Performance and emissions of CI engine by using blended fuel

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ABSTRACT

Increase in energy demand, stringent emissions norms and depletion of oil resources led to find alternate fuels for internal combustion engines. Many alternate fuels like alcohols, bio diesel, LPG, CNG,, etc.it have been already commercialized in the transport sector. In the context, Pyrolysis of solid waste is currently fuel receiving renewed interest. The disposal of waste tyres can be simplified to some extent by pyrolysis for getting oil. This pyrolysis oil can be blended with diesel and used as an alternate fuel for CI engines.In the present work the performance characteristics and emissions are evaluated on a single cylinder 4 stroke diesel engine fuelling with 20%, 40%,60% and 75% pyrolysis oil-diesel blends by volume. These results are compared with the results obtained when fuelled with pure diesel oil.

***Keywords**: Brake thermal efficiency[1], Diesel engine[2], Hartridge smoke unit (Hsu) emissio [2], Hydrocarbon emission [3].*

INTRODUCTION

Compression ignition engines are employed particularly in the field of heavy transportation and agriculture on account of their higher thermal efficiency and durability. However, diesel engines are the major contributors of oxides of nitrogen and particulate emissions. Hence more stringent norms are imposed on exhaust emissions. Following the global energy crisis in the 1970s and the increasingly stringent emission norms, the search for alternative renewable fuels has intensified.

1.1. Pyrolysis

Pyrolysis, in general, is the thermal degradation in the absence of oxygen agent. Relatively low temperature is employed (500°C – 800°C). Three products are usually produced: gas, pyrolysis oil and char, the relative proportions of which depends upon pyrolysis method, characteristics of biomass and the reaction parameters. Fast or flash pyrolysis is used to maximize either gas or liquid products according to the temperature employed.More than 30 major pyrolysis products have been proposed, designed, patented, licensed or built over the years but none have yet been commercially successful. The limiting factors are capital cost of the facilities and the products from pyrolysis do not have sufficient value in the market. Developments of less costly techniques or processes for higher value added products would enable pyrolysis to become a profitable alternative for waste tyre recycling. In addition to this pyrolysis is known for low emission to the environment.

1.2. **Experimental setup**

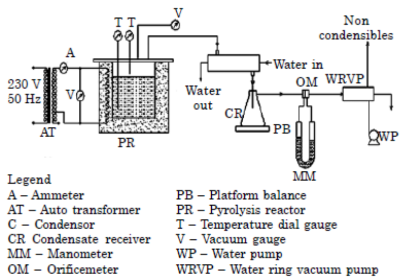

Figure 1. Schematic of experimental set up of 2 kg batch pyrolyser

The used tyres are mechanically processed in order to separate their components, which are rubber, steel and fabric. The rubber is taken over and cut into small pieces of approximately 1–inch length, 0.25*0.25-width size cleaned off from dust and other impurities. The rubber is subjected to pyrolysis; it undergoes thermal decomposition under the absence of oxygen in the reactor. Figure 1 illustrates the details of experimental setup emphasized to carry out the work. The system consists of a cylindrical M.S vessel, lined inside with insulating material and with a cavity volume of about 2 it in which the raw material for pyrolysis is charged. The reactor is heated electrically using 230V AC power supply through an autotransformer. Weight of pyrolysis oil received from the condenser, monitored by a platform balance as function of time. The reactor is closed by a flat plate with provision for insertion of dial gauges for measurement of bed, vapour temperatures. The reactor was connected to a water ring vacuum pump for generating vacuum in the reactor. The pyrolysis products of hydrocarbon are fed to a water-cooled shell and tube condenser where the condensed heavy hydrocarbon as pyrolysis oil was collected in a receiving vessel. Selected properties of pyrolysis oil obtained under different process conditions such as heat load, vacuum level, etc.were evaluated and presented in the Table 1.

Table 1. properties of tyre pyrolysis:

Properties	**Diesel**	**Tyre Pyrolysis oil**
Kinematic viscosity at 40 ^{0}C (cSt)	3.52	3.2
Density at 15 [degrees] C (kg/m^3)	830	840
Flash point (^{0}C)	49	60
Calorific value (kJ/kg)	42000	40838.6
Total sulphur (% by mass)	0.01	0.06
Ash (% by mass)	0.01	0.14

2.EXPERIMENTAL SETUP & PROCEDURES

2.1 Introduction

The details of the experimental set up are presented in this chapter the alternationsmade to the instrumentation are also described .The experimental setup is fabricated to fulfill the objective of the present work. The various components of the experimental set up including modification are presented in this chapter.

2.2 Experimental set up

The experimental set up consists of engine, an alternator, top load system, fuel tank along with immersion heater, exhaust gas measuring digital device and manometer.

Engine:

The engine which is supplied by M/s Alimgar Company. The engine is single cylinder vertical type four stroke, Air-coo1ed, compression ignition engine. The engine is self governed type whose specifications are given in Appendix 1.is used in the present work.

2.3 Reasons for selecting the engine

The above engine is one of the extensively used engines in industrial sector in India. This engine can with stand the peak pressures encountered because of its original high compression ratio. Further, the necessary modifications on the cylinder head and piston crown can be easily carried out in this type of engine. Hence this engine is selected for the present project work.

Dynamometer
The engine is coupled to a generated type electrical dynamometer which is provided for loading the engine.

Fuel injection pump
The pump is driven by consuming some part of the power produced by the engine; it will provide the required pressure to the injector. The pump is BOSCH fuel injection pump.

Fuel injector (BOSCH)
A cross sectional view of a typical BOSCH fuel injector
The injector assembly consists of
- i. A needed valve
- ii. A compression spring
- iii. A nozzle
- iv. An injector body

U-tube manometer
The one of end of the U-tube manometer is connected to the orifice of the air tank and the other end is exposed to the atmosphere, the manometer liquid used is water.

Digital thermometer
It consists of a temperature sensing element connected to the electronic digital display which is operated by battery

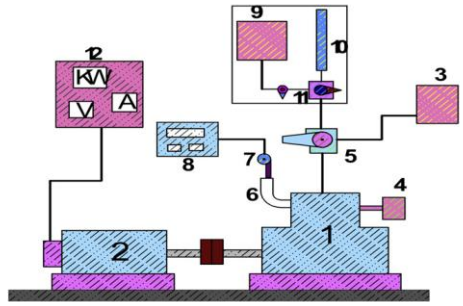

Figure 2.expermental set up

2.4 various parts of expermental setup
1. Alamgir engine
2alternator
3. Diesel tank
4. Air filter
5. Three way valve
6. Exhaust pipe
7. Probe
8. Exhaust gas analyser
9. Alternative fuel tank
10. Burette
11. Three way valve
12. Control panel

2.5 Instructions:
1. Measurement of brake power:
The power developed by the engine is measured by using electrical dynamometer. The pump is run by using the power developed by the engine. The total power is obtained by adding pump power to the product of voltage and current.
2. Measurement of fuel:
The fuel flow is measured by volume through a burette tube which isfixed between fuel tank and fuel pump. A T-joint prepared and one side of it is connected to the fuel measuring tube. The remaining two sides of the joints are connected to the fuel tank and the fuel pump respectively. Fuel flow is measured by nothing the time taken for 10cc of fuel consumption by stop watch.
3. Measurement of air flow:

Air flow is measured by using a viscous flow air meter. A paper element filter is an integral part of the meter .The meter consists of an orifice, the pressure drop across the orifice is measured by manometer, is ensured that there are no leakages in the connecting tubing.

4. Load on engine:

Initially fuel tank and auxiliary fuel tanks are filled with right amount of required fuel. The instruments such as NOx meter and CO/HC analyzer are connected to the exhaust pipe.The engine is started and allowed to run for 20 minutes to attain steady state condition.

5. Measurement of emissions:

Engine emissions are measured by using exhaust gas analyzer.

6. Measurement of exhaust gas temperature:

The temperature of exhaust gas is measured by using digital electronicdevices. It gives the exhaust gas temperature directly.

2.6 Experimental Procedure

Before starting the engine, the fuel injector is separated from the fuel system. it is clamped on the fuel injection pressure tested and operates the tester pump. Observe the pressure reading from the dial. At which the injector starts spraying. In order to achieve the required pressure by adjusting the screw provided at the top of the injector .This procedure is repeated for obtaining the various required pressures.

As first said, diesel alone is allowed to run the engine for about 30 min, so that it gets warmed up and steady running conditions are attained. Before starting the engine, the lubricating oil level in the engine is checked and itis also ensured that all moving and rotating parts are lubricated.

The various steps involved in the setting of the experiments are explained below

1. The Experiments were carried out after installation of the engine
2. The injection pressure is set at 200 bar for the entire test.
3. Precautions were taken, before starting the experiment.
4. Always the engine was started with no load condition
5. The engine was started at no load condition and allowed to work for at least 10 minutes to stabilize.
6. The readings such as fuel consumption, spring balance reading, cooling water flow rate, manometer reading etc., were taken as per the observation table.
7. The load on the engine was increased by 20% of FULL Load using the engine controls and the readings were taken as shown in the tables.
8. Step 3 was repeated for different loads from no load to full load.
9. After completion of test, the load on the engine was completely relieved and then the engine was stopped.
10. The results were calculated as follows.

The above experiment is repeated for various loads on the engine.The experimental procedure is similar as foresaid. While starting the engine, the fuel tank is filled in required fuel proportions up to its capacity. The engine is allowed to run for 20 min, for steady state conditions, before load is performed.

We have not been observed any separation in the blend of 50% Fish oil & 50%diesel. We have heated the blend about 70°C in order to reduce the viscosity for easy flowing in the tubes, better injection and atomization in the cylinder the viscosity is nearly equal to the viscosity of diesel.

Finally, the engine is run by blend (200atm) at various injection pressures the corresponding observations are noted.

The test is carried on the Alimgar Engine for the following fuel blends:

1. 100% Diesel
2.20% Tyre Oil + 80% Diesel
3.40% Tyre Oil + 60% Diesel
4.60% Tyre Oil + 40% Diesel

5.75% Tyre Oil + 25% Diesel

3. CALCULATIONS AND GRAPHS

The parameters that are determined at differ rent loads are as fallows

1. Brake Power, B.P = $\frac{VI\cos\emptyset}{\eta_{tran} \times \eta_{gen} \times 1000}$ kw

Where,

V = voltage, volts

A = current, amperes

Cos Ø = Power factor = 1

η_{tran} = Transmission Efficiency = 0.98

η_{gen} = Generator Efficiency = 0.9

Brake Power, B.P =

2. T.F.C = $\frac{20 \times 0.85 \times 3600}{t \times 1000}$ Kg/h

Where,

T.F.C = Total Fuel Consumption, Kg/h

Specific gravity of diesel = 0.85

t = Time taken for 20 c.c fuel, seconds

T.F.C =

3. Brake Specific Fuel Consumption, bsfc = $\frac{T.F.C}{B.P}$ Kg/kwh

Brake Specific Fuel Consumption, bsfc = Kg/kwh

4. Heat Input = T.F.C X C.V kW

Where,

C.V = Calorific Value of Fuel, kJ/kg k

Hear Input =

5. Frictional Power, F.P = kW (from graph by William's line method)

6. Indicated Power = B.P + F.P

Indicated Power = kW

7. Mechanical efficiency = $\frac{B.P}{I.P}$ x 100 %

= –x 100 %

Mechanical Effciency, η_{mech} = %

8. Brake thermal efficiency = $\frac{B.P}{\text{Heat Input}}$ x 100 %

= –x 100 %

Brake thermal efficiency, η_{bth} =

9. Indicated thermal efficiency = $\frac{I.P}{\text{Heat Input}}$ x 100 %

Indicated thermal efficiency, η_{bth}=

10. Brake Mean Effective Pressure, bmep = $\frac{B.P \times 60}{L \times A \times n \times k}$

Where L = lenth of the stroke, m n = speed of the engine = 1500/2

A = Area of the cylinder, m² k = no. of cylinders

$$= \frac{3.1188 \times 60}{0.116 \times \frac{\pi}{4} \times (0.102)^2 \times \frac{1500}{2} \times 1}$$

Brake Mean Effective Pressure, bmep =

11. Indicated Mean Effective Pressure, Imep = $\frac{I.P \times 60}{L \times A \times n \times k}$

$$= \frac{4.6688 \times 60}{0.116 \times \frac{\pi}{4} \times (0.102)^2 \times \frac{1500}{2} \times 1}$$

Indicated Mean Effective Pressure =

12. Volumetric efficiency, η_{vol}= $\frac{\text{actual volume flow rate of air}}{\text{the rate at which volume is displaced}}$ x 100 %

$$= \frac{\text{area of inlet pipex velocity of air}}{\left[\begin{matrix}\text{are of}\\\text{the cylinder}\end{matrix}\right] \times \left[\begin{matrix}\text{length of}\\\text{the stroke}\end{matrix}\right] \times \left[\begin{matrix}\text{revolutions}\\\text{per second}\end{matrix}\right]} \times 100\%$$

$$=\frac{\frac{\pi}{4}x(0.035)^2 \text{ x } 9.27}{\frac{\pi}{4}x(0.102)^2 \text{ x } 0.116 \text{ x } \frac{1500}{2 \text{ x } 60}} \text{ x } 100\%$$

3.1 MODEL CALCULATIONS

Sample Calculations for 100% diesel

1. Brake Power, B.P $=\frac{VIcos\emptyset}{\eta_{tran} \text{ x } \eta_{gen} \text{x } 1000}$ kw

Where,

V = voltage, volts

A = current, amperes

Cos Ø = Power factor = 1

η_{tran} = Transmission Efficiency = 0.98

η_{gen} = Generator Efficiency = 0.9

$= \frac{230 \text{ x } 12 \text{ x } 1}{0.98 \text{ x } 0.9 \text{ x } 1000}$

Brake Power, B.P =3.118

2. T.F.C $= \frac{20 \text{ x } 0.85 \text{ x } 3600}{t \text{ x } 1000}$ kg/h

Where,

T.F.C = Total Fuel Consumption, Kg/h

Specific gravity of diesel = 0.85

t = Time taken for 20 c.c fuel, seconds

$= \frac{20 \text{ x } 0.85 \text{ x } 3600}{39.65 \text{ x } 1000}$ kg/h

T.F.C =1.5427 kg/h

3. Brake Specific Fuel Consumption, bsfc = $\frac{T.F.C}{B.P}$ Kg/kwh

=1.5427

Brake Specific Fuel Consumption, bsfc =.4946 kg/kWh

4. Heat Input = T.F.C X C.V kW

Where,

C.V = Calorific Value of Fuel, kj/kg k

= 4.2857 x 42000

Hear Input =17.999kW

5. Frictional Power, F.P = 1.5 kw (from graph by William's line method)

6. Indicated Power = B.P + F.P

= 3.1188 + 1.5

Indicated Power = 4.6688 kW

7. Mechanical efficiency = $\frac{B.P}{I.P}$x 100 %

= $\frac{3.118}{4.668}$x 100 %

Mechanical Effciency, η_{mech} = 66.78%

8. Brake thermal efficiency = $\frac{B.P}{\text{Heat Input}}$ x 100 %

= $\frac{3.118}{17.999}$x 100 %

Brake thermal efficiency, η_{bth} = 17.32%

9. Indicated thermal efficiency = $\frac{I.P}{\text{Heat Input}}$ x 100 %

= $\frac{4.6688}{17.9999}$x 100 %

Indicated thermal efficiency, η_{bth}=25.93%

10. Brake Mean Effective Pressure, bmep = $\frac{B.P \text{ x } 60}{L \text{ x } A \text{ x } n \text{ x } k}$

Where L = lenth of the stroke, m n = speed of the engine = 1500/2

A = Area of the cylinder, m^2 k = no. of cylinders

$$= \frac{3.1188 \times 60}{0.116 \times \frac{\pi}{4} x(0.102)^2 \times \frac{1500}{2} \times 1}$$

Brake Mean Effective Pressure, bmep $=263.26 kN/m^2$

11. Indicated Mean Effective Pressure, Imep $= \frac{I.P \times 60}{L \times A \times n \times k}$

$$= \frac{4.6688 \times 60}{0.116 \times \frac{\pi}{4} x(0.102)^2 \times \frac{1500}{2} \times 1}$$

Indicated Mean Effective Pressure $=394.1 kN/m^2$

12. Volumetric efficiency, $\eta_{vol}= \frac{\text{actual volume flow rate of air}}{\text{the rate at which volume is displaced}} \times 100\ \%$

$$= \frac{\text{area of inlet pipex velocity of air}}{\left[\begin{matrix}\text{are of}\\ \text{the cylinder}\end{matrix}\right] \times \left[\begin{matrix}\text{length of}\\ \text{the stroke}\end{matrix}\right] \times \left[\begin{matrix}\text{revolutions}\\ \text{per second}\end{matrix}\right]} \times 100\%$$

$$= \frac{\frac{\pi}{4} x(0.035)^2 \times 9.27}{\frac{\pi}{4} x(0.102)^2 \times 0.116 \times \frac{1500}{2 \times 60}} \times 100\%$$

Volumetric efficiency, η_{vol} $=65.77\%$

4.RESULTS AND DISCUSSIONS

Experiments were conducted when the engine was fuelled with tyre pyrolysis oil and their blends with diesel in proportions of 20:80, 40:60, 60:40 and 75:25 (by volume) which are generally called as T-20, T-40, T-60 and T-75 respectively. The experiment covered a range of loads.

The performance of the engine was evaluated in terms of brake specific fuel consumption, brake thermal efficiency and volumetric efficiency. The emission characteristics of the engine were studied in terms exhaust gas temperature, concentration of HC, CO, CO_2 and O_2. The results obtained for tyre pyrolysis oil and their blends with diesel were compared with the results of diesel.

4.1 brake specific fuel consumption:

The result for the variations in the brake specific fuel consumption (BSFC) is presented in the fig.3. For all the fuels the BSFC falls with increasing load. The differences of BSFC are very small when using different fuels. The maximum BSFC values are 0.91 kg/kW hr for diesel, 0.77 kg/kW hr for T-20, 0.975 kg/kW hr for T-40, 1.039 kg/kW hr for T-60, 1.112kg/kW hr and for T-75, 1.095 kg/kW hr. The higher BSFC values in the case of pure tyre pyrolysis oil are due to their low energy content.

6.2 brake thermal efficiency:

The variation of brake thermal efficiency with load is shown in fig. 4. Brake thermal efficiency gives an idea of the output generated by the engine with respect to heat supplied in the form of fuel. For all the fuels the brake thermal efficiency increases with load. The brake thermal efficiency values at full load are 12.54% for diesel, 16.42% for T-20, 16.6% for T-40, 16.50 for T-60 and for T-75, 16.57%. The brake thermal efficiencies of T-20 are very close to the brake thermal efficiencies of diesel at all loads. This may be due to their low heat input requirement for higher power output at a given load.

4.3 volumetric efficiency:

The fig.5. shows the variation of volumetric efficiency with load for various blends. Volumetric efficiency is a measure of success with which the air supply, and thus the charge, is inducted in to the engine. It indicates the breathing capacity of the engine. From the figure it is evident that the volumetric efficiency values of T-20, T-40, T-60 and

T-75 are exceeding the volumetric efficiency values of diesel at all loads.

4.4 brake mean effective pressure:

Fig.6. shows the variation of Brake Mean effective pressure with load. Mean effective Pressure is the average Pressure inside the cylinders of an internal combustion engine based on the measured output. From the figure it can be seen that, Brake mean effective values of tyre pyrolysis oil and its diesel blends are slightly less than diesel.

4.5 indicated mean effective pressure:

Fig.7. shows the variation of Indicated Mean effective pressure with load. Mean effective Pressure is the average Pressure inside the cylinders of an internal combustion engine based on the measured output. From the figure it can be seen that, Brake mean effective values of tyre pyrolysis oil and its diesel blends of T-20 and T-40 are slightly less than diesel.

6.6 exhaust gas temperature:

Fig.8 shows the variation of exhaust gas temperature with load for various test fuels. It is observed that the exhaust gas temperature increases with load because more fuel is burnt at higher loads to meet the power requirement. It is also observed that the exhaust temperature increases for T-20 and T-40 blends at all loads. This may be due to the oxygen content of the tyre pyrolysis oil, which improves combustion and thus may increase the exhaust gas temperature. But the exhaust gas temperatures of T-20 blend and T-60 are very close to the exhaust temperature of diesel.

6.7 mechanical efficiency:

Mechanical efficiency indicates how good an engine is inverting the indicated power to useful power Fig.9. Shows the mechanical efficiency behind is less than the pure diesel. Because higher fuel injection pressures increases the decrease of atomization. The fitness of atomization Reduces ignition lag.

6.8 hc emission:

Unburned HC emissions in exhaust gas expresses non-utilizable lost chemical energy as in CO emission. The reason of HC among combustion products are disability to reach the ignition temperature or disability of fuel to be oxidized because of lack of oxygen and semi oxidation the main reason of reduction in HC emission shows that oxygen in tyre pyrolysis oil supplies is sufficient for oxidation in mixture parts of rich air–fuel. Fig.10 shows that The HC emissions Resulted from tyre pyrolysis oil and its blends were lower than that of diesel fuel. The lowest HC emissions were observed for T-40, while the higher Ones were obtained for diesel. With the increase in the amount of tyre pyrolysis oil that of containing blends HC emissions decreased accordingly. The reason for the HC decrease for biodiesel and its blends is that biodiesel contains extra oxygen which helps to improve combustion.

6.9 CO emission

Fig. shows the reduction of CO emission with the addition of tyre pyrolysis oil to diesel. CO is predominantly formed due to the lack of oxygen. Since oxygenated fuel, it leads to better combustion of fuel resulting in decrease in CO emission. The CO emission is found to be lower for T-20.

6.10 CO_2 emission:

One of the main factors of global warming, and the most important environmental problem of the world is the increase of CO_2 emission which eliminates greenhouse effect in the atmosphere. Some researchers think that CO_2emission released into the atmosphere with the use of diesel combines with photosynthesis cycle CO_2 existing among exhaust products is an important parameter for expressing the complete combustion. When the engine load increases, in Fig. 4.10, it has been thought that oxygen in tyre pyrolysis oil contributes significantly to reactions of oxygen-fuel in engine cylinder. The CO_2 emissions of the diesel engine fuelled with the tyre pyrolysis oil and its blend fuels and pure diesel fuel are shown in Fig. 4.11. CO_2 emissions resulting from tyre pyrolysis oil and diesel blends were lower than that of diesel. The lowest CO_2 emissions were found for T-60.

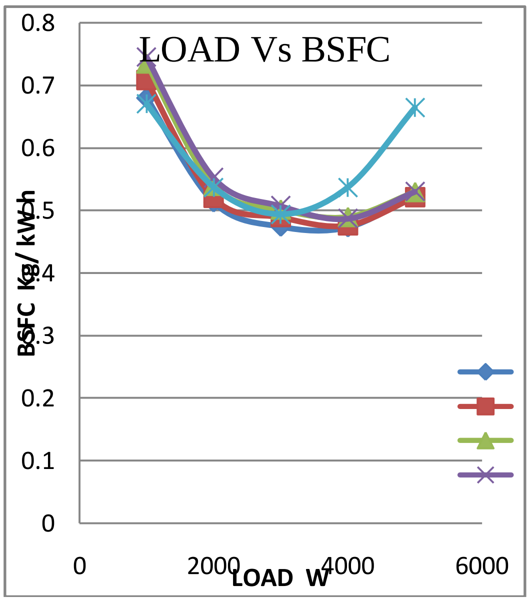

Figure 3.load Vs BSFC

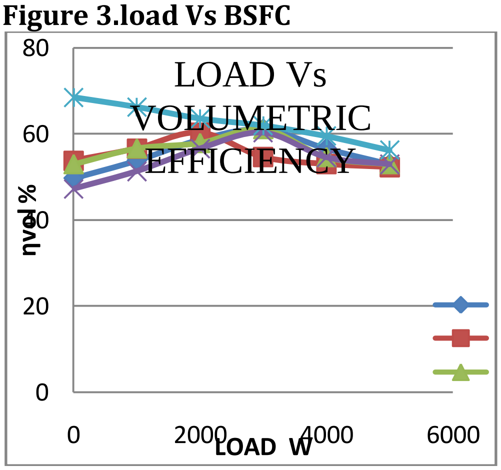

Figure 4 LOAD Vs VOLUMETRIC EFFICIENCY

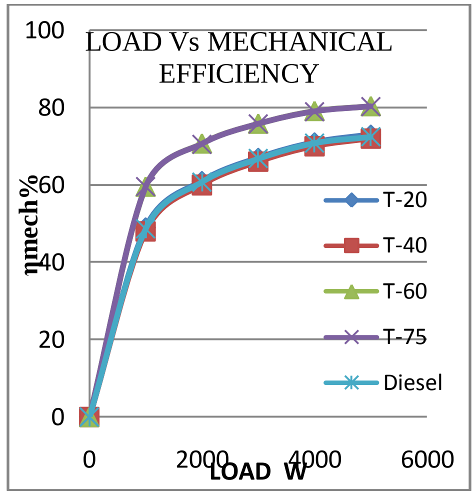

Figure 5 LOAD Vs MECHANICAL EFFICIENCY

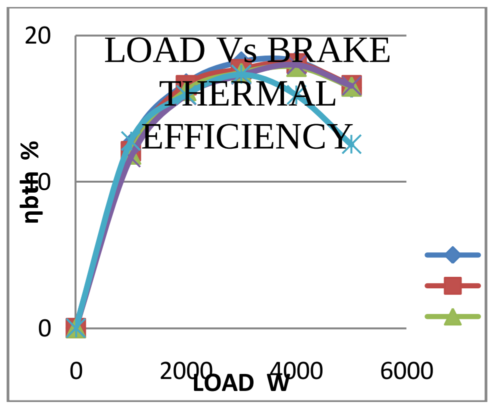

Figure 6 LOAD Vs BTH

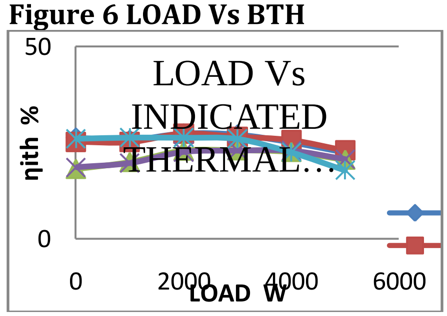

Figure 7 LOAD IMEP

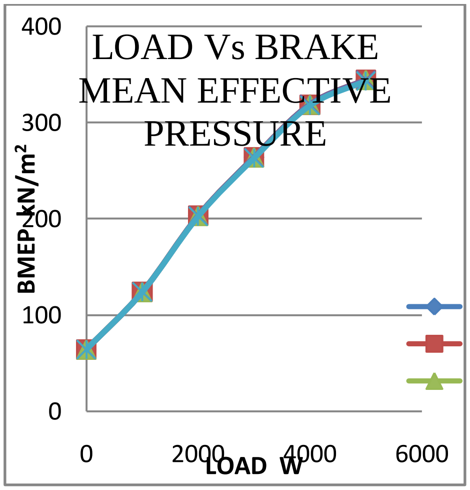

Figure 10 LOAD Vs BMEF

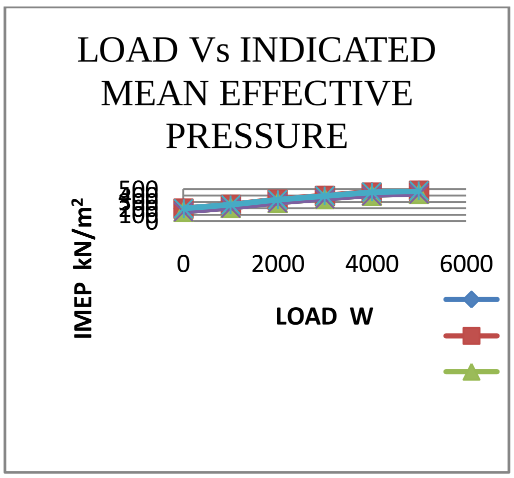

Figure 11 LOAD Vs IMEP

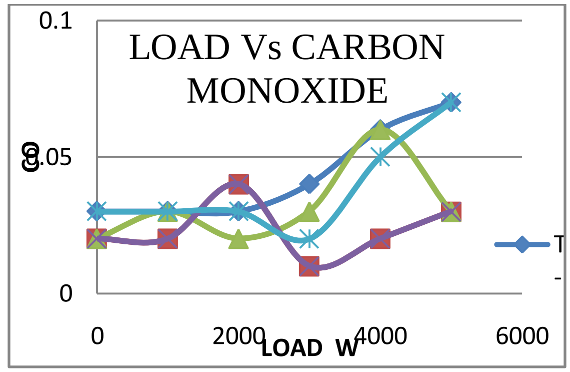

Figure 12 LOAD Vs CO

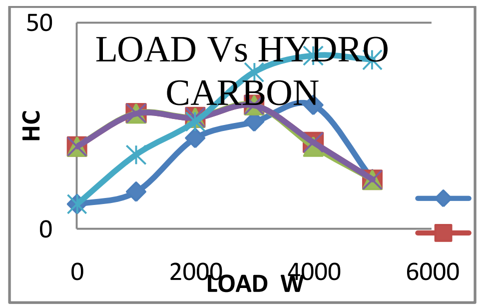

Figure 13.LOAD VSHC

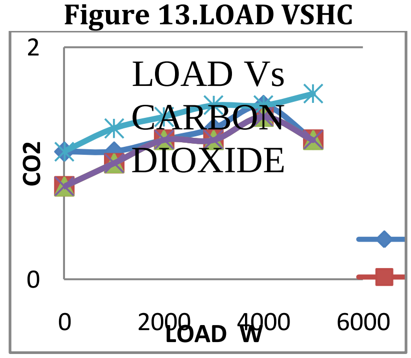

Figure 14 LOAD Vs CO2

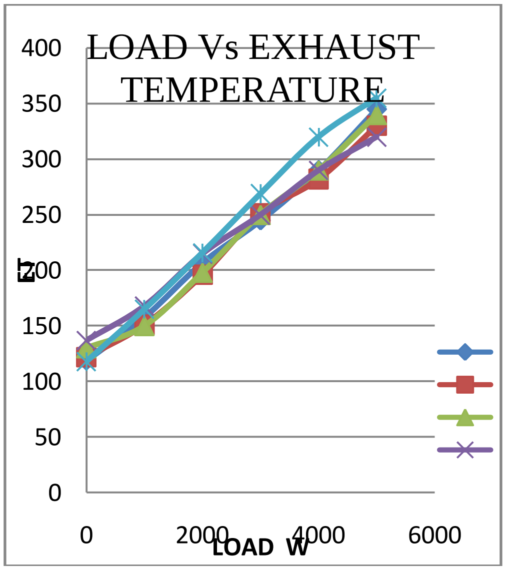

Figure 15 LOAD Vs EXHAUST TEMPERATURE

CONCLUSIONS

Following are the conclusions based on the experimental results obtained while operating single cylinder air cooled diesel engine fuelled with methyl fish oil and its diesel blends.
*Tyre Pyrolysis oil and its diesel blends can be directly used in diesel engines without any modifications.
* Brake specific fuel consumption values of tyre pyrolysis oil and its diesel blends are slightly higher than diesel.
* Brake thermal efficiency of T-40 blend is very close to the brake thermal efficiency of diesel at all loads.
* Volumetric efficiency values of tyre pyrolysis oil and its diesel blends are exceeding the volumetric efficiency values of diesel at all loads.
* CO emission decrease with increase in percentage of tyre pyrolysis oil in the fuel.
* CO_2 emissions of tyre pyrolysis oil and its diesel blends are slightly lower than that of diesel.
* HC emissions of tyre pyrolysis oil and its diesel blends are lower than that of diesel
* O_2 emissions of tyre pyrolysis oil and its diesel blends are slightly higher than that diesel
* From the above analysis the main conclusion is tyre pyrolysis oil and its diesel blends are suitable substitute for diesel as they produce lesser emissions than diesel and have satisfactory combustion and performance characteristics.

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Table.2. Performance and emission test Results at Pure Diesel:

S. NO	Load	Speed	Time	B.P	TFC	I.P	F.P	Heat Input	BSFC	η_{bth}	η_{ith}	η_{mech}	η_{vol}	BMEP	IMEP	CO	HC	CO_2
	W	rpm	Sec	kW	kg/s $*10^{-4}$	kW	kW	kW	Kg/ kW h	%	%	%	%	kN/m^2	kN/m^2	%vol	ppm	%vol
1	0	1500	80.21	0.762	2.119	2.312	1.5	8.9	0.91	8.56	25.98	32.97	68.48	64.32	195.22	0.02	10	1.1
2	1000	1500	55.68	1.469	2.736	3.019	1.5	11.491	0.67	12.76	26.27	48.67	66.23	124	254.84	0.02	18	1.3
3	2000	1500	47.9	2.401	3.586	3.951	1.5	15.062	0.537	15.94	26.23	60.77	63.58	202.69	333.53	0.02	26	1.4
4	3000	1500	39.67	3.118	4.285	4.668	1.5	17.999	0.494	17.32	25.93	66.78	61.96	263.23	394.1	0.02	30	1.5
5	4000	1500	30.21	3.765	5.627	5.351	1.5	23.634	0.537	15.93	22.47	70.84	59.51	317.87	451.75	0.015	35	1.5
6	5000	1500	22.63	4.068	7.511	5.618	1.5	31.547	0.664	12.54	17.8	72.44	56.25	343.39	474.23	0.01	41	1.6

Table.3. Performance and emission test Results at T75:

S. N0	Load	Speed	Time	B.P	TFC	I.P	F.P	Heat Input	BSFC	η_{bth}	η_{ith}	η_{mech}	η_{vol}	BMEP	IMEP	CO	HC	CO_2
	W	rpm	Sec	kW	kg/s $*10^{-4}$	kW	kW	kW	Kg/ kW h	%	%	%	%	kN/m^2	kN/m^2	%vol	ppm	%vol
1	0	1500	**81.57**	0.765	0.838	1.765	1	9.524	1.095	8.03	22.5	43.3	47.28	64.64	149.15	0.025	6	0.7
2	1000	1500	**61.46**	1.473	1.112	2.475	1	12.63	0.745	11.66	21.4	59.56	51.36	124.4	209.15	0.025	14	0.9
3	2000	1500	**51.35**	2.409	1.332	3.409	1	15.13	0.552	15.92	20.6	70.6	56.62	203.57	288.08	0.025	22	1
4	3000	1500	**43.06**	3.129	1.588	4.129	1	18.04	0.507	17.3	19.5	75.78	60.33	264.42	348.92	0.023	26	1.1
5	4000	1500	**37.17**	3.778	1.840	4.778	1	20.91	0.487	18	18.5	79	54.62	319.26	403.77	0.02	31	1.2
6	5000	1500	**31.56**	4.081	2.167	5.081	1	24.62	0.530	16.57	15.5	80.3	52.99	344.87	429.38	0.018	37	1.2