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EXPERIMENTAL STUDY ON THE PERFORMANCE OF BIODIESEL FUELLED CI ENGINE USING EXHAUST GAS RECIRCULATION

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ABSTRACT

Environmental degradation and depletion of fossil fuel reserves are matters of great concern around the world. Diesel is one of the main transport fuel used in sector and India depends heavily on oil import. Recent concerns over the environment, increasing fuel prices and shortage of its supply have promoted the interest in development of the alternative sources for petroleum fuels. It is observed by several researchers that with biodiesel fuelled compression ignition (CI) engine; the exhaust emission is lower than that of diesel, whereas the NO_x emission increases due to the excess oxygen content and high in-cylinder temperature of biodiesel. So, the exhaust gas recirculation (EGR) technique may be employed to lower the NO_x emissions form CI engines. Although, EGR in CI engine has a number of benefits on the combustion process and emissions, its effect on the performance of the engine should be critically evaluated. Keeping this in mind, only the performance characteristics of a double cylinder, water cooled, four stroke, direct injection compression ignition engine fuelled with jatropha biodiesel-diesel blends (10%, 15%, 20%, 25%, 30% and 50%) have been investigated and compared to mineral diesel with 15% exhaust gas recirculation. The lower blends of biodiesel increases the thermal efficiency and reduces the fuel consumption. Exhaust gas recirculation improves brake thermal efficiency and fuel consumption.

Keywords: Biodiesel, Jatropha, Exhaust Gas Recirculation, Efficiency, BSFC.

1. INTRODUCTION

The diesel engines lead the field of commercial transportation and agricultural machinery due to its ease of operation, high thermal efficiency, high power outputs and low maintenance cost. Fuel consumption of diesel engine is quite a few times higher than that of petrol. Due to the shortages of petroleum products and its increasing cost, efforts are onto develop alternative fuels, mainly to the diesel oil for full or partial replacement. It has been found that the biodiesel are potential fuels because of their properties, which are similar to that of diesel and are produced easily and renewably from the crops. At the present time biodiesel have traditional important attention both as a possible renewable alternative fuel and as an additive to the petroleum based fossil fuels. Biodiesel show some merits when compared to that of the conventional petroleum fuels. Using biodiesel instead of diesel fuel reduces emissions such as the overall life cycle of carbon dioxide (CO_2), particulate matter (PM), carbon monoxide (CO), sulfur oxides (SO_x), volatile organic compounds (VOCs), and unburned hydrocarbons (HC). Hence, in order to meet the strict vehicular exhaust emission norms worldwide, several exhaust pre-treatment and post treatment techniques have been employed in modern engines. Exhaust Gas Recirculation (EGR) is a pre-treatment technique, which is being used widely to reduce and control the oxides of nitrogen (NO_x) emission from diesel engines [1]. Various researchers reported that exhaust gas recirculation technique is a useful technique to lower the NO_x emission from diesel engines.

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The application of EGR on compression ignition (CI) engine has a number of effects on the combustion process and emissions. Firstly, preheating effect in this process the inlet charge temperature increased when the hot EGR mixed with air/fuel mixture. Secondly, dilution effect due to the introduction of EGR leads to a considerable reduction of the oxygen concentration. Thirdly, heat capacity effect in this process affects the total heat capacity of the mixture of EGR, air and fuel will be capacity owing to the higher heat capacity of carbon dioxide and water vapor. This will lead to a reduction of the gas temperature at the end of the compression stroke. Fourthly, as chemical composition of the intake mixture changes, the combustion characteristics of the charge become different from the non-EGR condition. In this work, experiments have been carried out in a double cylinder, water cooled, four stroke, direct injection diesel engine with exhaust gas recirculation technique (EGR) using diesel blended with jatropha biodiesel at different ratios as fuels and the performance characteristics of the engine have been studied.

2. EXHAUST GAS RECIRCULATION

Exhaust Gas Recirculation is an effective method for NO_x control. The exhaust gases mainly consist of carbon dioxide, nitrogen and water vapour. When a part of this exhaust gas is re-circulated to the cylinder, it acts as diluent to the combusting mixture. The specific heat of the EGR is much higher than fresh air; hence EGR increases the heat capacity (specific heat) of the intake charge, thus decreasing the temperature rise for the same heat release in the combustion chamber. Re-circulated exhaust gas displaces fresh air entering the combustion chamber with carbon dioxide and water vapour present in engine exhaust. As a result of this air displacement, the amount of oxygen in the intake mixture is decreased. Reduced oxygen available for combustion lowers the effective air–fuel ratio which affects exhaust emissions substantially. In addition, mixing of exhaust gases with intake air increases the specific heat of intake mixture, which results in the reduction of flame temperature. Thus combination of lower oxygen quantity in the intake air and reduced flame temperature reduces rate of NO_x formation reactions. The EGR (%) is defined as,

$$\%EGR = \frac{volume\ of\ EGR}{\text{total intake charge into the cylinder}} \times 100 \qquad (1)$$

Another way to define the EGR ratio is by the use of CO_2 concentration [2],

$$\text{EGR ratio} = \frac{[CO_2]_{\text{intake}} - [CO_2]_{\text{ambient}}}{[CO_2]_{\text{exhaust}} - [CO_2]_{\text{ambient}}} \qquad (2)$$

It is difficult to employ EGR at high loads due to drop in diffusion combustion and this may result in an excessive increase in smoke and particulate emissions. But at low loads, unburnt hydrocarbons contained in the EGR re-burn in the mixture, leading to lower unburnt fuel in the exhaust and thus improved brake thermal efficiency. Apart from this, hot EGR would raise the intake charge temperature, thereby influencing combustion and exhaust emissions. Implementation of EGR in diesel engines has problems like (a) increased soot emission, (b) introduction of particulate matter into the engine cylinders. When the engine components come into contact with high velocity soot particulates, particulate abrasion may occur. Sulphuric acid and condensed water in EGR also cause corrosion. If the exhaust gas is re-circulated directly to the intake, it results in increased intake charge temperature i.e. hot EGR. An increase in inlet charge temperature always results in shorter ignition delay and may improve thermal efficiency [3]. If the exhaust gas is cooled before recirculation to combustion chamber, then it is called cooled EGR. Cooling of EGR increases the charge density therefore improves volumetric efficiency of the engine. The engines using EGR emit lower quantity of exhaust gases compared to non-EGR engines because part of the exhaust gas is re-circulated [4]. It is also found that with the employment of EGR technique the thermal efficiency also increases [5]. EGR was also used in a direct injection spark ignition engine as an effective way of improving fuel economy [6-8]. But the addition of biodiesel to petro-diesel increased brake specific fuel consumption because of lower calorific value of biodiesel, but the thermal efficiency remained almost the same [9].

3. PROPERTIES OF JATROPHA BIODIESEL

Jatropha oil can be converted to its methyl esters via transesterification process in the presence of catalyst. The purpose of the transesterification process is to lower the viscosity of the oil. Ideally, transesterification is potentially a less expensive way of transforming the large, branched molecular structure of the bio-oils into smaller, straight chain molecules of the type required in regular diesel combustion engines. Biodiesel from jatropha oil is free from sulphur and still exhibits excellent lubricity, which is an indication of the amount of wear that occurs between two metal parts covered with the fuel as they come in contact with each other. It contains very small amount of phosphorus and sulphur and therefore emission of oxides of sulphur (SO_x) is almost negligible. In comparison with commercial petro-diesel, jatropha biodiesel has higher density and cetane number. In addition, the higher flash point (more than 100ºC) of jatropha biodiesel makes the storage and transportation issues less important. It is a much safer fuel than diesel because of its higher flash and fire point. The cloud filter pour point is generally higher than that of diesel and this may involve some complications for the operation in cold weather. The amount of carbon residue from the hot decomposition of vegetable compounds with higher molecular weight is greater than that of commercial diesel oil. Some of the important fuel properties of jatropha biodiesel and

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conventional petro-diesel are compiled from the previous works of Graboski and McCormick [10], Ramos *et al.* [11], Pradeep and Sharma [12], Barnwal and Sharma [13] are presented in the tabular form for ready reference in table 1 for comparison.

setup three known diameter (1 inch/ 2.54 cm) orifice meters are installed for measuring the re-circulated exhaust gas flow (orifice 1), ambient air flow (orifice 2) and mixing flow (orifice 3).

Table 1: Properties of Jatropha Oil, Jatropha Biodiesel and Diesel

Property	Jatropha Oil	Jatropha Biodiesel	Diesel	Biodiesel Standards	
				ASTM D 6751-02	DIN EN 14214
Density(15°C, kgm^{-3})	940	880	850	-	860-900
Viscosity (mm^{2}s^{-1})	24.5	4.8	2.6	1.9-6.0	3.5-5.0
Flash Point (°C)	225	135	68	>130	>120
Pour Point (°C)	4	2	-20	-	-
Water Content (%)	1.4	0.025	0.02	<0.03	<0.05
Ash Content (%)	0.8	0.012	0.01	<0.02	<0.02
Carbon Residue (%)	1.0	0.20	0.17	-	<0.30
Acid value (mgKOHg^{-1})	28.0	0.40	-	<0.80	<0.50
Calorific Value (MJkg^{-1})	38.65	39.23	42	-	-

4. EXPERIMENTAL DETAILS

The experiment was carried out on a double cylinder, constant speed, direct injection diesel (compression ignition) engine. The specification of the engine is given in table 2. The engine was directly coupled to a hydraulic dynamometer of maximum load capacity 20 kgf. The experiment was conducted at a rated speed of 1500 rpm. The load was varied by adjusting load wheel on the top of the engine. Water pressure was kept constant at 1.5 kg/cm^2. The torque and the fuel consumption rates were measured for different loads and fuel blends. The fuel blends were prepared by mixing different percentages of biodiesel with pure diesel. The percentages of biodiesel in the blends were taken to 0% (pure diesel, B0), 10% (B10), 15% (B15), 20% (B20), 25% (B25), 30% (B30) and 50% (B50).

Table 2: Engine Specifications

Manufacturer	Kirloskar Oil Engines Ltd.
No of cylinder	2
Model	AV2
Engine No	11.1001/81801
Type	Four stroke, Water Cooled
RPM	1500
BHP	10 (7.35KW)
SFC	199 G/bhp-hr
Bore Diameter	80 mm.
Stroke Length	110 mm.
Rated Power	7.35 kW @ 1500 RPM
Injection Timing	23° before TDC
Method of Loading	Hydraulic Dynamometer

4.1 Exhaust Gas Recirculation Setup

A general cross sectional view of exhaust gas recirculation technique is shown in fig. 1. Figure 2 shows the schematic diagram of experimental setup for exhaust gas recirculation technique. From the figure, it is clear that a part of exhaust gas is controlled by control valve 1 and re-circulated then mixed at the (A) indicating portion with the incoming fresh air and go to the intake for combustion. Percentage of EGR is controlled by the control valve 1. In the experimental

Figure 2 also shows the schematic of orifice meter. Every orifice meters connected with U tube manometer for the flow measurement. In the setup the thermometer is used to determine the mixing temperature. The U tube manometer adjusts with the centimetre scale from which we can measure the pressure difference for the particular flow. From the pressure difference, the flow rate is calculated.

5. RESULTS AND DISCUSSION

The calorific values of different biodiesel-diesel blends are calculated using weighted average method. The specific gravity for each blends are measured by a hydrometer. The calorific values and specific gravity for different blends are shown in table 3. Brake power, brake thermal efficiency (BTE) and brake specific fuel consumption (BSFC) of the engine for different blends of the fuel at different loads are calculated using standard formulae. The variations of different performance parameters with load (brake power) are presented graphically in the next section.

Table 3: Calorific value and Specific gravity of different blends of fuel

Blends	Calorific value (kJ/kg)	Specific gravity (g/ml)
Pure Diesel (B0)	42500.00	0.8350
B10	42190.36	0.8377
B15	41933.63	0.8400
B20	41821.19	0.8410
B25	41717.18	0.8419
B30	41533.20	0.8435
B50	40820.89	0.8500

5.1 Brake Thermal Efficiency

The brake thermal efficiency ratio is defined as the ratio of the output in the brake power to that of the chemical energy input in the form of fuel supply. The brake thermal efficiency gives an idea of output power generated by the engine with respect to heat supplied in the form of fuel. It is the true indication of the efficiency with which the thermodynamic input is converted into mechanical work.

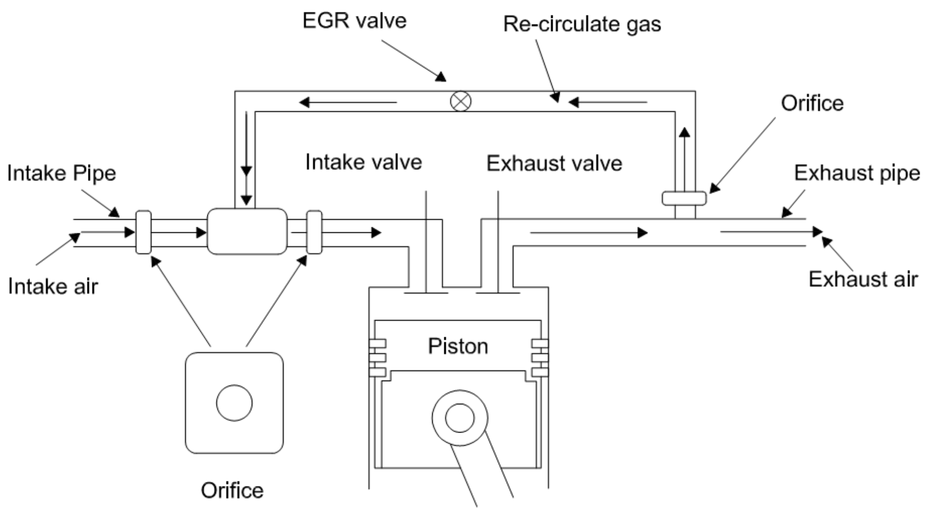

Fig.1. General View of Exhaust Gas Recirculation Technique

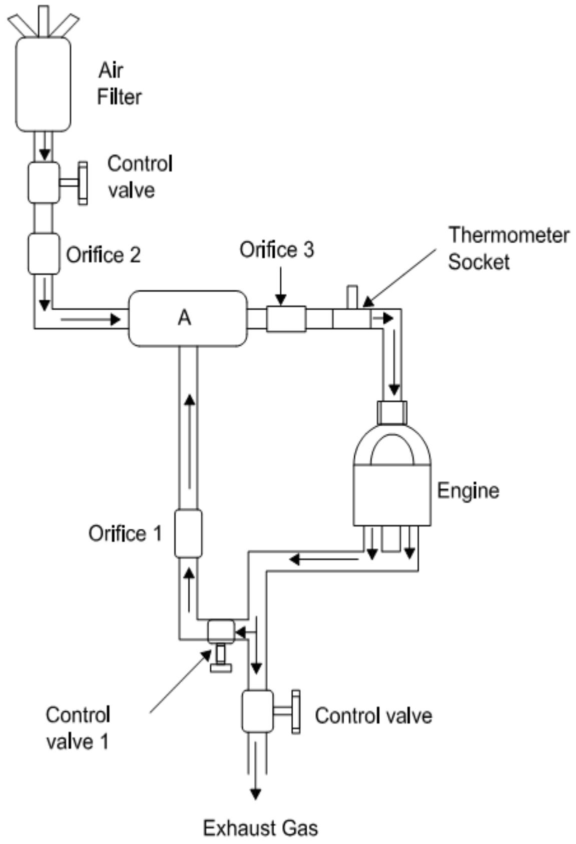

Fig.2. Schematic Diagram of Exhaust Gas Recirculation Setup

Figure 3 and figure 4 represent the effect of brake power on the brake thermal efficiency for without EGR and with EGR condition respectively for all fuel blends. It is observed that, initially with increasing brake power, the brake thermal efficiencies for each fuel (diesel, diesel-biodiesel blends) increased due to reduction in heat loss and increase in power developed with increase in load. The maximum brake thermal efficiency was obtained at around 5.0 to 5.5 kW and then tends to decrease with further increase of load.

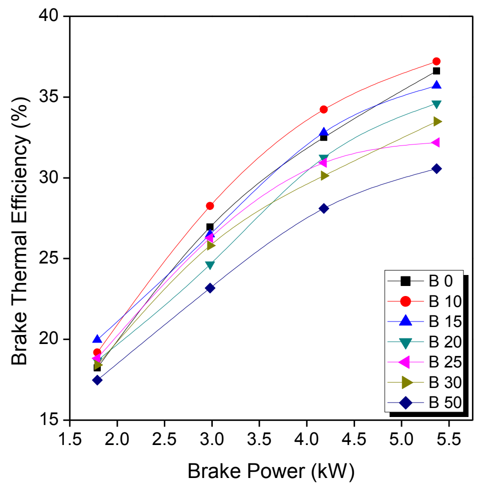

Fig.3. Effect of Brake Power on Brake thermal efficiency for without EGR condition.

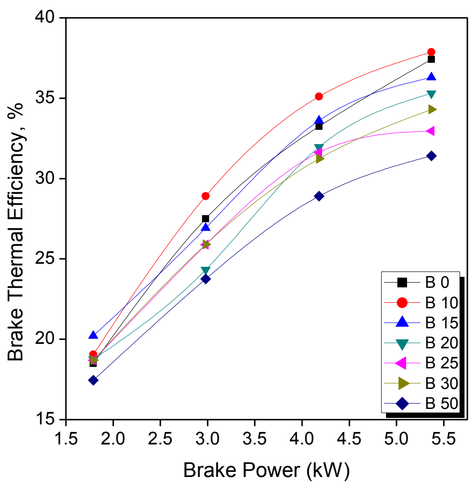

Fig.4. Effect of Brake Power on Brake thermal efficiency for with EGR condition.

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For the B10 fuel the brake thermal efficiency is improved compared to pure diesel and for B15 fuel, it is almost same of pure diesel. The possible reason for this improvement is the molecules of biodiesel contain some amount of excess oxygen which takes part in the combustion process. It is noticed from the figures that, with EGR the brake thermal efficiency increases a bit, due to increase of the combustion velocity, caused by higher intake charge temperature. Disassociation of carbon dioxide to form free radicals can also be credited to this improvement in efficiency.

5.2 Brake Specific Fuel Consumption

The brake specific fuel consumption defined on the basis of brake output of the engine. It is a clear indication of the efficiency with which the engine develops power from fuel or it is the important parameter that reflects how good the engine is.

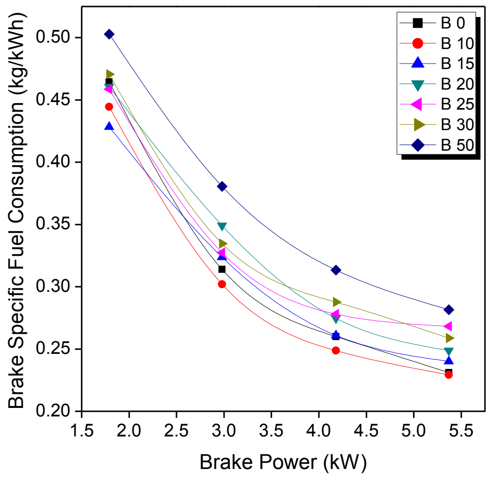

Fig.5. Effect of Brake Power on Brake specific fuel consumption for without EGR condition.

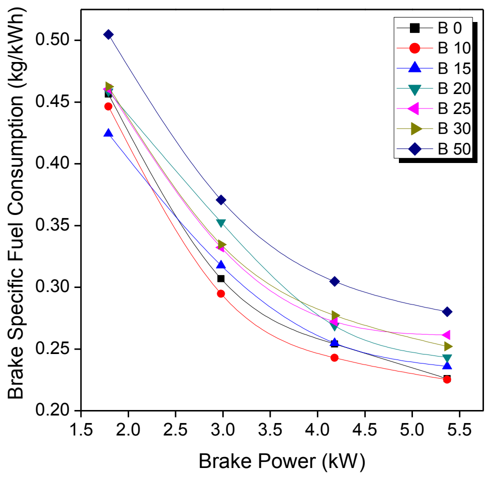

Fig.6. Effect of Brake Power on Brake specific fuel consumption for with EGR condition.

Figure 5 and figure 6 represent the variations of brake specific fuel consumption (BSFC) with brake power for all fuels at without and with EGR condition respectively.

For all fuel blends, BSFC is found to be decrease with increase in load. This is due to the higher percentage increase in brake power with load as compared to the increase in fuel consumption. Using B10, BSFC of the engine is lower than that of diesel for all loads and for B15, BSFC almost same. In general the BSFC is found to be greater with biodiesel because of its lower calorific value. It is noticed from the figures that, with EGR condition reduces the BSFC. It could be the effect of re-burning of charge, increase of intake charge temperature which increases the combustion velocity.

5.3 Indicated Thermal Efficiency

The indicated thermal efficiency gives an idea of the power generated by the engine within the cylinder with respect to heat supplied in the form of fuel. Figure 7 and 8 represent the effect of brake power on the indicated thermal efficiency at without and with EGR condition respectively.

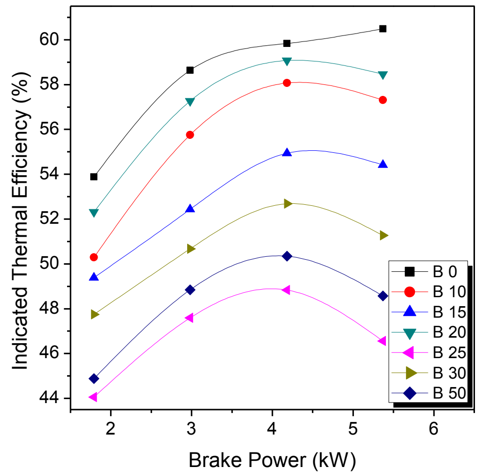

Fig.7. Effect of Brake Power on Indicated thermal efficiency for without EGR condition.

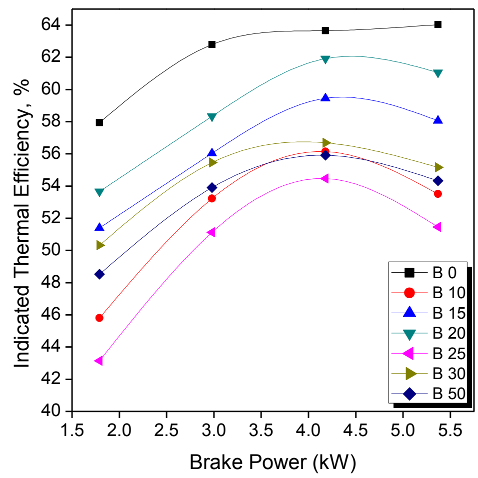

Fig.8. Effect of Brake Power on Brake thermal efficiency for with EGR condition.

The figures show that, with the increase in brake power the indicated thermal efficiency for each fuel has increased. Also, with the increase of biodiesel percentage in blended fuel, the indicated thermal efficiency gradually decreases. It is due to the lower calorific value of biodiesel. Form the figures, it is clearly seen that, with EGR condition, indicated thermal efficiency increases because of the increase in intake charge temperature and combustion velocity which is caused by EGR itself.

6. CONCLUSIONS

As it is already established that biodiesel and EGR improve the emission characteristics of the engine, this work involves only the energetic performance study of biodiesel fueled diesel engine using exhaust gas recirculation technique. From the experimental study it can be concluded that biodiesel can be used as suitable alternative to diesel. The brake thermal efficiency and the brake specific fuel consumption improve slightly with B10 fuel. For other blends, both the above parameters decrease to some extend. However, the indicated thermal efficiency decreases with all blends of biodiesel. This may be attributed to the better lubricating property of biodiesel. It is also noted that exhaust gas recirculation technique does not change the efficiencies or brake specific fuel consumption of the engine appreciably. Finally, it can be said that B10 can be used as fuel in Compression ignition engine employing EGR technique for optimum performance considering both energetic and environmental aspects.

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