Experimental Study on the Effect of Biodiesel and Turpentine Oil a DI diesel Engine Characteristics

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Abstract : This article analyses an experimental investigation into the viability of mixing biodiesel with turpentine. Using the use of dual fuel mode, the performance, emission, and combustion characteristics of a diesel engine were investigated. Diesel was introduced into the engine by a standard fueling device known as an injector in varied proportions, blended with 20% biodiesel. The study showed that all performance and emission criteria are better in Turpentine blends than those of diesel fuel. Compared to diesel, hazardous gases like CO and HC are a little more prevalent. Turpentine mixes mode results in reduced smoke. At all loads, it is discovered that the pollutant Nox is higher than that of diesel. This study has demonstrated that a dual fuel mode with minimal engine modification allows for a 15% diesel replacement with turpentine.

***IndexTerms* - Biodiesel; Emission; Combustion; Turpentine.**

I. INTRODUCTION

Researchers and scientists from all over the world have been driven to create a workable alternative to a non-petroleum fuel due to the world's ever rising energy needs, the oil reserves' rapid depletion, and growing concerns over serious environmental issues. Vegetable oils, which are derived from renewable sources, have been reported to be an appropriate alternative fuel by numerous researchers (Deivajothi et al 2017). Besides that, such alternative fuels could be straight used for the engines or even with minor modifications. High viscosity, the reaction of unburned hydrocarbons, which creates fuel injection nozzle foulants and carbon accumulation on cylinder walls, gum creation in the oxygen-rich fuel, gelatin all through stockpiling, the amount of free fatty acids, and lube oil stiffening are all issues. Over the years, investigators have attempted diligently to address a few of these problems (Sukumar et al 2017). Transesterification of bio-fuel can solve the issue of gum creation, and lubricating oil added to bio-fuel can protect oxidative damage. One of most common issue experienced to vegetable oil working is high viscosity, which causes inadequate atomization and therefore incomplete combustion. One technique of reducing overall higher solubility of biodiesel is to mix it with a certain percentage of diesel through quantity. Some other way to improve atomization and reduce viscosity is to pre-heat vegetable oil prior to actually introducing it into a combustion chamber.

Biodiesels are additionally thought of as an acceptable alternative to traditional diesel, with thorough research by now completed in this area. However, the issue with elevated viscosity had also demonstrated to be a serious obstacle to making sure its applicability as a perfect diesel substitute (Karikalan et al 2016). To solve this problem, numerous research studies have been released on the mixture of low viscosity biofuel production such like methyl esters, plant-based biofuels including essential oils and turpentine with conventional diesel or highly viscous biodiesel like Karanja, Cashew Nut Shell oil, Jatropha (Manieniyan et al 2015). In this particular instance. Tamilselvan et al. (2017) investigated the performance, combustion and emission character of a diesel engine powered by a mixture of 92% Karanja methyl ester and 10 percentage methanol. The results of this study revealed a 4.2% rise in highest heat efficiency at 80 percentage load, in addition to a decrease in NOx and smoke emission levels. Dubey et al (2017) investigated the implications of engine technology, performance, and emissions when using Jatropha esterification as a fuel. Enhancing the compression ratio and injectors pressure increased performance of the engine, he found. Brake thermal efficiency risen by 8.9% at a higher compression of 18 and inlet pressure of 250 bar, while HC emission and temperature of exhaust gases risen whereas CO and smoke reduced.

Turpentine, such as organic solvents and some other plant-based biodiesel, has already been discovered to be appropriate for use in diesel engines. Turpentine is a biofuel obtained from coniferous tree wax with a lower density, volatility, and a marginally greater calorific value than petrodiesel. Pravinkumar et al. (2017) proved that a 31% turpentine/diesel mixture engine shows lower HC, NOx, CO, smoke emissions and particulate matter. He discovered that turpentine blends of 42% and 51% resulted in lower brake thermal efficiency and though lesser exhaust emissions. According to Panneerselvam et al. (2016), whenever the engine has been operate in dual fuel operation (turpentine like a main resource and diesel like an ignition source), most characteristics as well as emission characteristics enhanced with the exception of UBHC and CO emission, whilst also NOx emission managed to remain identical to that of diesel fuel. In dual fuel mode, he claims a 40-45% reduction in smoke.

This research thoroughly investigated the performance and emission aspects of the engine fueled with Jamun Seed Oil-turpentine. By studying the impact of fuel properties whereas differing the turpentine percentage in the blend, the objective of this investigation is to identify an appropriate, so that greater combustion and lesser exhaust emission can be achieved.

II. MATERIALS AND METHODS

In this study, a single cylinder diesel engine was fueled with Jamun Seed Oil-turpentine blends to assess the effect on engine performance and emission parameters and to find a suitable blend. Jamun Seed Oil and turpentine oil were obtained from commercial stores. Transesterification was used to convert Jamun Seed Oil oil into Jamun Seed Oil methyl ester. Transesterified Jamun Seed Oil oil generally meets ASTM standards for properties. Blends were created by combining Jamun Seed Oil methyl ester and turpentine oil in the following volumes: DIESEL+B20+T5, DIESEL+B20+T10, and DIESEL+B20+T15. Some of the major properties of the resulting blends were tested in the laboratory in accordance with the Standard astm testing process.

III. EXPERIMENTAL SETUP

The tests were performed on a four-stroke, single-cylinder, direct injection, water-cooled, naturally aspirated engine. Table 1 shows the specifications of an engine. For applying brake load, an eddy current dynamometer is coupled to the engine. The engine has a rated capacity of 3.5 kW at 1500 RPM and a compression ratio of 17.5. A piezoelectric sensor is installed in the engine to measure pressure. The output values from across all load, temperature, and pressure sensors are routed to the control panel. To keep track of exhaust gas emissions in AVL DI gas. It can measure CO, NO_X and HC emissions. Smoke was measured using an AVL smoke meter. To reduce the effects of fluctuations, the experiment was repeated three times to select average peak values.

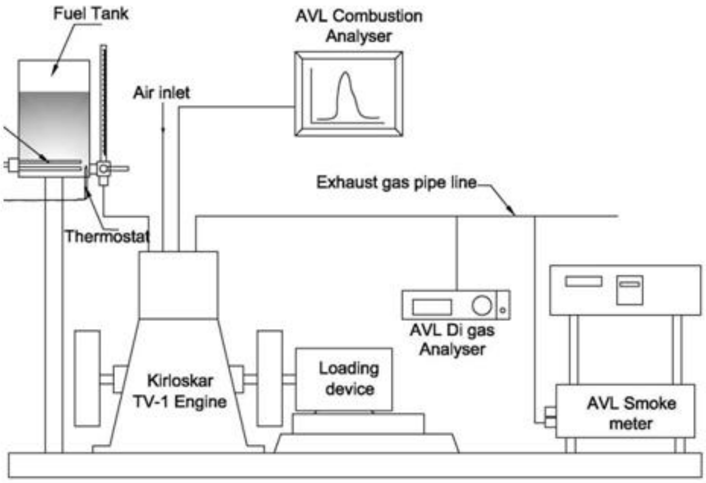


Figure 1 Experimental setup

Table 1 Specification of the Engine

Injection timing	:	23° before TDC
Compression ratio	:	17.5:1
Speed	:	1500 rev/min
Bore	:	87.5 mm
Type	:	Vertical, Water cooled, Four stroke
Stroke	:	110 mm
Dynamometer	:	Eddy current
Maximum power	:	5.2 kW
Injection pressure	:	220 kgf/cm^2
Number of cylinder	:	One

IV. RESULTS AND DISCUSSION

Performance characteristic:

Figure 2 depicts the variation of SFC with brake power output. Because turpentine has a higher heat content than diesel fuel, dual fuel mode results in a very small rise in fuel consumption. At full load in dual fuel mode, there is a significant increase in fuel consumption. The main cause of the improved fuel consumption is the start of knock at 80% load. Knock occurrence is indicated by an unusual increase in peak pressure, increased cylinder pressure fluctuation, and pinking noise (Silambarasan et al 2017). Improper rate combustion at maximum loading results in poor combustion efficiency and rising fuel consumption. The DIESEL+B20+T10 blend shown lower SFC compare to all other blends.

Figure 3 depicts the relationship between brake thermal efficiency and brake power output. At all loads, the dual fuel mode engine's brake thermal efficiency is clearly comparable to that of the DIESEL+B20+T10. When compared to other blends, the maximum efficiency of a dual fuel mode engine is 32% and the maximum efficiency of a DIESEL+B20+T10 is 31% at all loads. The occurrence of knock is the cause of the decrease in brake thermal efficiency as the blend is increased. More octane fuel admission at full load increases the likelihood of knock (Manieniyan et al 2015). As a result, break thermal efficiency is low at all loads.

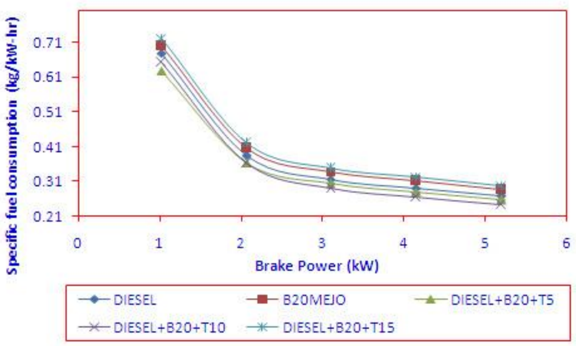


Fig 2 specific fuel consumption with brake power

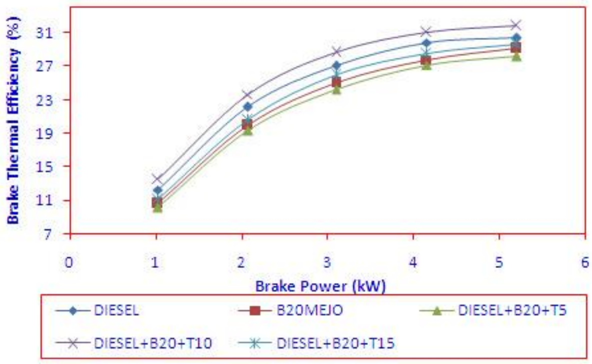


Fig 3 Brake thermal efficiency with brake power

Emission characteristic

The emission performance of turpentine DIESEL+B20 blends at various load conditions is shown in Fig. 4. From 0% to full load, the CO emission of the turpentine-diesel dual fuel mode engine gradually increases. Higher fumigation rates and oxygen scarcity are the causes of increased CO emission at lighter loads. The flame quenching as well as vented surface of control close to the wall will also cause a rise in CO emission. The incidence of knock and greater fuel consumption lead to a substantial rise in CO emissions at full load. When compared to other blends, the DIESEL+B20+T10 blend emits approximately 35% less CO at 100% load. Because irregular combustion decrease oxygen supply, a huge rise in CO at full load leads to greater CO emissions (Suresh Kumar et al 2018).

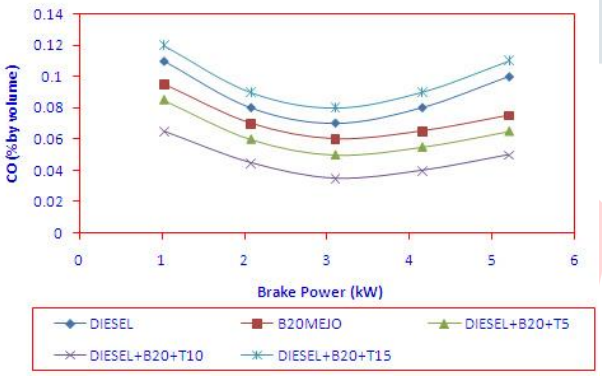


Fig 4 Carbon monoxides with brake power

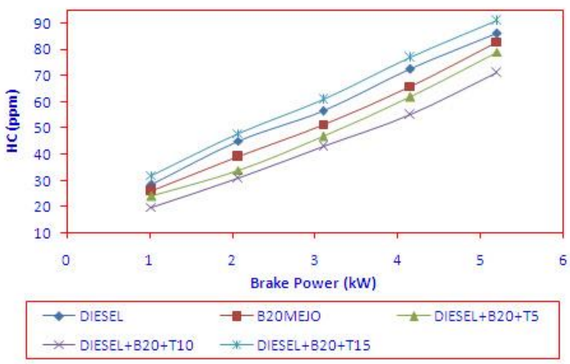


Fig 5 Hydro carbon with brake power

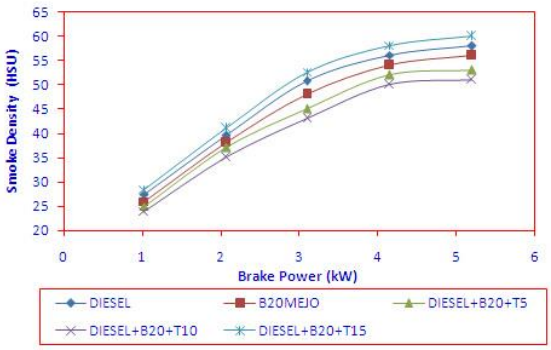
From 0% to 100% load, the HC emission of turpentine DIESEL+B20 blends gradually increases. Higher fumigation rates and oxygen scarcity are the causes of increased HC emissions. The burn cavitation and warmed layer of control close to the barrier will cause higher HC emission. Despite the fact that this is traditional dual fuel method engine behaviour on the part, the substantial increase in HC emission at full load is because of insufficient combustion and increased fuel economy. The HC emission level is lower at lighter loads due to charge homogeneity and higher oxygen availability. When compared to other blends, the DIESEL+B20+T10 blend emits about 15% less CO at all loads (Santhanakrishnan et al 2016).

Figure 6 depicts the variation of smoke emission with brake power. Turpentine with DIESEL+B20 blends reduces smoke more effectively. The presence of such a premixed and uniform control within the engine prior to combustion records for the lesser smoke emission. Other reasons for lower smoke levels include higher turpentine heat content, peak combustion temperature, extensive heat release, and faster flame propagation. The pilot fuel was used in smaller quantities, which may have contributed to the reduction in smoke emissions (Sukumar et al 2015).

Figure 7 depicts the variation of NOx with brake power. At all loads, the NOx of DIESEL+B20+T10 closely follows the diesel. Higher heat release during in the second stage of combustion, maximum combustion temperature, as well as greater turpentine heating value all lead to higher NOx. Increased NOx may also be caused by increased diesel engine ignition delay, which promotes diffusive combustion. Because of the abnormal combustion, there is a significant increase in NOx level above 80% load. This results in extremely high gas temperatures, increased fuel consumption, and inefficient heat utilization. The availability of hot gas within the cylinder creates an ideal surroundings for nitrogen cycle to react to oxygen (Manieniyan et al 2015).

Combustion characteristic

Figure 8 depicts the relationship between cylinder pressure and crank angle in turpentine with DIESEL+B20 blends. The figure shows that the engine's ignition delay is significantly longer than that of diesel. This is due to the fact that pilot fuel is infused in a less oxygen-rich surroundings, and the mixture's internal temperature at the time of injection is less than the temperature of the diesel operation. As a consequence of absorbing the heat from the combustion process, turpentine oil evaporates within the cylinder. The longer the ignition delay at increased load ranges, the higher the maximum pressure and the greater the pressure variability (Nallusamy et al 2017).



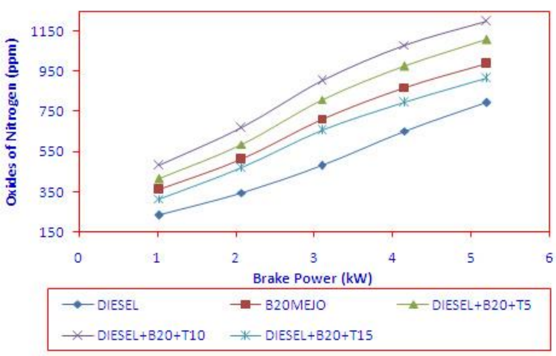
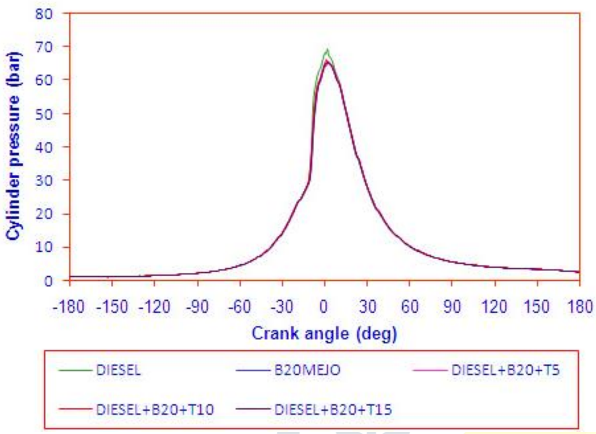


Fig 6 Smoke density with brake power

Fig 7 Oxides of nitrogen with brake power

The heat release rate of turpentine is compared to that of DIESEL+B20 blends in Fig. 9. It demonstrates that diesel mode has a longer ignition delay than turpentine with DIESEL+B20. The majority of heat release occurs only during the diffusive combustion phase. A higher ignition interruption allows for more heat release during diffusional burning process. As a consequence, the period of heat release in turpentine with DIESEL+B20 is also longer, leading to a rise in EGT. The prolonged combustion is caused primarily by the slow burning of infused fuel as well as flame propagation burning of instated turpentine (Ashok et al 2017).



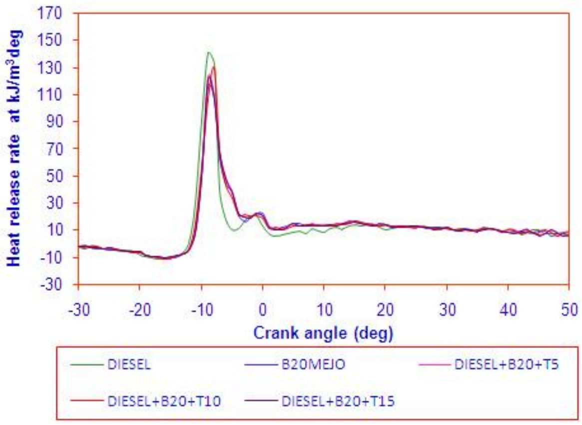


Fig 8 Cylinder Pressure with Crank angle

Fig 9 Heat release rate with Crank angle

V. CONCLUSION

From the detailed test conducted with turpentine, B20 and diesel blends the following conclusions are arrived:

The DIESEL+B20+T10 blends have a 5% higher brake thermal efficiency than other turpentine blends at all loads. Due to the presence of incomplete combustion, higher turpentine bends result in increased SFC at full load. EGT and NOx levels in DIESEL+B20+T10 are higher. When compared to other bends, the DIESEL+B20+T10 blend emits approximately 20% less CO. At full load, the dual fuel engine emits 18% more HC and emits 25% less smoke. Because of steadily increasing octane fuel admitted, the engine starts to knock at full load. The above test comes to the conclusion that it was under all load conditions, DIESEL+B20+T10 substitute of diesel to turpentine is really quite feasible. Exception of rising CO and HC emissions, the DIESEL+B20+T10 blend outperformed the DIESEL+B20+T10 blend in every other aspect of emission variables such as smoke and NOx, as well as performance parameters such as brake thermal efficiency.

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