Experimental Investigation on mixing of two Biodiesels and Additive blended with Diesel as Alternative Fuel for single cylinder Diesel Engine

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Abstract :

Biodiesel poses a very promising scenario of using diesel fuel as an alternative fuel. These features can be correlated positively with the diesel engine-specific inner combustion engine fuels. Alcohol is considered to be a suitable fuel substitution for diesel engines because it allows the diesel fuel to burn completely due to the presence of more oxygen which improves engine characteristics. The present investigation is to evaluate the performance and emission characteristics of different features of dual biodiesel mixing together with each blend having n-butanol as an additive.The experiment uses a unique cylinder, four stroke, constant speed, water-cooled, direct injection VCR diesel engine. In the case of biodiesel fuel without EGR, experimental results showed better oxides of nitrogen emissions. NOx emissions were lowered when the engine was operated with EGR. The EGR level was optimized as 20% based on a considerable reduction in NOx emissions, minimum smoke, CO, HC emissions and similar brake Thermal efficiency. Among all the blends D80-B20 Blend has shown increased engine performance and emission control, D80-B20 is given as the optimum blend and EGR with 10 and 20 percent is used to lower emissions. Thermal efficiency, Between all the blends D80-B20 Blend has shown better performance in terms of engine performance and emission control,D80-B20 is taken as the optimum blend and EGR with 10% and 20% is used to reduce emissions.

KEY WORDS–**Dual biodiesel, Castor oil, Rubber seed oil, n-butanol,Exhaust Gas Recirculation(EGR), alternative fuels,Diesel engine, performance and emissions.**

I. INTRODUCTION

Energy preservation is now declining for a few days, and it is alleged to result in demand for energy. Alternative fuels have been acquired and recognized as important over the past two decades. A potential biodiesel replaces diesel oil, composed of ethyl ester of fatty acids formed with the assistance of a catalyst through the transesterification process of vegetable oils and ethanol triglycerides. Furthermore, in terms of very limited sulfur content, biodiesel is better than diesel fuel and also has better flash and fire point temperatures than diesel fuel. A lot of research work has shown that biodiesel has attracted considerable attention and is a feasible alternative fuel. Biodiesel And its diesel blends were used as a diesel engine gasoline without any changes to the current engine.

M. Arunkumar, M. Kannan, G. Murali The author investigated the Castor biodiesel to determine the efficiency of the CI engine when it is fuelled with Castor biodiesel and is used as another diesel fuel; the use of castor biodiesel decreases CO emissions, HC emissions and significantly decreases NOx emissions. The author's experimental result describes that castor biodiesel use decreased CO by 9%, HC by 8.8%, and NOx. But the issue is that Castor biodiesel mixture in diesel has improved the specific fuel consumption 4% and reduced the break thermal efficiency 2.2. **Atmanli et al.** conducted an engine test using RSM on varied operating conditions at different blends of diesel fuel, cotton oil, and n-butanol. Homogeneity and no phase separation were noted. BMEP, brake power, and thermal efficiency of blend were reduced; however, BSFC improved marginally. The emissions of the blends were reduced, namely HC, NOx, and CO.

Zhu et al. conducted experiments on n-butanol blends, EGR rates, and timing of injection on a modified diesel engine. The results suggested that NOx emissions would decrease with higher EGR rates, but increase emissions from smoke. As the n-butanol proportion increased, smoke emissions decreased with a small.

Castor oil and rubber seed oil are the most appropriate biodiesels for diesel engines. In this paper, CBD and PBD mixture (CRBD) were blended with diesel fuel and 5% of additive in different proportions forming different mass proportions of fuel blends in this paper. The properties were analyzed and compared to the diesel fuel. Fuel properties such as density, calorific value, viscosity and flash point and fire poiny were tested. Experimental tests were performed to evaluate the performance and exhaust emissions of various dual biodiesel blends on the diesel engine.

II.MATERIALS AND METHODS

The process of transesterification is selected from castor and rubber seed oils to produce biodiesel. Raw oil is filtered to remove water, strong particulate matter and heated at 105^0C temperature using surgical cotton to extract all water content. In 99% pure methanol, 120 ml per liter of oil is added to the heated oil in acid treatment and stirred for ten minutes. For each liter of petroleum, two milliliters of 98% sheer sulfuric acid are added. The mixture is heated and stirred in a closed conical beaker for one hour at 60^0C. The blend can stay in a decanter for four hours and remove glycerin from the methyl ester. 200 ml of methanol (20% by volume) and 6.5 grams of 97% pure NaOH (Sodium Hydroxide) are introduced to the base treatment for each liter of oil. The mixture is carefully stirred until it forms a transparent "Sodium Methoxide" solution. Added to the oil, this solution is heated to 60^0C and preserved in a closed container at the same temperature with stirring at 500 to 600 rpm. When the solution transforms into a color brown silky, which shows the completion of the entire reaction. The upper portion of the glycerin is removed from the biodiesel after the blend has been settled in the decanter. In order to avoid soaps and unreacted alcohol, the shaped methyl ester is bubble washed with distilled water for about half an hour.

Washing is performed with transparent water until the methyl ester is removed. Collected methyl ester is heated to remove water and formed biodiesels CBD, RBD and blends prepared with diesel fuel are taken for characterization.

III.ENGINE SETUP AND PROCEDURE

For experimental testing, Kirloskar single cylinder water cooled variable diesel compression engine integrated with EGR is used. For loads on the engine, the Eddy current dynamometer is used. To apply loads to the engine, the Eddy current dynamometer is connected to the flywheel. To inject the fuel, an injection pressure of 200 bar is maintained. The pressure of the cylinder is evaluated by the piezo-sensor installed on the head of the engine cylinder and the angle of the crank encoded on the fly wheel. The normal engine is supplied with 0 to25oBTDC injection point variation. The HC, CO, CO2, UBHC and NOX emissions are evaluated using the fire gas analyzer AVL-DIGAS 444. AVL smoke meter measures the opacity of the smoke.

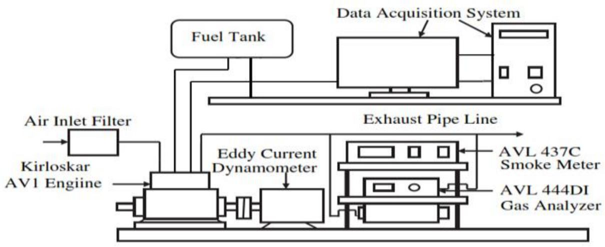

Fig.1. Schematic diagram of experimental set up

Fig.2. Complete Experimental engine setup

Engine make & Type	VCR Engine test setup 1 cylinder, 4 stroke, Diesel with EGR (Computerized) & Kirloskar
EGR	Water cooled, ss 304, Range 0-20%
Displacement	661 cc
Bore & Stroke	875 mm & 110 mm
Compression ratio	12 to 18
Rated Speed	1500rpm
Dynamometer	Eddy current, water cooled with loading unit

Table 2: Test Engine specifications

3.1Test procedure

The current research focuses on the use of caster oil and rubber seed oil as fuel in the current direct injection (DI) diesel engine. Some of these two biodiesel characteristics are closer to diesel oil. As a result, castor oil and rubber seed oil biodiesel is selected as an alternative fuel and is experimented with 10% and 20% implementation of EGR to investigate the highest possible diesel substitute in a DI diesel engine. However, since the combination of these two biodiesel mixed with diesel has a reduced amount of cetane.

By decreasing its viscosity, we used this caster oil and rubber seed oil. Also engine will be produced to operate with blends of diesel and caster oil and rubber seed oil biodiesel consisting of 5%, 10%, 15%, 20% and 25% in diesel fuel and after testing the results shown combination B20-D80 is the optimum blend based on its performance and emission features

BLENDS	BLENDS PROPORTIONS with 5% ADDITIVE
D95-B05	2.5% CO + 2.5% RSO + 95% Diesel
D90-B10	5% CO + 5% RSO + 90% Diesel
D85-B15	7.5% CO + 7.5% RSO + 85% Diesel
D80-B20	10% CO+ 10% RSO + 80% Diesel
D75-B15	12.5% CO + 12.5% RSO + 75% Diesel
D80-B20	10% EGR
D80-B20	20% EGR

Table.2: Blending proportions table

IV.RESULT AND DISCUSSIONS

4.1 Performance Analysis:

4.1.1 Brake specific fuel consumption:-

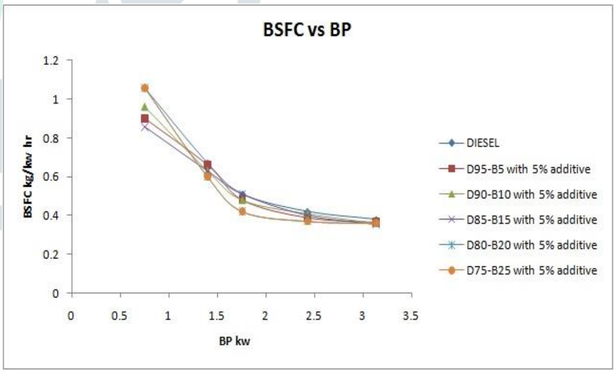

Fig 3. Variation of BSFC with Brake power

Figure 3 shows the differences in brake-specific fuel consumption of various blends with regard to brake power. The plot reveals that the BSFC is diminishing as the load rises. The BSFC obtained under full load conditions is 0.374 kg / kw-hr, 0.356 kg / kw-hr, 0.362 kg / kw-hr, 0.353 kg / kw-hr and 0.350kg / kw-hr for diesel fuels, D95-B5,D90- B10,D85- B15,D80- B20 andD75- B25 respectively. The percentage variation of BSFC of all diesel blends at different load environments is reduced by 0.964%, 0.951%, 0.921%, 0.896%, and 0.949%. For the D80-B20 blend, which is 0.34 kg / kw-hr, the minimum brake-specific fuel consumption is achieved, i.e. 0.897 % reduces with 10 kg diesel fuel. The BSFC for B20D80 is achieved lower compared to all oil blends, but for D95-B5, D75-B25, the BSFC is closer to diesel.

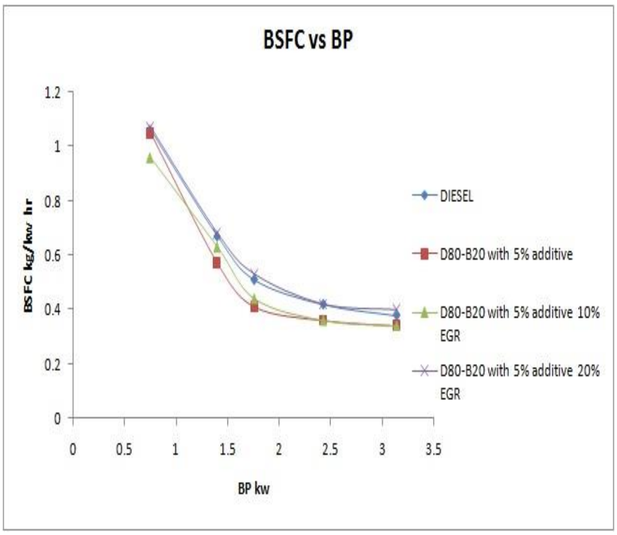

Fig 4. Variation of BSFC with Brake power for various EGR percentages

Figure 4. shows the brake-specific fuel consumption with respect to brake power impact of exhaust gas recirculation. The plots reveal that as the EGR percentage enhanced brake-specific fuel consumption due to dilution of clean air with exhaust gas resulting in improper oxidation, combustion and reduced power output, more fuel is needed to generate the same power output as diesel fuel. Compared to other fuel blends, BSFC for B20-D80 with 20 percent EGR is noted to be greater for the entire load spectrum. It may be due to greater heat value and reduced mixed fuel viscosity. The BSFC is 0.379 kg / kw hr, 0.34 kg / kw hr, 0.34 kg / kw hr, 0.34 kg / kw hr and 0.4 kg / kw hr at complete load circumstances. BSFC's percentage variation on diesel at distinct EGR percentages is 0.89%, 0.89% and 1.055% respectively.

4.1.2 Brake thermal efficiency (BTE):-

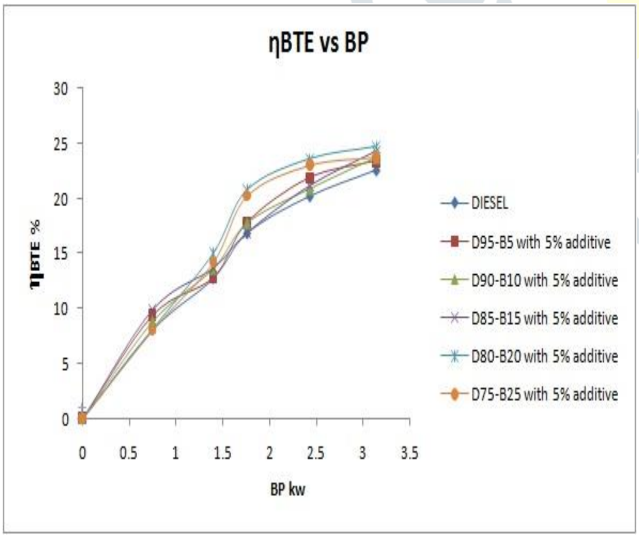

Fig 5. Variation of ηBTE with Brake power

The Brake Thermal Efficiency Variations regarding Brake Power as shown in the figure 5. The thermal efficiency of the brake generally improves with increased load. As long as the load increases the brake energy and decreases the loss of bsfc and heat which improves the thermal efficiency of the Brake. The Brake Thermal Efficiency was 22.58 percent, 23.45 percent, 23.68 percent, 24.23 percent, 24.73 percent and 23.78 percent respectively for fuel blends such as Diesel, D95-B5,D90-B10, D85-B15,D80- B20, D75-B25 from graphs at 10 kg load circumstances.The percentage variation of Brake thermal efficiency of all diesel blends at different load circumstances is improved by 1.038%, 1.048%, 1.07%, 1.095% and 1.053%. The maximum brake thermal efficiency for the B20-D80 blend is 24.73 percent, i.e. 1,095 percent higher for diesel fuel, and blends such as D75-B25 reduced as compared to D80-B20 due to an rise in the proportion of castor oil and rubber sedd oil in diesel fuel.

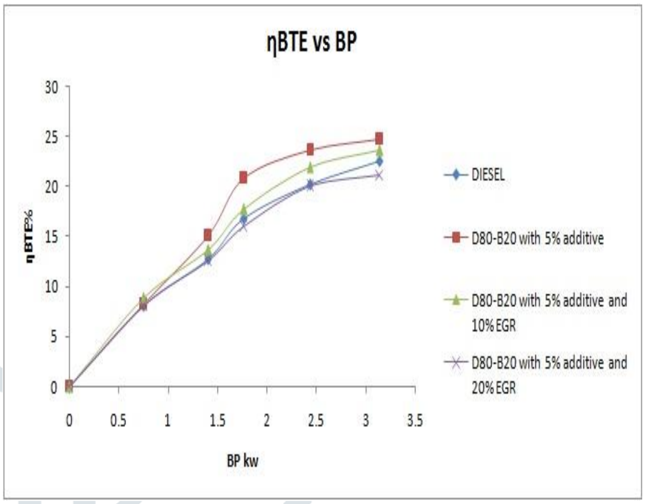

Fig.6. Variation of ηBTE with Brake power for various EGR percentages

Figure 6. shows the brake thermal efficiency impact of exhaust gas recirculation is shown in figure.With the rise in the EGR flow rate, the brake thermal efficiency reduces marginally. The brake thermal efficiency is 22.58 %, 24.73 %, 23.63 % and 21.25 % for fuel blends such as diesel, B20-D80 without EGR, B20-D80 with 10 % EGR and B20-D80 with 20 % EGR, from the graph at 10 % load condition we absorbed. The decrease in brake thermal efficiency is due to elevated percentages of EGR resulting in a defect in the concentration of oxygen in the combustion system and a greater substitution of warm air from exhaust gases. Higher exhaust gas flow rate decreases the average combustion temperature in the combustion chamber resulting in reduce brake thermal efficiency at all loads.The percentage variation of brake thermal efficiency of all blends at different EGR rates with respect to diesel are 1.095%, 1.04% and 0.94% respectively.

4.1.3 Mechanical Efficiency (Mechη):-

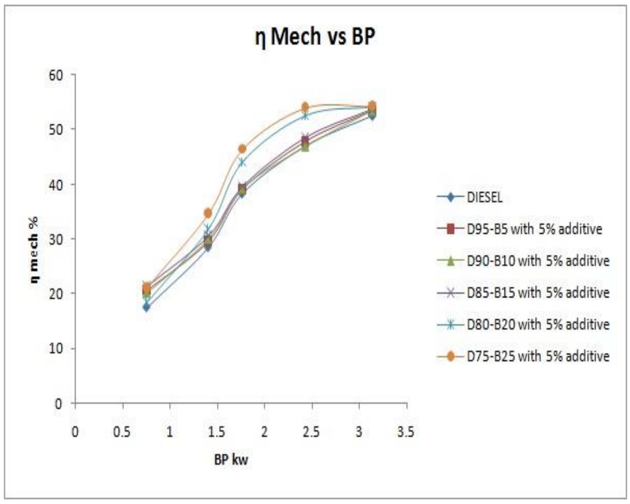

Figure 7. Variation of Mechη with Brake power

Mechanical efficiency is the measure of the engine's helpful job with fuel and energy input. The graph below shows mechanical efficiency variations for all fuel blends under different load conditions. From the graph it is noted that with growing loads for pure diesel to all blends, there is an increase in mechanical efficiency. At 10 kg load conditions, the Mechanical Efficiency values for fuel blends such as Diesel, D95-B5,D90- B10, D85-B15,D80-B20, D75-B25were 52.51 %, 53.46%, 53.51 %, 53.73 % and 54.11 % .The percentage variation in the mechanical efficiency of all diesel blends at different load conditions is increased by 1.018%, 1.019%, 1.023%, 1.030% and 1.034%. The mechanical efficiency is improved from the diesel fuel to Blend D75-B25 due to rises in the blending ratios. The maximum mechanical efficiency is achieved by 56.94 % in the blend D75-B25 i.e. 1,08 % higher than diesel.

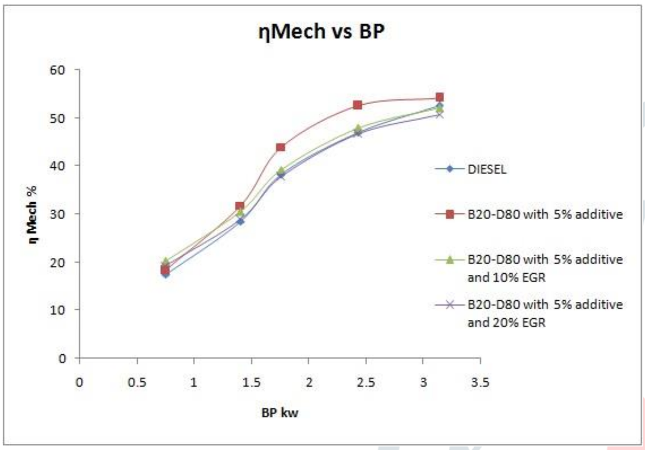

Fig 8. Variation of ηMech with Brake power for various EGR percentages

Figure 8. shows the effect of exhaust gas recirculation on brake power with respect to mechanical efficiency.As we all know, it is the measure of the engine's helpful job with fuel and energy input. From the plotted graph it is noted that the nature of mechanical efficiency is declining while reducing the flow of exhaust gasses into the engine cylinder by varying the EGR proportion for the fuel blend B20-D80 with 10 percent and 20 percent. The reason for reducing mechanical efficiency is due to poorer atomization of fuel blends with exhaust gasses. We observed readings such as 52.5 %, 54.1 %, 52 % and 50.8 % for fuel blends such as diesel, D80-B20 without EGR, B20-D80 with 10 % EGR and D80-B20 with 20 % EGR respectively at 10 kg load conditions.

4.2 Emission Analysis

4.2.1 Oxides of nitrogen in emissions (NO_X):-

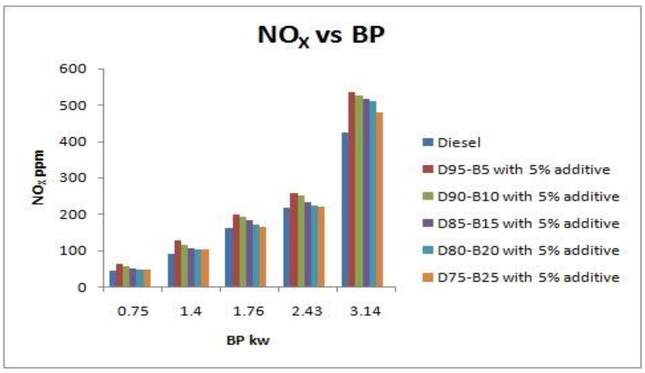

Fig 9. Variation of NO_X emissions with Brake power

In the graph shows, the variation of nitrogen oxides with varying blends at varying brake power is represented graphically. The increase in temperature causes the formation of nitrogen oxides in the exhaust. But in this case the engine temperature is equivalent to standard diesel for D85-B15 and D80-B20 and the blend D75-B25 is nearer to diesel. This is due to the fact that addition of additive to blends has more oxygen content than standard diesel fuel, although the engine temperature is less.This provides surplus oxygen, resulting in excess nitrogen oxide release. That's why we received too much NOX emissions. It is depicted from the graph that all the other than diesel such as D95-B5,D90- B10, D85-B15,D80- B20and D75-B25 produce greater NOX emissions. Among them,D75-b25 produce NOX emissions, i.e. 478 ppm, which is much closer to diesel fuel, and the D95-B5 blend provides peak NOX emissions, i.e. 534 ppm

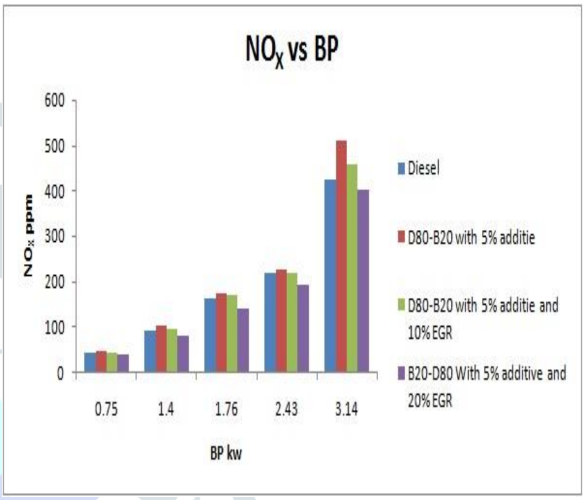

Fig 10. Variation of NOx with Brake power for various EGR percentages

.Figure 10. shows the variation at varying EGR conditions of NOx emissions with brake power. NOx emissions reduced through the use of the exhaust gas recirculation method. This is because the intake charge is diluted and the flame temperature reduced. NOx emissions were found to be 423ppm, 509ppm, 458ppm and 403ppm at 10 kg load conditions for diesel, D80-B20 without EGR, B20-D80 with 10% EGR and D80-B20 with 20% EGR respectively.

4.2.2 Carbon monoxide emissions:-

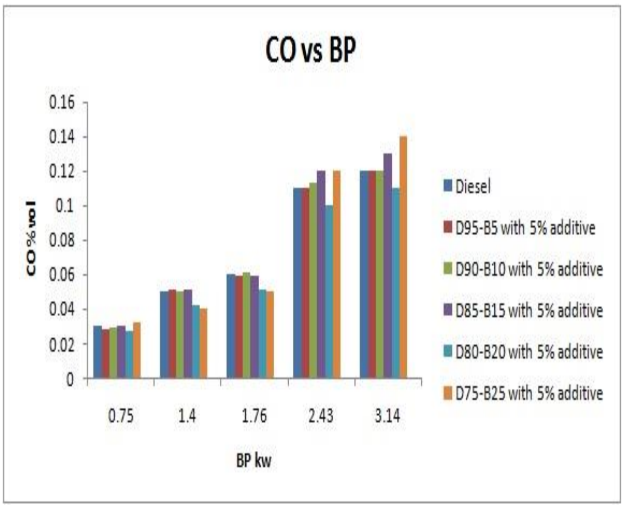

Figure 11. Variation of CO emissions with Brake power

Carbon monoxide (CO) is a slightly less dense, colorless, odorless, and tasteless gas than air. When found in levels above about 35 ppm, it is poisonous to hemoglobin animals (both invertebrates and vertebrates, including humans). The graph shows compares the emissions of carbon monoxide (CO) to the the brake power. In particular, when there is incomplete fuel burning, carbon monoxide can be seen in the exhaust.The blend D80-B20 and D75-B25 yield greater CO emissions from the graph at different load conditions and the D80-B20 produces lower CO percent by volume compared to all blends and diesel due to the oxygen needed for complete combustion. The CO emissions obtained for the blends such as Diesel,D95-B5,D90- B10, D85-B15,D80-B20 and D75-B25are 0.12%, 0.12%, 0.12%, 0.13% and 0.11%

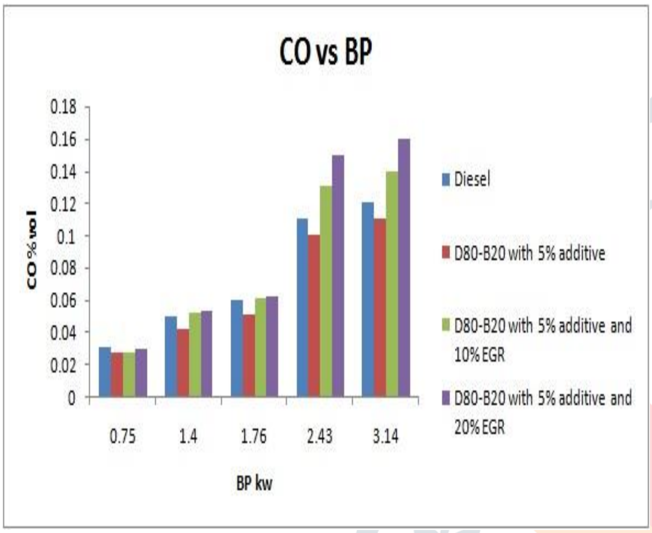

Fig 12. Variation of CO with Brake power for various EGR percentages

The graph shows the variation of CO emissions with Brake power at various EGR conditions.There is an increase in CO emissions due to a reduction in the air-fuel proportion when exhaust gas is circulated. The CO emissions for diesel, D80-B20 without EGR, D80-B20 with 10% EGR and D80-B20 with 20% EGR were found to be 0.12% vol, 0.11% vol, 0.14% vol and 0.16% vol respectively at 10 kg load conditions.

4.2.3Carbon dioxide emissions:-

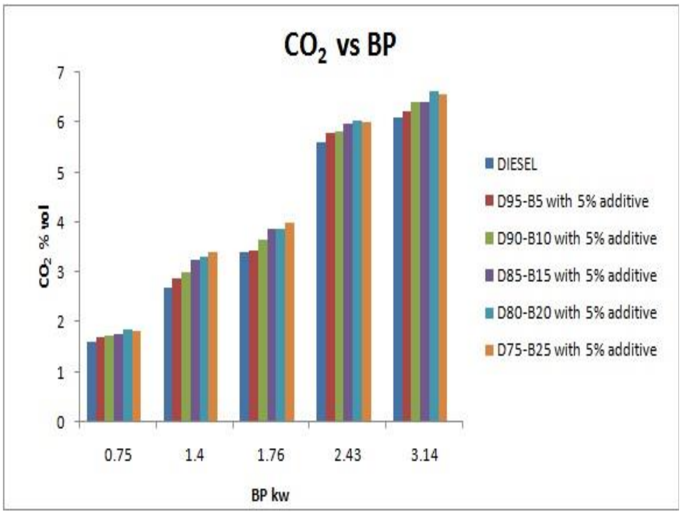

Fig 13. Variation of CO_2 emissions with Brake power

Figure 13.show variation of carbon dioxide with different blends with varying brake power is represented graphically. Only when complete combustion occurs in sufficient oxygen supply can carbon dioxide be seen in engine exhaust. The castor oil and rubber seed oils we used may have oxygen in excess. This gives excess oxygen that ultimately releases big amounts of exhaust carbon dioxide. In reality, carbon dioxide in the diesel engine was seen as pollution along side lived green house gas.The Blend D75-B25 and Diesel emits a lower CO2 emission of about 5.84 % and 6.1 % from the graph under different load conditions, whereas the D80-B20 mix provides 6.62 % of the CO2 emissions due to more amount of oxygen present in the fuel which make to bur the fuel completely than other fuel mixtures.

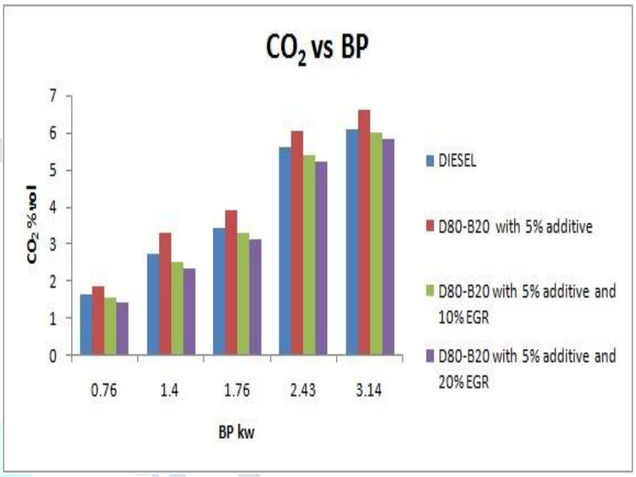

Figure 14. Variation of CO_2 with Brake power for various EGR percentages

Figure 14. shows the variation in CO2 emissions at different EGR conditions with Brake power. Carbon dioxide is the primary component of the recirculation of exhaust gas because it has greater heat carrying ability and serves as heat absorbing agent in the combustion process.The CO2 emission level for diesel, B20-D80 without EGR, D80-B20 with 10% EGR and D80-B20 with 20% EGR was found to be 6.1% vol, 6.62% vol, 6% vol and 5.8% vol respectively at 10 kg load conditions. This is due to the instability of combustion and oxygen deficiency, which causes CO2 to decrease and CO emissions to rise.

4.2.4 Unburnt HydroCarbons emissions (UHC):-

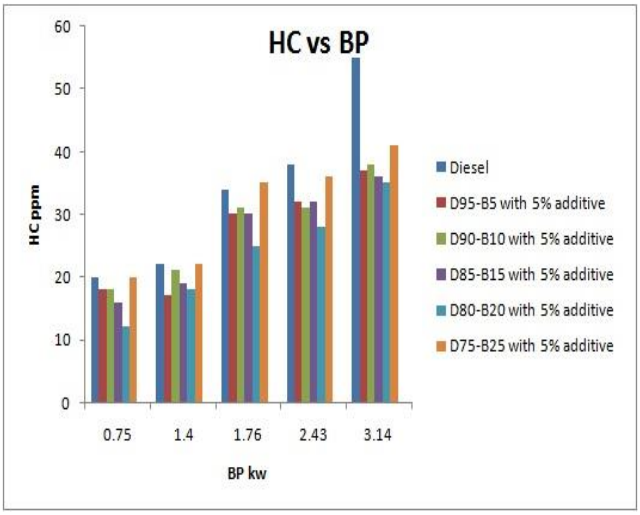

Fig15. Variation of HC emissions with Brake power

The Figure 15. shows the variety of unburned hydrocarbons with distinct blends at varying brake power. It is generally possible to see hydrocarbons in the exhaust when the fuel has a greater amount of cetane and there is not enough combustion. Compared to standard diesel, the cetane number of castor oil and rubber seed oil is lower, hence the magnitude of combustion rises and hydrocarbon emissions reduces. From the graph it is noted that all the blends other than diesel blends such as D95-B5,D90- B10, D85-B15,D80- B20 and D75-B25produce lower UHC emissions. The emissions of D85-B15 and D80-B20 are lower, i.e. 36 ppm and 35 ppm.

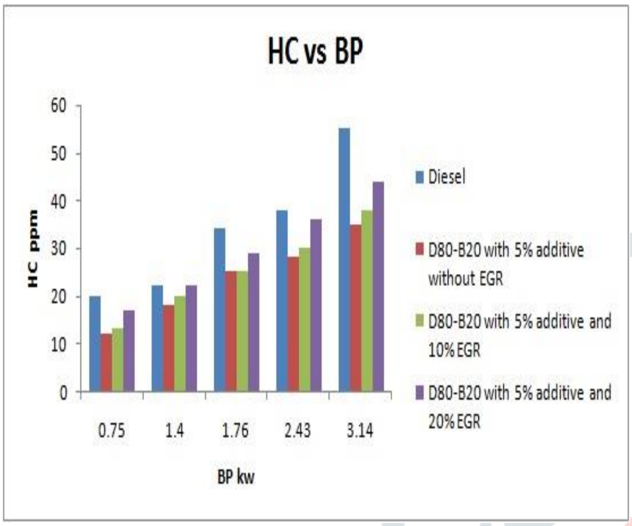

Figure 16. Variation of HC with Brake power for various EGR percentages

Figure 16. shows the variation in HC emissions at different EGR conditions with Brake power. The changes in unburned hydrocarbons follow a near trend with an rise in EGR leading to increased emissions of HC. The HC emission levels for diesel, D80-B20 without EGR, D80-B20 with 10% EGR and D80-B20 with 20% EGR respectively were found at 10 kg load conditions as 55ppm, 35ppm, 38ppm and 44ppm. The graph shows that with the rise in EGR concentration HC emissions rise but not up to diesel fuel emissions. The reasons for this exhaust emission are due to the reduction of oxygen in the inlet with EGR in the engine cylinder responsible for HC emission formation

4.2.5 Smoke Opacity:-

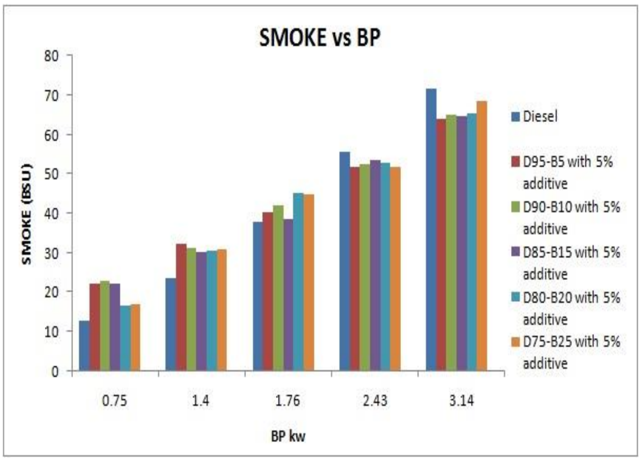

Fig17.Variation of Smoke opacity with Brake powerOpacity of smoke rises with load rises

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Figure 17. shows a graph plotted between the opacity of smoke and the engine brake power fuelled with blends i.e. Diesel, respectively D95-B5,D90- B10, D85-B15,D80-B20 and D75-B25. In general, smoke opacity of diesel fuel is higher than other fuels. Due to more oxygen content, smoke opacity reduces with an rise in the percentage of the mixture. The minimum opacity for smoke is in the blend D85-B15 (47.8BSU) while considering the fuel and the maximum in the blend D75-B25 (68.6BSU). Opacity of smoke in diesel fuel is noted at 71.8 BSU. It can be observed that the amount of smoke is lower than diesel for all blends. The reason for the lowered smoke is the availability within in the engine of premixed and homogeneous charge well before the start of combustion. The other reasons for reduced smoke are higher combustion temperature, longer combustion length and fast flame propagation. However, there was a noticeable white smoke emission at a greater load range due to lack of sufficient air and abnormal combustion. Because of the oxygen current in castor oil and rubber seed oil, another reason for reduced smoke may be better and complete combustion of fuel.

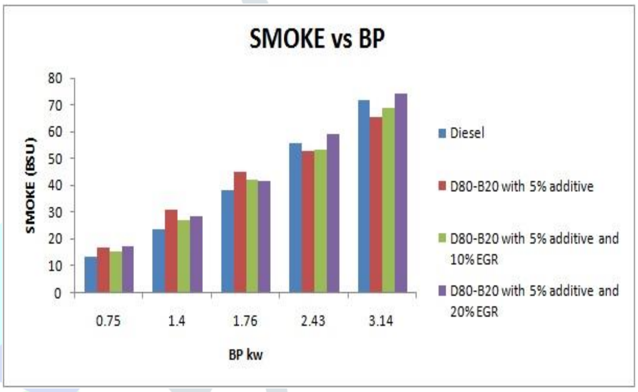

Fig 18. Variation of Smoke opacity with Brake power for various EGR percentages

Figure 18. shows the variability of Brake power and smoke emissions under different EGR conditions. Smoke emissions for diesel, D80-B20 without EGR, D80-B20 with 10% EGR and D80-B20 with 20% EGR were found at 10 kg load conditions as 71.8 BSU, 65.4 BSU, 68.9 BSU and 74 BSU respectively. The rise in concentration of smoke is due to partial exhaust gas replacement of air resulting in instability of combustion.

V. CONCLUSION

The following conclusions were taken on the basis of the experimental investigation into the performance and emission characteristics of a four-stroke single-cylinder VCR diesel engine fuelled with straight diesel and dual biodiesel in the varying blending ratios.coming to the Brake thermal efficiency of the engine increases with the increase in blending proportions from D95-B5 to D80-B20 and decreases at D75-B25. The maximum brake thermal efficiency obtained at blend D80-B20is 24.73% and minimum value is obtained at Diesel (D100) of about 22.58%. By using EGR brake thermal efficiency decreased and minimum Brake specific fuel consumption is obtained at blend D80-B20and maximum value is obtained at Diesel.The mechanical efficiency increased with increase in fuel blend proportions.The CO emission decreases with increase in the blending proportions where as at higher loads the CO emission is increased for the blends D75-B25i.e. 0.14 ppm. The minimum CO emissions occurred at D80-B20 blend of about 0.11 ppm.D80-B20 blend gives 6.62% of CO2 emissions.The UHC emissions decrease with increase in the blending proportions. The blendD80-B20 and D85-B15 gives lesser UHC emissions The formation of nitrogen oxides in the exhaust is due to the increased temperature. But in these case the engine temperature for D85-B15 and D80-B20 is equal to the conventional diesel and the blend D75-B25 is closer to diesel. Among all the blends D80-B20 and D75-B25 gives NO_X emissions very closer to diesel fuel and the blend B5-D95 gives maximum NO_X emissions In, general diesel fuel has more smoke opacity than other fuels. Smoke opacity decreases with increase in blend proportion due to more oxygen content. While considering the fuel the minimum smoke opacity is obtained for the blend D85-B15 and D80-B20.The blend B20-D80 gives lesser smoke opacity than diesel.By inceasing EGR concentarion the CO, HC and SMOKE emissions increased when compared with all the other blends.

Hence,D80-B20 is chosen as an optimum blend given to stronger results and can be used as a replacement for diesel fuel.

Nomenclature

Symbol	Abbrevation
BTE	Brake Thermal Efficiency
PPM	Parts Per Million
SFC	Specific Fuel Consumption
CO	Carbon Monoxide
HC	Hydrocarbons
CO_2	Cabon dioxide
NOx	Oxides Of Nitrogen
EGR	Exhaust Gas Recirculation

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