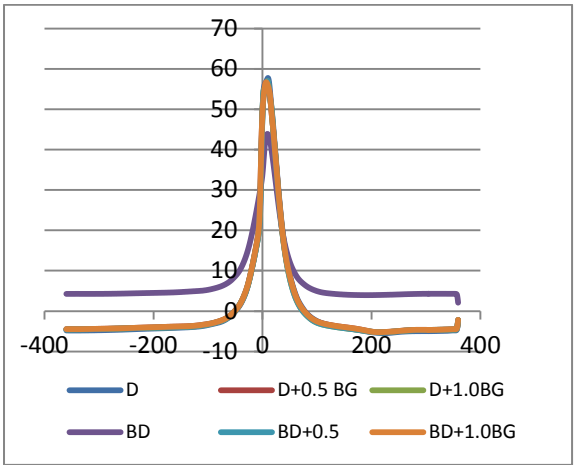

Fig. 3comparison of ICP vs CA (3.5 k W, 1500 rpm)

Brake power

Variation of brake power with % of full load is shown in fig 4. It is observed that the dual fuel mode develops a brake power equivalent to the diesel dual only mode. At lower loads the power developed in different modes of operation is nearly the same. At higher loads it is seen that the dual fuel mode develops somewhat higher power than that of diesel being the difference very small.

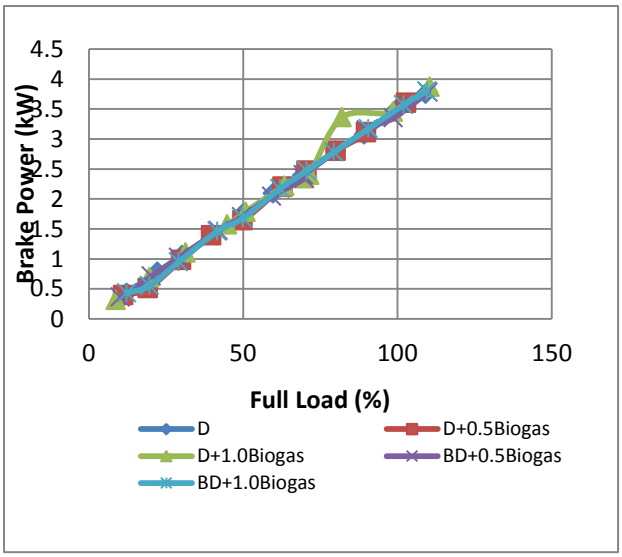

Fig4. Variation of Brake power with % full load

Mechanical Efficiency

The variation of mechanical efficiency of the engine with % of full load is show in fig 5 was operated on diesel and dual fuel mode. In dual fuel mode, the engine develops an equivalent amount of power as developed in diesel mode. A keen observation shows that the mechanical efficiency is slightly higher than diesel except at few points. A maximum mechanical efficiency of 70.61% was obtained at slightly overloaded condition.

Brake Thermal Efficiency (%)

The variation of brake thermal efficiency of the engine with % of full load is show in fig 6 was operated on diesel and dual fuel mode. The maximum brake thermal efficiency was found at 90% of full load in dual fuel mode.

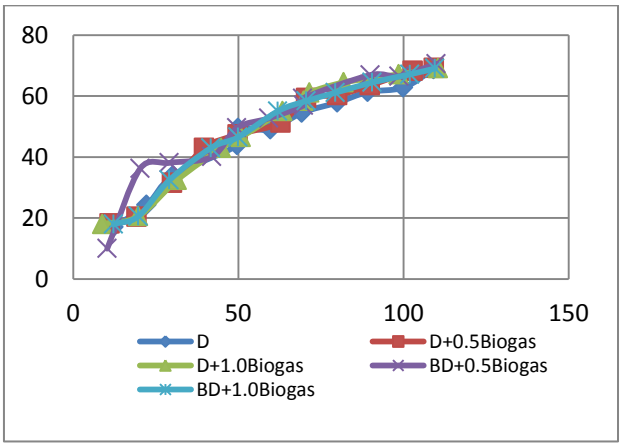

Fig.5 Variation of Mechanical Efficiency (%) with % full load

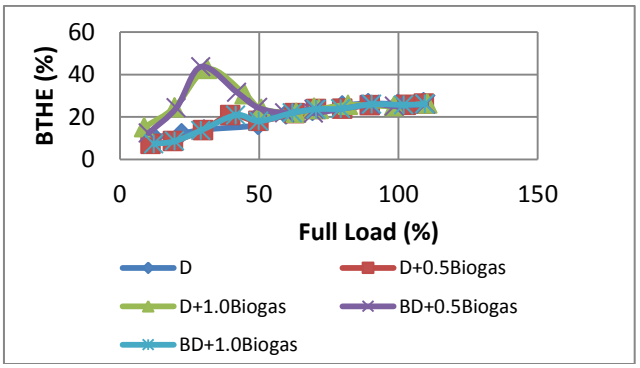

Fig. 6 Variation of BTHE (%) with% full load

The flow rate for this particular case was 1.0m3/hr. This is achieved because of the biogas which is having 95% methane. Brake thermal efficiency in dual fuel mode is higher than that in diesel only mode at all the points. This is the advantages of methane present in biogas. It is clean fuel, and burns completely. This is in accordance with all engines operating in dual fuel mode with a gas as primary fuel.

Volumetric efficiency

The variation of volumetric efficiency of the engine with % of full load is show in fig 7 was operated on diesel and dual fuel mode. In all cases it decreases with increase in the load. For both mode of operation, i.e. diesel only mode as well as dual fuel

mode, the volumetric efficiency is decreasing with increase in load. For biogas flow rate of 0.5m³/hr, it is minimum. This is because of the nature of biogas induction method. For biogas flow rate of 0.5m³/hr, it is minimum. This is because of the nature of biogas induction method. As biogas is directly fumigated into the induction manifold using a T-joint mixer, it directly replaces/displaces the amount of air that is supposed to enter in the cylinder. This may affect other parameters like emissions, maximum peak pressure, and indicated work as amount of air required for complete combustion may decrease.

Hydrocarbon (HC) emissions

The variation of HC emission of the engine with brake power is show in fig 8 was operated on diesel and dual fuel mode. Unburnt hydrocarbons are the result of incomplete combustion. The figure shows that unburnt HC emissions are lowest in diesel only operation, and increases with higher flow rates of biogas. As biogas has methane as its main and only constituent, it is obvious that the unburnt part of the fuel will be more methanous in nature. A biogas flow rate of 1.0 m3/hr records maximum HC emissions.

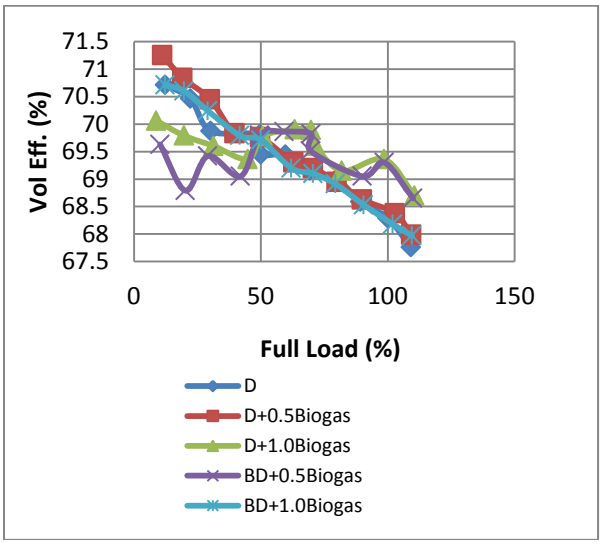

Fig. 7 Variation of Vol eff. (%) with % full load

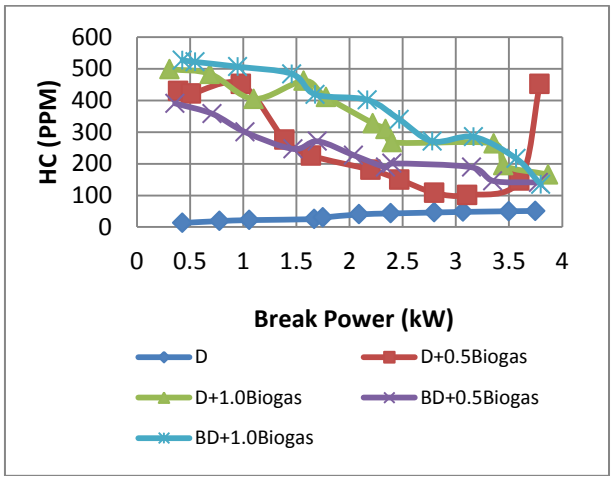

Fig. 8 Variation of HC with Break power

Carbon monoxide (CO) emissions

The variation of CO emission of the engine with brake power is show in fig 9 was operated on diesel and dual fuel mode. There are two major causes of formation of CO emissions. The first one is the incomplete combustion due to insufficient supply of oxygen to combustion chamber and second one is the poor mixture formation.

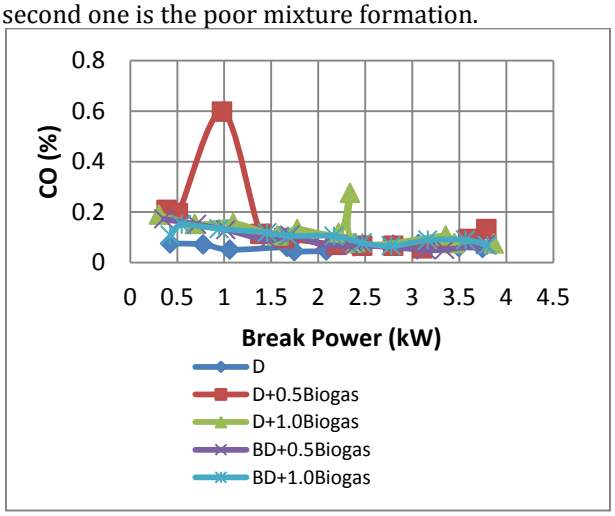

Fig. 9 Variation of CO with Break power

Carbon dioxide (CO_2) emissions

The variation of CO_2emissionof the engine with brake power is show in fig 10 was operated on diesel and dual fuel mode. Figure shows the biogas flow rate of 1.0m3/hr shows a reduced level of CO2 emissions. However, under this mode, the performance of engine was noisy, harsh, and intermittent. For other flow rates of biogas, the CO_2 level is higher in the emissions. This is because the biogas contains CO_2, and the same is found in the emissions also.

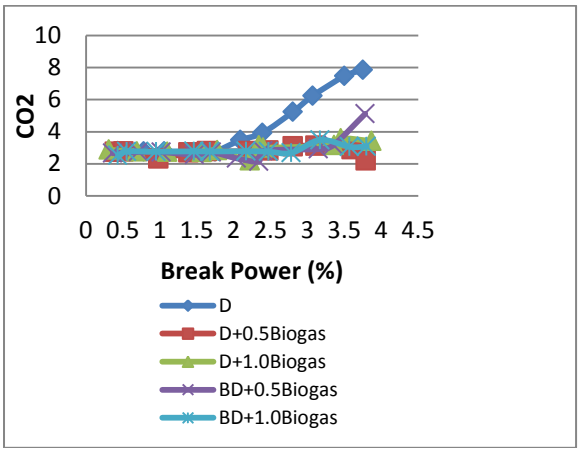

Fig. 10 Variation of CO_2 with Break power

Conclusions

The objective of this work was to study the performance of a single cylinder diesel engine when operated in biogas diesel dual fuel mode. It was studied by observing different parameters of diesel engine, following pars reports the short pres conclusions of this work.

It was found that the engine when operated on biogas-diesel and biogas-biodiesel dual fuel mode develops nearly same power as that of when operated on neat diesel mode. Near full load condition, the brake power developed in dual fuel mode is same as that of developed by neat diesel operation.

Mechanical efficiency the engine was also tested. It was noted that the biogas-diesel and biogas-biodiesel dual fuel mode has same as the mechanical efficiency over neat diesel mode. This trend is similar for part load as well as full load operation.

Volumetric efficiency of the engine was found decreasing with increasing loads. It further decreased in fuel mode as biogas is introduced into the induction manifold direct and is replacing some amount of air being sucked.

Thermal efficiency of engine was higher in diesel dual fuel mode as compared to neat diesel mode. This is due to the methane from biogas, which has good combustion characteristics. Biogas forms a homogeneous mixture before entering the combustion chamber. This is helpful for achieving near complete combustion of the charge. This is major advantage of this system.

Emission measurements were noted for neat diesel mode and biogas-diesel and biogas-biodiesel dual fuel mode. Neat diesel operation has a record minimum and fairly constant value throughout the operating range. Subsequently, biogas-diesel and biogas-biodiesel dual fuel mode represents reducing nature of HC emission with increasing loads. CO emissions have a similar trend. A like HC emissions CO emissions are also notably less for neat diesel operation, where in they decreases initially and then rises near full load. For biogas-diesel and biogas-biodiesel dual fuel mode, the trend shows that HC emissions reduce with increasing loads. For other flow rate of biogas CO_2 emissions are greater that neat diesel operation. This is because biogas inheritable contains 2 as one its constituents, which is found in the exhaust emissions also.

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HP31705109-Combined effect of EGR and Sesame oil biodiesel blends on CI engine: An experimental investigation

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Abstract

In this study, the biodiesel is produced from transesterification of sesame oil. Properties of biodiesel are very close to diesel hence it can be used in diesel engine without any modification but use of biodiesel in CI engine will leads to higher NO_X emission. Exhaust Gas Recirculation (EGR) is one of the most effective methods to reduce NO_X emission. The study is carried out to investigate the emission and performance characteristics of single cylinder, four stroke, direct injection and water cooled CI engine to observe the effect of different EGR rates and blends of sesame oil biodiesel. The rated power and speed of engine were 3.5 KW and 1500 rpm respectively. The engine performance parameters and exhaust emissions are evaluated. The results in each cases are compared with baseline data. Improvements have been observed in both performance parameters of the engine as well as exhaust emissions. It has been observed that B10 blend and 5% EGR is the best combination. For above combination the brake thermal efficiency is increased by 4.45% at partial load. With application of EGR at high load the reduction in NO_X is approximately 400 ppm but HC and CO emissions were increased by 1 ppm and 0.09 % only. The result shows that sesame oil Biodiesel and EGR both can be employed together in CI engines to obtain reduction of NO_X emissions

Keywords: *Transesterification, SOME, CI engine, EGR, NO_X.*

. Introduction

National interest in generating fuels for internal combustion engines continues to be strong to fulfill the energy demand of the world. The search for energy independence and concern for a cleaner environment have generated significant interest in biodiesel, despite its shortcomings. After United States, China and Japan India is the world's fourth largest consumer of crude and petroleum products. The net oil import dependency of India rose from 43 % in 1990 to 71 % in 2012 that resulted in a huge strain on the current account. India's energy security would remain weak until alternative fuels are developed to substitute or supplement petro-based fuels. Biodiesel is an alternative diesel fuel which can be obtained from the transesterification of vegetable oils or animal fats and methyl or ethyl alcohols in the presence of a catalyst (alkali or acidic). Rudolph Diesel demonstrated the first use of vegetable oil in compression ignition engine in 1910. He used peanut oil as fuel for his experimental engine.

There are reasons that justify the development of biodiesel.

1. It helps to provide a market for excess production of vegetable oils and animal fats.
2. It can decreases, although will not eliminate, the country's dependence on imported petroleum.

1

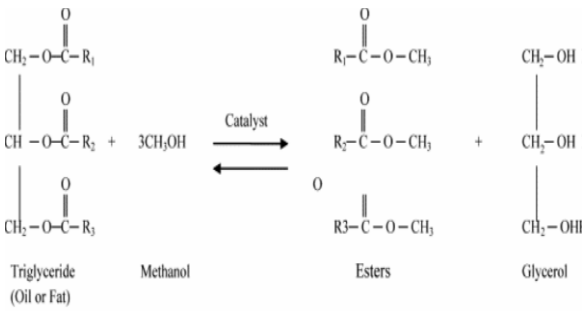
Biodiesel is renewable and does not contribute to global warming due to its closed carbon cycle

1. The exhaust emissions of carbon monoxide, unburned hydrocarbons, and particulate emissions from biodiesel are lower than with regular diesel fuel. Unfortunately, most emissions tests have shown a slight increase in oxides of nitrogen (NO_X).
2. When it added to regular diesel fuel in certain amount, it can convert fuel with poor lubricating properties into an acceptable fuel.

NO_X emissions can be reduced by lowering the cylinder temperatures. This can be done by three ways 1) Enriching the air fuel mixture 2) Lowering the compression ratio and retarding ignition timing 3) Reducing the amount of Oxygen in the cylinder that inhibits the combustion process. The first two methods reduce the efficiency of combustion and so the best way is to reduce the amount of Oxygen. This is done by recirculating of little amount of exhaust gas and mixing it into the engine inlet air into the intake manifold. This process is known as Exhaust Gas Recirculation (EGR).

Transesterification: The fatty acid triglycerides themselves are esters of fatty acids and the chemical splitting up of the heavy molecules, giving rise to simpler esters, is known as transesterification. The triglycerides are reacted with a suitable alcohol (Methyl, Ethyl, or others) in the presence of a catalyst

under a controlled temperature for a given length of time. The final products are alkyl esters and glycerin. The alkyl esters has favorable properties as fuels for use in CI engines, are the main product and the glycerin is a byproduct. The chemical reaction of the triglyceride with methyl alcohol is shown in figure 1. It can be seen from the reaction that one mole of the heavy triglyceride and three moles of methyl alcohol yields one mole of glycerol and three moles of lighter fatty methyl esters. Without the use of a catalyst the reactions would be very slow and also incomplete. A temperature of 60°C to 70°C would be needed for the reactions to become effective. Also a vigorous agitation of the reactants would be needed and so a mechanized stirrer in the reaction vessel becomes necessary. Various catalysts can be used. The most common are the alkalies, like NaOH and KOH. For transesterification any alcohol can be used. The most popular is methyl alcohol.



NOx formation mechanism

Oxides of nitrogen is produced in very small quantities can cause pollution. Oxides of nitrogen are dangerous to health in a prolonged exposure. Oxides of nitrogen which occurs only in the engine exhaust are a combination of nitric oxide (NO) and nitrogen dioxide (NO_2). Nitrogen and oxygen react at relatively high temperature. Due to post-flame combustion process in the high temperature region of combustion chamber NO formed. The high peak combustion temperature and availability of oxygen are the main reasons for the formation of NOx. At high combustion temperature in the present of oxygen inside the combustion chamber following chemical reactions will takes place behind the flame [12].

$N_2+O_2 = 2NO$

$N_2+2H_2O = 2NO+H_2$

Calculation of chemical equilibrium shows that a significant amount of NO will be formed at the end of combustion. The most of the NO formed will however decompose at the low temperatures of exhaust. But, due to very low reaction rate at the exhaust temperature, a part of NO formed remains in exhaust. The NO formation will be more in lean mixtures than in rich mixtures [1]. The concentration of oxides of nitrogen in the exhaust is closely related peak combustion temperature inside the combustion chamber.

Exhaust Gas Recirculation (EGR)

Instead of using after treatment systems to comply with exhaust emission legislation, it is also possible to avoid the formation of emissions during the combustion. The raw emissions are reduced and thus no after treatment is needed. It is common practice nowadays, to use EGR to reduce the formation of NO_X emissions. A portion of the exhaust gases is recirculated into the combustion chambers. This can be achieved either internally with the proper valve timing, or externally with some kind of piping, Figure 2 shows this schematically.

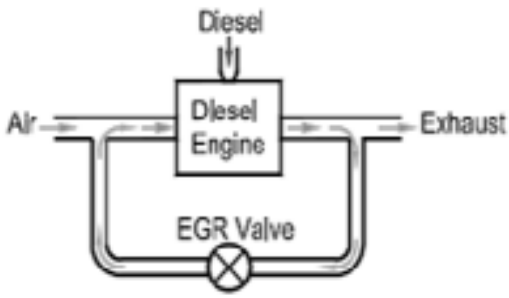


Fig.2 Exhaust Gas Recirculation [3]

Exhaust gases does not participate in the combustion reaction, they act as an inert gas in the combustion chamber. This leads to a reduction of the combustion temperature by different effects. The fuel molecules need more time to find a oxygen molecule to react with, as there are inert molecules around. This slows down the combustion speed and thus reduces the peak combustion
temperature, as the same amount of energy is released over a longer period of time.

The energy is also used to heat up a larger gas portion than it would without EGR. As the air is diluted with exhaust gas, the mass of a gas portion containing the needed amount of oxygen gets bigger. Another effect is the change in heat capacity. Exhaust gas has a higher specific heat capacity than air, due to the CO_2-molecule's higher degree of freedom. Gas mass containing EGR will get a lower temperature than pure air due to the same amount of combustion energy. NOx formation reduces directly due the lower temperature in a combustion chamber

EGR ratio is defined as the ratio of mass of recycled gases to the mass of engine intake. Also %EGR is.

$$\% \text{ EGR} = \frac{A - B}{C}$$

Where

A=Mass of air admitted without EGR
B = Mass of air admitted with EGR
C = Mass of air admitted without EGR

2. MATERIALS AND METHODS

This section provides a description of the materials and methodology used for production of biodiesel and CI engine test rig. used for performance and emission characteristics of TOME with EGR.

2.1Materials

The sesame oil used in this study was purchased from reliance mart hadapser, pune. The commercial diesel fuel was purchased from petrol pump which is nearer to Imperial College Of Engineering and Research (ICEOR) Wagholi, pune. Other chemicals (Methanol, KOH Catalyst) were procured during experimentation from D Haridas& Company, katraj, pune.

2.2Methods

The Biodiesel was produced by transestrification of the sesame oil using 6:1 molar ratio of methanol and 1.5 % of KOH.

Fig.3 Transestrification set up

The transesterification process was carried out as per the procedure given below:

1 kg refined sesame oil was taken in a 1000 ml capacity conical flask and heated at 55°C selected reaction temperature for 30 min preheating time. Then methyl alcohol was taken to obtain molar ratio of 6:1 and 10 g of Potassium hydroxide (KOH) was mixed thoroughly in it. This mixture was added to 1000 g preheated sesame oil and the mixture was placed on magnetic stirrer to carry out reaction for a period of 90 minutes in at the 60°C reaction temperature. After that the liquid which was a mixture of biodiesel (SOME) and glycerol is poured in a separating funnel and left it for the settling down for separation of biodiesel and glycerol. The glycerol settled at the bottom of separating funnel was separated by draining it. Then the biodiesel Remaining in the funnel was washed with distilled water for and allowed it to settle down. The water accumulated along with traces of glycerol at the bottom of the separating was drained. Washing of biodiesel is carried out three times to remove the remaining glycerol, alcohol and KOH in the biodiesel

Fig.4 Biodiesel glycerol separation

Fig.5 Water washing of biodiesel

At last the washed biodiesel was dried by silica gel it absorbs moisture and final biodiesel was ready to use.[7].Viscosity, density and calorific value were measured by redwood viscometer, hydrometer and bomb calorimeter respectively. The fuel properties of sesame oil methyl ester
diesel are summarized in Table 1.Methyl ester was compared with diesel fuel.

Table 1 Properties of Diesel and SOME

Property of oil	ASTM std	Diesel	sesame oil biodiesel
Density (kg/m3)	----	830	869
Kinematic viscosity (cSt)	1.9 to6.0	3.5	4.2
Flash point (°C)	>130	56	170
Fire point, (°C)	>153	62	187
Cloud point(°C)	-3 to -12	-10	-17

Pour point, (°C)	-15 to 10	-6	-5
Calorific value(kJ/kg)	> 33000	42000	39895

3. EXPERIMENTAL SETUP

In the present experimental work single cylinder, four stroke and CI engine was used. The engine was Kirloskar Make and water cooled. The engine is connected to Eddy current dynamometer for measurement of brake power. Engine torque was measured using load cell. The specifications of engine are given in Table 2.

Table 2 Engine Specification

Make	Kirloskar Engine
Model	TV1
No of cylinders	1
No of strokes	4
Cylinder Bore	87.5mm
Stroke length	110
Type of cooling	Water cooled
Power	3.5 KW
Rated Speed	1500 rpm
Compression Ratio	18:1
Loading device	Eddy current dynamometer

The experimental set up is shown in fig.6. It has stand-alone panel box consisting of air box, fuel tank, manometer, fuel measuring unit, transmitters for air and fuel flow measurements, RTD and thermocouples are used for air and water temperature measurement at various points. Signals from sensors are interfaced to computer through high speed data acquisition

Fig.6 Experimental Setup

device. Rotameters are used for cooling water and calorimeter water flow measurement. The exhaust gas emissions were measured by using AIRREX HG-540 exhaust gas analyser. Water cooled EGR cooler connected to engine by appropriate plumbing for cooling and recirculation of exhaust gas. The quantity of exhaust gas recirculation can be controlled by valve fitted in EGR path.

4. EXPERIMENTAL PROCEDURE

To achieve objective of experimental study engine was run at normal condition. Tests were conducted at 1500 rpm engine speed. B10 and B20 blends of sesame oil methyl ester were prepared on volumetric basis. Engine was started at no load and varied to rated load in number of steps. Set of reading is obtained without EGR and with 5% and 10% of EGR for pure diesel fuel [9]. Similar set of reading is obtained for B10 and B20 blends of SOME. Engine performance parameter like Brake Thermal Efficiency (BTE), Specific Fuel Consumption (SFC), and emission parameters such as Nitrogen monoxide (NO_X), Carbon monoxide (CO), Unburned Hydrocarbons (HC) were measured during test. Then these engine parameters were compared for different blends and EGR combinations.

5. RESULTS AND DISCUSSION

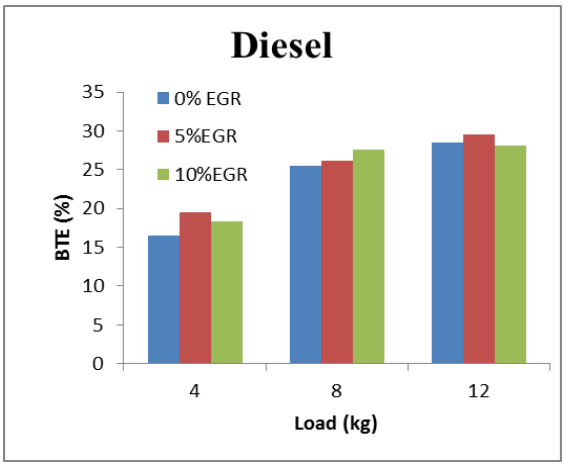
In this section fuel properties are discussed. Engine performance and emission data is analyzed and presented in graphical from for thermal efficiency, SFC, HC, CO, NO_X, emissions.

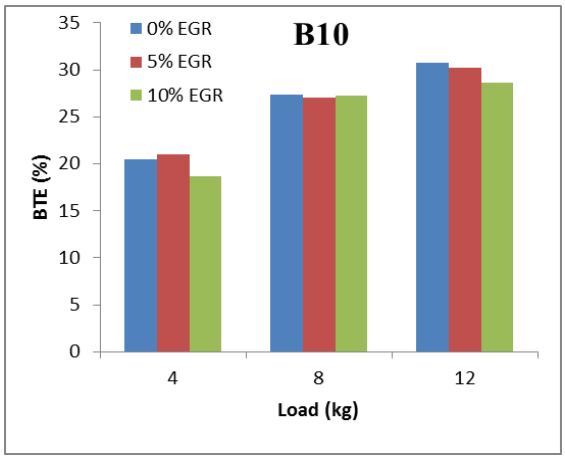
5.1 Fuel properties

The experimental results indicated that the density of sesame oil methyl ester is slightly high to that of diesel. The kinematic viscosities of diesel and sesame oil methyl ester were found as 3.5 and 4.2 cSt at 40°C. The calorific values of diesel and sesame oil methyl ester were found as 42 and 39.9 MJ/kg respectively. The calorific value of sesame oil methyl ester is 5 % less as compared to diesel fuel. The sesame oil methyl ester was found to have higher flash and fire point than diesel fuel hence it can be stored and transported safely. Cloud point of sesame oil methyl ester is higher than that of diesel.

5.2Effects on engine performance

5.2.1 Brake thermal efficiency (BTE)





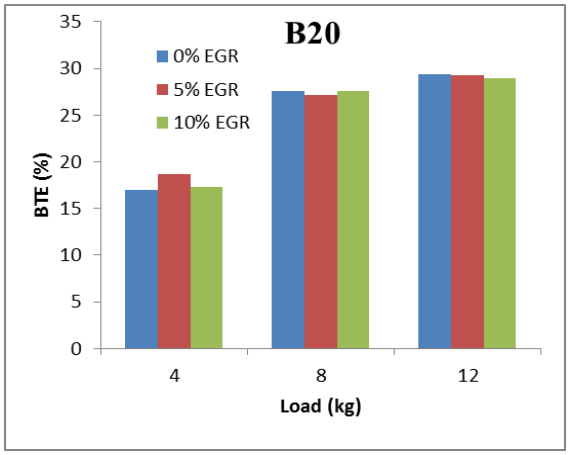
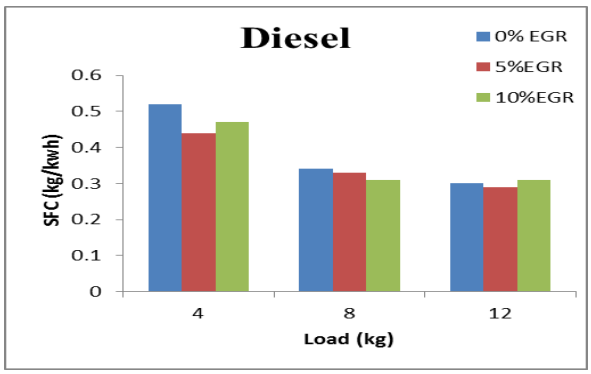
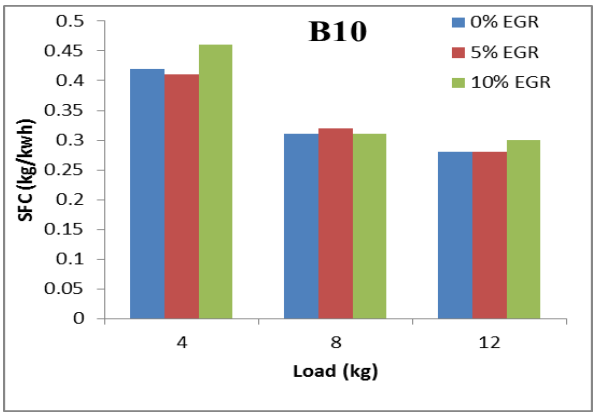


Fig.7 Variation of Brake thermal efficiency with load

Fig.7 indicates that efficiency slightly increases at lower load with EGR but it will not affects significantly at higher load this is because at lower load exhaust gas contains higher amount of oxygen hence when exhaust gas recirculated to cylinder unburned hydrocarbons in exhaust gas will get sufficient oxygen for burning but at higher loads due to less oxygen concentration re-burning of unburned hydrocarbon is not possible. Maximum brake thermal efficiency found 30.17 % at full load for B10 blend with 5% of EGR. Maximum increment in BTE is 4.45% is also found at same blend and EGR at lower load.

5.2.2 Specific Fuel Consumption (SFC)





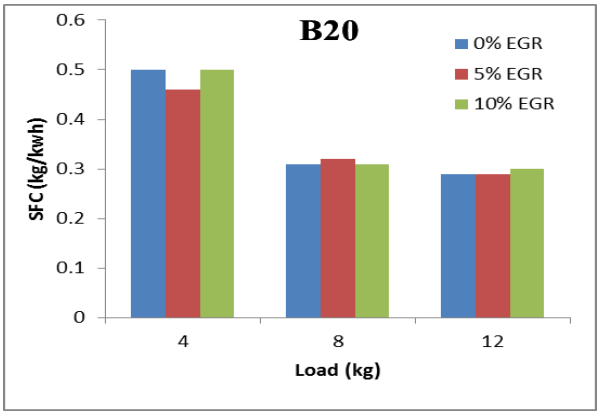
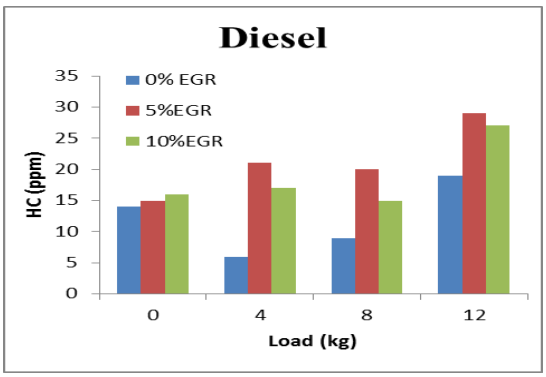


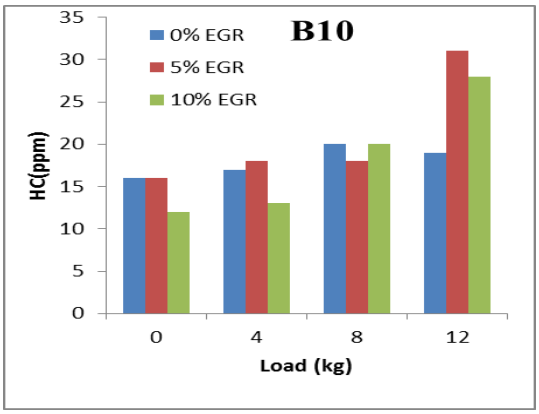
Fig.8 Variation Specific Fuel Consumption with load

Fig.8 represents variation of SFC with load at different blends with and without EGR. The results show that the SFC decreases with increasing load. SFC fuel consumption is reduced with application of EGR at lower loads but at higher load there is no considerable change in SFC. It is also seen that if blend percentage increases the effect of EGR on SFC decreases. Maximum reduction in SFC 15.39 % has been found at low load for diesel with 5 % EGR as compared to without EGR.

5.3Effect on engine emission

5.3.1Unburned Hydrocarbon emission





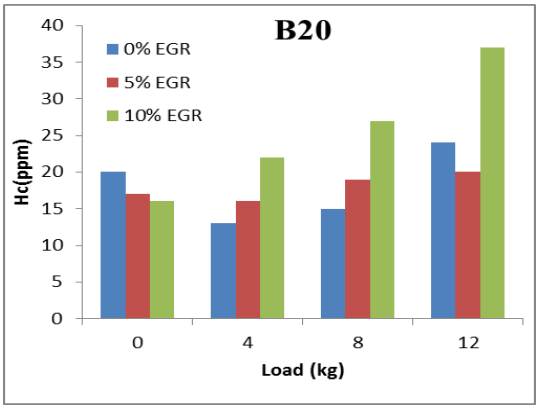
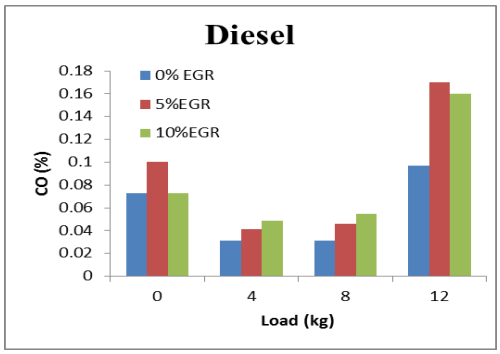
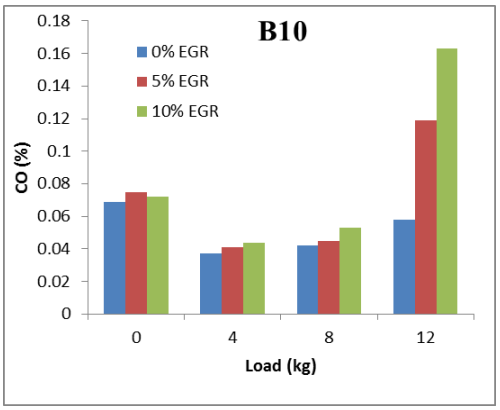


Fig.9 Variation of Unburned Hydrocarbon emissions with load

Effect of EGR on unburned hydrocarbon emission is represented in Fig.9. It indicates that HC emission increases as load increases. It is also observed that HC emission are higher with EGR as compared to without EGR for all at high load but at low load there is no significant effect of EGR on HC emission. This may be due to lower amount oxygen in re-circulated exhaust gas at higher load which causes incomplete combustion as explained earlier.

5.3.2 Carbon monoxide emission





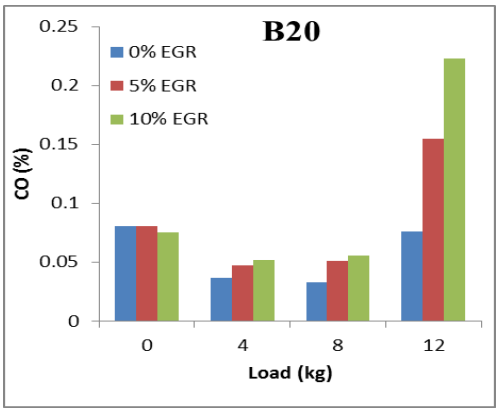
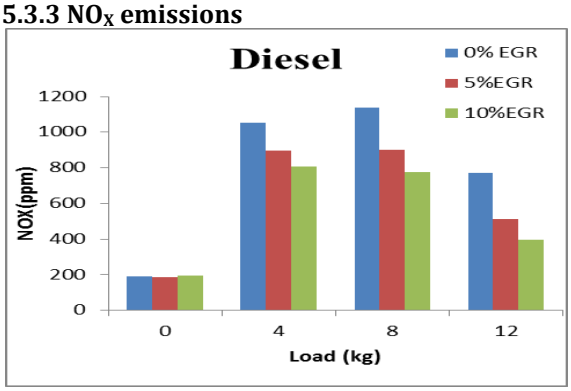
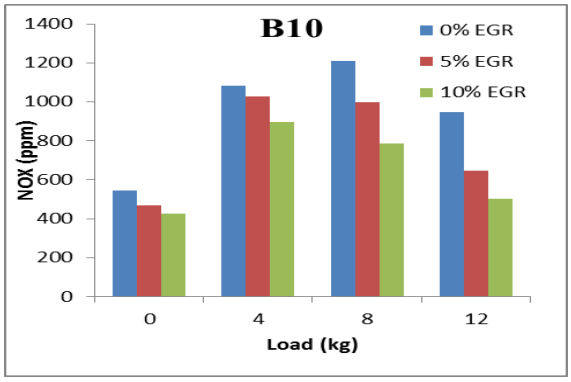


Fig.10 Variation of Carbon monoxide emission with load

From Fig.10 It is observed that at high load CO emissions are higher with the effect of EGR. Less oxygen concentration in exhaust gas at higher load cause incomplete combustion and results in higher CO emission. At full load condition with 10% EGR CO emission increases by 0.087 %, 0.09 % and 0.14 % for diesel, B10 and B20 respectively.

5.3.3 NO_X emissions





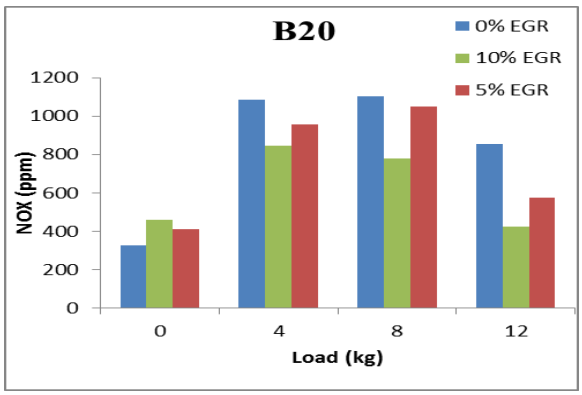


Fig.11 Variation NO_X emissions with load

Fig.11 shows effect of EGR on reduction in NO_X emission which is main advantage of EGR. It has been seen that NO_X emission increases with increment in load and blend percent of biodiesel. But NO_X emission decreases with EGR for all blends. EGR reduces the NO_X emissions by decreasing combustion temperature and lowering O_2 concentration of intake air. NO_X reduction due to EGR is higher at high load but as load decrease reduction in NO_X emission decreases because high oxygen concentration in exhaust gas at low load. With application of EGR NO_X emission was reduced approximately by 100 ppm for low loads and 400 ppm at full load condition.

6. CONCLSIONS

Based on the above outcomes, following conclusions can be made.

1. The brake thermal efficiency increases slightly at low load with EGR. When B10 and 5 % EGR used at partial load, the maximum increment in brake thermal efficiency 4.45% is observed.
2. With application of EGR at lower loads SFC is reduced but at higher load there is no considerable change in SFC. When diesel with 5% EGR is used at lower load.
3. SFC is reduced by 15.39 % HC and CO emission is higher for blends and EGR.
4. CO emission increases considerably with EGR from partial load to full load condition. At full load condition with 10% EGR CO emission increases by 0.087 %, 0.09 % and 0.14 % for diesel, B10 and B20 respectively.
5. EGR reduces the NO_X emission. This reduction is higher at high loads for all blends.
6. With application of EGR NO_X emission was reduced approximately by 400 ppm at full load and 100 ppm for low loads condition.
7. Effect of EGR on performance of the engine increased slightly at low load and remains very close normal at high load. Where on emission is more at high load and less at low load. Hence EGR can be applied at high load to reduce NO_X emission.

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HP31706002-Experimental Investigation on CI Diesel Engine Using Simarouba Biodiesel, Hippe Biodiesel and Al_2O_3 Nano Additive Blended Biodiesel

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Abstract

In this present work experiments were conducted to determine performance, emissions and combustion characteristics of a single cylinder, four stroke VCR diesel engine using pure diesel , simarouba oil methyl ester(SOME) biodiesel , Hippe oil methyl ester (HpOME) biodiesel, and aluminium oxide nanoparticles were added to HpOME-20 as an additive in mass fractions of 25 ppm,50 ppm and 75 ppm with the help of a mechanical Homogenizer and ultrasonicator with cetyltrimethyl ammonium bromide (CTAB) as the cationic surfactant. It was observed from results that HpOME(B20) biodiesel at 3.5kw Brake power(BP) gives 4.7% more BTE, 3.33% reduction in BSFC, 0.437% increase in Volumetric efficiency, reduced hydro carbon (HC) emissions(7.69%),reduction in Carbon Monoxide (Co) with slightly increased in Nox emissions in comparison with SOME (B20) biodiesel. Further experiments were conducted using different aluminium oxide nanoparticles ANP-blended biodiesel fuel (HpOME20 + ANP25, HpOME20 + ANP50 and HpOME20 + ANP75) and the results obtained were compared with those of pure diesel and Hippe oil methyl ester(HpOME20). The results shows a substantial enhancement in the brake thermal efficiency and a marginal reduction in the harmful pollutants (such as CO, HC and sNox) for the nanoparticles blended biodiesel.

Keywords: *Aluminium oxide nanoparticle(ANP), Hippe oil methyl ester (HpOME), Simurouba oil methyl ester (SOME), Transesterification, Combustion, Emission, Mechanical Homogenizer.*

1. Introduction

Over a hundred years ago Rudolf Diesel, the inventor of the C.I engines that still bear his name, demonstrated at a World Fair that agriculturally produced seed oil (peanut oil) may be used as fuel . The use of these agriculturally derivative oils as a fuel was phased out by petroleum-based diesel fuels that became more widely available because they are cheaper in price as a result of government subsidies in the 1920's . In the present scenario with the depletion of the petroleum-based diesel, the demand for alternatives to petroleum-based fuels continues to increase. The increase in the awareness of these alternative biofuels is not only because of the depletion of fossil fuels, but also because these bio-energy resources have lower emissions than conventional fuels and more over they are made from renewable resources. Biofuels refer to any kind of fuel generated which is made mostly from biomass or biological material collected from living or recently living resources. Transportation sectors have shown particular interest in biofuels because of the potential for rural development. In a country like India where it is observed that biodiesel can be a viable alternative automotive fuel. Biodiesel is a fastest growing alternative fuel and India has better resources for its production.Owing to the depletion of the fossil fuels day by day, there is anecessity to find out an alternative resolution to fulfil theenergy requirement of the world. Petroleum fuels play an vitalrole in the fields of transportation, industrial development and agriculture[1,2]. Fossil fuels are fast depleting because of increased fuel consumption. Steady with the estimation ofthe International Energy Agency, by 2025 global energy utilization will increase by about 42% [3]. Many research works are going on to substitute the diesel fuel with an appropriate alternative fuel such as biodiesel. Biodiesel is one of the best available sources to fulfil the energy requirement of the world [4]. Nonedible sources such as cotton seed oil, pongamia oil, Mahua oil, Jatropha oil, and Karanja oil have been investigated for biodiesel fuel production [5]. In the modern years, severe efforts have been made by many researchers to use various sources of energy as feed in existing diesel engines. The make use of straight vegetable oils (SVO) is inadequate due to some unfavorable physical and chemical properties, particularly their viscosity and density. Because of higher viscosity, SVO causes incomplete combustion, poor fuel atomization and carbon deposition on the valve and injector seats ensuing in severe engine

problems. When diesel engines are fuelled with straight vegetable oil as fuel, it leads to incomplete combustion. The potential methods to overcome the problem of higherviscosity were blending of vegetable oil with diesel fuel in the proper proportions and transesterification of vegetable oils to produce biodiesel [6–8]. The transesterification process has been established worldwide as a successful means for biodiesel production and viscosity reduction of vegetable oils [9].Transesterification is the process, by means of an alcohol (e.g. either ethanol or methanol) in the presence of catalyst to break the triglyceride molecules of the raw vegetable oil into ethyl or methyl esters (fatty acid alkyl esters) of the vegetable oil with glycerol as a by-product [10]. Ethanol is one among the chemical preferred for transesterification process when compared to methanol because it is derived from renewable sources (agricultural waste) and is biologically non-harmful for the environment. Mechanism of transesterification process is shown in Fig. 1. In general, methyl esters of vegetable oil propose the reduction of harmful exhaust emissions from the diesel engine such as CO, HC and smoke but it increased the NOx emissions [11–17]. The NOx emission is the most dangerous parameter that has an effect on the environment through acid rain, human diseases, etc. Furthermore, CO and NO are primary pollutants in the formation of atmospheric ozone, which is an important greenhouse gas [18,19] . Many researchers have found that the B20 biodiesel blend gives greater thermal efficiency and emission parameters compared with other biodiesel blends [22]. Among the different techniques accessible to reduce exhaust emissions from the diesel engine while using biodiesel, the use of fuel-borne metal catalyst is presently focused because of the advantage of an enhancement in fuel efficiency while reducing harmful exhaust emissions and healththreatening chemicals [23]. Aluminium oxide nanoparticles at high temperatures dissociate into Al2O and oxygen:

$$Al_2O_3 \longrightarrow Al_2O + 2O$$

Al2O3 is unstable at high temperatures during combustion

in the combustion chamber, so it also decomposes as follows:

$$Al_2O \longrightarrow 2Al + 2O$$

Many researchers found that the combustion behaviour of methyl esters with the addition of nanosize energetic materials as an additive improves the combustion and engine performance of diesel engines. In addition, due to the small size of nanoparticles, the stability of fuel suspensions should be noticeably improved [24–26].In this investigation, aluminiumoxide nanoparticles were added in different proportions (25, 50and 75 ppm) to a biodiesel blend (HpOME20) which is found to be better than SOME20 to investigatethe performance, emission and combustion of the single cylinder, four stroke VCR diesel engine without any Modification.

2. Biodiesel production

Simarouba oil and Hippe oil which is also known as Mahua oil is heated to a temperature of 100–120 ^{0} C to remove water contents present in raw vegetable oil which is followed by filtration. The raw vegetable oil is processed by one base-catalysedtransesterification method where it is mixed with 200 ml of methanol and 7 g of potassium hydroxide (KOH) pellets per litre of vegetable oil and placed on a hot plate magnetic with stirring arrangement for 1–1.5 h up to 60 ^{0}C and then it is allowed to settle down for about 6–8 h to obtain biodiesel and glycerol. The biodiesel obtained is further washed with distilled water two to three times for the removal of acids and heated above 100 ^{0}C to remove the moisture present in the biodiesel.

2.1Process of extracting Biodiesel

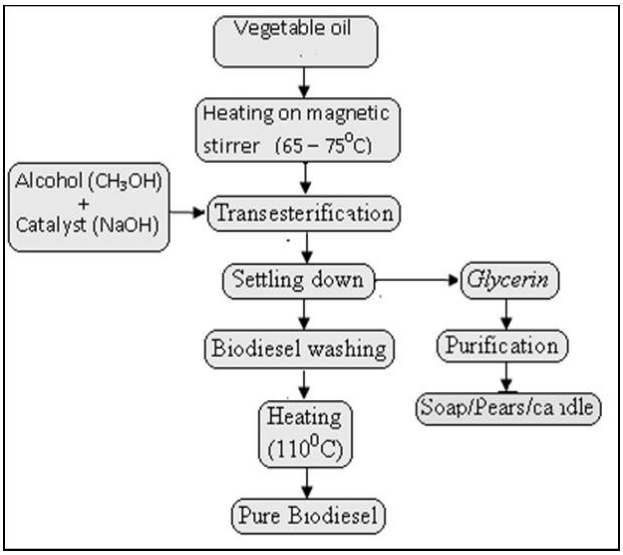


Fig.1 Flow chart ofBiodiesel production.

2.2Transesterification Process

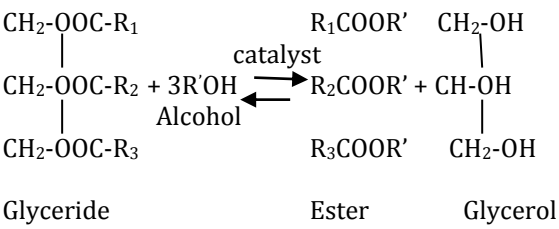


Fig.2 Transesterification of triglycerides with alcohol in presence of catalyst, where R isthe alkyl group.

3. BLEND PREPARATION

3.1. Biodiesel blends preparation.

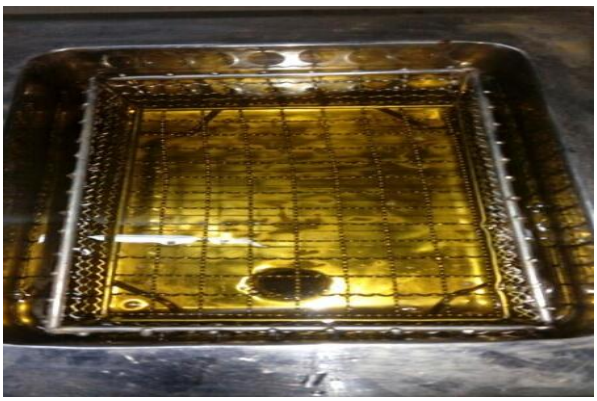
The Simarouba oil methyl ester biodiesel obtained by transesterification is blended with neat diesel on volume basis as per the requirement to form blends like SOME10,also called as B10(10 % SOME + 90 % Diesel),B20(20% SOME +80 % Diesel),B30 and B40. Similarly blends of Hippe oil methyl ester biodiesel are

prepaired (HpOME10,HpOME20,HpOME30 and HpOME40).

3.2. Nanofluid preparation.

The unique hybrid biodiesel-nanoparticle blends for the present study were prepared by using with the help of homogenizer and ultrasonicator.Aluninium oxide is selected as nanoparticle additives to HpOME20 because of their improved properties like higher thermal conductivity, mechanical and magnetic properties.The mean size of the nanoparticles varies from 32 to 48 nm. The nanoparticles were diffused in the solvent with the help of a homogenizer. Nanoparticles usually have a high surface contact area and therefore surface energy will be high. Nanoparticles clustered together to form a micro molecule and start to sediment. To make nanoparticles be steady in a base fluid, it should need to surface modification. Cetyltrimethyl ammonium bromide (CTAB) is a cationic surfactant and it creates an envelope on the surface of the nanoparticles and makes the surface as a negative charge. Hence the particle sedimentation was controlled. In order to disperse the nanoparticle to the base, the magnetic stirrer procedure was followed. A known quantity of aluminium oxide nanoparticles (25, 50 and 75 ppm) and CTAB (100 ml for 1Lt)was weighed and poured in the ethanol solvent and magnetically stirred for 2 h. Then it forms an even nanofluid.

3(a)



3 (b)

Fig.3 (a) & (b) Ultrasonicator setup.

3.3. Hippe oil methyl ester-nanofluid blend preparation

The aluminium oxide nanofluid was added to the Hippe oilmethyl ester blend (HpOME20) in three different proportions (25, 50 and 100 ppm). After the addition of aluminium oxide nanofluid, it is shaken well. And then it is poured into signification apparatus where it is agitated for about 30–45 min in an ultrasonic shaker, making a uniform HpOME20-ANP blend. The properties of Simuroba oil methyl ester (SOME) , Hippe oil methyl ester and ANPs blended Hippe oil methyl ester blend are determined as per ASTM standards and is listed below.

Table.1 Properties of Diesel, biodiesel and biodiesel nanoparticles blended samples.

Description	Viscosity @40^{0}c(Cst)	Density (kg/m$^{3)}$	Calorific value(MJ/kg)	Flash point (^{0}C)
Diesel	3.4	830	42.86	63
SOME 10	2.68	827	42.46	60
SOME 20	2.83	831	41.45	67
SOME 30	3.07	836	40.90	70
SOME 40	3.92	840	39.46	75
HpOME 10	3.52	831	42.43	74
HpOME 20	3.59	833	41.62	76
HpOME 30	3.79	836	41.78	77
HpOME 40	3.84	840	41.25	77.5
HpOME20 +ANP 25	3.42	825.75	41.65	74
HpOME20 +ANP 50	3.37	827.5	41.66	71
HpOME20 +ANP 75	3.34	828	41.67	68

4..Experimental set up

The experiments were conducted on fully computerized Kirloskar TV1, four stroke, single cylinder ,water cooled diesel engine. The rated power of the diesel engine was 3.7 kW. The engine was operated at a constant speed of 1500 rpm by maintaining the injection pressure from 210 to 220 bar at various load conditions. The engine was at the start fuelled with neat diesel to provide the baseline data, and then it was fuelled with SOME, HpOME biodiesel blends and then HpOME20, HpOME20 + ANP25 , HpOME20 + ANP50 and
HpOME20 + ANP75. Details of the engine specification are given in Table 2. Eddy current dynamometer was used for loading the engine. HG-540 AIRREX

(approved by ARAI) five-gas analyzer was used to measure HC, CO and NOx emissions.
In-cylinder pressure and heat release rate were measured by using data acquisition system interfaced with dual core processor. The experimental set-up is indicated in Fig. 4.

Table.2. Engine Specification**.**

Sl.No.	Parameter	Specification
1	Engine Supplier	Apex innovations Pvt. Ltd Sangli, Maharashtra, India
2	Type	TV1 (Kirloskar made) VCR
3	Software used	Engine soft
4	Nozzle Opening Pressure	200-205 bar
5	Governor type	Mechanical centrifugal
6	No. of cylinder	Single cylinder
7	No. of strokes	Four stroke
8	Fuel	H.S Diesel
9	Rated power	3.7 kw (5HP)
10	Cylinder diameter (bore dia)	87.5 mm
11	Stroke length	110 mm
12	Compression ratio	17.5
13	Speed	1500 rpm
14	Arm length	180 mm

Fig.4. Experimental Setup

5. Results and Discussions

The operation of diesel engine using Simuroba oil methyl ester, Hippe oil methyl ester blends and ANPs added Hippe oil methyl ester fuel blends was found to be very smooth throughout the rated load, without any operational trouble.
The performance characteristics such as brake specific fuel consumption (BSFC), brake thermal efficiency (BTE), volumetric efficiency and the emission characteristics such as NOx, HC, CO are plotted against the brake power. Based on the combustion data, cylinder pressure and heat release rate are plotted against crank angle.
Initially experiments were conducted with neat diesel, SOME10,SOME20,SOME30 and SOME40,and Simillary for HpOME10, HpOME20, HpOME30 and HpOME40. In both the cases blend B20 (SOME20 and HpOME20) found better than their counter partners.

5.1 Engine performance

5.1.1Brake thermal efficiency

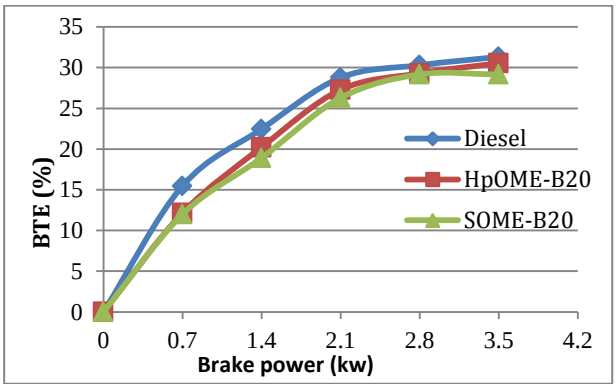


Fig. 5(a)BTE (%) against Bp (kw) for Diesel,SOME20 and HpOME20

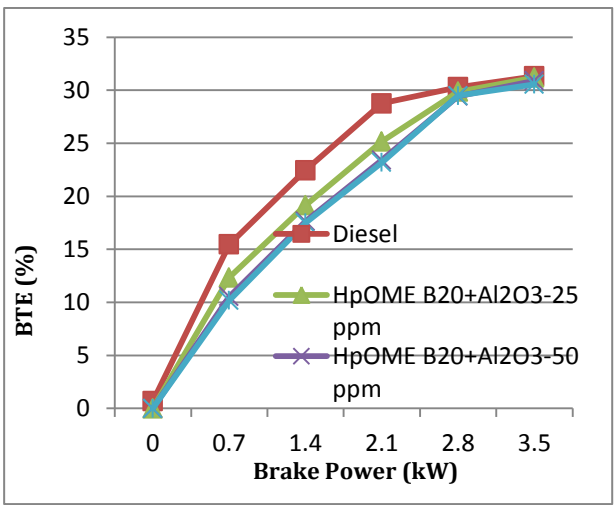


Fig. 5(b)BTE(%) against Bp (kw) for Diesel, HpOME20 +ANP25,HpOME20+ANP50 and HpOME20 + ANP75.

From Fig. 5(a), it can be observed that the brake thermal efficiency (BTE) increases with the load for diesel, SOME and HpOME biodiesel. The BTE of HpOME20 (30.54 %) is better than that of SOME20 (29.16 %) at full load.From Fig. 5(b),it is observed that BTE is further improved with addition of ANP25 PPM to HpOME20 (31.2 %) this could be attributed to the better combustion characteristics of ANP. The catalytic activity of ANP might have improved because of the existence of high active surfaces.

5.1.2 Brake Specific fuel consumption (BSFC)

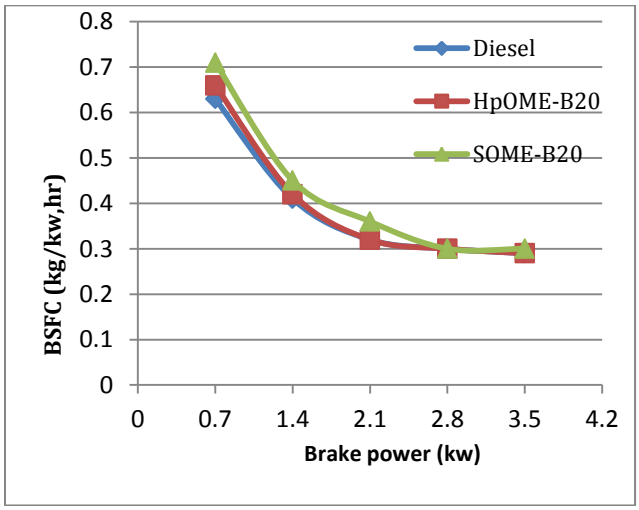


Fig. 6(a) BSFC against BP for Diesel,HpOME20 and SOME20 biodiesel.

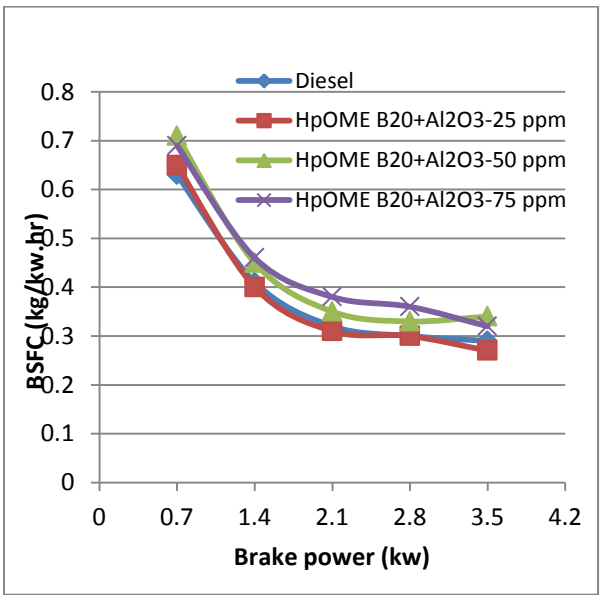


Fig. 6(b) BSFC against BP for Diesel, HpOME20 + ANP 25,HpOME20 + ANP 50 and HpOME20 + ANP 75

It is observed from the above Fig. 6(a) the BSFC for HpOME20 is less than SOME20 and same as that of Diesel at full load condition.

It is observed from Fig. 6(b) that with addition of ANP to HpOME20 there is marginal decrease in BSFC. For ANP25 BSFC is less compared to ANP50 and ANP75 at full condition.

5.1.3Volumetric efficiency

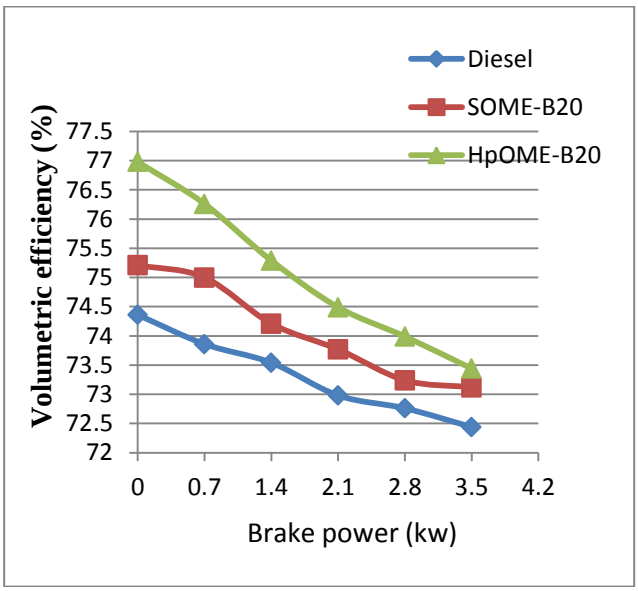


Fig. 7(a) Vol.efficiency against BP for Diesel,SOME20 and HpOME20 biodiesel.

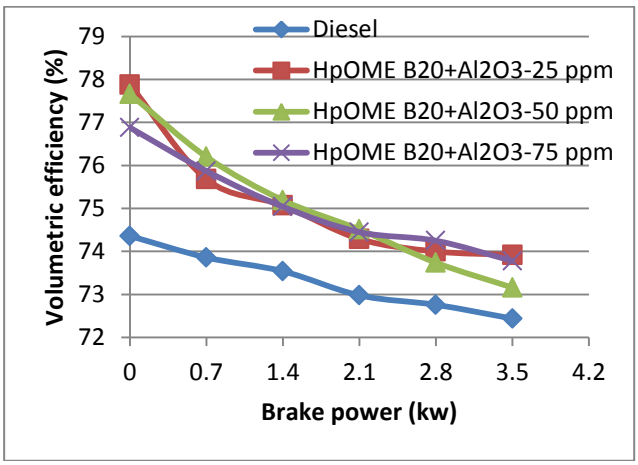


Fig. 7(b) Vol.efficiency against BP for Diesel,HpOME20 + ANP25,HpOME20 + ANP50 and HpOME20 +ANP75 ppm.

It is observed from Fig. 7(a) and Fig. 7(b) that the Volumetric efficiency for HpOME20 is more than SOME20 and also HpOME20 + ANP25 ppm has efficiency compared to other blends.

5.2Emission parameters

5.2.1Hydrocarbon (HC)

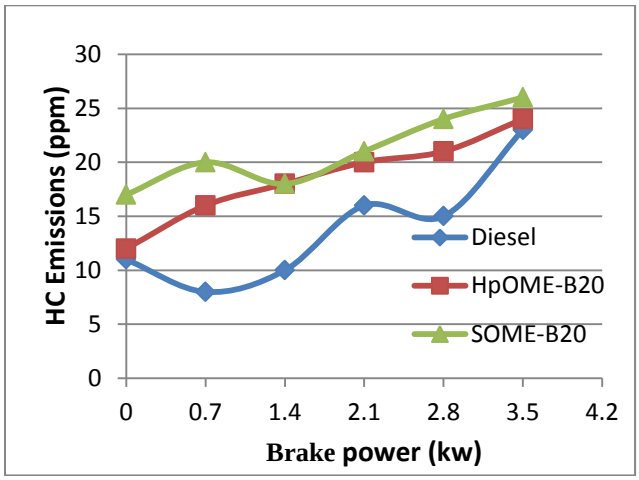


Fig. 8(a) HC emissions against BP for Diesel,SOME20 and HpOME20 biodiesel.

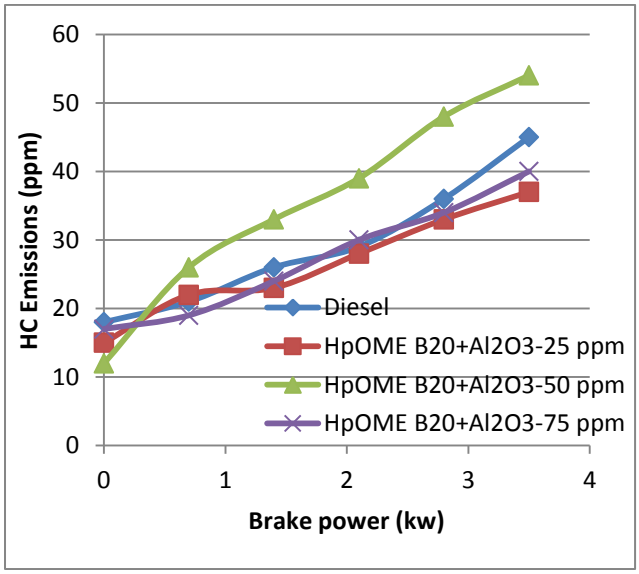


Fig. 8(b) HC emissions against BP for Diesel,HpOME20 + ANP25, HpOME20 + ANP50 and HpOME20 + ANP75 ppm blended biodiesel.

It is observed from Fig. 8(a) and Fig. 8(b) that as the load increases the HC emissions steadily increases for all cases. Many authors' results show a significant reduction in HCemissions when replacing diesel fuel with biodiesel [17,31]. The higher cetane number of biodiesel blend (HpOME20) reduces the combustion delay period and the reduction has been connected to decreases in the HC emissions. Further addition of aluminium oxide nanoparticles reduces the hydrocarbon emissions, because ANP supplies the oxygen for the oxidation of hydrocarbon and CO during combustion.

5.2.2 Carbon monoxide (CO)

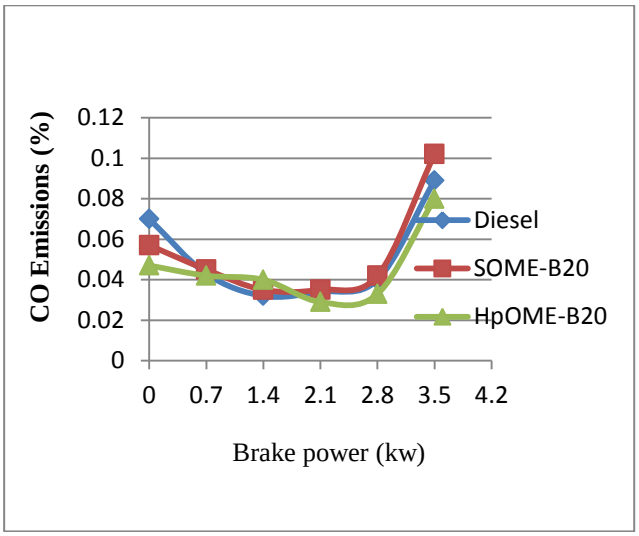


Fig. 9(a) CO emissions against BP for Diesel, SOME20 and HpOME20

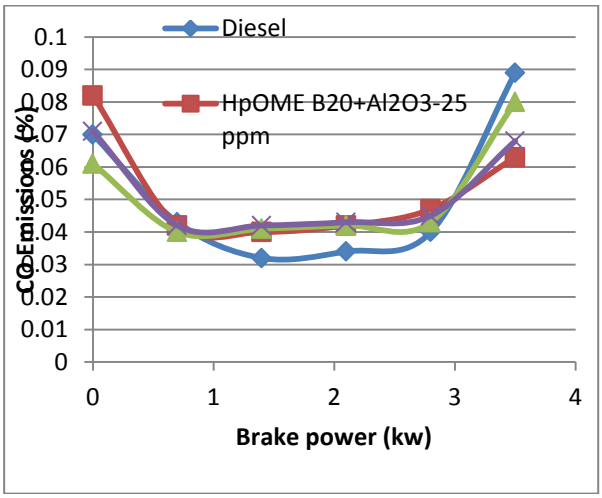


Fig. 9(b) CO emissions against BP for Diesel, HpOME20 + ANP25, HpOME20 + ANP50 and HpOME20 + ANP75 ppm blended biodiesel.

It is observed from from Fig. 9(a) the CO emissions of HpOME20 is less than SOME20 and is almost same as diesel at full load. The effects of ANP with a biodiesel blend (HpOME20) on thecarbon monoxide emission at various engine loads have beenshown in Fig. 9(b) ANPs have high surface contact areas which raise the chemical reactivity which consecutively shortened the ignition delay period. From Fig. 9(b) it shown that CO emission found marginally less than diesel for HpOME20 + ANP25 ppm.

5.2.3 Oxides of nitrogen (NOx)

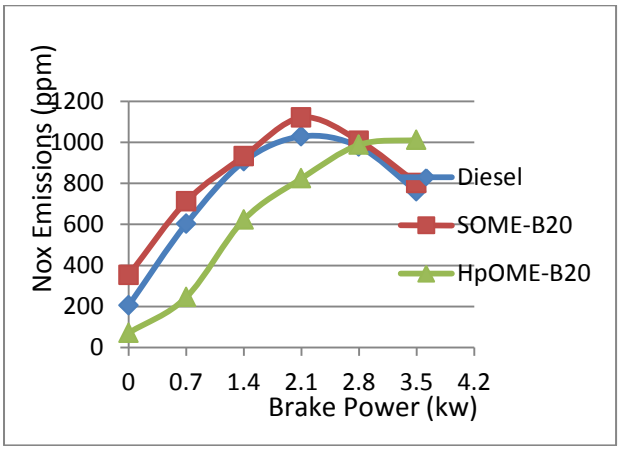


Fig. 10(a) No_X emissions against BP for Diesel, SOME20 and HpOME20

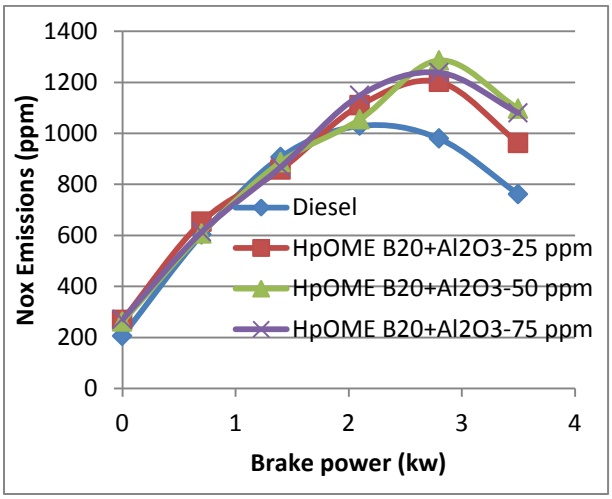


Fig. 10(b) No_X emission against BP for Diesel, HpOME20 + ANP25, HpOME20 + ANP50 and HpOME20+ANP75 ppm.

The reduced ignition delay and combustion timing are obtained when using the biodiesel blend. It is widely accepted that the shorter ignition delay may contribute to slightly increased NOx emissions with biodiesel blend (HpOME20). The addition of nano metal oxide particles leads to complete combustion because of the aluminium oxide nanoparticles acting as an oxygen-donating catalyst. NOx emissions increased for ANP blend due to maximum heat release rate and high peak pressure during the combustion. NOx emissions of the engine at different nanoparticle concentrations with a biodiesel blend with engine loads are shown in Fig. 10(b). From the figure, it is clear that the NOx emission noticeably increases by means of aluminium oxide nanoparticle additives.

5.3Combustion characteristics

5.3.1 Cylinder pressure

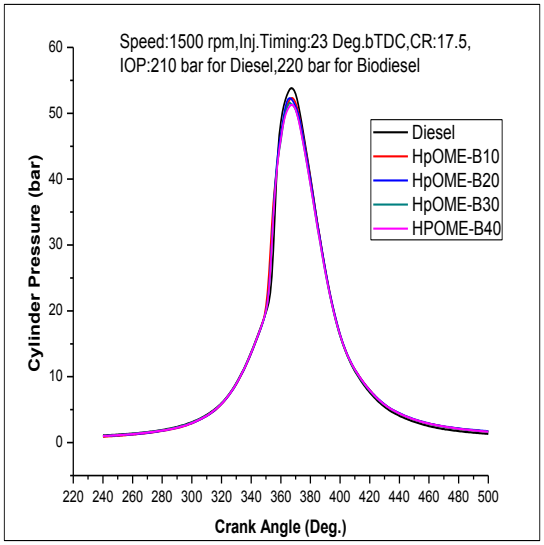


Fig.11(a) Cylinder pressure against crank Angle for Diesel and HpOME biodiesel.

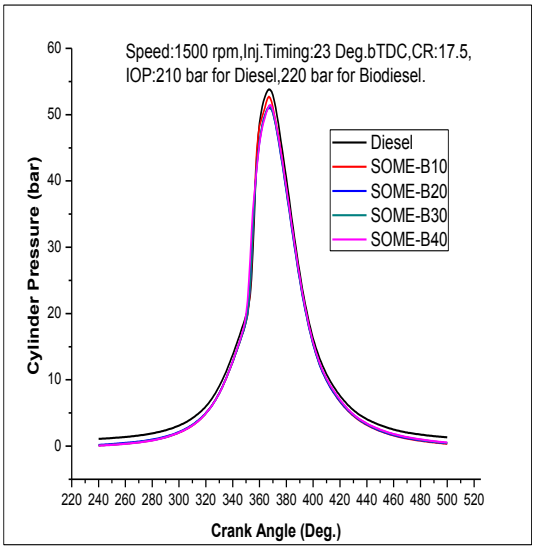


Fig.11(b) Cylinder pressure against crank Angle for Diesel and SOME biodiesel.

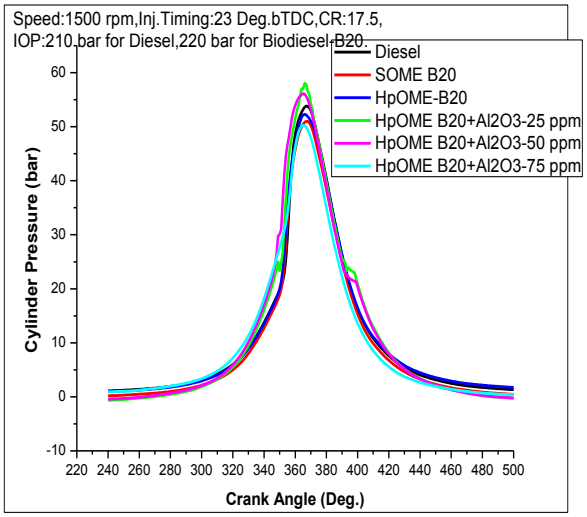


Fig. 11(c) Cylinder pressure against crank Angle for Diesel and HpOME20 with different blends of ANP(25, 50 and 75 ppm) biodiesel.

Fig. shows the variation of the in-cylinder pressure of the
engine with crank angle. From the figure 11(a), it is seen that the maximum pressure at 366° crank angle is 52.18 bar is obtained for HpOME20.
From the figure 11(b), it is seen that the maximum pressure at 366° crank angle is 51 bar is obtained for SOME20.
From the figure 11(c), it is seen that the maximum pressure at 366° crank angle is 58.0275 bar is obtained for HpOME20 + ANP25 ppm.

5.3.2 Heat release rate

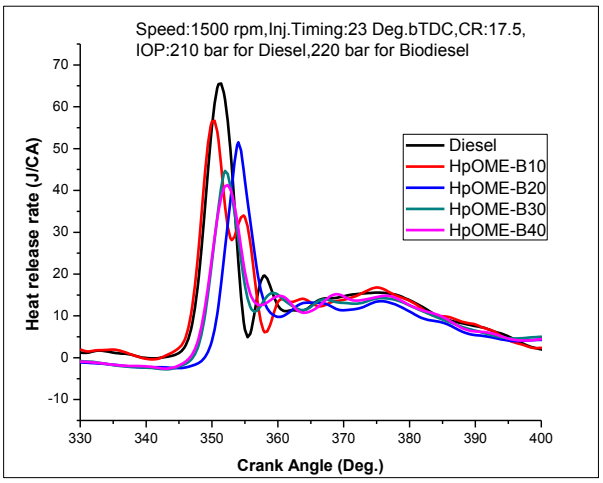


Fig. 12(a) Heat release rate against crank Angle for Diesel and HpOME biodiesel.

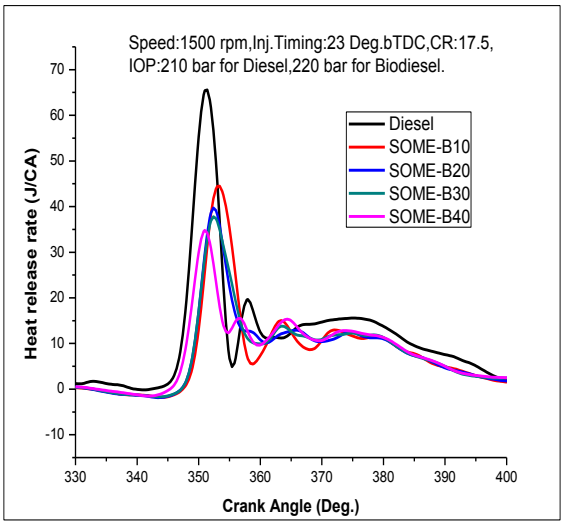


Fig. 12(b) Heat release rate against crank Angle for Diesel and SOME biodiesel.

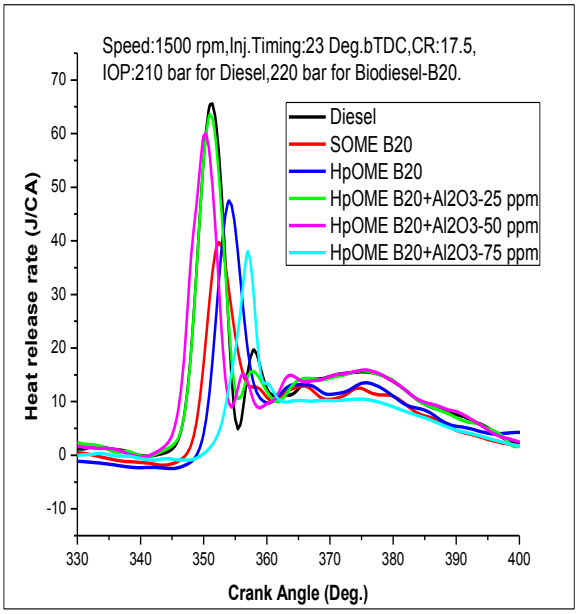


Fig. 12(c) Heat release rate against crank Angle for Diesel and HpOME20 + ANP(25,50 and 75 ppm) blends biodiesel.

Fig. 12(a, b and c) shows the heat release rate for diesel, Hippe oil methyl ester, Simarouba oil methyl ester biodiesel and HpOME20 + ANP(25, 50 and 75 ppm) blended biodiesel.It is seen that heat release rate is maximum for neat diesel this is due to high CV.
The heat release rate curves indicate the available heat energy, which can be rehabilitated into useful work. Hippe Oil methyl ester (HpOME) includes a small amount of diglycerides having high boiling points compared with the diesel fuel. The chemical reactions during the injection of the Hippe oil methyl ester blend (HpOME20) at very high temperature resulted in the breaking of the diglycerides. These chemical reactions produce the gases of monoglycerides. Gasification of

these monoglycerides in the tassel of the spray spreads out the fuel jet, and the volatile combustion compounds present in the fuel are ignited in advance and reduced the ignition delay period. The heat release rate in the combustion chamber during the starting of combustion is not enough to entirely combust the fatty acids.. The addition of ANP considerably increases the heat release rate of HPOME20

6. Conclusions

In this present investigation work, ANP-mixed Hippe oil methyl ester blend fuelled diesel engine performance, emission and combustion characteristics were studied, and based on the experiments the following conclusions were drawn:

- ANP-blended biodiesel (HpOME20+ANP25 HpOME20+ ANP50 and HpOME20+ ANP75) showed an improvement in the calorific value and a reduction in the flash point compared to HpOME20.
- Biodiesel has higher fuel consumption, because of its inferior heating value. With the addition of aluminium oxide nanoparticles, there is a considerable reduction in fuel consumption compared to biodiesel operation and for HpOME20+ANP25 ppm BSFC is found to be 6.896% reduction to Diesel.
- A minor reduction in BTE was observed in all the cases with minimum of 0.38 % (HpOME20 + ANP 25 ppm) w.r.t diesel.
- ANP reduced HC and CO emissions up to 17.77% and 29.21% compared with a biodiesel blend (HpOME20+ANP25), because ANP acts as an oxygen buffer catalyst and donates surface lattice oxygen for the oxidation of HC and CO. NOx emissions increase with the use of ANP and biodiesel blend compared to the diesel fuel.
- The peak pressure increases with the addition of ANP. The addition of ANP reduces the ignition delay period. The heat release rate also increases with the addition of ANP. The addition of ANP accelerates the hydrocarbon combustion and is the reason for the higher heat release rate when compared with neat diesel and biodiesel blend (HpOME20 and SOME2).

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HP31706006-Analysis on Performance Characteristics and Emissions of Diesel Engine using different Blends of Calophyllum Inophyllum, Cotton Seed Oil, Karanja.

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Abstract

Ecological concernment and energy extremity of the planet has led to the quest of feasible alternatives to the non-renewable fuel source, FAME (Fatty Acid Methyl Ester) is biodegradable, ecological, alternative, and safe, environmental friendly which has a high flash point and is also termed as Bio-Diesel. In upcoming years, in most of the regions of the world production and application of biodiesel has extrinsic fame. It is usually produced by the method trans-esterification. In this experiment, biodiesel from Calophyllum Inophyllum oil, Karanja Oil Methyl Ester and Cottonseed Oil has been produced using the trans-esterification process. Engine trail have been executed in water cooled, 4- stroke diesel engine. Investigational analysis has been conducted to study the performance and emission on different biodiesel blends of Cottonseed Oil, Karanja Oil Methyl Ester and Calophyllum Inophyllum oil for unequal injection pressures. From the evaluation of obtained results, it can be deduced that the engine operation process is considerably become better with noteworthy subdual in emissions of the CO and HC.

Keywords: *Bio fuel, Calophyllum Inophyllum, Cotton Seed Oil, Karanja*

1. Introduction

The demand for energy utilization in automobiles and agricultural segment in India has been expanding along with the economic advancement. India is facing difficulties in regard to the fuel necessity for increased transportation pressure and is importing of about 70 % of its petroleum demand. Domestic consumption of mineral diesel in India accounted for approximately 16% of the total imports valued about INR 3.4lakh crores in 2015-16 (PPAC). This indicates the economic stress on the country due to diesel consumption. Since pricing of petroleum products like Diesel, Domestic LPG and kerosene continues to be regulated and subsidized. The subsidy / under recovery on diesel alone accounted for 57.78% of the under recoveries (173,523 crores) during the span of two years.

2. Why Bio-Fuel

Alternative fuels giving guarantee of sustainable advancement with security of supply and lesser ecological implications, are needed. For transport part confronts extra difficulties sufficient energy density and lower pollutant emission potential because their exhaust items are transmitted straightforwardly into the ambient air, which influences human wellbeing. The execution of major project for advancement of bio diesel in India is possible into following favorable conditions.

5) Biodiesel shows perfect diesel characteristics, with none or minor hardware modifications within the engine it shows a probability of it being utilized as substitute.
6) Utilization of biodiesel is useful in decreasing the greenhouse gas emissions.
7) Biodiesel may be manufactured from regional available feedstock resources. Thus advancement of biodiesel industry would fortify local industrialization.
8) Blending it with diesel, it is able to adjust for the decrease in the lubricity in low sulphur content diesel on the grounds that substance is being diminished in diesel fuels, to make them perfect with EURO- IV or higher measures.

3. Literature Review

We know that in village place, where the electric power is supplied for only 3 hours in a day time and 6-8 hours at night time. This is the main reason, which drives me to overlook the biofuels. At whatever point I consider fuel, an image of villages of developing countries overshadowed in darkness, poverty, no planned mechanization, agriculture depending on rain comes before my eyes. Mainly

these petroleum prices are directly proportional to the inflammation rate. As the price of petroleum products increases the daily needs prices also increases.

C K Reddy the thought of utilizing vegetable oil as fuel has been around from the conception of diesel motor. Rudolph diesel, the inventor of the engine that stands his name, experimented with fuels starting from powdered coal to shelled nut oil.

Auld D L et al. Utilized rapeseed oil to study the impacts of utilizing an option Fuel as a part of diesel motors. An examination of the rapeseed oil demonstrated a relationship in the middle of consistency and unsaturated fat chain length. Motor power and torque results utilizing rapeseed oil were like that of diesel fuel. Consequences of the fleeting tests showed further long haul testing was expected to assess motor durability when oil was utilized.

Goering C E et al. studied the characteristic properties of eleven vegetable oils. To determine which oils would be best suited for use as an alternative fuel source. Of the eleven oils tested, corn, rapeseed, sesame, cottonseed, and soybean oils had the most favorable fuel properties.

4. Methodology

Biodiesel which may be delivered from consumable and non- consumable oils which is derived from vegetables, reused vegetable oils and creature fat (Auld D. L. B. L. Bettis and C. L. Peterson 1982) This bio degradable substances can be converted by using trans esterification process by changing over tri-glycerides into unsaturated fats alkyl-esters which is one of best alternative fuel, which may be utilized as a fractional substitute for fossil based diesel. Trans esterification is a reaction in which tri-glycerides exhibit within the raw material vegetable oils with essential alcohols in presence existence of a catalyst, which delivers essential glycerol and esters.

4.1 Cottonseed Oil Methyl Ester Production

The procedure trans esterification of cottonseed oil was performed by using the catalyst of 5 g potassium hydroxide and 200 ml methyl liquor for each 1L (G. R. Kannan K. Rajasekhar Reddy and Velmathi 2006-2009). To begin with, the cottonseed oil was heated to around 700C in a reactor then; the catalyst was blended with methyl alcohol to disintegrate and added to the heated cottonseed oil in the reactor. After that blend was blended for 1 hour at an temperature of about 700C, it was exchanged to another container and the partition of the glycerol layer was permitted. Once the glycerol layer was settled down, the methyl ester layer framed at the upper piece of the compartment was exchanged to another vessel. Thereafter, a washing procedure was completed to evacuate some un reacted rest of methanol and catalyst utilizing refined water and the blown air. At that point, a distillation procedure at around 1100 C was employed for evacuating water contained in the esterified cottonseed oil. At the last moment the produced cottonseed oil methyl ester was left to cool down (R. Anand 2006-2009)

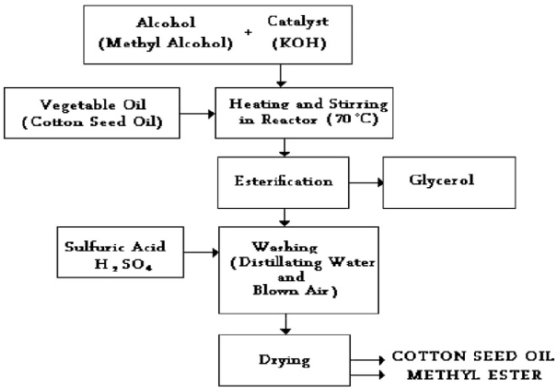

Fig.1 Flow chart of COME

4.2 Calophyllum Inophyllum Production

The oil is initially warmed to 50ºC and then 0.7% (by wt. of oil) sulphuric acid is to be added in oil and methyl alcohol about 1:6 molar proportion (by molar mass of oil) is added. Methyl alcohol is added in excess quantity to accelerate the reaction. This reaction was continuing with stirring at 650 rpm and temperature was controlled at 55-57ºC for 90 min. The fatty ester is isolated after regular cooling (P. L. Naik and D. C. Katpatal 2013). At second level, the isolated oil from the isolating funnel needs to experience trans esterification. Meth- oxide (methanol + sodium hydroxide) is added in the above ester and it is heated to 65°C The same temperature is kept for 2 hr. with connected stirring and after that, it undergoes natural cooling for 8 hr. Glycerol will be stored at the bottom of the flask, and it is isolated out by a separating pipe. The remainders in the flask is the esterified vegetable oil (biodiesel) (S. Patil 2013).

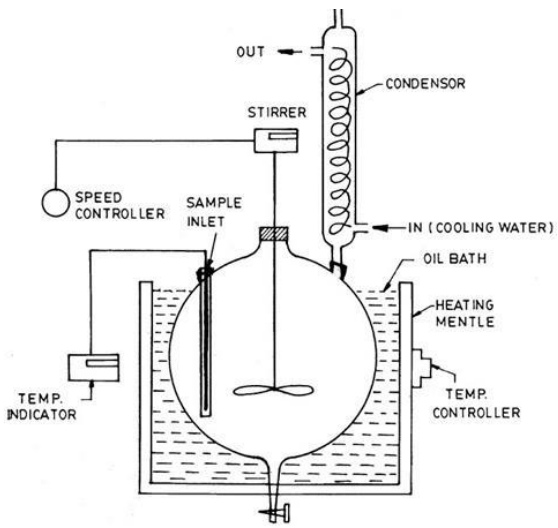

Fig.2 Experimental set up for trans esterification of Honne and Karanja Crude oil

4.3 Karanja Production

The trans esterification procedure is the reaction of triglyceride (fat/oil) with an alcohol in the vicinity of acidic, alkaline or lipase as a catalyst to form mono alkyl ester that is biodiesel and glycerol. However the vicinity of strong acid or base quickens the transformation. It is accounted that alkaline catalyzed trans esterification is quickest and require basic set up hence, in current study the oil of pongamia pinnata were trans esterified with methyl alcohol in presence of strong alkaline catalyst like sodium hydroxide or potassium hydroxide in a batch type trans esterification reactor.

To prepare biodiesel from pongamia crude oil first sodium hydroxide was added in to the methyl alcohol so as to form sodium Methoxide and at the same time oil was heated in a separate vessel of tranesterifaction reactor and it is subjected to heating and stirring. At the point when temperature of oil came to at 60oC then sodium Methoxide was mixed in to the oil and reaction mixture was stirred for one and half hour. After reaction completion, there action mixture was moved in separating funnel. The mixture of glycerol and methyl ester was permitted to settle for 8hours. In the wake of settling for 8 hours glycerol and methyl esters was isolated manually. The methyl ester was the washed with heated water to uproot hints of sodium hydroxide polluting influence. The washed biodiesel then refined to evacuate moisture and final good quality biodiesel was subjected for chemical analysis.

5. Properties of Biofuels

Table 1 Physical-chemical Properties of Biofuels

Properties	Unit	Diesel	Honne Oil	Karanja Oil	Cotton Oil
Density	gm/cc	0.84	0.895	0.865	0.85
Viscosity (at 40 0c)	cst	2.5	4.43	4.78	4.35
Calorific Value	KJ/kg	43,560	39,650	38,540	39,648
Specific Gravity		0.84	0.9	0.925	0.91
Flash Point	0c	52	173	225	207
Fire Point	0c	61	181	236	219

Table 2 Blending percentage of fuel

Notation	Fuel Quantity (Liter)	Bio-Diesel Quantity(ml)			Diesel Quantity (ml)
		Honne	Karanja	Cotton	
O1	1	300	400	-	300
O2	1	100	500	-	400
O3	1	200	600	-	200
O4	1	300	-	400	300
O5	1	100	-	500	400
O6	1	200	-	600	200

6. Experimental Setup

The experiment is completed at constant rated speed, comparing the performance of C.I engine by varying its Injection pressure on Diesel and Using Different Blends of Calophyllum, Inophyllum, Cotton Seed Oil & Karanja. The specimens are arranged by utilizing 1000 ml measuring container and a graduated test tube.

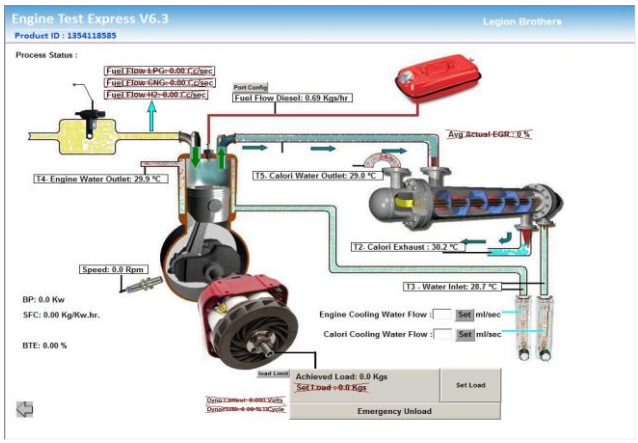

Fig.3 Schematic Diagram of the Experimental Set-up.

7. Result and Discussion

7.1 Brake Thermal Efficiency

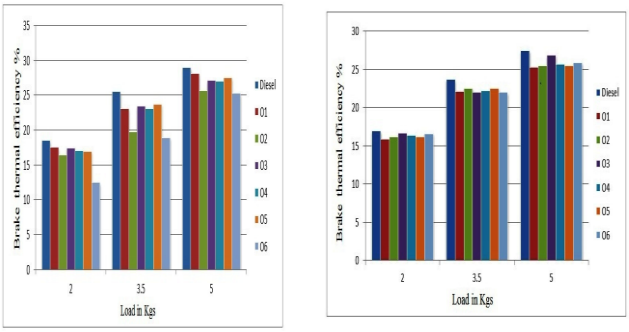

Fig.4 Brake thermal efficiency VS load for Injection Pressure180 bar and 220 bar respectively

Above graph displays the variation of brake thermal efficiency versus load for Injection Pressure 180bar and 220 bar respectively. It is found that the brake thermal efficiency is steadily increased with the

increases in load. The thermal efficiencies of biodiesel fuel blends are reducing with comparison to diesel fuel. This is mainly because of the lower heating value and inferior combustion of the Bio fuels.

7.2 Exhaust Gas Temperature

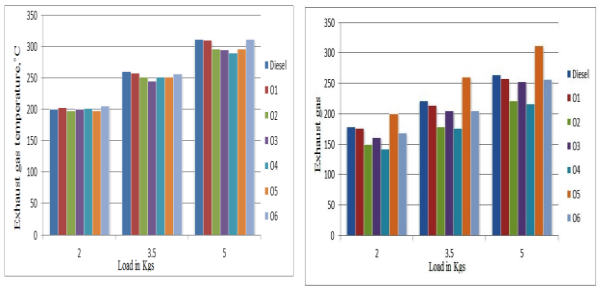

Fig.5 Exhaust gas temperature versus load for Injection Pressure 180bar and 220 bar respectively

Above graph gives the variation of Exhaust gas temperature versus load for Injection Pressure 180bar and 220 bar respectively. The exhaust gas temperature reduces with increase in the bio fuel blend percentage and the values are smaller in comparison to diesel fuel as shown in the graphs. The main reason for lower exhaust gas temperatures for Bio fuels-diesel blends is Lower viscosity. Because lower the viscosity lower the penetration of the fuel into the combustion chamber, which results in the smaller amount of heat is produced.

7.3 Carbon Monoxide Emission

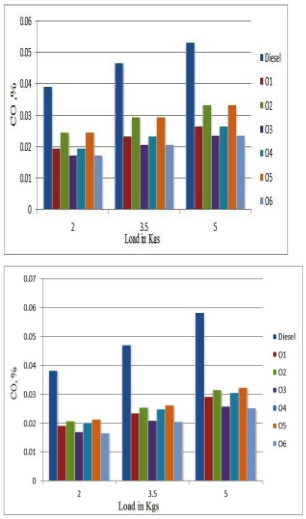

Fig.6 Percentage Carbon Monoxide Vs Load for Injection Pressure 180 bar and 220 bar respectively.

Above graph demonstrates the variations of %CO with load for Injection Pressure 180bar and 220 bar respectively. The emission of carbon monoxide get increases for Injection Pressure of 180bar to the injection Pressure 2200bar .This is because of fuel air mixture fills inside the cylinder is very lean and some

of the mixtures nearer to the wall and crevice volume, the flame will not propagate.

7.4 Hydrocarbon Emission

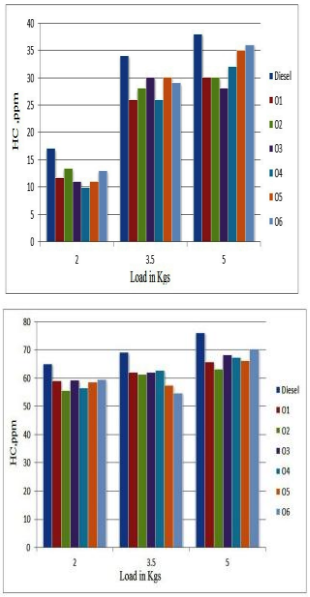

Fig.6 Hydrocarbon VS load for Injection Pressure 180bar and 220 bar respectively.

Above graphs shows the variation of Hydrocarbon opposite of load for Injection Pressure 180bar and 220 bar respectively. The Unburnt hydrocarbon discharge is mainly due to the incomplete combustion. By comparing Injection Pressure 180bar to the emission of hydrocarbon increases for Injection Pressure 220 bars. This is because of two reasons. First one is, the fuel spray doesn't spread deeper in the combustion chamber and second one is gaseous HC's will remain alongside the cylinder wall and the crevice volume and left Unburnt.

Conclusion

The complete study is based on the exhaust emission and engine performance of Cotton seed oil, Karanja, Honne biodiesel were performed. The following conclusions can be made

1) In biofuel blend O2, BTE was dropped by 13 % compared to 100% Diesel fuel at 180 bars.
2) In biofuel blend O5, BTE was dropped by 11.32 % as compared to 100% Diesel fuel at 180 bars.

3) In terms of BTE O5 is somewhat better than O2 at 180 bars.
4) In biofuel blend O2, BTE was dropped by 15.44 % compared to 100% Diesel fuel at 220 bars.
5) In biofuel blend O5, BTE was dropped by 13.25 % as compared to 100% Diesel fuel at the 220 bars.
6) In terms of BTE O5 is better than O2 at 220 bars.
7) It was observed that with the increase in the blend, exhaust temperature get increased.
8) Emission of NOx found to be increased with the increase in the blending percentage.
9) But, CO and HC found decreased with the increase in the percent of bend.

From the end of work we can deduce that the bio fuels namely cotton seed oil, Karanja and Honne can be used as Alternative fuel. The values of BTE and BSFC are nearer to diesel fuel values. As the overall emissions of biodiesel is less than that of diesel, they are more Eco-friendly. As compared to Honne - Karanja biodiesel, Honne – Cotton seed oil biodiesel is preferred because they show better performance characteristics. From the economy point of view Honne – Cotton seed oil biodiesel has less cost compared to Honne - Karanja, as Cotton seed oil biodiesel is produced from waste Cotton seed.

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HP31706602-Enhancing performance Of Internal Combustion Diesel Engine by Oxygen Enriched Air combustion by Membrane technology.

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Abstract

IC engine is the device which is used to produce mechanical energy from fossilfuel like petrol, Dieseletc. Those are different than external combustion engines because of burning of fuel inside the engine & heat release of the same. The air charge before combustion and the burned gases after combustion are actual working fluids. Heat release gives mechanical work as output. In recent years, that the IC engine is responsible for too much pollution, which is dangerous to human health and environment. Thus it is important to take some decision in the path of reducing pollutants without losing power and fuel consumption. The main challenge to control emission to conform the legal law & orders to protect the environment. The concept of oxygen enrichment by using Membrane technology associated at limited increase in the Oxygen in air by membrane to achieve high performance & low emission levels as well. Because of the increased oxygen content, additional fuel is burned. The resulting increase in power output is a beneficial offshoot, though it is not attempted for its own sake. Oxygen-enrichment of combustion air provides an opportunity to achieve ignition with minimum amounts of mixed fuel.

Keywords: *Oxygen Enrichment, Membrane Technology, Oxygen Enriched Combustion.*

1. Introduction

Today's conventional internal combustion engine uses only air as it is necessary for combustion process. Air is mixture of various gases which results in loss of heat energy produced by combustion due to undesired combustion of gases. This results in loss of efficiency of that particular internal combustion engine. Due to low cost of diesel fuel diesel engine are more economical as compared tothe gasoline engines. Diesel engines are widely use in field where both high power and high torque is required. But diesel engines are suffers from inherent higher particulate matter and nitride oxide emissions. Oxygen enriched combustion is one of the attractive combustion technologies to control pollution and improve combustion in diesel engines (P.Baskar et al,2016). Diesel engines are measure contributors of air polluting exhaust gasses such as particulate matter, carbon monoxide, oxides of nitrogen and other harmful compounds. by controlling the flow rate of oxygen and carbon dioxide, the intake air components of the engine are adjusted to optimize the engine performance, save energy and reduce emissions(Bingyuan Han et al,2011).All polluting exhaust gases form due to incomplete combustion of diesel fuel. To avoid pollution and heat loss it is necessary to have complete combustion of fuel which can be accomplished by availing extra oxygen for combustion. Diesel engine manufacturers face major challenges to improve performance characteristics of diesel engine by achieving proper combustion of diesel fuel. To improve performance and lower exhaust emission further one of the least exploited variables has been oxygen concentration in combustion air. Use of oxygen enriched air was compared with different level of oxygen enrichment to evaluate combustion parameters.The oxygen-enriched air improves combustion, promotes the stability and reduces CO and HC obviously, and also results in a little higher NOx. Nevertheless the rise of NOx keeps the lower level in the start process(K.rajkumar et al,2010).Among all the methods, the oxygen enrichment and diesel particulate trap without modifying engine design play a vital role in improvement of Combustion parameters in diesel engine. Use of oxygen enriched air was compared with different level of oxygen enrichment to evaluate combustion parameters.The use of oxygen enriched combustion air will have a direct affect on the combustion process and on the overall engine thermodynamics(Li Shengqin et al,2010). A number of experimental studies have demonstrated benefits of applying Oxygen Enriched Combustion in diesel engine.there is considerable improvement in power output and thermal efficiency of the engine with increased oxygen concentration and injection pressure while NOx emission increased pro rata with the oxygen added(P.Baskar et al,2013). There are different methods for the enrichment of the oxygen such as use of the zeolite, use of the different additives etc. Though many researches are being conducted on diesel engine oxygen enriched air most of them simulated oxygen enrichment by mixing air stream with pure oxygen stored in cylinder. In the present work separate oxygen cylinder was use to enrich oxygen level in intake air the

small mixing chamber was provided before inlet manifold. But providing oxygen which is stored in tank is not economical and efficient way so other approach is by enriching oxygen, so for this polysulfone hollow fiber membranes tested on oxygen/nitrogen mixtures, where the effect of the feed pressure and feed flow rate on the oxygen enrichment by the air separation membrane technology.(M.I.suhaina et al,2014).

2.Experimental Methods

2.1 Experimental Process Flow

This Experimental setup is used to find out this research perspective as shown in figure. Oxygen is added during the intake by membrane technology in the air box for correct mixing measured by O2 analyzer. Flow control valve is used to control the flow of oxygen from membrane at volume fraction from 21% base value to 27 % high value.

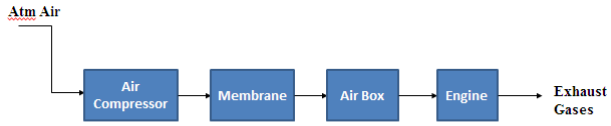

Fig 1 Process flow for experimentation

2.2 Engine Specification

A single Cylinder,four stroke, compression Ignition, Direction injection, vertical diesel engine having following specification as follows in table 1 was used to conduct the experiment.

Table 1 : Engine Specification as follows

Product	Engine test setup 1 cylinder, 4 stroke, Diesel (Computerized)
Product code	224
Engine	Make Kirloskar, Model TV1, Type 1 cylinder, 4 stroke Diesel, water cooled, power 5.2 kW at 1500 rpm, stroke 110 mm, bore 87.5 mm. 661 cc, CR 17.5
Dynamometer	Type eddy current, water cooled, with loading unit
Propeller shaft	With universal joints
Air box	M S fabricated with orifice meter and manometer
Fuel tank	Capacity 15 lit with glass fuel metering column
Calorimeter	Type Pipe in pipe
Piezo sensor	Range 5000 PSI, with low noise cable
Crank angle sensor	Resolution 1 Deg, Speed 5500 RPM with TDC pulse.
Data acquisition device	NI USB-6210, 16-bit, 250kS/s.
Piezo powering unit	Make-Cuadra, Model AX-409.

The oxygen concentration is measured with an Oxygen analyzer fitted between air box and inlet manifold of the engine. The test engine is coupled to Saj Test Plant Pvt. Ltd. (model-AG10) eddy current dynamometer. The main measuring instruments used were; a mass balance with an accuracy of 0.01 g to measure the fuel flow rate, theoretical constant is taken for the calorific value of fuel, a thermocouple to measure the temperature of exhaust gas, inlet air, a TDC marker (a magnetic pickup) and an rpm indicator. A PCB Piezotronics, INC.HSM111A22 and M108A02 electric transducer measures the combu09stion chamber pressure (it is a mean value of 50 consecutive cycles) with an increment of 1° crank angle using an data acquisition system.(NI USB-6210 Bus Powered M Series).

2.3 Experimental procedure

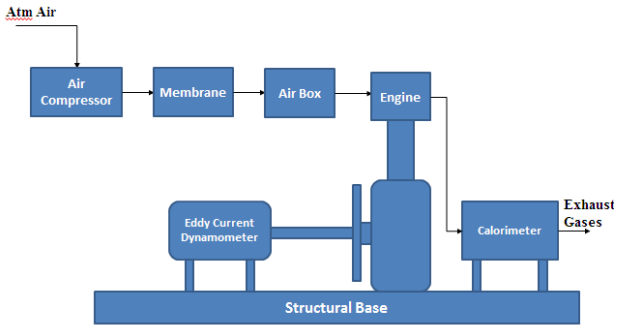

Fig 2 Experimental Setup

The engine parameters, air flow rate, fuel flow rate are measured using the above instruments. Test conditions were designed to investigate the effect of oxygen concentration on engine performance and combustion characteristics. Tests were carried out at different loadings starting from no load to the rated capacity of the engine with an incremental loading of 20%, at a constant speed of 1500 RPM. Consistency and repeatability of the engine operating conditions were ensured by first running it for approximately 10 minut.es at 1500 rpm at 50% load until exhaust gas temperature reached 250 °C. Once these conditions are achieved, the test engine was brought to the required test condition and then allowed for at least two minutes before collecting the data. Four different levels of oxygen concentration, 21% (ambient air), 23%, 25% and 27% by volume, were used for the inlet air. The fuel injection timing and injection pressure were maintained at original setting while adding oxygen to the intake air.

Results & Discussion

The prime objective of this research is to investigate the engine performance parameters affected by the use of oxygen enrichment. The performance values are reported at four operating performance points.

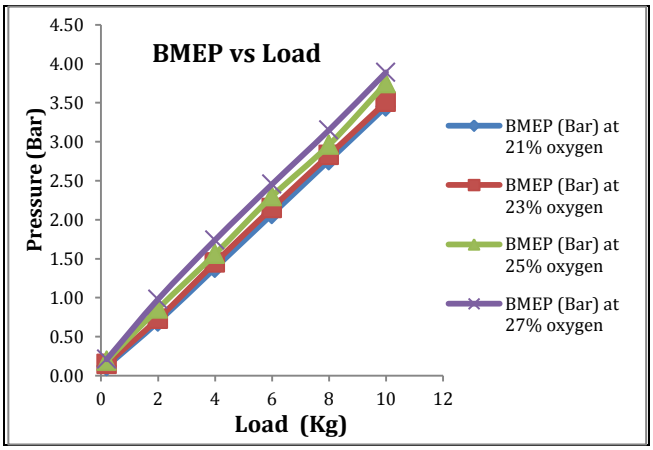

Fig 3 BMEP VS Load

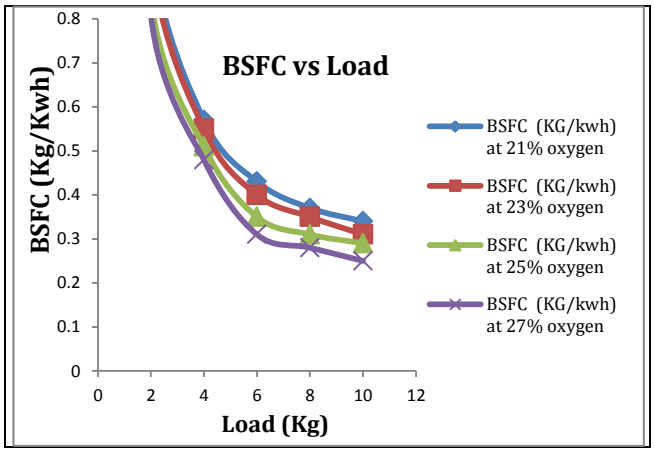

Fig 4 BSFC VS Load

Fig. illustrates Brake the mean effective pressure developed after combustion in the cylinder the maximum of 1 to 4 percent increase in peak cylinder pressures will be achieved in 23 to 27 percent oxygen enriched air than ambient air at part load conditions. These indicate a feasibility of increasing the net engine power by reasonable level. There is formation of local stoichiometric mixtures rather than rich premixed mixtures, which leads to rising in cylinder temperature and pressure.

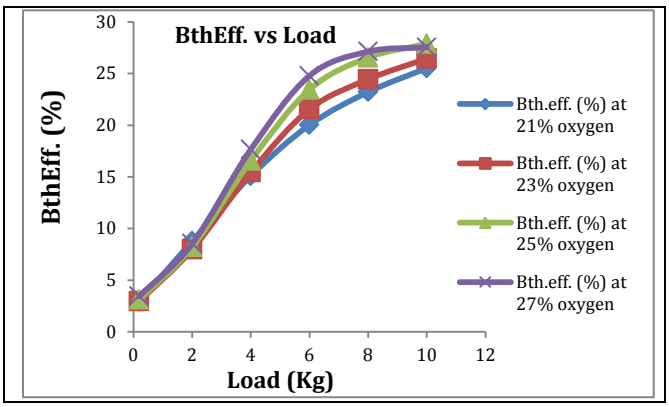

Fig 4 Bth.Eff. VS Load

The brake thermal efficiency, which is the ratio between the measured brake power to the product of the fuel flowrate and its calorific value, were calculated and plotted against different loads as shown in Fig. 3. From an ideal perspective, the brake thermal efficiency is affected by compression ratio and the thermodynamic properties of the working mixture. Compression ratio is fixed in this study; thermodynamic properties of the mixture however changed due to the addition of oxygen. An increase in oxygen concentration increases the mixture ratio of specific heats, which in essence increases the potential to convert the mixtures thermal energy to work energy.Brake thermal Efficiencies at the normal air composition that is on 21% of oxygen content, There is about 4 to 8 percent increased in brake thermal efficiency throughout all levels of oxygen enrichment as per the research is concern.

The brake specific fuel consumption is the ratio of rate of fuel consumption to break power produced, animportant parameter that reflects how good the engine Performance is. Stoichiometric air–fuel ratio decreases when the oxygen concentration in air increases. This means that less air is required for complete combustion of diesel fuel. When air mass flow is constant, as in these experiments, the additional oxygen was used to burn diesel and improves combustion. There is about 5 to 8 percent decrease in specific fuel consumption with increase in oxygen concentration from 21 to 27.

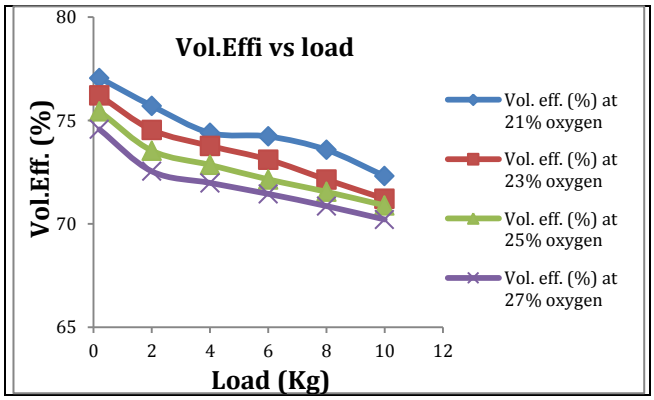

Fig 5 Vol.Effi VS Load

Volumetric Efficiency at the various air composition that is on 21%-27% of oxygen content, There is about 4 to 6 percent decreased in Volumetric efficiency throughout all levels of oxygen enrichment as per the Experimentation results.

Conclusions

The main effect of oxygen enrichment on the engine performance characterised and it is been seen that there is increased brake thermal efficiency& and reduced brake specific fuel consumption.

1. The research show that, as the oxygen concentration in the intake air is increased, the maximum value of cylinder pressure is increased, and the peak heat release rate is increased too. A feature of

this study was to consider the power output and combustion performance for the different levels of oxygen enriched combustion air on engine.

2. As far as the membrane technology is concern there are lots of polymer fibers are available for oxygen enrichment and its Experiments and simulation are performed to evaluate the separation performance of for an example of polysulfone hollow fiber membrane for oxygen enrichment system.
3. It is easy to vary percentage of Oxygen from the atmospheric air by using Membrane technology, so it will be revolution to automobile if its get adopted.

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HP31706605-Development 0f Improved Cooling System for Tracked Vehicle

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Abstract
This research paper focuses on a development of radiator of tracked Vehicle in which heat transfer enhancement is carried out by designing a new radiator with different configuration of fins like Rectangular Offset serrated plate fins on Hot Side and Wavy Fins On Cold Side to improve its thermal performance .Detailed thermal analysis, design,(by Effectiveness- No of transfer Units (ε-NTU) method) experimental investigation and comparison of heat transfer enhancement is done of both the old and new radiator. The designed radiator is going to be used for 300 HP engine of Tracked Vehicle (BMP2) which has been used by Vehicle Research and Development Establishment (VRDE), Ahmednagar, Government of India.
Keywords: Thermal Analysis, Effectiveness, Heat Dissipation, Heat Reception, Pressure Drop, Ejector cooling

1. Introduction

Compact heat exchangers are extensively used in aerospace, vehicle and cryogenic manufacturing because of their compactness for necessary thermal performance, compact space, weight, energy requirement and less cost.
Radiator is the vital important component of automotive cooling system. Upwards of 33% of the energy produced by the engine through combustion is lost in heat (Frank et al, 1996). Insufficient heat dissipation results in overheating of the engine, which can turns in breakdown of lubricating oil, weakening, wear of engine parts leads in less productivity. To reduce the stresses of engine as a result of high heat generation, radiators must be more compact so as to maintain required levels of thermal performance. (Amrutkar et al, 2013).
Coolant surrounding engine passes through radiator. In radiator coolant flows through it, gets cooled down and it is re-circulated into system again and again. Radiator sizing is very important while designing cooling system. Radiator size depends on mainly heat load and space availability. Heat load depends on heat rejection requirement for keeping engine surface at optimum temperature (Yadav et al, 2011). Compactness, low pressure drop, low cost and new material should be considered in the radiator design. The radiator size will be increased so that more heat can be brought away from the engine (Bengt Sunden, (2010).
Design of fin plays an important role in heat transfer. There is a scope of improvement in heat transfer of air cooled engine cylinder fin if mounted fin's shape varied from conventional one. The fin geometry and cross sectional area affects the heat transfer co efficient. In High speed vehicles thicker fins provide better efficiency. Increased fin thickness resulted in swirls being created which helped in increasing the heat transfer. Large number of fins with less thickness can be preferred in high speed vehicles than thick fins with less numbers as it helps inducing greater turbulence and hence higher heat transfer takes place (Durai Raju et al, 2015).Generally Logarithmic mean temperature difference (LMTD) or Effectiveness-No of Transfer Units (ε-NTU) methods are useful for calculations of thermal performance heat exchanger. Both methods have its own merits and preferred according to availability of data. When radiator inlet and outlet temperatures are known, LMTD gives quick solution. When any of the temperature is unknown, LMTD method requires more iterations to find exact solution (Shah et al, 2003). In this project Effectiveness-NTU (ε-NTU) classical method is used for thermal analysis because of its accuracy (Kays and London, 1998). The heat sink design of the radiator must be analyzed using the Effectiveness-NTU method to find the theoretical effectiveness, overall heat transfer rate of the radiator, and outlet temperatures of both air and water. Experimental analysis was conducted on the radiator to compare and confirm the analytical results. Matthew Carl et al, (2012).
The surfaces with wavy patterns are one of the popular surfaces in Plate-fin heat exchanger due to sinusoidal curve. Friction factor has an effect on mass flow rate of air.The suction side of the wavy fin punched with Rectangular Winglet Pairs(RWPs) can increase Nusselt number by 1.2%–4.1%, and decrease friction factor by 2.7%–9.6% (Balanna et al, 2015). In Heat Exchanger for air-side heat transfer applications, special surfaces are often employed to obtain high rates of heat transfer within the imposed size constraints. One geometry that can be used to enhance heat exchanger performance is a sinusoid ally curved wavy passage. Wavy channels are easy to fabricate and can provide significant heat transfer enhancement if carried out in an

appropriate (transitional) Reynolds number regime. Calculating total volume of the heat exchanger is possible just at the end of the designing process and naturally after doing all calculations of related to pressure drops, heat transfer coefficients, heat exchanger efficiency and outlet temperature talking about total heat transfer area is possible (Masoud et al, 2013).

When engines run at high values of rpm to increase the

Speed of the vehicle, the heat generated in the parts of the engine also increases drastically. Hence, at higher speed the cooling process should also be effective in order to dissipate the heat to the atmosphere. It can concluded by this analysis that, even at higher speed the given dimensioned radiator with given number of fins attached to it works properly with slight compromise in the decrease in efficiency of the fins used in the radiator, (Mounika et al, (2016). Ejector cooling system is used In Military Vehicles by using compact heat exchangers,(Engineering Design handbook Power Plant Cooling, headquarters United States army Materiel Command, 1975).

Ejector Cooling System- The cooling system is a high temperature, liquid, closed and forced-circulation cooling system. (Compact Heat Exchanger - Radiator.)

Principal Of Cooling System-With high velocity of exhaust gaseous coming out of nozzle creates low pressure zone in the ejector tray, which causes suction of atmospheric air Inside the ejector tray passing over radiator and oil cooler. (Venturi effect) and hence the radiator is cooled.

2. Project Overview

Fig. 1 BMP2 Tracked Vehicle**.**

This project is sponsored by (DRDO-VRDE) Vehicle Research and Development Establishment, Ahmednagar, Government of India, there was requirement to design and development of radiator for 300 HP engine of BMP2 Tracked Vehicle.

To reduce the Overheating of BMP2 Vehicle engine was the main objective of the project. In this project work, based on size of radiator, the theoretical calculations have been made by using ε-NTU method. The experimentation made on experimental set up available at VRDE which is with proper arrangement of coolant and air supply, temperature measurement sensors for coolant and air. And then after thermal performance has been validated by experimental testing. Objectives of this project includes, design and development of new radiator, its thermal analysis, and to carry out the experiment to check the required effectiveness based on the availability of hardware and finally to reduce the temperature of cooling system from 120 degree Celsius to 105 to 110 degree Celsius**.**

3. Experimental set up layout

The heat dissipation performance is investigated by using experimental setup. The efficiency of cooling system of an Internal Combustion engine when fitted with radiator is judged by its heat transfer performance. Comparison between theoretical and experimental heat balance is done for heat transfer performance.

The thermal performance of radiator is experimentally investigated in laboratory at VRDE Ahmednagar. The test section is mainly divided into waterside circuits and airside circuits. The test section is shown in Fig.2.

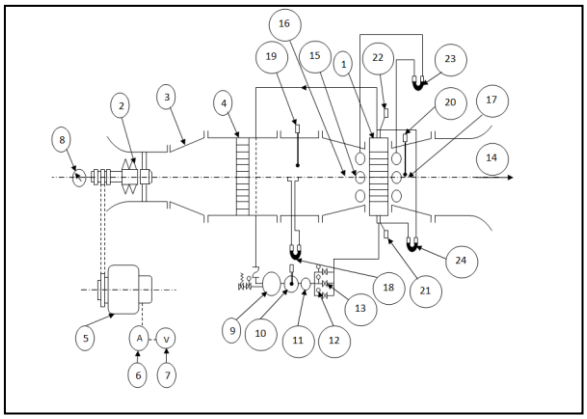

Fig.2 Experimental Setup

Experimental layout includes following components: 1)Test Radiator 2)Fan 3)Tunnel body 4)Rectifying Lattice 5)Shunt motor 6)Ampere Meter 7)Voltmeter 8)Speed Counter for Fan 9)Hot Coolant Tank 10) Additional Hot Coolant Tank 11)Coolant Pump And Motor 12)Coolant Flow meter 13)Coolant Flow Valve 14)Wind Direction 15)Connecting Tube 16)Upstream End 17)Downstream End 18) Liquid Column Gauge (Water) For Air Flow meter 19)Thermometer For Inlet Air Temp 20)Thermometer For Outlet Air Temp 21)Thermometer For Coolant inlet Temp. 22) Thermometer For Coolant outlet Temp. 23) Liquid

Column Gauge (Water) for Air Side Pressure Loss 24) Liquid Column Gauge (Mercury) for Water Side Pressure Loss.

Measuring Equipment's used are

i. Water Flow Meter: The water flow meter used with an accuracy of +2% of maximum scale.

ii. Air flow Meter: The minimum scale for liquid column is 1 mm on 30˚ inclined type manometer.

iii. Pressure Gauges: For waterside the liquid mercury column gauge have minimum 1mm accuracy. For the airside to measure pressure loss, the liquid column has 1mm accuracy.

iv. Thermometers: For measuring temperatures the thermometers used have +/- 0.1˚C accuracy for waterside and 1˚C accuracy for the airside.

4. Heat Transfer Calculations

Thermal analysis is determined by classical method first theoretically and then by experimental method which consists of heat rejection requirement, heat transfer requirement is decided as per engine specifications, engine operating conditions and vehicle operating conditions. Cooling system designed to fulfill all above requirements.

Table 1 Radiator dimensions

Parameter	Unit	Value
Total Heat Transfer	KW	105
Height	mm	733
Length	mm	1080
Depth	mm	140

Following parameters are considered for analytical approach.

Table 2 Inputs for Radiator thermal calculations.

Parameters	Hot Side	Cold Side
Fluid	Water	Air
Inlet temp (^oC)	120 (Th_1)	73 (Tc_1)
Outlet temp (^oC)	110.66 (Th_2)	114.70 (Tc_2)
Mean temp (^oC)	115.33	93.85
Mass flow rate (m) Kg/s	3.086	2.5
Density (ρ) Kg/m^3	1028.55	0.950
Specific heat (Cp) KJ/Kg-K	3.644	1.007
Dynamic viscosity (μ) $N\text{-}s/m^2$	0.00077	$19.8x10^{-6}$
Thermal Conductivity (k) W/m-K	0.37974	$28x10^{-3}$
Prandtl no. (P_r)	1.23	0.7214

5. Thermal Analysis

Thermal analysis of heat exchanger is to determine by doing the performance calculations to find out heat transfer rate (Rating Method). It is necessary to find out amount of heat transfer, outlet temperatures of both fluids. E-NTU method is based on concept of heat exchanger effectiveness.

Thermal analysis of Heat Exchanger -

A) Calculations for finding out required effectiveness

i. Coolant outlet temperature (hot)

$Th2 = Th1 - (Q/Ch)$ (1)

ii. Air outlet temperature (cold)

$Tc2 = Tc1 + (Q/Cc)$ (2)

iii. Coolant side heat capacity rate (hot)

$Ch = mh * Cph$ (3)

iv. Airside heat capacity rate (cold)

$Cc = mc * Cpc$ (4)

v. Heat capacity rate ratio:

$Cr = Cmin/Cmax$ (5)

vi. Required effectiveness

$\varepsilon reqd = [Ch * (Th1 - Th2)]/[Cmin * (Th1 - Tc1)]$ (6)

B) Selection of fins-calculations of free flow area (Aff), frontal area (A), heat transfer area per fin (As), Fin area (Af), Equivalent Diameter Dh, Heat Transfer area.

C) Heat Transfer coefficients and surface effectiveness of fins.

i. Core mass velocity, Gh = m / Affh (7)

ii. The Reynolds no. Re. = G Dhh/μ (8)

iii. The Colburn factor j given by correlation proposed by Joshi and Webb is

$j = 0.53\ Re^{-0.5} \times (1/Dhh)^{-0.15} \times \alpha^{-0.14}$ (9)

Iv. The Convective Heat Transfer Coefficient,

$hh = (jh \times cph \times Gh)/(Pr)^{0.667}$ (10)

v .The Fin Parameter is given by, M-

$M = \sqrt{2x\ hh / (kf \times t)}$ (11)

vi. Fin effectiveness is given by $\eta f = \tanh(Mlh)/(Mlh)$ (12)

Overall surface effectiveness is given by

$\eta oh = 1 - (af / as) \times (1 - \eta f)$ (13)

D) Pressure Drop Calculations-

i. Friction Factor f is given by correlation -

$f = 8.12\ Re^{-0.74} \times (1/Dh)^{-0.41} \times \alpha^{-0.02}$ (14)

ii. Pressure Drop $\Delta p = (4x\ f \times L \times G^2) / (2x\ Dh \times \varrho)$ (15)

Above calculations are done for both hot and cold sides.

E) Overall Heat Transfer coefficients and Number of Transfer Units (NTU)-

i. $1/UA = 1/(\eta overall \times hh \times A)\ hot + t/(Kw \times Aw) + 1/(\eta overall \times hc \times A)\ cold$ (16)

ii. Hot Side, Uoh = (UoAo) h / Aoh (17)

iii. Cold Side, Uoc = (UoAo) c / Aoc (18)

iv. Number of Transfer Units, Ntu = UoAo/Cmin (19)

F) Radiator effectiveness is calculated by

$$\varepsilon_{cal} = 1 - e\left((e(-NTU^{0.78} \times Cr) - 1) \times \frac{NTU^{0.22}}{Cr}\right) (20)$$

As the Cold Side (Air) is Critical, Compare this Pressure Drop with the Pressure Drop Of Old Radiator at Cold Side (Air)

Pressure Drop = 176.61 mm of Water Column with Old Heat Exchanger at cold side (Air).

=37 mm of Water Column with New Heat Exchanger at Cold Side (Air).

Hence From Above Comparison, it is proved that Design is Completely Safe.

From the results obtained, the heat dissipated from waterside (hot) has been calculated and this value is judged by heat received on airside (cold) simultaneously.

Mathematical expressions used for calculations;

i. Heat dissipated on coolant (hot):

$Qh = mh * Cph * (Th1 - Th2)$ (21)

ii. Heat received on airside (cold):

$Qc = mc * Cpc * (Tc2 - Tc1)$(22)

6. Results and Discussion

Comparison of analytical and experimental results at 180 lpm coolant flow rate and 35 m/s air velocity,

Table 3 Analytical Results

Parameter		Unit	Value
Heat dissipated by coolant	Qh	KW	105
Heat received by air	Qc	KW	105
Coolant inlet temperature	Th1	°C	120
Coolant outlet temperature	Th2	°C	110.66
Air inlet temperature	Tc1	°C	73
Air outlet temperature	Tc2	°C	114.70

Table 4 Experimental Results

Parameter		Unit	Value
Heat dissipated by coolant	Qh	KW	118.23
Heat received by air	Qc	KW	105.99
Coolant inlet temperature	Th1	°C	90
Coolant outlet temperature	Th2	°C	79.5
Air inlet temperature	Tc1	°C	36.8
Air outlet temperature	Tc2	°C	78.9

The above results it seems that both analytical and experimental results for heat dissipation from coolant are nearby matched with each other. Thus theoretical thermal analysis of radiator using ε-NTU method is validated using experimental method.

But from above experimental results it is shown that the heat dissipated by coolant is not received by air totally. The main reason behind this is, there are some radiation heat losses in the range of 9% to 11.3% and remaining heat losses are unaccounted heat losses.

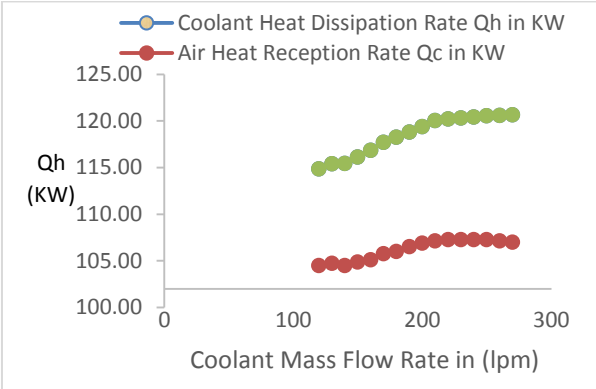

Fig. 3 Coolant Mass Flow Rate Vs Heat Transfer

From the above graph it is shown that the heat dissipated by coolant as well as heat received by air are simultaneously increasing with coolant mass flow rate.

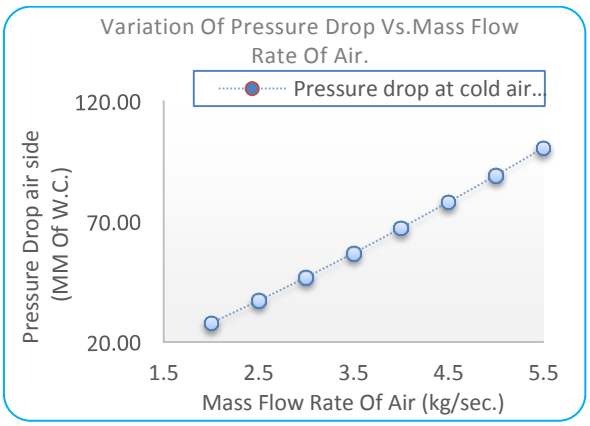

Fig. 4 Mass Flow Rate vs Pressure Drop

From the above graph it is found that as mass flow rate of air increases, pressure drop across heat exchanger also increases.

Motivation-The performance of new radiator can be understood by characteristics curve.

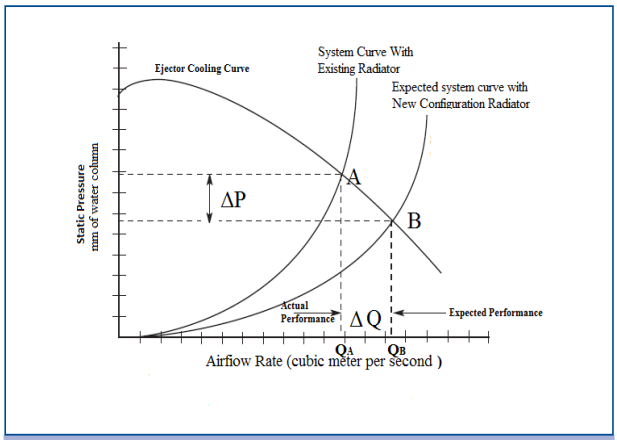

Fig. 5 Characteristic Curve for the Performance of Radiator.

Conclusions:

In this project work testing have been made with variation in mass flow rate of coolant. After completing all the tests the following conclusion can be made;

1) From the experimental analysis, it is found that the heat dissipated by coolant is not received by air totally. Some of the heats dissipated by coolant get lost while transferring from coolant to air.
2) The heat transfer losses are varied from 9 % to 11.3% over entire range of coolant inlet temperature change.
3) Out of total heat transfer losses radiative heat transfer has more contribution and the remaining heat losses are unpredictable heat losses.
4) It is found that due to reduction in pressure drop, heat transfer rate increases and the effective (working) temperature of coolant decreases and radiator gets cooled.
5) From the Characteristics curve (Motivation), it is also concluded that Mass flow rate of air Increased from Q_A to Q_B. Hence due to increase in mass flow rate heat transfer rate is increased by Equation,
   $Q = m.Cp.\Delta T$.

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HP31710001-Influence of N-Butanol additives with Terminalia Methyl Ester as a fuel on Engine Emission Characteristics

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Abstract

In the today's world increase of consumption of fossil fuels is on very large scale and resources are limited hence there must be some alternative source which could replace fossile fuels and also give desired output as per given by fossil fuels. The present research in this paper reviews the various aspects of biodiesel fuel derieved from non edible Terminalia oilwhich is converted to biodiesel by transesterification chemical process. The blend of biodiesel and diesel used where B00, D89B6NB5, D83B12NB5, D77B18NB5, D71B24NB5, D65B30NB5 and D59B36NB5. The Present performance test was conducted in a single cylinder, water cooled, four strokes, 3.5 KW at 1500 rpm, with VCR .The emission parameters such as CO, Nitrogen Oxide, HC and smoke opacity are discussed with different compression ratios of different blends at vary loads conditions. It was obtained that the Terminalia Oil with n-butanol additive blends with diesel are founding improved performance and emission characteristics outcome in a diesel engine with no any modification.

Keywords: *VCR, Terminalia oil, N-Butanol , transesterification, Brake power, Brake thermal efficiency.*

1. Introduction

It is a true fact that many countries around the world are still heavily dependent on fossil fuels for energy and basic needs. No doubt, these fuels are very effective when concerned the power production quality. when we think long run, they are not helpful. Fossil fuel is limited. They will undoubtedly finish one day. Therefore, before the critical phase comes up, Scientists and Engineers, around the world, are constantly functioning and researching in this regard and should attempt their level best to substitute fossils fuels with renewable energy sources. Choosing to utilize a renewable energy supply will not only convert into cost savings over the long-term but will also help keep the atmosphere from the risks of fossil fuel harmful emissions. Management of energy and utilizing renewable sources is the final goal of energy. Many vehicles run on gasoline (which is a fossil fuel). Gasoline will diminish one day and vehicle industry must alternative to some new class of energy. Thus, it is essential to substitute fossil diesel fuel by alternative fuel. In biofuels, the nation has a ray of expectation in providing energy security. The properties of biodiesel are very close and similar to the conventional fuels; therefore biodiesel becomes a very superior and little cost choice to the diesel fuel. Biodiesel is a biodegradable, harmless and renewable fuel which is produced from close by and locally accessible sources like vegetable oil and animal fats etc. the emission of unburned HC, CO, PM of biodiesel combustion is much lesser than conventional diesel fuel.

Murugesan et.al [3] reported that the very high flash point and high viscosity of vegetable oil are the major problems of the direct use of oil as fuel in compression ignition diesel engine. This problem is solved to convert into biodiesel by blending of vegetable oil with diesel fuel and transesterification chemical process.

Y.C. Sharma et.al [5] studied that general oil bearing plants and trees which are non-edible like Neem, Karanja, Mahua, jatropha, etc. These different species of forest-based seeds are reported as the possible resource of biodiesel feedstock in India. Heterogeneous catalysts such as calcium oxide, magnesium oxide and others are also being tried to decrease the catalyst amount and production cost of biodiesel.

The oil yields from this variety at current are unsatisfactory to meet the demand for raw material on great amount production of biodiesel. India is an agrarian nation and has rich plant biodiversity which can carry the growth of biodiesel. India has a huge geological area with farming lands and wastelands on which oil containing plants can be planted.

Maumita Chakraborty et.al [6] reported that Terminalia found locally obtainable raw materials could be right for biodiesel making in the north-eastern section of India and has the probable to suit an alternative for usual diesel.

K. Muralidharan et.al [7] evaluated the performance, combustion and exhaust emission levels of waste cooking oil biodiesel and diesel blends in a fully instrumented single cylinder, variable compression ratio multi fuel engine.

B. De et.al [9] evaluated that the effects of Jatropha oil combustion on the performance and pollutant

emissions on VCR diesel engine. They demonstrated that increase in compression ratio improves the performance in terms brake thermal efficiency and the increase of Jatropha oil concentration in the blends increasing the exhaust gas temperature and emission parameters like NOx , CO and decreases the thermal efficiency of the engine.

Gokhan Tuccar et.al [11] reported that the power and torque output of engine reduced slightly when butanol was added to the microalgae biodiesel blends and CO, NOx emission and smoke opacity values improved with butanol addition.

Terminalia Oil preferred for the current work of investigational research of performance and emission characteristic of VCR diesel engine. From the literature examination, it found that a more of study work has been passed out on evaluating the performance and emission characteristics of dissimilar grades of vegetable oils and biodiesels at a compression ratio diesel engine but the very minute study has been done in evaluating the performance of Terminalia Oil with the n-butanol additive. The outcome of compression ratio has not been analysed for the Terminalia Oil - Diesel blends with the n-butanol additive. Hence, the research work of the individuality of Terminalia Oil on the diesel engine for variable compression ratio with vary loads is very essential. In the current study, an outcome of variable compression ratio with varies loads of Terminalia Oil – Diesel blended fuels with n-butanol additive on the performance and emission characteristic of fuel has been studied.

The dissimilar blends of Terminalia Oil with n-butanol additive and usual diesel fuel are prepared and the following studies are passed out. The performance of a variable compression ratio engine using dissimilar blends at different compression ratios like 16:1, 17:1, and 18:1 and vary load like 0,4 and 8 in kg. It is compared with the result of standard diesel fuel.

The emission parameters such as CO, Nitrogen oxide and HC are discussed with different compression ratios of different blends at vary loads conditions.

It was obtained that the Terminalia Oil with n-butanol additive blends with diesel are founding improved performance and emission characteristics outcome in a diesel engine with no any modification.

2. Material and Method

Biodiesel is prepared from terminalia seed oil using alkali-catalyzed transesterification. The fatty acid profile of Terminalia oil obtainable 39.5% of the oil is saturated whereas, 60.5% is unsaturated fatty acid. The methyl ester formed in two stages. The first stage (acid catalysed) of the process is to decrease the free fatty acids (FFA) content in Terminalia oil by etherification with methanol (99% pure) and acid catalyst sulphuric acid (98% pure) in one hour time at 57 ^{0}C in a 2000 ml three necked round –bottom flask was used as a reactor. The Terminalia oil is initially heated to 50 ^{0}C and 0.5% (by wt) sulphuric acid is to be mixed in oil then methyl alcohol about 10% (by wt) added. Methyl alcohol is a mixed in the extra amount to fast the reaction. This reaction was carried with stirring at 650 rpm and the temperature was maintained at 55-57 ^{0}C for 90 min with regular analysis of FFA every after 20-30 min. When the FFA is controlled up to 1%, the reaction is finished. The main difficulty to acid catalysed esterification for FFA is the water formation. After dewatering the esterified oil was ready to the transesterification process.

1000ml of Terminalia oil was calculated using the measuring cylinder, then poured into a 2000 ml three necked round bottom flask. This oil was heated up to 60 ^{0}C. Cao has used as a solid base ecofriendly heterogeneous catalyst. The reaction conditions taken were as10:1 (alcohol: oil) molar ratio of 3.00wt % of CaO catalyst at 65±0.5°C for 2 hr of reaction time. The solution (methanol+Cao) was correctly stirred and mixed into preheated oil then heated upto 60 ^{0}C. Finally, FFA was checked and the solution was permitted to settle for 24 hours in a separating funnel. By settling and separation method glycerol and soap were collected from the bottom of separating funnel. The extra alcohol was removed by distillation then hot water was used to clean it and then permitted it to remain in separating funnel until the plain water was seen below the biodiesel in the separating funnel. The seven fuel sample designation of different composition blends of diesel and Terminalia Methyl Ester were premixed on volume basis after production of ester which is shown in Table I.

Table I: Seven Fuel Sample Designation of Different Composition

Fuel Sample Designation	Composition (By Volume %)
B00	100% Diesel Fuel
D89B6NB5	89%Diesel+6% Terminalia methyl ester+5% n-butanol
D83B12NB5	83%Diesel+12% Terminalia methyl ester+5% n-butanol
D77B18NB5	77%Diesel+18% Terminalia methyl ester+5% n-butanol
D71B24NB5	71%Diesel+24% Terminalia methyl ester+5% n-butanol
D65B30NB5	65%Diesel+30% Terminalia methyl ester+5% n-butanol
D59B36NB5	59%Diesel+% 36Terminalia methyl ester+5% n-butanol

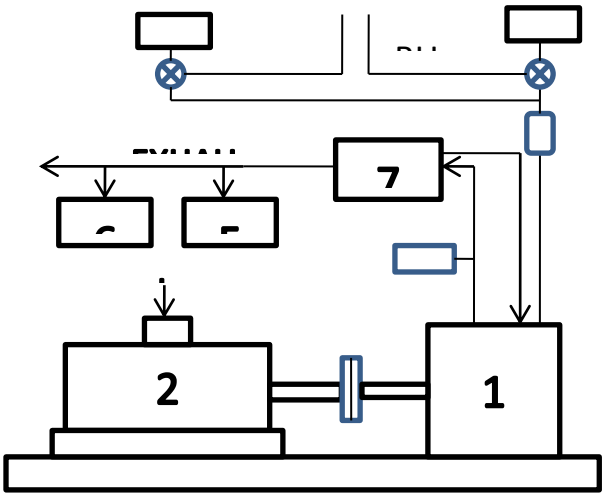
The parameter of diesel (B00) and dissimilar blends of diesel and TEME (B100) were determined as per ASTM D6751 which shown in Table II.

Table II: Properties of Diesel Fuel and Terminalia Oil

Parameter	Test Std. ASTM 6751	Reference		Diesel	Terminalia Biodiesel
		Unit	Limit	B00%	B100%

Density	D1448	gm/cc	0.800-0.900	0.830	0.875
Calorific value	D6751	MJ/kg	34-45	42.5	39.80
Cetane no.	D613	-	41-55	49	52
viscosity	D445	mm^2/sec	3-6	2.7	5.1
Flash point	D93	^0C	-	64	138

3. **Experimental Setup**



1. ENGINE TEST RIG, 2.

Fig.1 Experimental Setup

The setup used in this experiment as shown in Fig.1. The Present performance test was conducted in a single cylinder, water cooled, four strokes, 3.5 KW at 1500 rpm, with VCR shown in figure 1. Compression ratio was changed within the range of 12-18 and loading and unloading in KG with the help of loading unit manually. The different parameter measured with the help of various sensor which is mounted on the engine and then data stored using engine software. The detail specification of the engine is given in Table III. To obtain the baseline parameters, the engine was first operated on diesel fuel. Performance and emission tests are carried out on the diesel engine using Terminalia methyl ester, and its various blends.

Table III: Engine Specification

Sr.No.	Description	Specification
1.	Model and Make	Kirloskar and TV1
2.	No. of cylinder	Single
3.	Cycle	Four stroke
4.	Bore and stroke	87.5 mm and 110 mm
5.	Rated Power	3.5 kW at 1500 rpm
6.	Compression ratio	17.5, Modified to work in range of 12 to 18
7.	Dynamometer	Eddy current, water cooled, with loading unit
8.	Cubic capacity	0.661 liters
9.	Software	"EnginesoftLV"Engine performance analysis software

4. Result and Discussion:

Engine Performance

The performance parameters considered in the current work are BP and BTE. The different parameter with respect to load of 0, 4 and 8 in kg and compression ratio of 16, 17, and 18 are presented respectively. the seven blends tested i.e. B00, D89B6NB5, D83B12NB5, D77B18NB5, D71B24NB5, D65B30NB5 and D59B36NB5.

Brake Power

It can be seen from fig.2a-c that BP graph lines of different blends came closer to each other as CR was increased from 16 to 18.It was found that when increase the blending of Terminalia biodiesel in diesel, slightly decrease or close in BP. As compression ratio increases from 16 to 18 and load increases 0 kg to 8 kg, the BP increases for all biodiesel blends. The maximum brake power obtained for D77B18NB5 and diesel at 8 kg load and CR18 are 2.42 kW and 2.41 kW respectively.

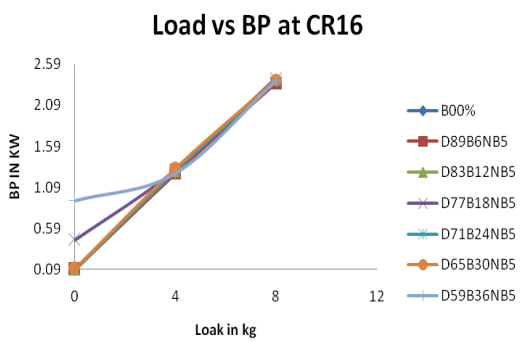


Fig. 2a Variation of load Vs BP at CR16

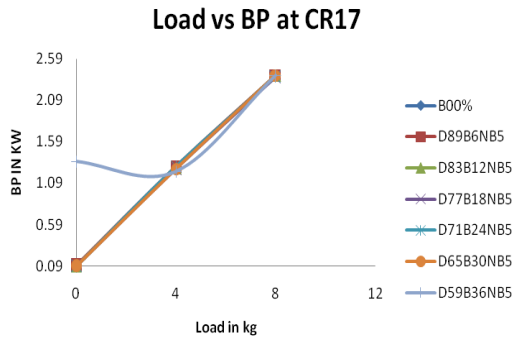


Fig. 2b Variation of load Vs BP at CR17

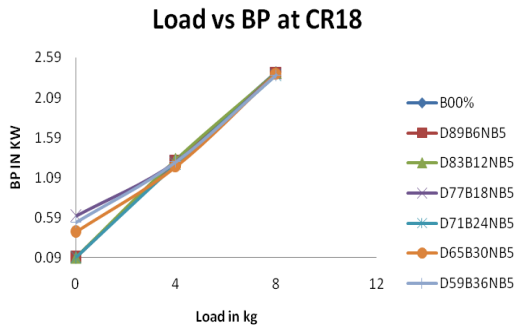


Fig. 2c Variation of load Vs BP at CR18

Brake Thermal Efficiency

It can be noted from Fig.3a-c that as the blend percentage increases the Brake thermal efficiency increases as compared to diesel. But it is found that the maximum BTE of D76B24NB5 is 27.01%. With the increase in CR from 16 to 18, the BTE increased for Terminalia bio diesel blends with respect to Diesel. The BTE increased for Terminalia biodiesel blends with the increasing load.

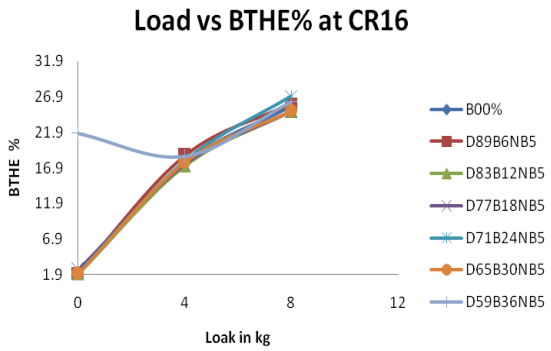


Fig. 3 a Variation of load Vs BHTE at CR16

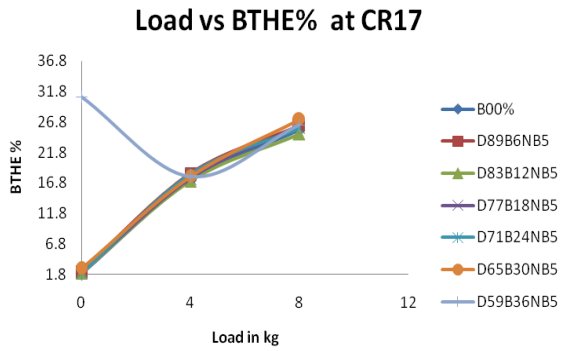


Fig. 3b Variation of load Vs BHTE at CR17

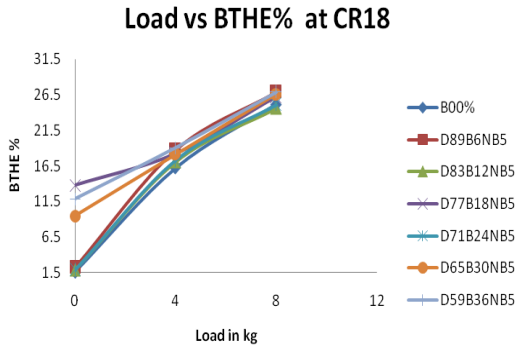


Fig. 3c Variation of load Vs BHTE at CR18

2. Emission Performance:

The variation of emission parameter with different compression ratio at 16,17 and 18 and increasing laod at 0 kg,4 kg and 8 kg considered are Carbon monoxide (CO), Unburned Hydrocarbon (HC) and Oxides of Nitrogen (NO).

Hydrocarbon Emission (HC) :

It is seen from the Fig.4a-c that HC emissions decrease with increase in CR from 16-18 for all the fuels tested which is due to complete combustion of fuel at higher CR as compared to diesel. It is also founded that HC decreases with the increase in blend percentage. This is due to better combustion of Terminalia biodiesel due to its oxygen content.

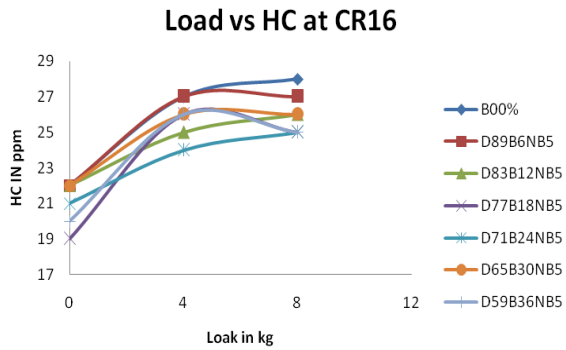


Fig. 4a Variation of load Vs HC at CR16

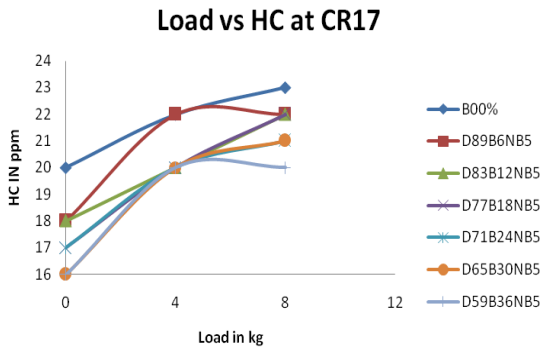


Fig.4b Variation of load Vs HC at CR17

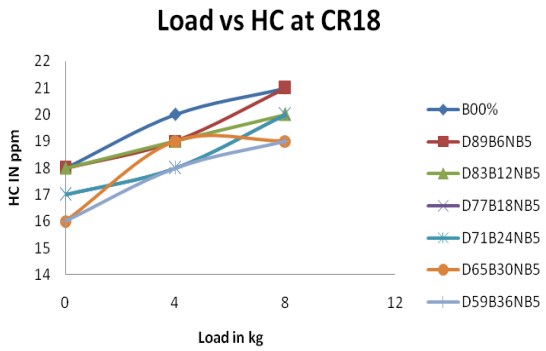


Fig. 4c Variation of load Vs HC at CR18

Carbon Monoxide Emission (CO):

It is seen from the Fig.5a-c that as the CR is increased, the CO emission is reduces for all Terminalia biodiesel blends. This is due to superior combustion of fuel at higher CR due to high air temperature in the combustion cylinder. It is observed that the CO emission decreases with increasing load as compared to diesel fuel. This decrease is mainly due to the oxygen content of biodiesel which makes the entire combustion. This is held due to addition of n- butanol to Terminalia diesel blend.

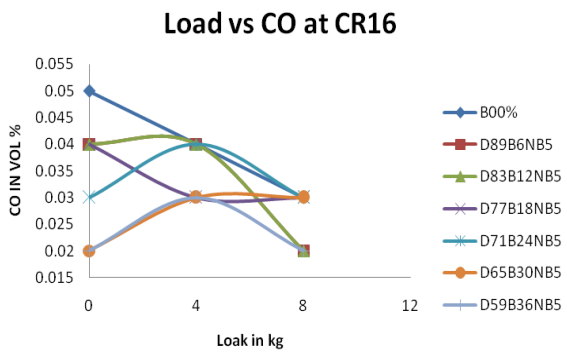


Fig. 5a Variation of load Vs CO at CR16

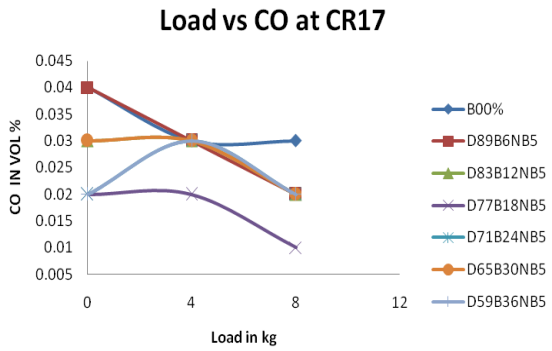


Fig. 5b Variation of load Vs CO at CR17

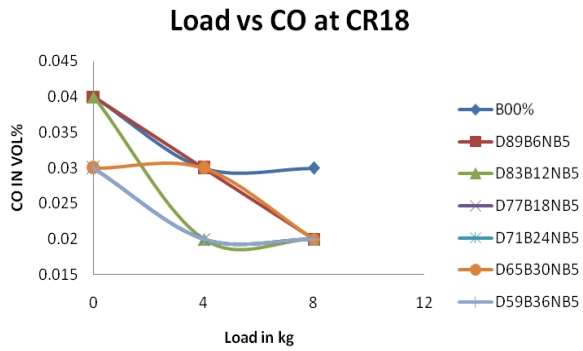


Fig. 5c Variation of load Vs CO at CR18

Nitrogen Oxide Emission (NO):

It is seen from the Fig.6a-c that as CR increases Nitrogen Oxides emissions increases for blends but it decreases for Diesel. At a CR of 16 and 17 and at load 8 Kg,it founded that the Nitrogen Oxides emissions for Diesel are higher as compared to Terminalia biodiesel. It is observed that the Nitrogen Oxides emission decreases with increasing load as compared to diesel fuel at CR 16 and CR17.

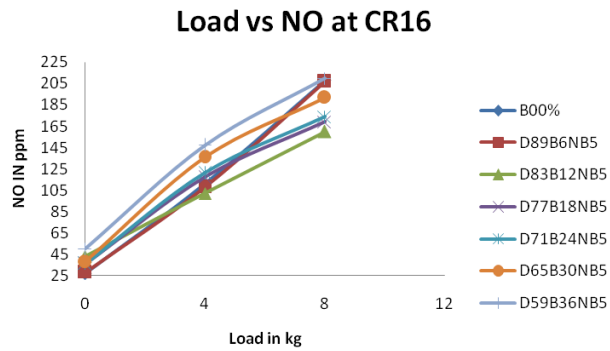


Fig. 6a Variation of load Vs NO at CR16

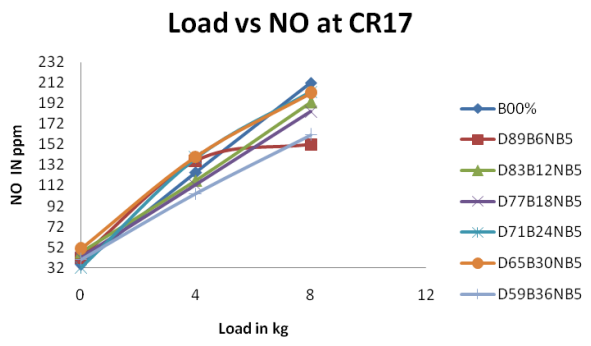


Fig. 6b Variation of load Vs NO at CR17

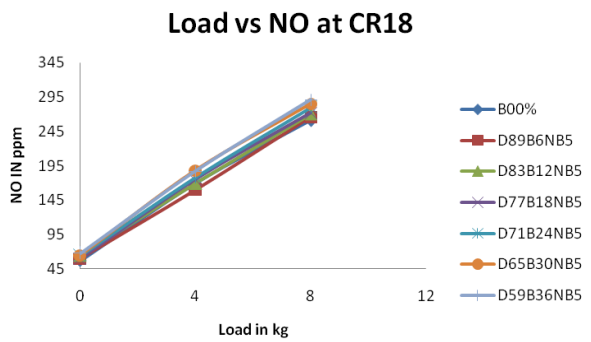


Fig. 6c Variation of load Vs NO at CR18

Conclusion:

Based on the performance and emission results, it is concluded that the brake power of biodiesel is close to standard diesel for all compression ratio in load operation. The Break Thermal Efficiency is increases with increasing load and compression ratios. At higher compression ratios (16 to 18), combustion of fuel is complete due to high temperature of compressed air. Due to which, the exhaust emissions are found to reduce at higher CRs. It is observed that the CO emission decreases with increasing load and CRs as compared to diesel fuel. It is observed that the Nitrogen Oxides emission decreases with increasing load as compared to diesel fuel at CR 16 and CR17.HC emission decreases with increasing CRs and Blend percentage. The exhaust emission tests revealed that CO and NOx emission values improved with butanol addition. CO and NOx emission values of biodiesel are decreased as compared to diesel fuel with addition of n-butanol in the blends because of n-butanol possesses higher heating value,higher cetane number, lower vapor pressure and good atomization.

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HP31710009-Performance and Emission Analysis of a Diesel Engine Fuelled with Waste Turmeric oil.

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Abstract

In the present work, a single cylinder variable compression ratio diesel engine was tested with different blends of biodiesel obtained from waste turmeric oil. The different blends of biodiesel used are B00, B9, B18, B27, B36, and B45. The Engine performance (brake power, mechanical efficiency) and emission parameter (carbon monoxide, nitrogen oxide and hydrocarbon emission) were measured by using different blends of biodiesel has been studied behavior of engine. Engine experimental result shows that reduction in carbon monoxide, nitrogen oxide and hydrocarbon emission compare with diesel fuel together with brake power and mechanical efficiency close to diesel. There are slight increases in smoke opacity as for biodiesel mixture higher than diesel fuel.

Keywords: *Waste turmeric oil, Engine performance, Emission parameter.*

1. Introduction.

The continuous increases in requirement of fossil fuel, consequent upon the increasing population in present day, has made reduces of conventional fuel resources in future more quite fact. Also the gas emission from these fuel affect the climate and causing global warming. In such situation it is need to find out an alternative fuel overcome to this effect. In addition the energy sources clean and renewable, it will reduce environmental effect. In this detection of an alternative energy sources, scientist select the biodiesel-diesel blend as alternative fuel because of research work show that properties of biodiesel prepared from vegetable oil have very close to fossil fuel.

The experimental result has been compared and analyzed with standard diesel, it show that considerable improvement in performance parameter as well as exhaust emission. Reduction of carbon monoxide, hydrocarbon, carbon dioxide at expense of nitrogen oxide emission(Muralidharan, et al, 2011). Experimental analysis shows that increases in mechanical efficiency and brake power at high compression ratio. The emission of CO, HC dropped with increases in blending ratio and compression ratio (Nagaraja et al, 2015). Evaluate the performance and emission of DI CI variable compression ratio engine fueled with honne oil. At 18 CR break thermal efficiency less than diesel and brake specific fuel consumption higher than that diesel fuel. Reduction in carbon monoxide and hydrocarbon as compare diesel fuel at compression ratio 18(Channapatana, el at, 2015). Test conducted by using hydro treated refined sunflower oil emission and performance result show that decrease in CO, HC, NOx, BSFC and increases in brake thermal efficiency (Hemanandh et al, 2015). The engine performance for various PFAD biodiesel blends at various loads comparatively close to diesel fuel.(Malvadeav, et al, 2013).

From the literature review it can be inferred that a lot of research work has been carried out on evaluating the performance and emission characteristics of different grade of vegetable oils and biodiesel but there is no anyone work has been found on waste turmeric oil. This oil has the potential that become an alternate for conventional diesel oil. Waste turmeric oil is produced from leaf of turmeric crop which is source of raw material. This crop cultivated in western maharashtra side. These leafs are crushed and mixed with steam to remove the oil. After that oil and steam is separated finally golden yellow colour oil is obtained. Waste turmeric oil and its blends with diesel fuel are selected as fuel for VCR single cylinder engine. The different blends of waste turmeric oil and standard fuel are prepared to carry the test.

1) Performance and emission characteristics of variable compression ratio engine using various blends at different load.
2) Compare its performance and emission parameter with diesel fuel.

2. Biodiesel production.

Vegetable oil has similar density, calorific value, cetane number, viscosity compared to pure diesel fuel. However this straight vegetable oil cannot be used directly in engine because it create problem in

compression ignition engine. Such as poor fuel atomization, piston ring sticking, clogging of fuel injector. Therefore it is needed to improve properties of vegetable oil such that it may be used as substitute to diesel fuel. Several techniques are available to reduce the viscosity of vegetable oil such as blending, pyrolysis, transesterification. Waste turmeric oil obtained from leaf of turmeric crop. This oil filtered to remove the solid material and preheated at 110ºC for the 30 minute to remove the moisture and wax.

2.1 Transesterification.

Transesterification is reaction of oil (triglyceride)with primary alcohol to form ester and glycerol. The oil was stirred and heated up to 60ºc at which mixture of alcohol 10% and 0.5% NaOH added and reaction continued for 90 minute. After the confirmation of completion of methyl ester formation, the heating was stopped and product was cooled and transfer to separating funnel. Two layers were observed clearly whenever it allowed settling for 24 hours in separating funnel. Top layer was biodiesel and higher denser layer settle at bottom was glycerin.Once the glycerin and biodiesel phases were been separated. The biodiesel was washed with distilled water.

2.2. Fuel properties.

The properties of diesel fuel, waste turmeric oil, its blend are given in table 1. It is shown that viscosity of biodiesel higher than that pure biodiesel. The density of pure biodiesel is 0.860 which slightly higher than that diesel fuel 0.830. Heating value of biodiesel is lower than that diesel fuel. Fuel with flash point above 50ºc is considered safe. Thus biodiesel with high flash point 90ºc as an extremely safe to handle and storage.

Table.1. Properties of Diesel and Pure Biodiesel**.**

Properties	Ref. Std. Astm 6751	Unit	Diesel B00%	Pure Biodiesel B100%
Density	D1448	gm/cc	0.830	0.86
Calorific value	D6751	MJ/kg	42.50	38
Cetane no.	D613	-	49.00	51
Viscosity	D445	mm2/sec	2.7	4.2
Flash point	D93	ºC	64	91
Fire point	D93	ºC	71	106

3. Experimental Procedure.

The single cylinder variable compression ratio direct injection coupled with eddy current dynamometer engine use for experimentation with required instrumentation and with computer interface. Detailed specifications of engine given in table 2. Engine has provision to change the compression ration by tilting block arrangement. The tilting block arrangement consists of tilting block with six allen bolt, compression ratio adjusting lock nut and compression ratio indicator. For setting the selected compression ration allen bolt are loosened. Then lock nut on to be loosened and adjuster to be rotated to set chosen compression ration by observing the compression ratio indicator and to be locked by using lock nut. Finally all allen bolt tightened.

3.1 Test Procedure.

The fuel used in this study includes diesel fuel, biodiesel blend. The test were carried out by using neat diesel fuel(denoted as B00), 9% biodiesel + 91% diesel fuel(B9), 18% biodiesel + 82% diesel fuel(B18), 27% biodiesel + 73% diesel fuel(B27), 36% biodiesel + 64% diesel fuel(B36), 45% biodiesel + 55% diesel fuel(B45) at different compression ratio 16, 17, 18 with varying load such as zero, half, full. The engine is started using diesel fuel and run for few minute to reach the steady state condition. The test conducted at constant speed. In every test exhaust gas emission such as CO, HC, NO, CO2 and O2are measured. Also the performance parameter BP, BTE, Mechanical efficiency with respect to compression ratio and load for different blend are recorded. Sane procedure repeated for different blend of waste turmeric oil.

Table 2. Engine Specification.

Sr. no	Description	Specification
1	Model and make	Kirloskar and TV
2	No. Of cylinder	Single
3	Cycle	Four stoke
4	Bore and stoke	87.5mm and 110 mm
5	Rated power	3.5 kw at 1500 rpm
6	Compression ration	17.5, modified to work range of 12 to 18.
7	Dynamometer	Eddy current, water cooled with loading unit
8	Cubic capacity	0.661 liters
9	Software	EnginesoftLV

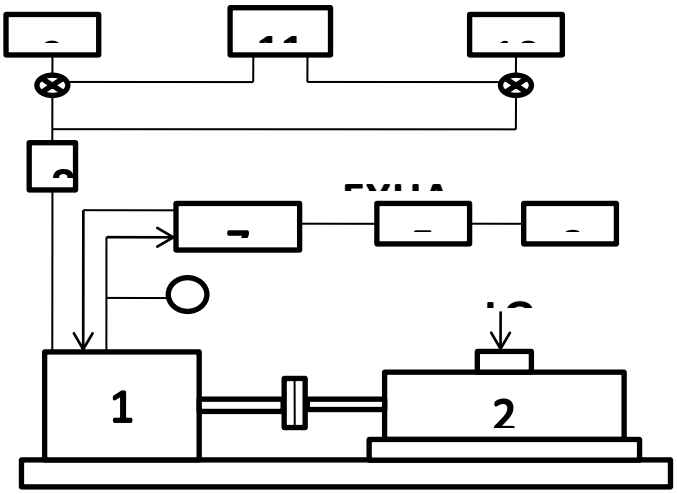

Fig:1 Engine set up.
1. Engine Test Rig, 2. Dynamometer, 3. Exhaust Gas Temp. 4. Test Bed, 5. Smoke Meter, 6. Exhaust Gas Analyser, 7. Calorimeter 8.Fuel Filter, 9. Oil,10. Diesel 11.Buretee

4.Result and Discussion.

4.1 Brake Power.

Fig. 2, 3 and 4shows that brake power with load for different blends and compression ratio. Graph indicate that the brake power increase with load. Brake power is lower at zero loads and high at full load condition. Brake power for all the blends are very close to pure diesel fuel for all compression ratios.

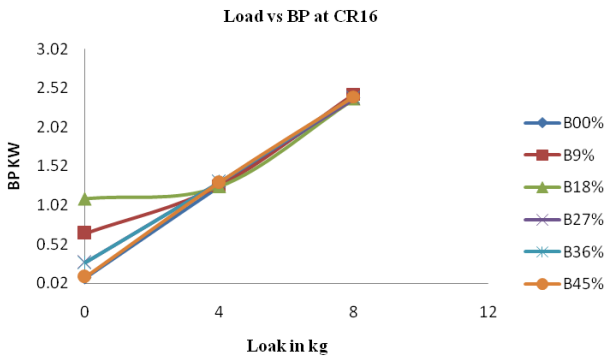

Fig: 2 Variation of brake power with load.

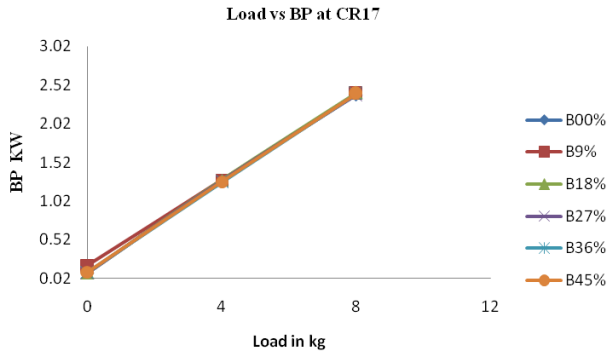

Fig:3 Variation of brake power with load.

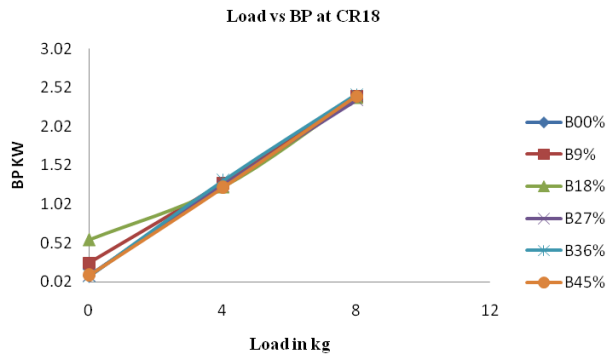

Fig:4 Variation of brake power with load.

4.2 Mechanical Efficiency.

Fig. 5, 6 and 7 show that variation of mechanical efficiency with load for different blends and compression ratio. It has been observed that there are increases in mechanical efficiency for all the blends as the load increase. Mechanical efficiency of the all blends is lower than that pure diesel. For high compression ratio mechanical efficiency of all blends close to pure diesel.

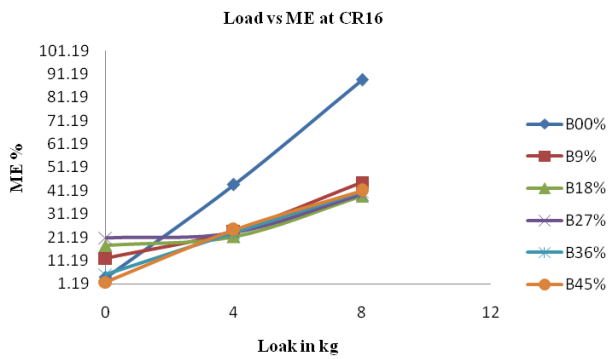

Fig:5 Variation of mechanical efficiency with load.

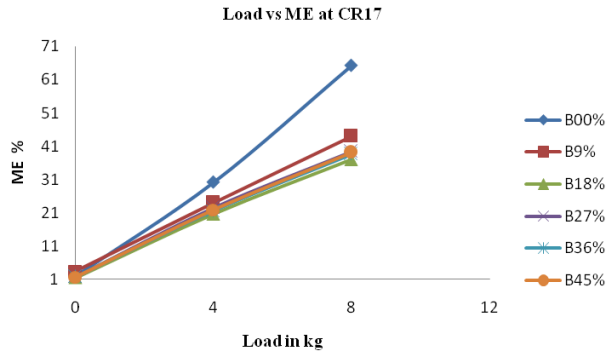

Fig: 6 Variation of mechanical efficiency with load.

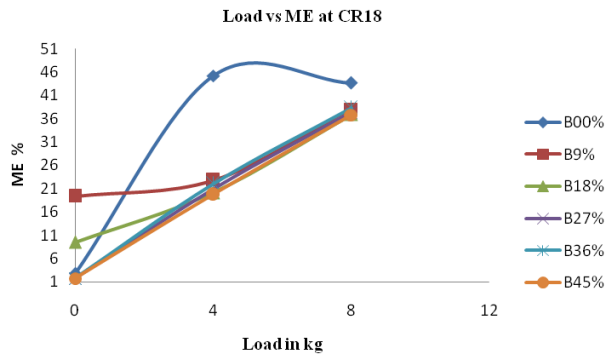

Fig:7 Variation of mechanical efficiency with load.

5. Emission Analysis.

5.1. Hydrocarbon emission.

Fig. 8, 9 and 10 shows variation of hydrocarbon emission with load for different blends and compression ratio. Graph indicates that hydrocarbon emission slightly increases with load for all blends and diesel. This due to the fuel rich mixture at higher load. There is reduction in HC emission for all blends as compare with diesel fuel.

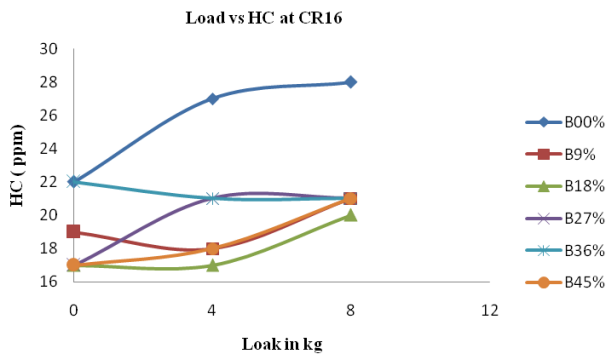

Fig:8 Variation of hydrocarbon emission with load.

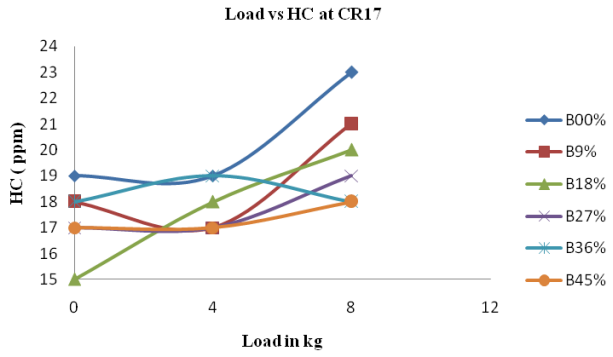

Fig: 9 Variation of hydrocarbon emission with load.

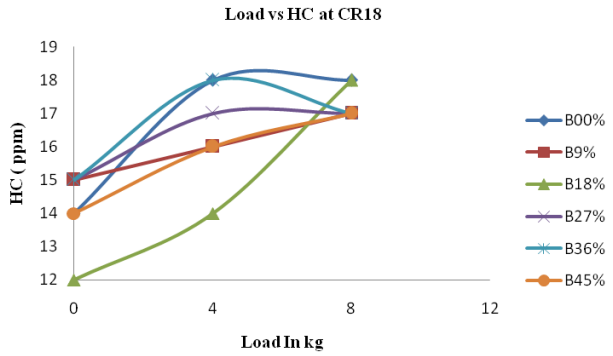

Fig: 10 Variation of hydrocarbon emission with load.

5.2 Carbon monoxide.

Fig. 11, 12 and 13 show that variation of carbon monoxide with load for different blends and compression ratio. Graph indicates that CO emission reduces with load. It can also observe that CO emission is lesser for all blends compared to diesel. This lower CO emission of biodiesel blends may be due to their more complete oxidation as compared to diesel. Some of CO produced during combustion of biodiesel might converted into carbon dioxide by taking up of extra oxygen in biodiesel and thus formation CO reduced.

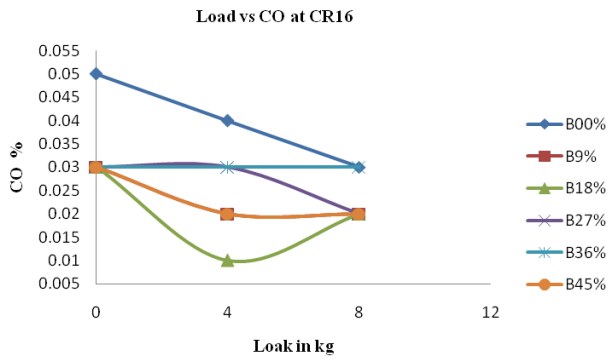

Fig: 11 Variation of carbon monoxide with load.

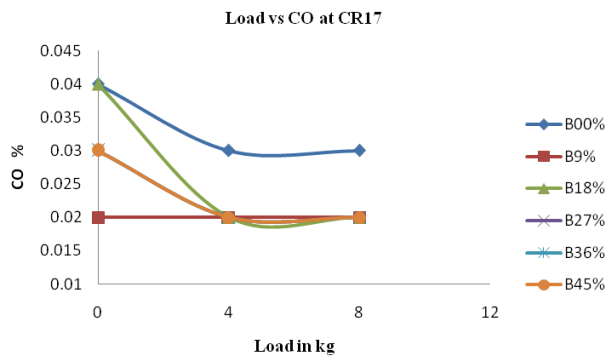

Fig:12 Variation of carbon monoxide with load.

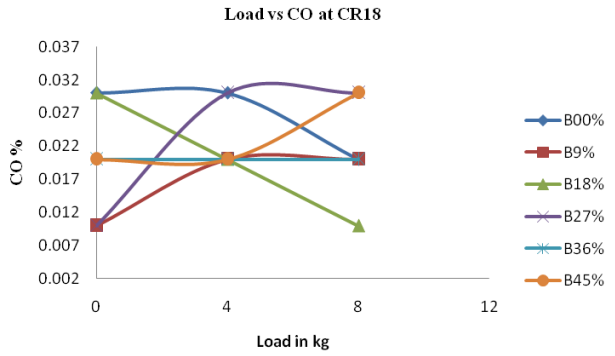

Fig:13 Variation of carbon monoxide with load.

5.3 Nitrogen Oxide.

Fig. 14, 15 and 16 show that variation of nitrogen oxide with load for different blends. NO emission for biodiesel blends is slightly lower than pure diesel fuel at higher load condition but it close at low load.

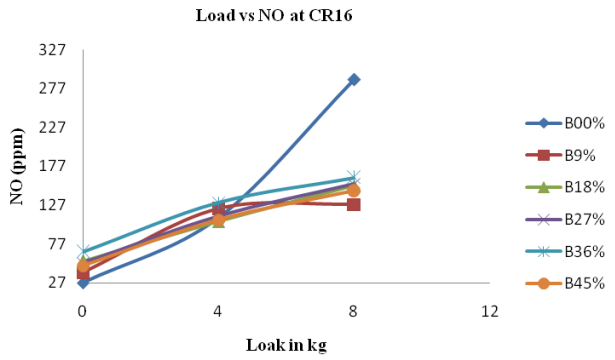

Fig: 14 Variation of nitrogen oxide with load.

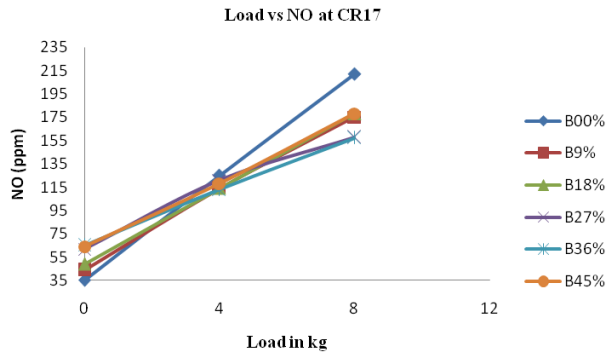

Fig: 15 Variation of nitrogen oxide with load.

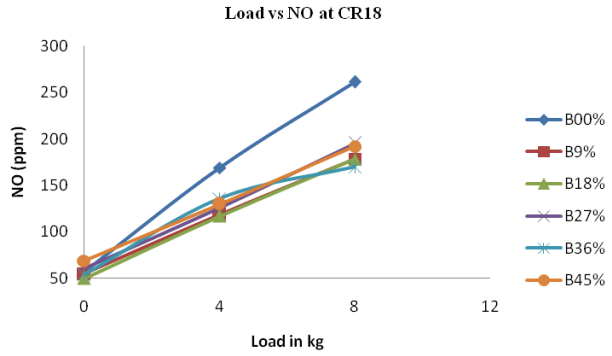

Fig: 16 Variation of nitrogen oxide with load.

Conclusion.

From the experimental result the following conclusions were made.

1. Brake power of waste turmeric oil is very close to standard diesel at all load condition.
2. With increases in load on engine, mechanical efficiency increases at all compression ratio.
3. There is reduction in hydrocarbon emission for all the blends of waste turmeric oil as compared with diesel fuel at all compression ratio and loads.
4. Carbon monoxide decreases for all the biodiesel blends than diesel fuel.
5. Nitrogen oxide emission for biodiesel blends close to diesel fuel but it increases with load.

From the above observation , it has been found that the waste turmeric oil blends show the better performance and emission characteristics compare to diesel fuel all compression ratio and full load condition.

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HP31710010-Experimental Investigation of Performance and Emission Characteristics of Diesel Fuelled With Mexicana Methyl Ester

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Abstract

The experimental work investigates the performance and emission parameters of single cylinder variable compression ratio (VCR) engine coupled to eddy current dynamometer fuelled with Mexicana biodiesel with addition of diethyl ether as additives and blend with diesel fuel under different load condition at compression ratio 16, 17 and 18.The blend of biodiesel and diesel used where D100, D90B5DEE5, D85B10DEE5, D80B15DEE5, D75B20DEE5, D70B25DEE5, D65B30DEE5.Use of diethyl ether as additives to improves the performance and emission parameters of the engine. All the Mexicana biodiesel blends shows that reduction in carbon monoxide, nitrogen oxide and hydrocarbon emission compare with diesel fuel and carbon dioxide emission is higher as compared to diesel fuel. The performance parameter like brake power and brake thermal efficiency close to diesel. There are slight increases in smoke opacity of Mexicana biodiesel higher than diesel fuel.

Keywords:*Mexicana biodiesel, additives- diethyl ether, engine performance, emission.*

1. Introduction

India's energy security would remain exposed until alternative fuels to substitute petro-based fuels produced renewable feedstock. Biodiesel is a renewable and eco-friendly alternative diesel fuel for diesel engine. The compression ignition engines are widely used due to its reliable operation and economy. Using optimized blend of biodiesel and diesel can help reduce some significant percentage of the world dependence on fossil fuels without modification of CI Engine, As the petroleum reserves are depleting at a faster rate due to the growth of population and the subsequent energy utilization, an urgent need for search for a renewable alternative fuel arise.

S.I mtenan et.al [3] described the emission and performance characteristics of the palm biodiesel with additives. The better blends contained 80% diesel, 15% palm biodiesel and 5% additive. Use of additives continuously improved brake power, reduced BSFC and improved BTE. Diethyl ether alternative potential additive can be formed from ethanol. It has got a very high oxygen content, high cetane number, low auto ignition temperature, high miscibility in diesel and broad flammability limits. NOx emission is increases by using a biodiesel blend, to overcome this problem to use of additives. Observed emission parameters additives showed relatively a good result of CO and NO emission.

Channapattana et.al [4] evaluated the emission and performance of nonedible honne oil is used on the DICI VCR engine. At 18 CR break thermal efficiency less than diesel and brake specific fuel consumption higher than that diesel fuel. Reduction in carbon monoxide and hydrocarbon as compare diesel fuel at compression ratio 18.Honne biodiesel at a CR18 results in lowest emissions but increases NOx emissions.

Panwar et.al [5] studied that the performance and emission parameters of castor methyl ester (CME) oil. Castor methyl ester (CME) blends showed performance characteristics close to diesel. The NOx emission of CME is similar to diesel fuel at lower loads and slightly higher at full loads.

As per literature survey, It is clear that much of the work has been done on the various biofuels such as Mahua biodiesel, Castor seed oil, Waste cooking oil methyl ester, Rapeseed plant oil, palm biodiesel, Calophyllum Inophyllum oil, Jatropha biodiesel, Palm Oil, Honne oil Methyl Ester, pinnai oil etc. In terms of finding various engine parameters like BP, ME, Brake thermal efficiency, Brake specific fuel consumption and its emissions. So, it has been found that less work has been carried out on the Mexicana biofuel, so it has been taken as an opportunity to explore its compatibility with diesel engine by undertaking experiments on various engine parameters.

India is an agrarian nation and has rich plant biodiversity which can carry the growth of biodiesel. These different species of forest-based seeds are reported as the possible resource of biodiesel feedstock in India. The current oil is unsatisfactory to meet the demand for raw material on great amount production of biodiesel. Mexicana Oil preferred for the current work of investigational research of performance and emission characteristic of VCR diesel engine. Mexicana oil is non-edible oil which is used for the production of biodiesel. These found on the road side, waste land and field. Mexicana plant belongs to

poppy family. It is commonly known as satyanashi and shialkanta in India.

In the present work investigate the performance and emission parameters of the Mexicana oil with using diethyl ether is additives. A diethyl ether additive is used to improve the performance of the biodiesel. Engine performance and emission analysis by using Mexicana biodiesel with CR 16, 17 and 18 with different load compare with diesel fuel. The fuel sample designation of different composition blends of diesel and Mexicana Methyl Ester with diethyl ether as an additive which is shown in Table I.

Table I: Fuel Sample Designation of Different Composition.

Fuel Sample Designation	Composition
D100	100% Diesel Fuel
D90B5DEE5	90%Diesel + 5% Mexicana methyl ester +5% diethyl ether
D85B10DEE5	85%Diesel + 10% Mexicana methyl ester +5% diethyl ether
D80B15DEE5	80%Diesel + 15% Mexicana methyl ester + 5% diethyl ether
D75B20DEE5	75%Diesel + 20% Mexicana methyl ester + 5% diethyl ether
D70B25DEE5	70%Diesel + 25% Mexicana methyl ester + 5% diethyl ether
D65B30DEE5	65%Diesel + 30% Mexicana methyl ester + 5% diethyl ether

2. Methodology

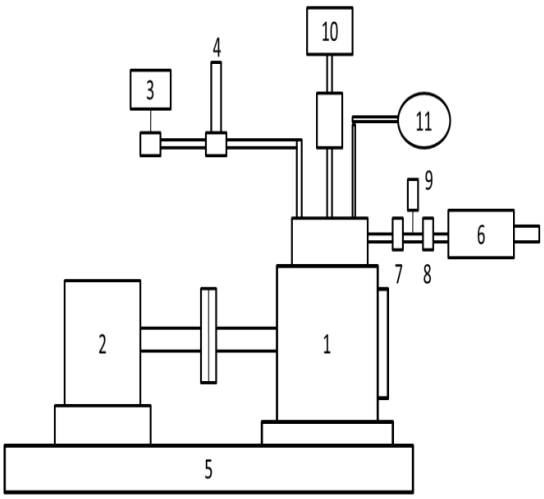
The Mexicana oil is first filtered to remove solid material then it is preheated at 110^{0}C for 30 min to remove moisture. The methyl ester is produced by chemically reacting Mexicana oil with an alcohol (methyl), in the presence of catalyst. A two stage process is used for the transesterification of Mexicana oil. The first stage (acid catalyzed) of the process is to reduce the free fatty acids (FFA) content in Mexicana oil by esterification with methanol (99% pure) and acid catalyst sulfuric acid (98% pure) in one hour time at 57^{0}C in a closed reactor vessel. The major obstacle to acid catalyzed esterification for FFA is the water formation. Water can prevent the conversion reaction of FFA to esters from going to completion. After dewatering the esterified oil was fed to the transesterification process. The catalyst used is typically sodium hydroxide (NaOH) with 0.5% of total quantity of oil mass. It is dissolved in the 10% of distilled methanol (CH3OH) using a standard agitator at 700 rpm speed for 20 minutes. When the methoxide was added to oil, the system was closed to prevent the loss of alcohol as well as to prevent the moisture. The temperature of reaction mix was maintained at 60 to 65^{0}C (that is near to the boiling point of methyl alcohol) to speed up the reaction. The recommended reaction time is 70 min. The stirring speed is maintained at 560-700rpm. Excess alcohol is normally used to ensure total conversion of the fat or oil to its esters. The reaction mixture was taken each after 20 min. for analysis of FFA. After the confirmation of completion of methyl ester formation, the heating was stopped and the products were cooled and transferred to separating funnel. Once the reaction is complete, it is allowed for settling for 8-10 hours in separating funnel. At this stage two major products obtained that are glycerin and biodiesel. Once the glycerin and biodiesel phases were been separated, the excess alcohol in each phase was removed by distillation. Once separated from the glycerin and alcohol removal, the crude biodiesel was purified by washing gently with warm water to remove residual catalyst or soaps. This is normally the end of the production process to remove water present in the biodiesel which results in a clear amber-yellow liquid with a viscosity similar to petro diesel. In some systems the biodiesel is distilled in an additional step to remove small amounts of color bodies to produce a colorless biodiesel. The properties of diesel and different Mexicana biodiesel blends were determined as per the ASTM6751 standards. The properties of Mexicana biodiesel and diesel are represented on table II.

Table II: Properties of Mexicana Biodiesel and Diesel

Sr. No.	properties	Ref. std. ASTM 6751	Reference Limit	Diesel D100 %	Mexicana Biodiesel B100%
1	Density gm/cc	D1448	0.800-0.900	0.830	0.876
2	Calorific value MJ/Kg	D6751	34-45	42.50	38.50
3	Cetane no.	D613	41-55	49.00	50.70
4	Viscosity mm2/sec	D445	3-6	2.700	5.2
5	Flash point ^{0}C	D93	_	64	149
6	Fire point ^{0}C	D93	_	71	158

3. Experimental Setup

The setup consists of single cylinder, four stroke, VCR (Variable Compression Ratio) Diesel engine connected to eddy current type dynamometer for loading. The compression ratio can be changed without stopping the engine and without altering the combustion chamber geometry by specially designed tilting cylinder block arrangement.



1. Test Engine, 2. Dynamometer, 3. Fuel Tank, 4. Fuel Burette, 5. Test Bed, 6. Silencer,
7. Smoke meter, 8. HC/CO/Nox/CO2/O2 Analyzer, 9. Exhaust Temperature sensor,
10. Air Flow Meter, 11. Stop Watch

Fig.1 Engine Setup

The detail specification of engine is given in above Table II. To obtain the baseline parameters, the engine was first operated on diesel fuel. Performance and emission tests are carried out on the diesel engine using Mexicana biodiesel using diethyl ether additives, and its various blends.

Table II: Engine Specification

Sr.No.	Description	Specification
1.	Model and Make	Kirloskar and TV1
2.	No. of cylinder	Single
3.	Cycle	Four stroke
4.	Bore and stroke	87.5 mm and 110 mm
5.	Rated Power	3.5 kW at 1500 rpm
6.	Compression ratio	17.5, Modified to work in range of 12 to 18
7.	Dynamometer	Eddy current, water cooled, with loading unit
8.	Cubic capacity	0.661 liters
9.	Software	"EnginesoftLV"Engine performance analysis software

4. Result and Discussion:

Engine performance and Emission

Performance parameters such as BP and BTHE and emission parameters such as HC, CO and NO has analysis by using Mexicana biodiesel with diethyl ether as additives with compression ratio 16, 17 and 18 with different load(0,4 and 8 kg) compare with diesel fuel.

1. Brake Power (BP)

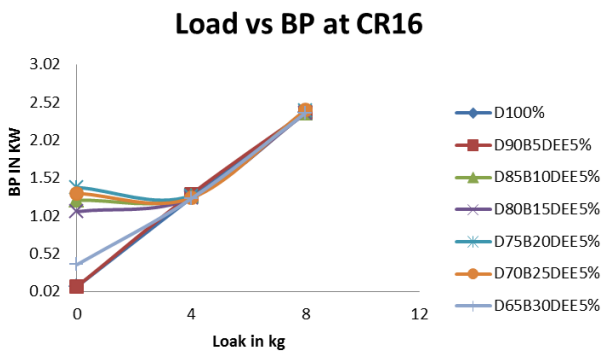


Fig 2: Variation of BP (KW) Vs. Load

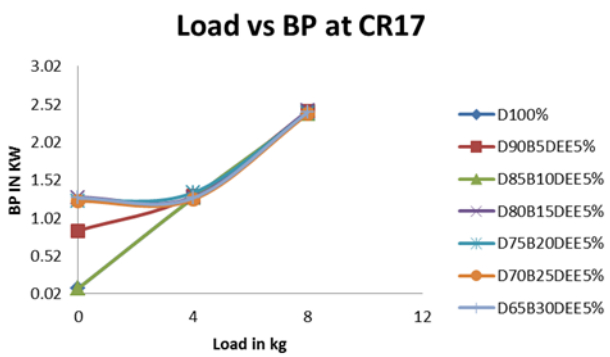


Fig 3: Variation of BP (KW) Vs. Load

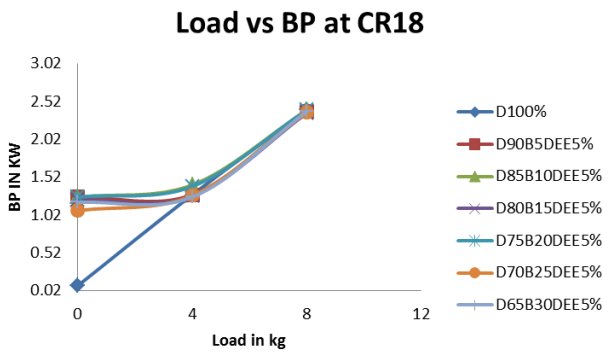


Fig 4: Variation of BP (KW) Vs. Load

Figure 2, 3 and 4 represent the variation of brake power versus load with compression ratio 16, 17 and 18 respectively.Brake power increases at increasing load with compression ratio 16, 17 and 18. Brake power of Mexicana biodiesel blend and diesel is close to each other due to enhanced combustion and high heat content in the biodiesel.

2. Brake Thermal Efficiency (BTHE):

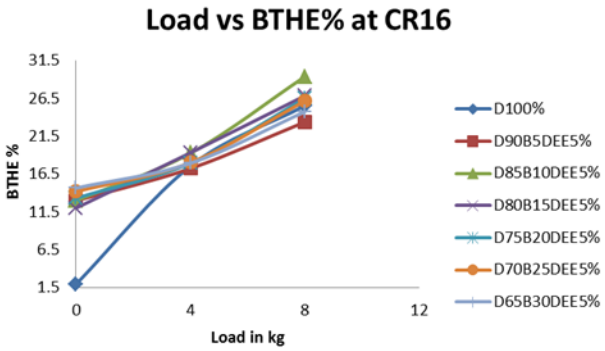


Fig 5: Variation of BTHE %Vs. Load

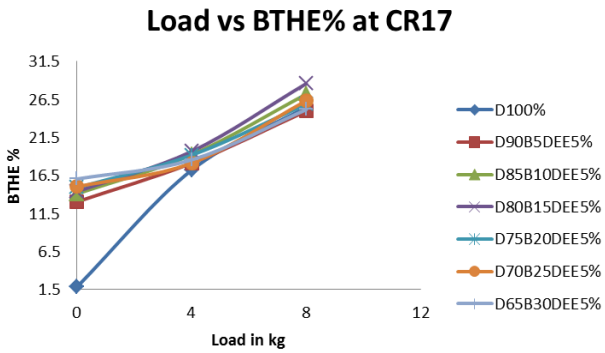


Fig 6: Variation of BTHE %vs. Load

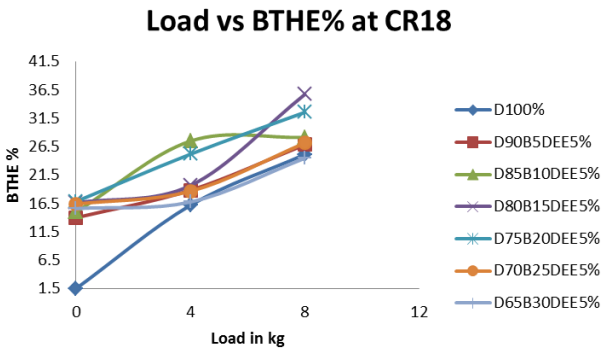


Fig 7: Variation of BTHE % vs. Load

From above figure 5,6 and 7 it is observed that Brake thermal efficiency of the biodiesel increases with increase in load. For compression ratio 16 and 17 brake thermal efficiency is close to diesel fuel, but compression ratio 18 brake thermal efficiency of biodiesel is high as compared to diesel fuel. Brake thermal efficiency of the blend D80 B15 DEE 5% and D75 B20 DEE 5% is high as compared to diesel with compression ratio 18.

3. Hydrocarbon Emission (HC) :

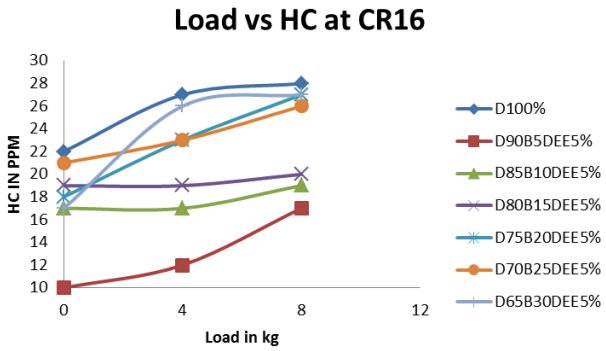


Fig 8: Variation of Hydrocarbon Emission Vs. Load

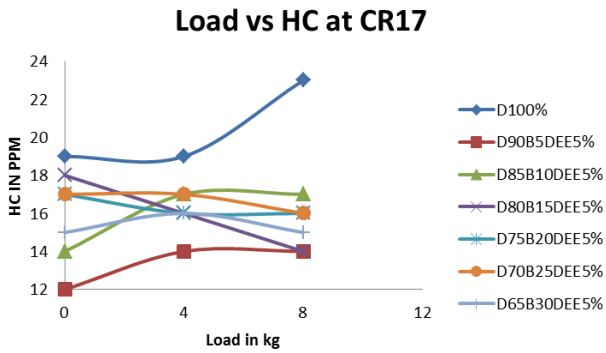


Fig 9: Variation of Hydrocarbon Emission Vs. Load

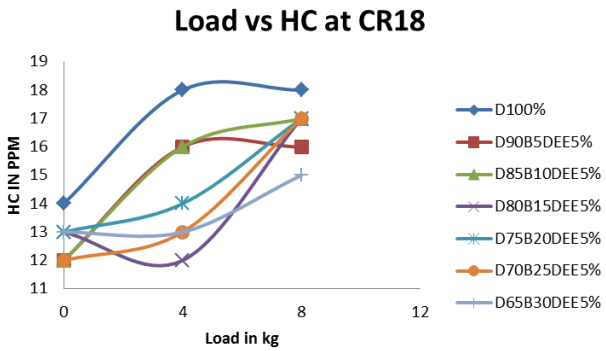


Fig 10: Variation of Hydrocarbon Emission Vs. Load

Above figure 8, 9 and 10 represents that the hydrocarbon emissions verses load with compression ratio16, 17 and 18. It is observed that hydrocarbon emissions of the all Mexicana blend with additives is decreases as compared to diesel fuel with compression ratio 16, 17 and 18. It is also seen that there is decrease in Hydrocarbon emission with increase in proportion of blend.

4. Carbon Monoxide Emission (CO) :

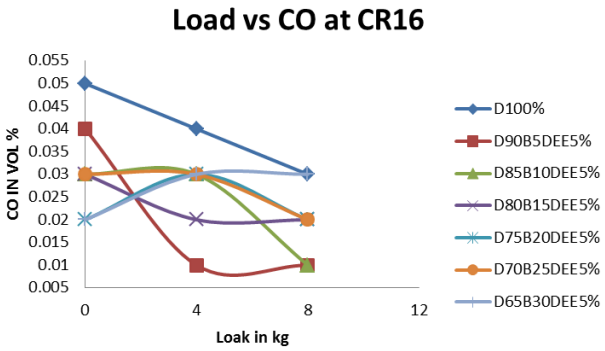


Fig 11: Variation of Carbon Monoxide Emission Vs. Load

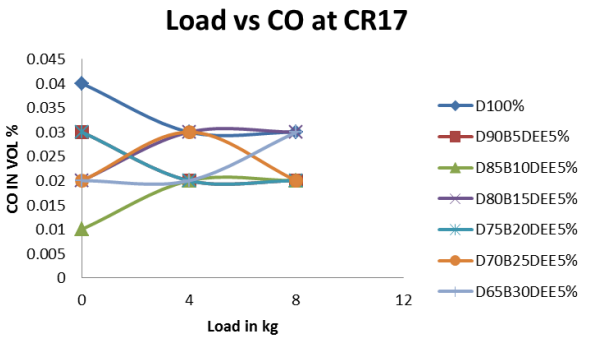


Fig 12: Variation of Carbon Monoxide Emission Vs. Load

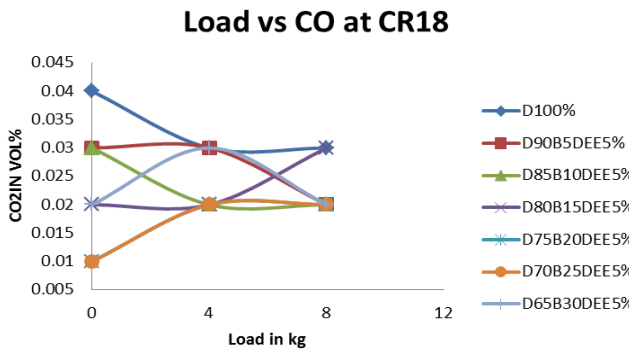


Fig 13: Variation of Carbon Monoxide Emission Vs. Load

From Figure 11, 12 and 13, it is observed that the carbon monoxide emission is decreasing with increase in compression ratio. This is due to the improved combustion rate at high air temperature in the cylinder. Carbon monoxide of blend D80 B15 DEE 5% and D75 B20 DEE 5% is close to diesel fuel.

5. Nitrogen Oxide Emission (NO):

From Figure 14, 15 and 16 represents the emission of nitrogen oxide versus load for different compression ratios. It is observed that nitrogen oxide emission of diesel fuel increases with increase in load, but all blend of the Mexicana oil is decreases as compared to diesel fuel.

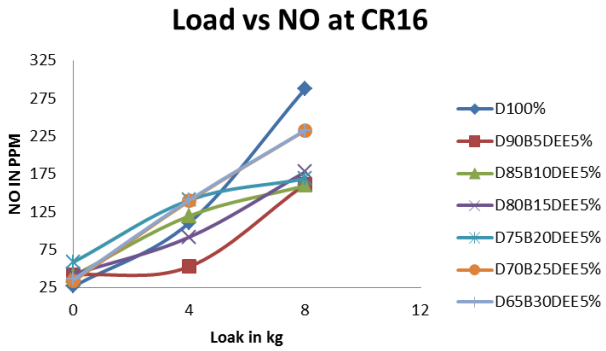


Fig 14: Variation of Nitrogen Oxide Emission Vs. Load

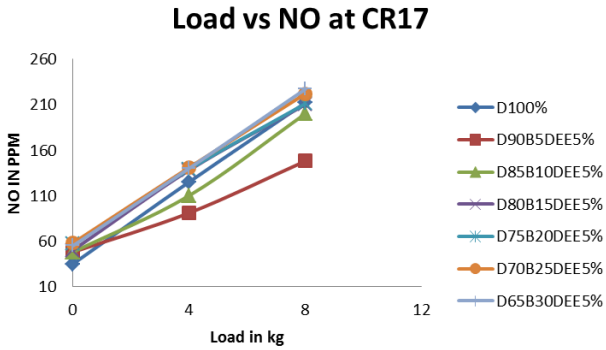


Fig 15: Variation of Nitrogen Oxide Emission Vs. Load

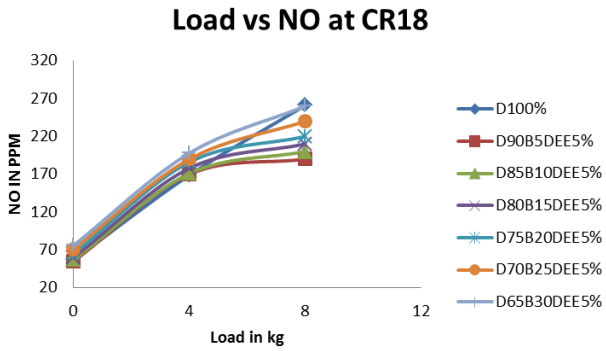


Fig 16: Variation of Nitrogen Oxide Emission Vs. Load

For the higher fuel bound oxygen in biodiesel nitrogen oxide emission is increases but addition of the diethyl ether which acquired high centane number, lowers auto ignition temperature and high oxygen content. Therefore it is observed that nitrogen oxide emission is decreases.

Conclusion:

From the above paper the following conclusions can be made in this work as:-

1. Brake power of Mexicana biodiesel blend and diesel is close to each other due to enhanced combustion and high heat content in the biodiesel.
2. Brake thermal efficiency of the blend D80 B15 DEE 5% and D75 B20 DEE 5% is high as compared to diesel with compression ratio 18.

3. It is observed that hydrocarbon emissions of the all Mexicana blend with additives is decreases as compared to diesel fuel with compression ratio 16, 17 and 18.
4. Carbon monoxide emission is decreasing with increase in compression ratio. This is due to the improved combustion rate at high air temperature in the cylinder.
5. Nitrogen oxide emission of diesel fuel increases with increase in load, but all blend of the Mexicana oil is decreases as compared to diesel fuel.

By addition of diethyl ether additives to improve the engine performance and emission characteristics of Mexicana oil because of by adding the diethyl ether in biodiesel which improved high centane number, lowers auto ignition temperature and high oxygen content.

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