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# Single Cylinder 4 Stroke VCR Diesel Engine Performance And Analysis At Various Blends Of Fuels Under Various Cooling Rates

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***Abstract:***
*Typical engine fuels are blends of various fuels species, i.e., multi component. Thus, the original single component fuel vaporization model was replaced by a multi component fuel vaporization model .The model has been extended to model diesel sprays under typical diesel conditions, including the effect of fuel cetane number variation .Necessary modifications were carried out at the various cooling rates. .Found the performance of the diesel engine under various cooling rates at various cetane numbers.*

**1.Introduction**

The improved model was applied to simulate diesel engine under various cooling rates. The ignition characteristics of a diesel fuel are assumed to be given by the local Cetane number (CN) of that fuel at each point in the combustion chamber. The higher the CN, the more readily the fuel ignites, and vice versa. The local composition of the fuel in the engine determines the CN which is used to determine the ignition rate. There have been several suggestions in the literature to account for the effect of CN on ignition rates Heywood suggested a correlation in which the ignition activation energy is adjusted for CN changes using:

$$\boxed{E_A = \frac{61884}{CN+25}}$$

The cetane number is evaluated directly as a function of fuel composition. By using a blend of cetane (hexadecane) which has a cetane number of 100, and iso cetane(a-methyl-naphthalene) which has a cetane number of 0, it is possible to achieve blends with a wide range of cetane numbers. Moreover, the cetane numbers used in the ignition model are the local values of cetane number within the engine since the two components could be vaporizing at different rates due to their volatility differentials.Hence, inter actions between turbulence and chemical reactions have to be considered. A laminar-and-turbulent characteristic-time combustion model was used for the present study. The local time rate of change of the partial density of species m, due to conversion from one chemical species to another, is given by

$$\boxed{\frac{dY_m}{dt} = -\frac{Y_m - Y_m^*}{\tau_c}}$$

Where Ym is the mass fraction of species m, $Y^*_m$ is the local and the instantaneous thermodynamic equilibrium value of the mass fraction, and is the characteristic time for the achievement of equilibrium. To predict thermodynamic equilibrium temperatures accurately, it is sufficient to consider seven species: fuel, 02, N2, CO2, CO, H2, and H20, and to assume that the characteristic time is the same for all seven. As was previously assumed for the ignition model, the two fuel species are combined into an "effective" fuel for the purposes of the present implementation of the combustion model.

**2.Experimental Procedure**

Series of several experimental cycles were conducted with varying conditions of cooling rates and iterations were done with varying cetane number and the results were compared.

The exhaust gas Analyzer used is MN-05 multi gas Analyzer (4 gas version) is based on infrared spectroscopy technology with signal inputs from an electrochemical cell. Non-dispersive infrared measurement techniques uses for CO, CO2, and HC gases. Each individual gas absorbs infrared radiation absorbed can be used to calculate the concentration of the sample gas. Analyser uses an electrochemical cell to measure oxygen concentration. It consist of two electrodes separated by an electrically conducted liquid or cell. The cell is mounted behind a polytetrafluorethene membrane through which oxygen can diffuse. The Device therefore measures oxygen partial pressure. If a polarizing voltage is applied between the electrodes the resultant current is proportional to the oxygen partial pressure.

The engine used in the present study is a Kirloskar AV-1, single cylinder direct injection, Water cooled diesel engine with the specifications given in Table 1. Diesel injected with a nozzle hole of size 0.15mm.the engine is coupled to a dc dynamometer. Engine exhaust emission is measured. The load was varied from 1 kilowatt to 2 kilowatts. The amount of exhaust gas sent to the inlet of the engine is varied. At each cycle, the engine was operated at varying load and the efficiency of the engine has been calculated simultaneously. The experiment is carried out by keeping the compression ratio constant i.e., 16.09:1.

*2.1.Table Of Engine Specifications*

| **Type** | **Four- stroke, single cylinder, Compression ignition engine, with variable compression ratio.** |
|---|---|
| Make | Kirloskar AV-1 |
| Rated power | 3.7 KW, 1500 RPM |
| Bore and stroke | 80mm×110mm |
| Compression ratio | 16.09:1, variable from 13.51 to 19.69 |
| Cylinder capacity | 553cc |
| Dynamometer | Electrical-AC Alternator |
| Orifice diameter | 20 mm |
| Fuel | Diesel and Biodiesel |
| Calorimeter | Exhaust gar calorimeter |
| Cooling | Water cooled engine |
| Starting | Hand cranking and auto start also provided |

*Table 1: Specifications Of Engine*

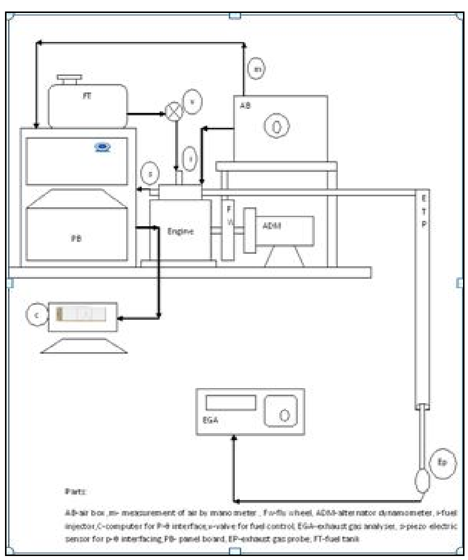


*Figure 1: Block Diagram Of The Experimental Setup*

*2.2.Parts*

AB-air box ,m- measurement of air by mano meter , Fw-fly wheel, ADM-alternator dynamometer, i-fuel injector,C-computer for P-θ interface,v-valve for fuel control, EGA-exhaust gas analyser, s-piezo electric sensor for p-θ interfacing,PB- panel board, EP-exhaust gas probe, FT-fuel tank

**3.Results**

Significant results were obtained after conducting of several experimental cycles with varying cooling rates and blends at different loads.

*3.1.Nomenclature*

| | |
|---|---|
| Blend1 | 95%diesel+5%kerosene (CN=48) |
| Blend2 | 90%diesel+10%kerosene (CN=45) |
| ηBTE | Brake thermal efficiency |
| ηvolumetric | Volumetric efficiency |
| BSFC | Brake specific fuel consumption |
| CR | Cooling rate in LPM |
| CN | Cetane number |

*Table 2: Nomenclature*

*3.2.Brake Thermal Efficiency*

From the graph below it is clear that pure diesel at cooling rate 6LPM has shown highest performance at all the loads where as blend 2 has shown a least performance which is to 41% of the performance of pure diesel at 6LPM. Figure 2 shows the comparative data of all the brake thermal efficiencies. To have some clear picture on effect of cooling rate on various blends Figure 3 considers the brake thermal efficiency at peak loads.

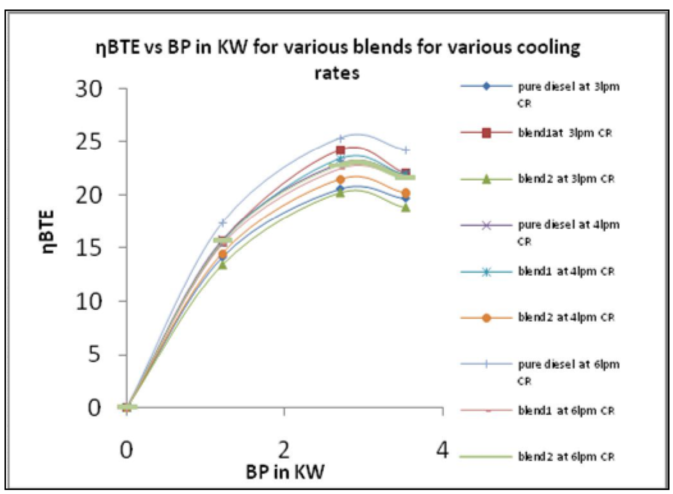


*Figure 2: Hbte Vs BP In KW For Various Blends For Various Cooling Rates*

From Figure 3 we infer that blend1 & blend2 was performing 10%more than that of the pure diesel at peak load conditions at 6LPM blend2 has shown drastically lowest performance proving that lower cetane number fuels are not suitable for hat conditions.

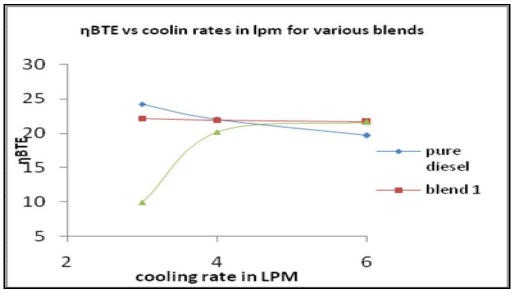


*Figure 3: Hbte Vs Coolin Rates In Lpm For Various Blends*

*3.3.Volumetric Efficiency*

Figure 4 gives the inference that pure diesel at 6LPM cooling rate has shown higher volumetric efficiency. Any how the trend of varying volumetric efficiency has stood very general , Figure 5 gives a clear picture of the effect of cooling rate on volumetric efficiency.

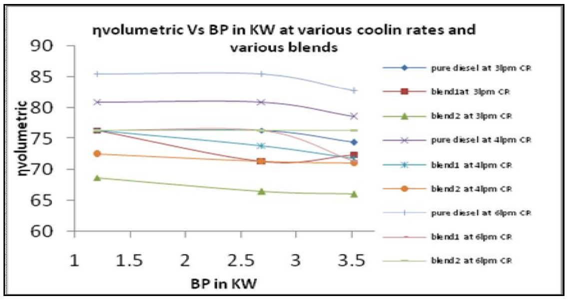


*Figure 4: Hvolumetric Vs BP In KW At Various Cooling Rates And Various Blends*

Figure 5 shows that at all cooling rates pure diesel has shown performance which is to mean of 21% more than that of blend1 & blend2.

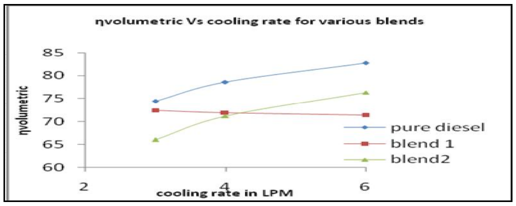


*Figure 5: Hvolumetric Vs Cooling Rate For Various Blends*

*3.4.A/F Ratio*

Air fuel ratio even at different blends and different cooling rates the variation was not so substantial except to that of blend2 at 6LPM cooling rate with 50% of the A/F ratio of other conditions as seen from Figure 6.

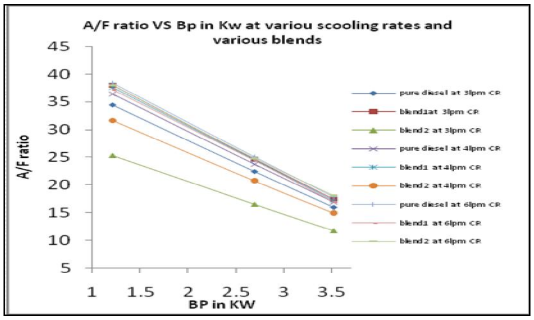


*Figure 6: A/F Ratio VS Bp In Kw At Various Cooling Rates And Various Blends*

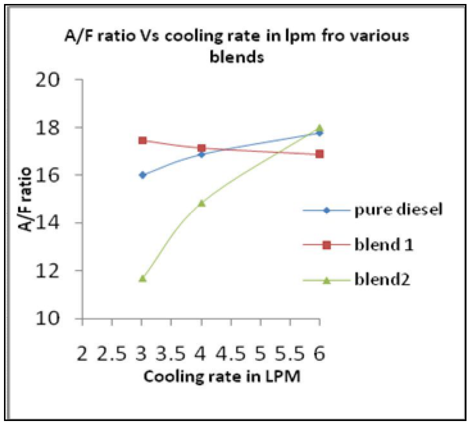


*Figure 7: A/F Ratio Vs Cooling Rate In Lpm At Various Blends*

*3.5.Brake Specific Fuel Consumption*

Figure 7 shows pure diesel at 6LPM shows lowest brake power and blend2 showing highest BSFC at 3LPM again showing the low cetane number blends are not suitable for hot or inadequate cooling condition.

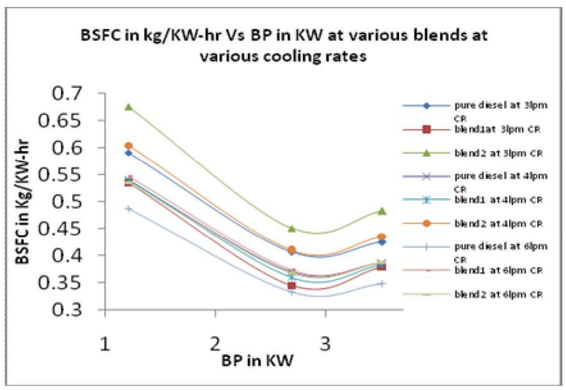


*Figure 8: BSFC In Kg/KW-Hr Vs BP In KW At Various Blends At Various Cooling Rates*

Taking only the values at peak loads Figure 9 shows pure diesel showing BSFC 24% less to that of blend2, where blend1 has shown no substantial rise or fall in the BSFC at all the cooling rate.

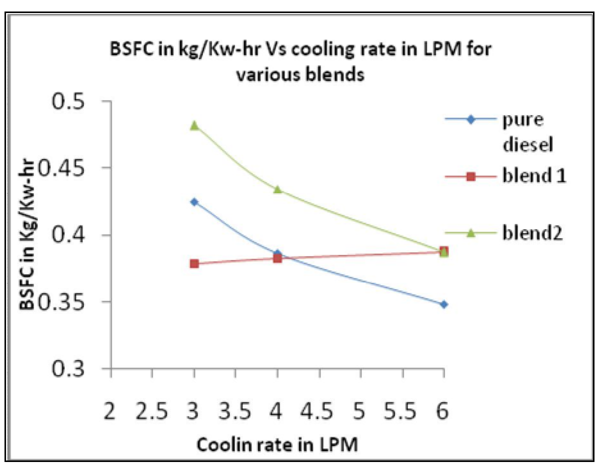


*Figure 9: BSFC In Kg/Kw-Hr Vs Cooling Rate In LPM For Various Blends*

*3.6.Emissions*

3.6.1.$No_x$ Emissions

From Figure 10 we infer that blend 1 has shown comparatively lower $NO_X$ emissions which by trend increases along with increased cooling rate as shown in the Figure 10.Blend1 at all cooling rates recorded a mean value of 44% less $NO_x$ in PPM compared to that of pure diesel & blend2.

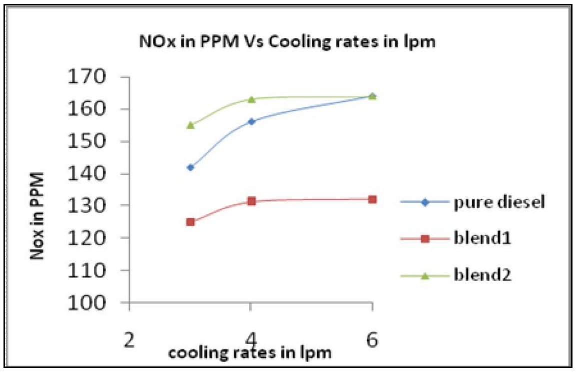


*Figure 10: Nox In PPM Vs Cooling Rates In Lpm*

3.6.2.CO Emissions

From Figure 11 we infer that blend1 has shown lowest emissions of CO when compared to that of pure diesel and blend2.

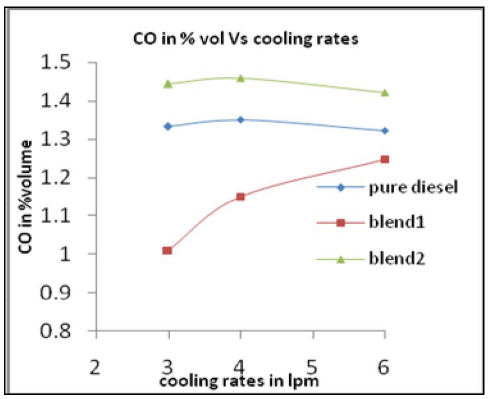


*Figure 11: CO In % Vol Vs Cooling Rates*

3.6.3.HC Emissions

From Figure 12 we infer that blend1 has over all of less HC emissions except at 6LPM cooling rate which is only of 7% more than that of diesel at 6LPM cooling rate. Where as blend2 goes higher more than23% of that of diesel.

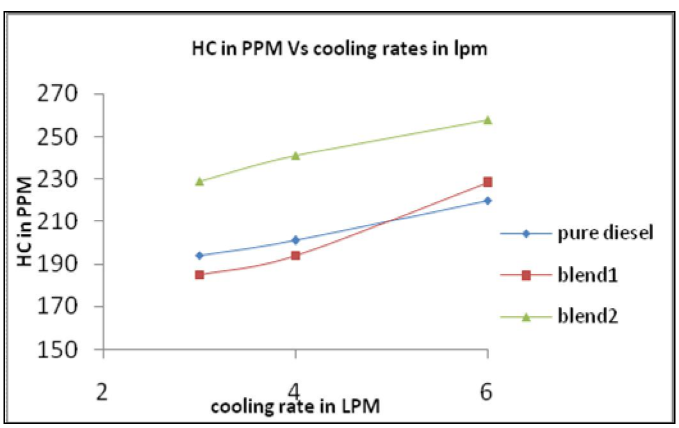


*Figure 12: HC In PPM Vs Cooling Rates In Lpm*

**4.Coclusion**

From the above obtained results the following conclusions were drawn

- Pure diesel at cooling rate 6LPM has shown highest performance at all the loads where as blend 2 has shown a least performance which is to 41% of the performance of pure diesel at 6LPM. Blend1 & blend2 was performing 10%more than that of the pure diesel at peak load conditions at 6LPM blend2 has shown drastically lowest performance proving that lower cetane number fuels are not suitable for hat conditions.
- Pure diesel at 6LPM cooling rate has shown higher volumetric efficiency. Any how the trend of varying volumetric efficiency has stood very general , Figure 5 gives a clear picture of the effect of cooling rate on volumetric efficiency.
- Air fuel ratio even at different blends and different cooling rates the variation was not so substantial except to that of blend2 at 6LPM cooling rate with 50% of the A/F ratio of other conditions.
- Pure diesel at 6LPM shows lowest brake power and blend2 showing highest BSFC at 3LPM again showing the low cetane number blends are not suitable for hot or inadequate cooling condition.
- Blend 1 has shown comparatively lower $NO_X$ emissions which by trend increases along with increased cooling rate as shown in the Figure 10.Blend1 at all cooling rates recorded a mean value of 44% less $NO_x$ in PPM compared to that of pure diesel & blend2.
- The blend1 has over all of less HC emissions except at 6LPM cooling rate which is only of 7% more than that of diesel at 6LPM cooling rate. Where as blend2 goes higher more than23% of that of diesel.

The above points make blend1 recommendable for usage under cold conditions more than that of pure diesel and blend2.

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