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Experimental Investigation on the Engine Performance and Emission Behaviour of Turpentine – Diesel Dual Fuel Operated Modified DI Diesel Engine

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ABSTRACT

The dual fuel technique offers the advantage of easy conversion of the diesel engine to work on the dual fuel mode with little engine modifications. In this mode, the primary fuel (turpentine) is inducted along with inlet air stream and the pilot fuel (diesel) is injected through the regular injection system. The primary fuel (turpentine) shared the maximum energy in the power production but the pilot fuel shared only a least part and act as an ignition source. The engine with

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this facility has been operated under various load conditions and at various turpentine energy shares. The energy share that provides better performance has been identified and compared with diesel fuel operation. From the obtained results it was proved that this engine offered higher BTE at full load and smoke free operation at all loads. The CO and HC emissions were higher and operated with very low NOx emission at all loads. The results also proved that the dual fuel operation successfully replaced 75% of diesel usage with turpentine fuel.

Keywords: *Turpentine, dual fuel engine, energy share, brake thermal efficiency, CO emission, HC emission, NOx Emission and Combustion*

Nomenclature

DBL	Diesel Baseline
CI	Compression Ignition
DI	Direct Injection
BTE	Brake Thermal Efficiency
EGT	Exhaust Gas Temperature
CO	Carbon Monoxide
UBHC	Unburned Hydrocarbon
NOx	Oxides of Nitrogen
TDI	Turpentine Direct Injection
BTDC	Before Top Dead Centre
FIP	Fuel Injection Pump
HRR	Heat Release Rate
HC	Hydro Carbon
VE	Volumetric Efficiency

Introduction

Ever increasing fuel price, continuous addition of on road vehicles, fast depleting petroleum resources and continuous accumulation green house gases are the main reasons for the development of alternative fuels. Many alternative fuels have been indentified and tested successfully in the existing engine with and without engine modification. However, still researches are continuing in this field to seek best alternatives which are offering best fuel characteristics.

Most of the alternative fuels suggested today are bio fuels and are proved to be a partial substitute for existing one [1]. However, the various admission techniques experimented earlier are giving good solution to apply larger fraction of replacing fuel in the existing engine. The primary advantage of this kind of fuel is renewable and eco-friendly. These fuels are identified well before the exploration of the other promising alternatives fuels [2].

Generally, bio fuels are obtained from plants and animals. Of which plants contribute more by supplying large quantity of biofuels. A plant generally yields two types of oils namely triglyceride oil (TG oils) and turpene oil (light oil). The triglyceride oils are obtained from plant seeds but turpene oils are obtained from all parts of the plant [3]. The TG oils are having higher viscosity but terpene oils are having lower viscosity and appreciable fuel properties. In addition, its availability in natural resources is more compared to triglyceride oil. Also, this kind of oils are available in abundant in some plant species namely eucalyptus, pine tree etc.

The present investigation used one such oil called Turpentine; a volatile fraction of oleoresin obtained from various specious of pine tree. The properties of turpentine fall in between the properties of petrol and diesel fuel. Also, few of them are also closer to that of diesel oil [4] (Table 1). The present investigation used turpentine as a primary fuel and diesel fuel as a pilot fuel.

Basically, dual fuel operation helps to admit low cetane fuel in CI engine and also this method permits more diesel replacement than bi-fuel technique. Hence, many researchers prefer this method to admit large fraction of low cetane fuel in CI engine. Dual fuel engine generally consists of two separate fuel admission devices such as carburetor and injector to admit pilot and primary fuel separately.

The fuel that is admitted through carburetor provides maximum energy share during combustion compared to the fuel that is used for ignition purpose. Also, this engine accommodates all kind of cetane fuels are the igniter and hence,

Table 1: Physical and Chemical Properties of Turpentine

Properties	Gasoline	Diesel	Turpentine
Formula	C_4 to C_{12}	C_8 to C_{25}	$C_{10}H_{16}$
Molecular Weight	105	200	136
Composition % Wt	C 88 H 15	C 87 H 16	C 88.2 H 11.8
Density kg/m^3	780	830	860-900
Specific Gravity	0.78	0.83	0.86-0.9
Boiling Point °C	30-220	180-340	150-180
Viscosity c St	0.4	3-4	2.5
Latent Heat of Vaporization kJ/kg	350	230	305
Lower Heating Value kJ/kg	43,890	42,700	44,000
Flash Point °C	- 43	74	38
Auto Ignition Temperature °C	300-450	250	300-330
Flammability limit % Volume	1.4	1.0	0.8
Cetane Number		40-55	20-25

worldwide researchers are using this method for testing various combinations of pilot and primary fuels. One such investigation conducted by [7] using vegetable oil and methanol showed the higher thermal efficiency and lower smoke emission compared to baseline operation. Also, in another test [5] showed better performance and emission characteristics in dual fuel engine using jatropha oil and orange oil. Similarly, [8] conducted a performance test using propane and diesel fuel and showed increased CO and HC emission and decreased NOx emission.

Most of the researchers used this method to admit gaseous fuel in CI engine and some of them are used this method for admitting liquid fuels. Invariably, all researchers obtained better performance and unusually high gaseous emissions such as CO and HC [6]. Other emissions such as smoke and NOx were found lower than diesel fuel operation.

Turpentine

Turpentine is a kind of volatile essential oil obtained from oleoresin exuded from the pine tree, when subjected to mechanical injury. Pine trees are the world's known tallest, biggest, oldest and most populated trees (even 5000 year old trees are known to exist!). Genus *Pinus* is one of the most widely distributed genera of trees in the northern and southern hemisphere, extending from the polar region to the tropics.

Pine trees (coniferous trees) naturally has a kind of resin, which is rich in chemical compounds such as terpenes, fatty acids, waxes, tannins and phenolics. This has been collected from the tree by bark chipping and borehole methods. The main function of the resin is to protect the tree against insect, pests and diseases, and act as energy reserves. The crude oleoresin is converted into its primary fractions called gum rosin and turpentine by steam distillation process. Pine trees can be easily cultivated in wastelands and they need very little or no water and human effort.

Experimental Setup

The engine setup shown in Figure 1 used for experimental investigation is a single cylinder, air cooled, and vertical and direct injection diesel engine. It is capable of developing 4.4 kW at a constant speed of 1500 rpm and coupled to an eddy current dynamometer. The inlet side of the engine consists of anti-pulsating drum, air heater and air temperature measuring device. The exhaust side of the engine consists of EGT indicator, exhaust gas analyzer and smoke sampler. This set-up also consists of two separate fuel admission device and measuring device for admitting and measuring diesel oil and turpentine oil separately.

A 64 bit DAQ system is also provided with the test rig to acquire crank angle and cylinder pressure data at stipulated intervals of crank angle.

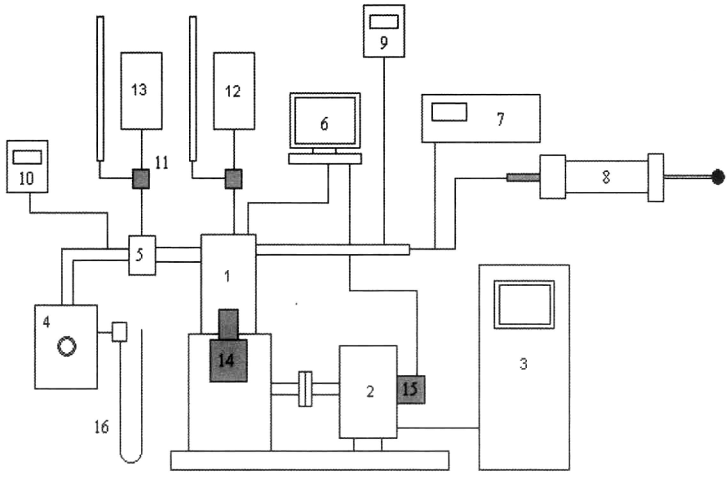

Figure 1: Experimental Setup for Dual Fuel Operation

1 Diesel Engine, 2 Eddy current Dynamometer, 3 Dynamometer Control, 4 Anti pulsating Drum, 5 Fumigator, 6 P-IV computer with DAQ, 7 Gas Analyzer Fumigator, 8 Smoke sampling pump, 9 Exhaust temperature indicator, 10 Air inlet temperature indicator, 11 Two way valve, 12 Diesel fuel, 13 Turpentine, 14 Fuel Injection Pump, 15 Crank angle encoder, 16 Manometer

Methodology

This method uses two separate fuel admission devices to admit pilot and primary fuel separately. In which, the diesel fuel was used as a pilot fuel and turpentine was used as a primary fuel. The primary fuel contributes more in energy production whereas the pilot fuel contributes less and used to initiate auto-ignition in the turpentine air mixture. The primary fuel (turpentine) was inducted through a throttleless carburetor and the pilot fuel (diesel) was injected through regular fuel injection system.

The engine modified for this method was started using diesel fuel and then the primary fuel turpentine was admitted gradually into the engine. The admission of primary fuel reduces the pilot quantity and keeps on running the engine. However, at one particular primary fuel opening the engine starts misfiring. This is the maximum level of primary fuel at that load condition. Similarly, the same experiment was repeated for various load conditions and its corresponding parameters were observed. At each energy share and at each load, the output parameters such as brake thermal efficiency, volumetric efficiency, exhaust gas temperature, CO, HC, NOx, smoke and combustion parameters were observed and used for the determination of performance and emission characteristics of dual fuel engine.

Results and Discussion

Determination of Optimum Energy Share of Turpentine

In this method, as the volatile primary fuel was inducted during suction stroke, removes considerable amount of heat from the cylinder due to evaporation. This reduces the heat available inside the cylinder for auto-ignition. Also, the amount of heat removed from the cylinder varies with respect to the quantity of induction. Hence, all inducted quantity never gives the same performance. Therefore, the optimum induction quantity that gives the maximum performance must be identified for each load.

Brake Thermal Efficiency

Figure 2 shows the variation of brake thermal efficiency with turpentine energy share at different loads. It is observed that the brake thermal efficiency increases with increase in turpentine flow rate reaches a maximum and thereafter decreases considerably.

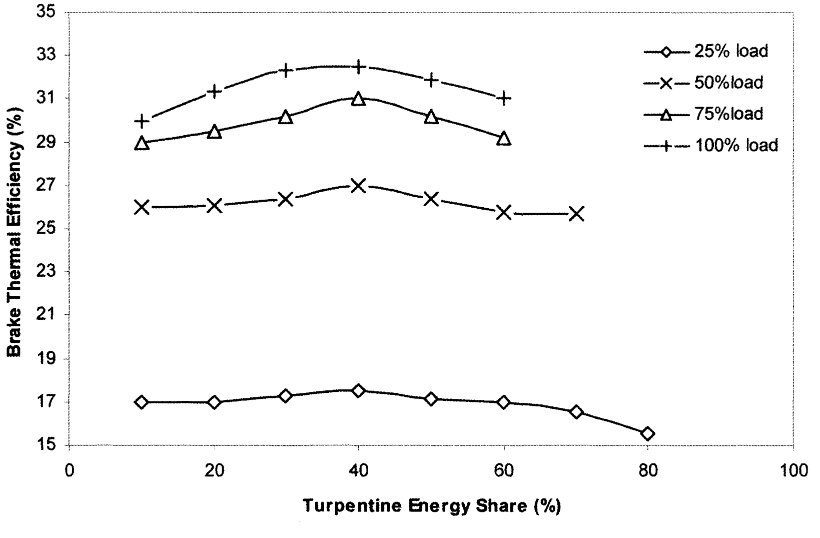

Figure 2: Variation of Brake Thermal Efficiency with Turpentine Energy Share

The increase in turpentine admission reduces the pilot fuel quantity and increases homogeneous mixture. This leads to better combustion and causes the increasing trend of brake thermal efficiency upto certain turpentine flow rates. However, the high turpentine flow rate affects the performance by the way of poor ignition or knocking.

At low loads, the quantity of fuel required is less and hence, the amount of pilot fuel required is less. At this stage, the induction of more turpentine leading to poor ignition. This is the main reason for low thermal efficiency at low loads at high turpentine flow rates.

At high loads and at high turpentine admission rates, the brake thermal efficiency falls due to rapid combustion of turpentine.

Comparison of Dual Fuel Operation with Standard Diesel Operation

In dual fuel operation, at each load, turpentine induction quantity was varied from 0% to 80% of the total volume of fuel required at that load. The performance level at 40% turpentine energy share showed maximum performance compared to other shares. Hence, it is considered for comparison with standard diesel operation. 4.2.1 Brake thermal efficiency

Figure 3 shows the variation of brake thermal efficiency with engine load. The brake thermal efficiency was higher at high loads and lower at low load than that of diesel fuel operation. The higher thermal efficiency is normally the case

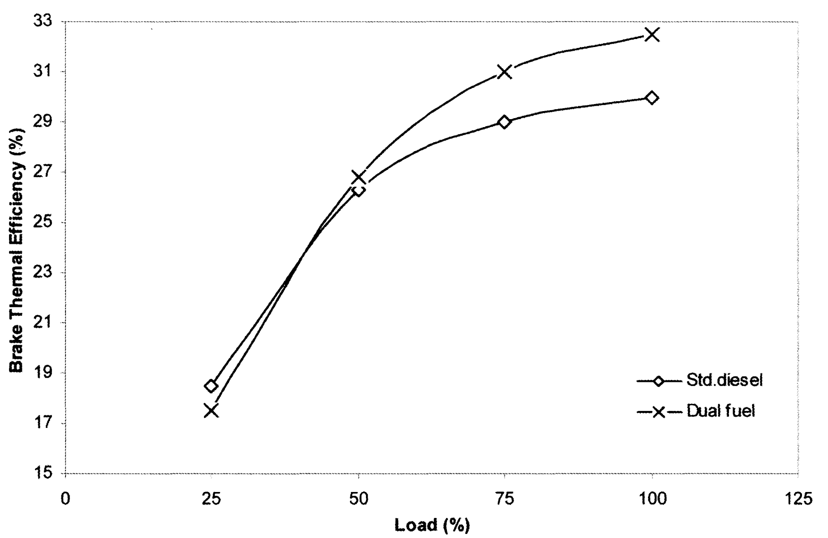

Figure 3: Variation of Brake Thermal Efficiency with Engine Load

with all dual fuel engines at maximum output. The higher flame speed due to presence of rich mixture and rapid rate of heat release due to flame propagation could be the reasons for the higher brake thermal efficiency.

At low loads, due to the presence of lean mixture a sluggish combustion occurred and causes poor ignition. This may be the reasons for low thermal efficiency at low loads.

The brake thermal efficiency of dual fuel engine at full load is 32.5% and it is 8% higher than that of DBL operation.

Volumetric Efficiency

Figure 4 shows the variation of volumetric efficiency with engine load. The volumetric efficiency of the dual fuel mode decreases with respect to engine load. In all dual fuel engines the inducted fuel displaces considerable portion of air and causes reduction of volumetric efficiency.

The preoccupation of cylinder by the retained exhaust gas also considered as the main reason for the reduction of volumetric efficiency. This effect is dominant at high outputs because of the high rate of combustion and high turpentine induction.

The lowest volumetric efficiency recorded in this mode is 76%. This is 5% lower than DBL operation.

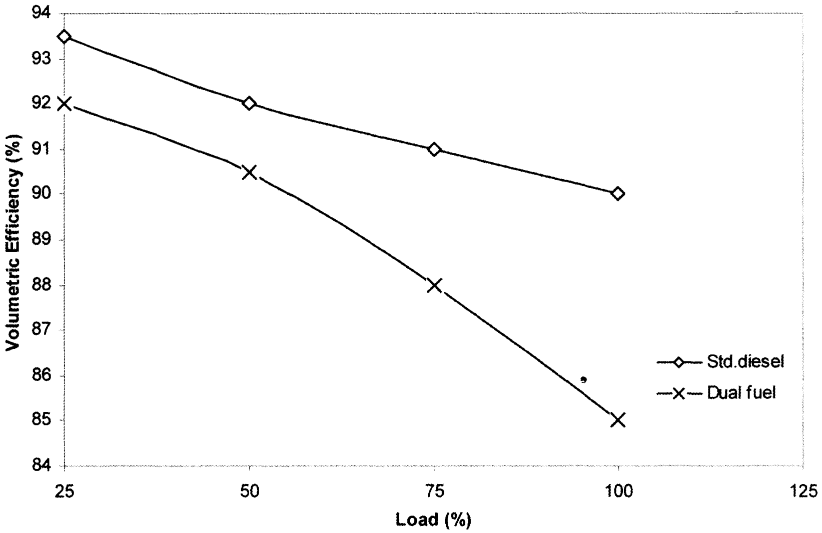

Figure 4: Variation of Volumetric Efficiency with Engine Load

Exhaust Gas Temperature

Figure 5 shows the variation of exhaust gas temperature at different loads. The exhaust gas temperature of dual fuel mode is higher than standard diesel operation at all loads. This is due to the sluggish combustion developed in the dual fuel mode. There are many reasons for the sluggish combustion of which, the important reasons are the lower concentration of oxygen, insufficient ignition source and presence of rich mixture at higher loads.

The exhaust temperature recorded at the time of full load is 310°C, which is 10°C higher than that of reference fuel.

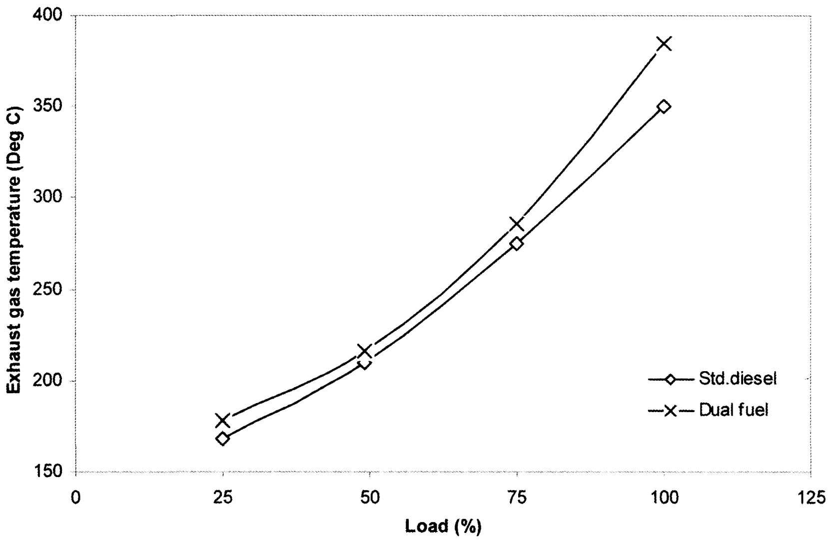

Figure 5: Variation of Exhaust Gas Temperature with Engine Load

CO Emission

Figure 6 compares the CO emission of dual fuel mode with standard diesel operation. It shows higher CO for dual fuel operation than that of standard diesel operation. The Partial oxidation of turpentine vapour due to the lack of oxygen is considered as the main reason for the liberation of more CO. Generally, in all dual fuel modes the primary fuel is inducted along with inlet air stream. This causes considerable displacement of air and resulting in lower volumetric efficiency. During this time, the injected pilot fuel encounters a charge, which has low concentration of oxygen. This leads to reduced flame temperature and lowered flame speeds. These may be the reasons for increased CO emission.

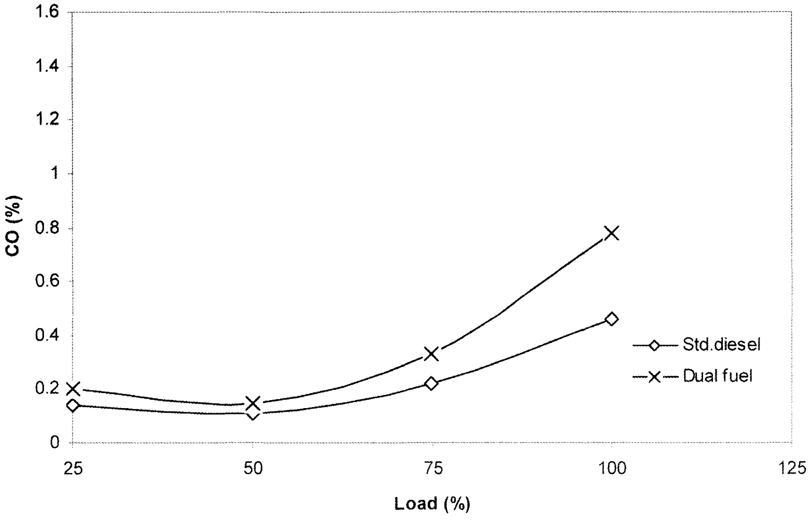

Figure 6: Variation of CO Emission with Engine Load

HC Emission

Figure 7 indicates the HC emission of dual fuel mode at various loads. Usually at high loads, the quantity of fumigated fuel inside the cylinder is more than

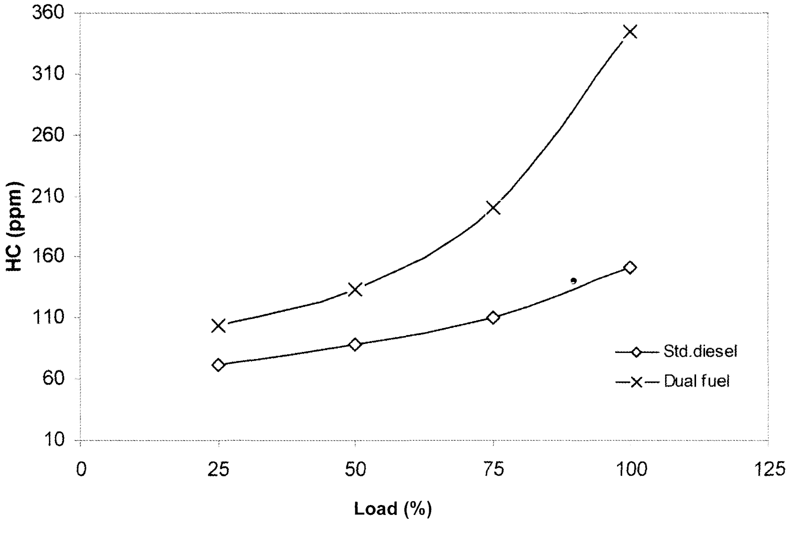

Figure 7: Variation of HC Emission with Engine Load

the pilot fuel. This absorbs a considerable portion of heat from the cylinder and keeps the charge relatively at low temperature. Hence, during the combustion, the flame front propagating from the diesel ignition centers do not extend to all regions of the cylinder and flame extinction occurs. This causes liberation of more HC at the time of higher loads.

At lighter loads, as the quantity of fumigated fuel inside the cylinder is less causes better charge preparation and better combustion. This may be the reason for lower HC emission at time of lower loads.

The HC emission of dual fuel engine at the time of full load is approximately 25% higher than reference fuel.

NOx Emission

Figure 8 compares the NOx emission of dual fuel mode with standard diesel operation. Usually, production of NOx is more associated with the intake charge temperature and the availability of oxygen. It also varies exponentially with combustion temperature (Barata., 1995).

From the figure, it is observed that the NOx emission of dual fuel mode is lower than standard diesel operation at all loads. This is due to the production of lower combustion temperature as a result of evaporative cooling and the sluggish combustion caused by lower concentration of oxygen.

The maximum NOx emission of DF mode was 1000 ppm. It is approximately 100 ppm lower than standard diesel operation.

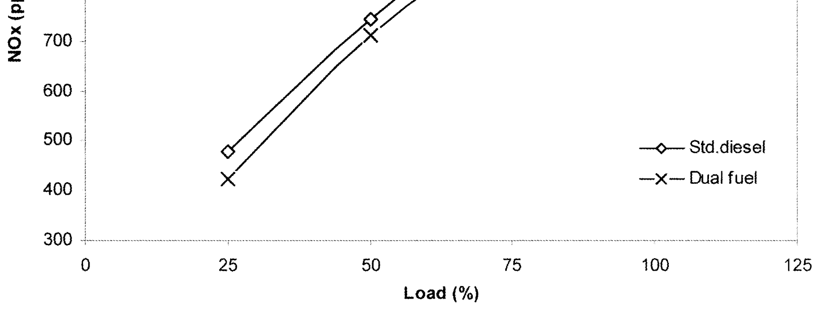

Figure 8: Variation of NOx Emission with Engine Load

Smoke Intensity

Figure 9 compares the smoke intensity of dual fuel mode with standard diesel operation. It shows lower smoke emission for dual fuel operation than that of standard diesel operation at all loads. More specifically, at low loads, it offered negligible smoke. This may be attributed to the reduced amount of diffusion burning and homogeneous mixer of turpentine oil.

At high loads, the smoke level increases due to the extended duration of combustion and more diffusion burning. However, it is lower than reference fuel. The maximum smoke level observed from the dual fuel mode is 3.5 BSU, which is 0.8 BSU lower than reference fuel.

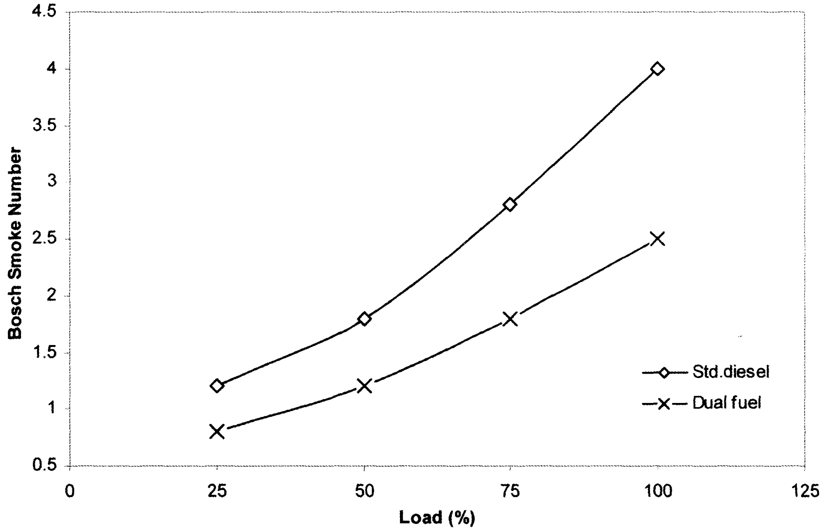

Figure 9: Variation of Smoke Intensity with Engine Load

Ignition Delay

Figure 10 compares the cylinder pressure diagram of dual fuel mode with DBL operation at full load. From the figure, it is observed that the peak pressure of dual fuel mode is higher than DBL operation. It occurs approximately 5 degrees later than reference fuel. The maximum cylinder pressure obtained from the dual fuel mode is 67bar, which is 3bar higher than standard diesel operation.

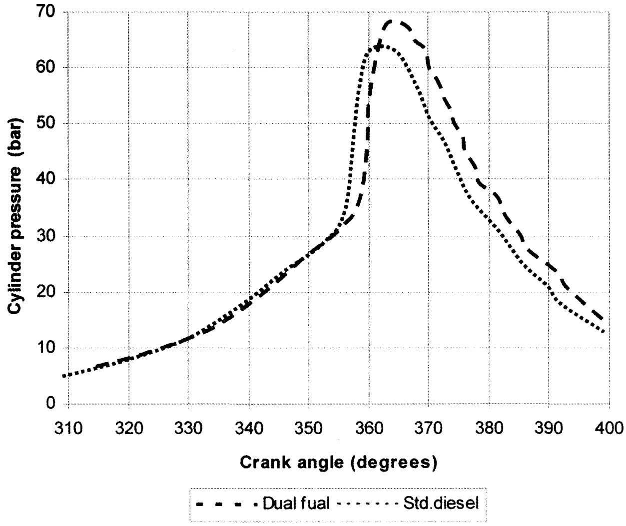

Figure 10: Variation of Cylinder Pressure with Crank Angle at Full Load

Rate of Pressure Rise

Figure 11 indicates the variation of rate of pressure rise with engine load. From the figures, it is observed that the value of parameter is lower than reference fuel

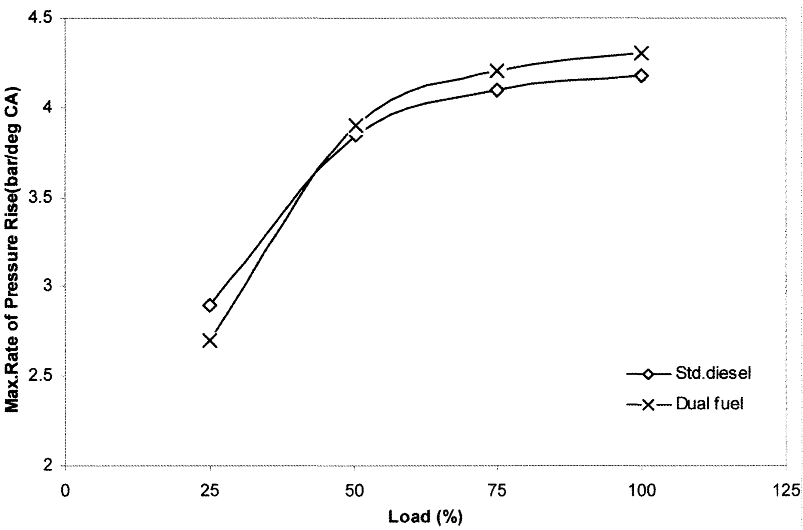

Figure 11: Variation of Rate of Pressure Rise with Engine Load

upto 50% load and then increases considerably. The sluggish combustion due to lean turpentine mixture may be the main reason for lower rate of pressure rise at lower loads. However, after 50% load, the presence of rich turpentine vapour causes rapid rate of pressure rise and higher cylinder pressure.

Heat Release Rate

Figure 12 indicates the variation of heat release rate of dual fuel mode at full load. From the figure, it is observed that the dual fuel mode offers higher premixed phase than that of reference fuel. This is due to the injection of more pilot fuel (60% of total fuel consumption) over the fumigated turpentine inside the cylinder and causes longer ignition delay. Usually, longer ignition delay accumulates more fuel inside the cylinder and causes rapid rate of heat release and higher peak pressure. This is the main reason for higher brake thermal efficiency at higher loads.

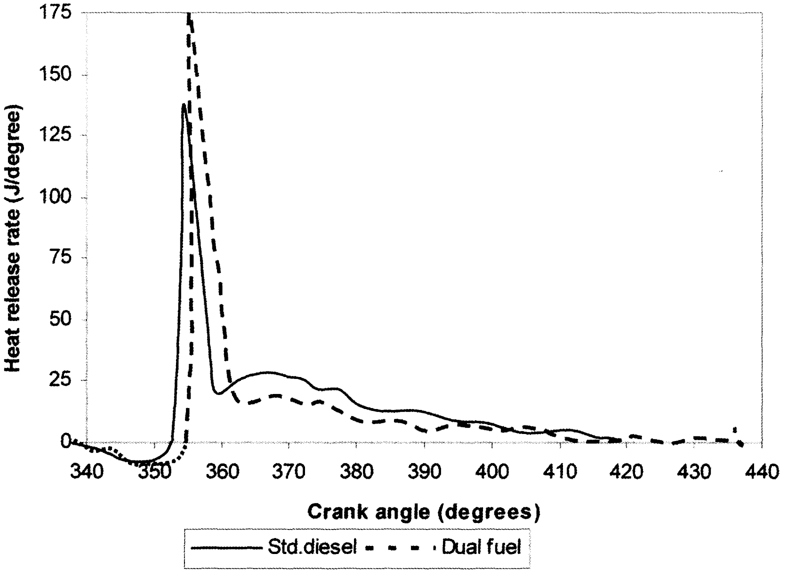

Figure 12: Variation of Net Heat Release Rate at Full Load

Conclusion

Based on the experimental investigations conducted on a dual fuel using Turpentine and diesel fuel the following major conclusions are arrived.

1. Results showed that the admission of turpentine through dual fuel mode increases brake thermal efficiency by 1-2% from the reference fuel.
2. Simultaneous reduction of NOx and smoke achieved in this mode.
3. Emissions such as CO and HC were increased considerably.
4. Almost 50% smoke free operation was achieved in this mode at full load.
5. The results also proved that the dual fueling increases the application quantity of turpentine to a maximum extent of 75%.

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