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Performance and emission characteristics of turpentine–diesel dual fuel engine and knock suppression using water diluents

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SUMMARY

In the present work, a normal diesel engine was modified to work in a dual fuel (DF) mode with turpentine and diesel as primary and pilot fuels, respectively. The resulting homogeneous mixture was compressed to a temperature below the self-ignition point. The pilot fuel was injected through the standard injection system and initiated the combustion in the primary-fuel air mixture. The primary fuel (turpentine) has supplied most of the heat energy. Usually, in all DF engines, low-cetane fuels are preferred as a primary fuel. Therefore, at higher loads these fuels start knocking and deteriorating in performances. Usually, DF operators suppress the knock by adding more pilot-fuel quantity. But in the present work, a minimum pilot-fuel quantity was maintained constant throughout the test and a required quantity of diluent (water) was added into the combustion at the time of knocking. The advantages of this method of knock suppression are restoration of performance at full load, maintenance of the same pilot quantity through the load range and reduction in the fuel consumption at full load. From the results, it was found that all performance and emission parameters of turpentine, except volumetric efficiency, are better than those of diesel fuel. The emissions like CO, UBHC are higher than those of the diesel baseline (DBL) and around 40–45% reduction of smoke was observed at 100% of full load. The major pollutant of diesel engine, NO_x, was found to be equal to that of DBL. From the above experiment, it was proved that approximately 80% replacement of diesel with turpentine is quite possible. Copyright © 2006 John Wiley & Sons, Ltd.

KEY WORDS: turpentine; biofuel; dual fuel; alternate fuel; emission analysis; knocks suppression; water diluent; combustion analysis

1. INTRODUCTION

Depleting petroleum reserves and increasing cost of the petroleum products demand an intensive search for new alternative fuels. Biofuels are proved to be the best substitutes for the

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existing petrofuels. But they require little engine modification or fuel modification. Generally biofuels are the oils obtained from the living plant sources. These oils may be obtained from resin and plant seeds. Plant oils are renewable and have low sulphur in nature. As biofuels are more expensive than fossil fuels, the widespread use of biofuel was restrained from its use in I.C. engines (Mayer-Pitroff, 1995; Choi *et al*., 1997). The use of vegetable oil in diesel engine was identified well before the exploration of the other promising alternative fuel alcohol. But the problems associated with vegetable oil are the high viscosity and low volatility. These properties have an adverse effect on fuel injection system and may lead to heavy carbon deposits in the engine combustion chamber (Choi *et al*., 1997; Masjuki *et al*., 1997; Husna *et al*., 1995). In the present work, turpentine is used as an alternate fuel to diesel.

Turpentine is also a biofuel. It is the volatile fraction of resin extracted from pine tree. The pine tree comes under the plant class of conifers. Most of the conifers will exude resin if wounded or naturally from branches. The distillation of pinus resin yields two products—turpentine and rosin. Turpentine was used in early engines without any modification. The abundant availability of petrofuels had stopped the usage of turpentine in I.C. engines. But the increasing cost of petrofuel prevailing today reopens the utility of turpentine in I.C. engine. Turpentine oil has low cetane number; it could be used in direct injection diesel engines in the form of turpentine–diesel blends (Rickeard and Thompson, 1993) and dual fuel (DF) mode. The specific objectives of this study are to analyse the performance and emission characteristics of turpentine in direct injection diesel engine and to analyse its feasibility as a fuel in a D.I diesel engine. From the authors' earlier study (Karthikeyan and Mahalakshmi, 2005), it was identified that 20% turpentine and 80% diesel is the optimum blend in terms of performance and emission characteristics.

In the present work, a normal diesel engine was modified to work in a DF mode with turpentine and diesel as primary and pilot fuels, respectively. The resulting homogeneous mixture was compressed to a temperature below the self-ignition point. The pilot fuel was injected through the standard injection system and initiated the combustion in the primary-fuel air mixture. The primary fuel supplied most of the heat energy. It has been reported that the DF operation at lighter load is less efficient than its diesel counterpart. However, beyond half load, the efficiency of DF operation is improved sufficiently and can even become better than that of diesel engine. However, at higher loads due to the occurrence of knock, a poor performance was recorded. This can be solved by the addition of water diluent into the cylinder at the time of knocking. Water, a very good diluent, is inducted along with the primary fuel during the knocking. The inducted water will evaporate by absorbing the heat from the combustion chamber and is mixed into the charge for keeping the mixture below self-ignition temperature. The important advantages of this method of knock suppression are restoration of performance at full load, maintenance of the same pilot quantity throughout the load range and reduction in the fuel consumption at full load. From the results, it was found that all performance and emission parameters of turpentine except volumetric efficiency are better than those of diesel fuel. Parameters like thermal efficiency, volumetric efficiency, smoke, UBHC, CO and NO_x were evaluated. Peak pressure, ignition delay and combustion duration were obtained from pressure crank angle data. The CO and UBHC emissions were found higher in DF modes at time of higher loads due to occurrence of knocking, poor fuel utilization and admission of water diluent, whereas other emissions like NO_x and smoke were considerably reduced.

Advantages of turpentine are:

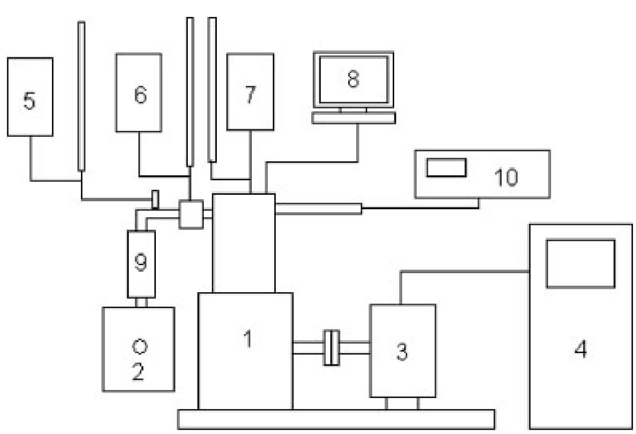
1. It is a renewable fuel and biofuel, obtained from pine tree.
2. Self-ignition temperature—close to diesel fuel.
3. Boiling point—almost equal to diesel fuel.
4. Calorific value—slightly higher than diesel fuel.
5. Viscosity—almost equal to diesel fuel.
6. Offers 11–15% higher calorific value compared to other biofuels (biodiesel and neat vegetable oil).

2. EXPERIMENTAL APPARATUS AND PROCEDURE

2.1. Experimental set-up

Figure 1 shows the schematic diagram of experimental set-up. The test engine is Kirloskar TAF 1, single cylinder, constant speed (1500 rpm), direct injection, with a bore of 87.5 mm and a stroke of 110 mm (Table I). The rated output of the engine is 4.4 kW at 1500 rpm. The compression ratio is 17.5:1 and the manufacturer recommended injection timing and injection pressure, 26° BTDC and 190 bar, respectively. The combustion chamber is the direct injection with a bowl-in piston. The engine is coupled to an eddy current dynamometer to provide a brake load and it is controlled by a control system provided in the control panel, which also consists of a speed indicator and a load indicator.

Two separate fuel-metering systems were provided to meter both primary fuel and pilot fuel. Fuel consumption of an engine was measured manually by a graduated burette. The primary fuel (turpentine) was allowed into the engine through primary-fuel spray system, which consists



1.Diesel Engine, 2.Air box, 3.Eddy current Dynamometer, 4.Dynamometer Control, 5.Diluent (water), 6.Turpentine, 7.Diesel fuel, 8.Data acquisition system, 9.Air pre heater, 10.Gas analyser

Figure 1. Experimental set-up.

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Table I. Engine details.

1. General details	Single cylinder, water cooled, compression ignition, constant speed engine
2. Make	Kirloskar TAF 1
3. Cubic capacity	661 cc
4. Bore	87.5 mm
5. Stroke	110 mm
6. Compression ratio	17.5:1
7. Speed (constant speed)	1500 rpm
8. Rated power	4.4 kW
9. Dynamometer	Eddy current
10. Pressure pickup	Piezotronics HSM111A22 Quartz pressure transducer

of a throttle-less gasoline carburetor, whose fuel flow rate is controlled manually by a fuel flow adjustment screw. The pilot fuel was admitted through conventional fuel system. Time taken for fuel consumption was measured with the help of a digital stopwatch.

An orifice meter attached to an anti-pulsating drum measures air consumption of an engine with the help of U tube manometer. The anti-pulsating drum fixed in the inlet side of an engine maintains a constant suction pressure to facilitate constant airflow through the orifice meter.

Exhaust emission from the engine was measured with the help of QRO TECH, QRO-402 gas analyser and smoke intensity was measured with the help of Bosch smoke meter. Bosch smoke meter usually consists of a piston-type sampling pump and a smoke level measuring unit. Two separate sampling probes were used to receive sample–exhaust gases from the engine for measuring emission and smoke intensity, respectively. A 2 inch diameter filter paper was used to collect smoke samples from the engine, through smoke sampling pump for measuring Bosch smoke number. A k-type thermocouple and a temperature indicator were used to measure EGT.

Combustion diagnosis was carried out by means of a PCB Piezotronics HSM111A22 quartz pressure transducer fitted on the engine cylinder head and a crank angle encoder was fixed on the output shaft of the engine. The pressure and crank angle signals were fed to a data acquisition card fitted with Pentium4 personal computer. The combustion parameters, such as peak pressure, heat release rate, mean gas temperature and ignition delay, were computed.

2.2. Procedure

1. The whole test was conducted for the standard engine injection pressure and injection timing.
2. Induction manifold was extended outward to accommodate primary-fuel spray system and diluent spray system.
3. Two separate fuels metering systems and one diluent metering system were provided with the test rig to measure fuel consumption and diluent consumption, respectively.

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4. The first test was conducted using 100% low sulphur diesel fuel to establish baseline readings for emission, fuel consumption and performance.
5. The engine was then started and its no-load rack position was locked.
6. First, when 25% load was applied consequently the speed of the engine decreased.
7. Then, the rated speed of the engine was restored by addition of turpentine through primary-fuel-regulating device.
8. 50, 75 and 100% loads were applied separately and for each load, step 7 was repeated.
9. Whenever the pinking noise was observed, the diluent was admitted into the engine through suction manifold by diluent spray system.
10. The diluent flow rate was adjusted in such a way that the pinking noise should be arrested.
11. Emission, fuel consumption and cylinder pressure were measured at each load.
12. The performance and emission of the DF mode were compared with the diesel baseline (DBL) readings.

2.3. *Detection of onset of knock*

The initial introduction of small quantities of turpentine with the air neither affected the usual pressure diagram nor the power developed. As the turpentine concentration slowly increased, the engine developed higher power at the same speed. A significant increase in the ignition delay was observed during this process. Further, increase in turpentine admission resulted in considerable increase in power output with higher cylinder pressures. The turpentine admission could be further increased up to a point beyond which combustion became very rapid and a slight increase in the turpentine admission led to 'knocking'. As soon as it occurred, the audible sound of the engine changed considerably. Simultaneously, the shape of the pressure crank angle diagram changed, exhibiting a very sharp pressure rise with much higher maximum pressure accompanied by oscillations on the expansion curve.

The transition from 'normal' to 'knocking' operation was quite sharp and could be achieved by small changes in mixture strength. The abrupt change in the shape of the pressure diagram was used as a means for detecting the onset of knock.

2.4. *Instrumentation*

Table IV provides the range, accuracy, measurement technique and percentage uncertainties of various instruments involved in this experiment for observing various parameters.

2.5. *Error analysis*

Errors and uncertainties in the experiments can arise from instrument selection, condition, calibration, environment, observation, reading and test planning. Uncertainty analysis is needed to prove the accuracy of the experiments. An uncertainty analysis was performed using the method described by J. P. Holman (book entitled 'Experimental techniques').

Percentage uncertainties of various parameters like total fuel consumption, brake power, specific fuel consumption and brake thermal efficiency were calculated using the percentage

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uncertainties of various instruments given in Table IV.

Total percentage uncertainty of this experiment is

$$
\begin{aligned}
&= \text{Square root of}\{(\text{uncertainty of TFC})^2 \\
&\quad + (\text{uncertainty of brake power})^2 \\
&\quad + (\text{uncertainty of specific fuel consumption})^2 \\
&\quad + (\text{uncertainty of brake thermal efficiency})^2 \\
&\quad + (\text{uncertainty of CO})^2 + (\text{uncertainty of CO}_2)^2 \\
&\quad + (\text{uncertainty of UBHC})^2 + (\text{uncertainty of NO}_x)^2 \\
&\quad + (\text{uncertainty of Bosch smoke number})^2 \\
&\quad + (\text{uncertainty of EGT indicator})^2 \\
&\quad + (\text{uncertainty of pressure pickup})^2\} \\
&= \text{square root of } \{(1)^2 + (0.2)^2 + (1)^2 + (1)^2 + (0.2)^2 \\
&\quad + (0.15)^2 + (0.2)^2 + (0.2)^2 + (1)^2 + (0.15)^2 + (1)^2\} \\
&= \pm 3\%
\end{aligned}
$$

Using the calculation procedure, the total uncertainty for the whole experiment is obtained to be $\pm$ 3%.

2.6. *Test fuels*

The test fuels for the present study were turpentine (primary fuel) and high-speed diesel with sulphur contents 0.04 ppm. The world turpentine oil resource is shown in Table II. Table III shows the physical and chemical properties of turpentine oil.

3. RESULTS AND DISCUSSION

The performance, combustion and emission analyses are presented in the following sections.

3.1. *Specific fuel consumption*

The variation of SFC with brake power output before diluent admission (with knock) and after diluent admission is shown in Figure 2. Since the heat content of the turpentine is close to that of diesel fuel, a minimum difference in fuel consumption was observed between DBL and DF mode up to 75% of load. But in DF mode, a markedly increased fuel consumption was recorded beyond 75% load due to the occurrence of knock. However, proper quantity of diluent admission reduces the SFC and brings it closer to DBL. The diluent admission at the time of knock prepares comparatively lean mixture inside the cylinder by diluting the air fuel mixture with the help of diluent resulting in a sluggish combustion and knock-free operation.

Table II. Estimated world production of crude resin, rosin and turpentine.

	Production (tonnes)			
	Year	Crude resin	Rosin	Turpentine
World total production of which:		976 000	717 000	99 400
People's Republic of China	1993	570 000	430 000	50 000
Indonesia	1993	100 000	69 000	12 000
Russia	1992	90 000	65 000	9000
Brazil	1993	65 000	45 000	8000
Portugal	1992	30 000	22 000	5000
India	1994	30 000	21 000	4000
Argentina	1993	30 000	21 000	4000
Mexico	1991	30 000	22 000	4000
Honduras	1992	8000	6000	1000
Venezuela	1993	7000	5000	800
Greece	1993	6000	4000	600
South Africa	1993	2000	1500	200
Vietnam	1990	2000	1500	200
Others		6000	4000	600

Table III. Property comparison of turpentine with existing petrofuels.

	Gasoline	Diesel	Turpentine
Formula	C_4–C_{12}	C_8–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
Pour point (°C)	−40	−23	
Boiling point (°C)	30–220	180–340	150–180
Vapour pressure (kPa)	48–103	<1	<1
Viscosity c St at 30°C		3–4	2.5
Latent heat of vapourization ($kJ\,kg^{-1}$)	350	230	285
Lower heating value ($kJ\,kg^{-1}$)	43 890	42 700	44 400
Flash point (°C)	−43	74	38
Auto-ignition temperature (°C)	300–450	250	305
Flammability limit % volume	1.4	1.0	0.8
Stoichiometric air–fuel ratio	14.7	14.7	
Flame speed ($m\,s^{-1}$)	4–6		
Cetane number		40–55	

Fuel consumption of DF mode before and after diluent admission with brake power output is clearly depicted in Figure 11. It shows that almost constant fuel supply is maintained by pilot-fuel supply system, irrespective of load. But the primary-fuel (turpentine) consumption gradually increases when load changes from 25 to 100%. Beyond 75% load, a sudden hike in primary-fuel fuel consumption was recorded due to occurrence of knock and this is indicated by the bar, primary-fuel consumption before diluent admission (BDA). The primary-fuel

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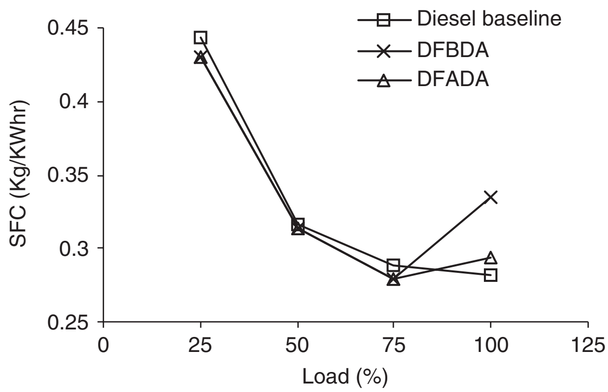

Figure 2. SFC before and after diluent admission with brake power output.

consumption after suppressed knock is indicated by another bar, primary-fuel consumption after diluent admission (ADA). From Figure 11, it is also clear that a considerable reduction in fuel consumption was recorded after diluent admission. The maximum diesel displacement by turpentine at full load is approximately 80% of total fuel consumption.

3.2. *Diluent consumption*

In DF mode, the charge comes to the auto-ignition at around 75% load. Hence, the diluent admission begins at around 75% load. The inducted diluent evaporates inside the cylinder by absorbing heat from the combustion chamber and makes the charge lean, which reduces the tendency of auto-ignition of the mixture. The maximum quantity of diluent required at the time of full load is 0.737 lit h^{-1}. Usually, the occurrence of knock in the engine can be observed by pinking noise of engine or steep pressure rises in the P–θ diagram.

3.3. *Brake thermal efficiency*

The variation of brake thermal efficiency with brake power output before diluent admission (with knock) and after diluent admission is shown in Figure 3. Usually, in gaseous fuel DF mode, the brake thermal efficiency at lower loads will be lower than that of DBL due to sluggish combustion of more diluted fuel air mixture. But in the present case, the brake thermal efficiency of DF mode at lower loads was observed to be very close to DBL. From the figure it is clear that the brake thermal efficiency of the DF engine is very close to DBL up to 75% load, beyond which it starts decreasing from the DBL. The maximum efficiency of DF engine is 29% at 75% of full load. The reason for the decrease of brake thermal efficiency beyond 75% load is the occurrence of knock (abnormal combustion). However, induction of water diluent, at the time of knocking, dilutes the mixture and keeps the charge slightly below the self-ignition temperature. This eliminates the occurrence of knocking and improves the break thermal efficiency. The improved brake thermal efficiency due to diluents admission is 28% at 100% load. The loss of brake thermal efficiency at the time of full load is due to heat carried by the diluents from the combustion chamber and reduced volumetric efficiency. However, the brake thermal efficiency obtained after diluent admission is higher than that of brake thermal efficiency obtained in the same load with knocking without diluent admission.

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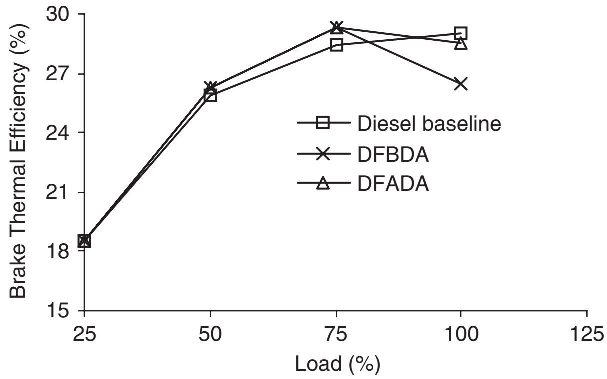

Figure 3. Brake thermal efficiency before and after diluent admission with brake power output.

Poor fuel utilization and increased fuel consumption at the time of knocking were also the other reasons of reduced brake thermal efficiency. However at 75% load, the break thermal efficiency of DF mode is better than that of DBL. This is due to higher heating value of the turpentine oil and more charge homogeneity at the end of the compression stroke.

3.4. Volumetric efficiency

The variation of volumetric efficiency with brake power output before diluent admission (with knock) and after diluent admission is shown in Figure 4. The volumetric efficiency of DF mode is lower than that of DBL at all loads. The reason for the decreased volumetric efficiency is induction of turpentine along with the inlet air. The inducted turpentine vapourizes inside the cylinder by absorbing heat from the cylinder wall and reduces the space available for fresh air entry. Hence, the volumetric efficiency decreases as the load increases or the percentage of turpentine admission increases. In DF mode, the drastic reduction of volumetric efficiency was observed beyond 75% load due to the occurrence of knock. At full load, approximately 11% drop in volumetric efficiency was obtained with DF mode. But this could be reduced to large extent by admission of proper quantity of water diluent. Hence, the resulting volumetric efficiency at full load after diluent admission is only 6% lower than that of DBL.

3.5. Exhaust gas temperature

The variation of exhaust gas temperature (EGT) with brake power output before diluent admission (with knock) and after diluent admission is shown in Figure 5. The primary fuel, inducted well before the start of combustion, evaporates sufficiently by absorbing heat from the combustion chamber. Hence, in DF mode, the combustion takes place at relatively lower temperature. This is one of the major reasons for lower EGT of DF mode at all load conditions, except at full load. More solid injection, poor fuel utilization and higher ignition delay due to occurrence of knock were the other reasons for high EGT after 75% load. However, proper quantity of diluent admission solves the above-said problem to a large extent.

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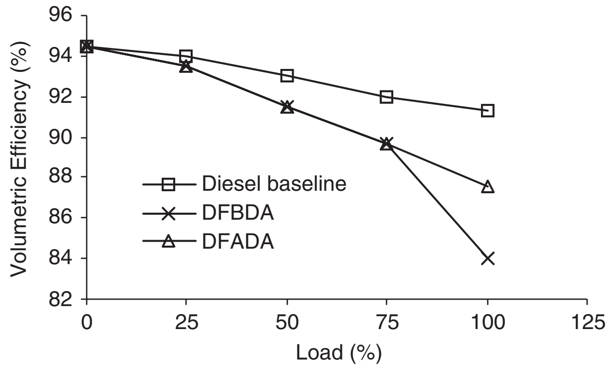

Figure 4. Volumetric efficiency before and after diluent admission with brake power output.

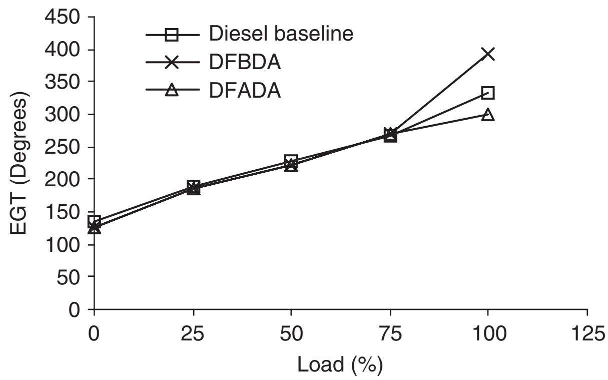

Figure 5. EGT before and after diluent admission with brake power output.

3.6. *P–θ diagram*

Figure 6 shows the variation of *P–θ* diagram before diluent admission (with knock) and after diluent admission. In DF mode, due to severe knock at the time of 90% load, the peak pressure jumps beyond the limit and makes the expansion line spikier. This could be reduced and brought closer to DBL with the help of proper quantity of diluent admission. Diluent admission at the time of knock helps to prevent auto-ignition of mixture by diluting the charge. This results in a sluggish combustion and knock-free operation. Hence, the peak pressure occurring at the time of 90% load of DF modes is slightly lower than that of DBL. Since the injection timing is not varied, the peak pressure of DF mode onsets far beyond TDC. The peak pressure of DF mode after diluents admission is 89 bar.

3.7. *Emission*

The variation of CO emission with brake power output before diluent admission (with knock) and after diluent admission is shown in Figure 7. As the primary fuel (turpentine) inducted along with the inlet air vapourizes inside the cylinder by absorbing heat from the combustion

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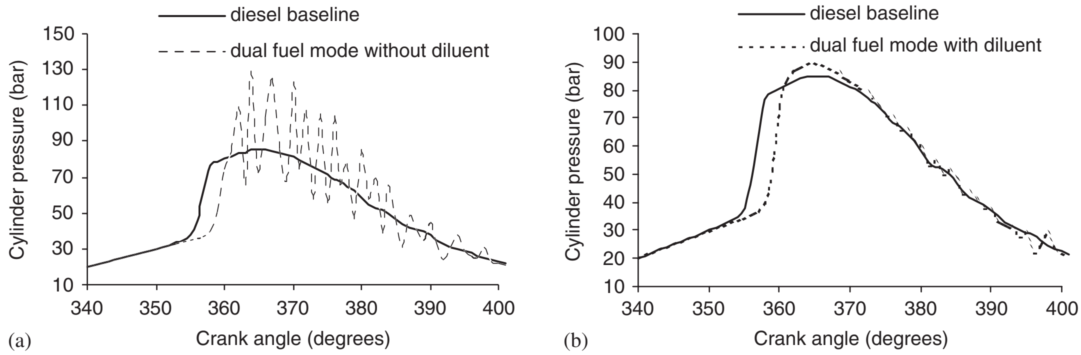

Figure 6. (a) Variation of cylinder pressure with crank angle before diluent admission at full load; and (b) variation of cylinder pressure with crank angle after diluent admission at full load.

chamber, the combustion occurs at relatively lower temperature. During the combustion, the lean air–fuel mixtures are not combusted fully due to flame quenching and the rich mixtures do not find oxygen for its complete combustion. These are the major reasons for higher CO emission. Usually, DF mode offers higher CO emission than DBL. But this is drastically increased after 75% load due to the presence of knock. However, this could be reduced to large extent by admission of proper quantity of diluent.

The variation of CO emission with brake power output before diluent admission (with knock) and after diluent admission is shown in Figure 8. The UBHC emission of DF mode gradually increases from 0% load to 75% load. The reason behind increased UBHC emission is higher fumigation rate and scarcity of oxygen. The flame quenching and cooled layer of charge near the wall were the other reasons for the increased UBHC emission. Though this is a conventional behaviour of the DF engine, the drastic increase of UBHC emission at higher loads is due to the occurrence of knock. However, this could be reduced to large extent by the admission of proper quantity of diluent. Approximately 148 ppm of higher UBHC emission was observed after diluent admission at full load of DF mode.

The variation of CO emission with brake power output before diluent admission (with knock) and after diluent admission is shown in Figure 9. The NO_x emission of DF mode closely follows the DBL up to 75% load, beyond which it starts increasing from DBL. The reason for drastic increase in NO_x level beyond 75% load is the occurrence of knock. The knock generated inside the cylinder increases the mean gas temperature and creates a conducive ambient for reaction of atmospheric nitrogen with oxygen. Hence, NO_x emission increases at higher load ranges. However, proper quantity of diluent admission keeps the cylinder temperature well below reaction temperature of nitrogen and oxygen resulting in lower NO_x emission. The maximum NO_x emission of DF mode at the time of full load is 1033 ppm.

Figure 10 compares the Bosch Smoke Number of DF and DBL with respect to various loads. Usually, DF mode offers reduced smoke emission. The reasons for the reduced smoke emission are the availability of premixed and homogeneous charge inside the engine, higher heat content of turpentine, rapid flame propagation and very small quantity of pilot-fuel admission. However, at higher load ranges due to the non-availability of sufficient air and abnormal combustion, a visible white smoke emission was observed. But this could be eliminated with the

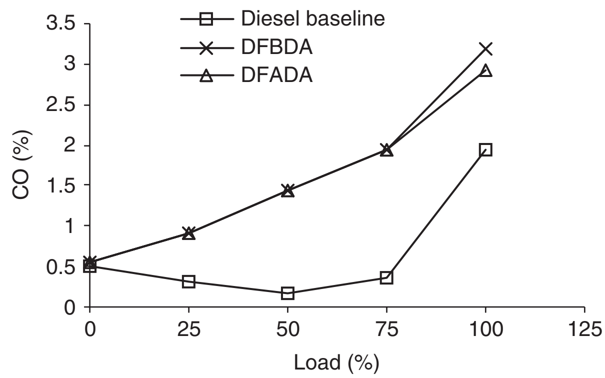

Figure 7. CO emission before and after diluent admission with brake power output.

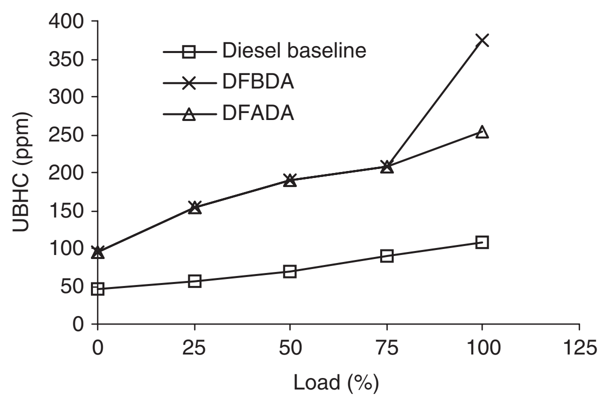

Figure 8. UBHC emission before and after diluent admission with brake power output.

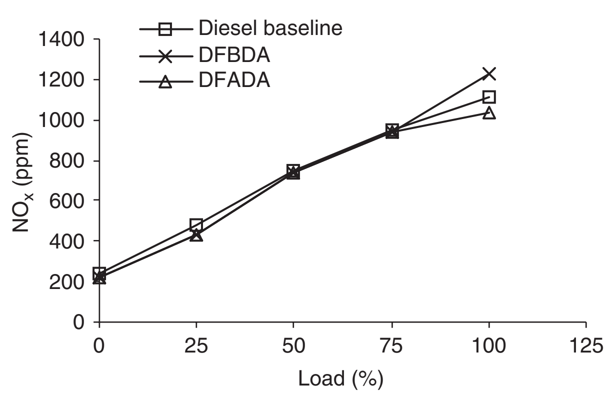

Figure 9. NO_x emission before and after diluent admission with brake power output.

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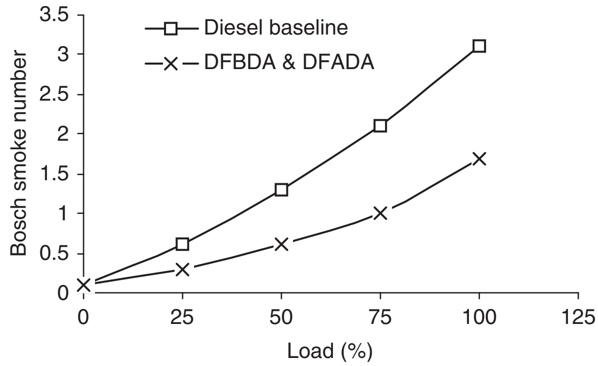

Figure 10. Bosch smoke number before and after diluent admission with brake power output.

help of proper quantity of diluent admission. The smoke number of DF mode at 75% of full load is 1.1 and this is 45% lower than that of DBL.

4. CONCLUSIONS

The following conclusions are drawn based on the experimental investigations on a turpentine diesel dual fuel engine and knock suppression with the addition of water diluent.

1. The brake thermal efficiency of dual fuel engine at 75% load is 1–2% higher than that of diesel baseline.
2. Increased SFC was observed at full load due to the presence of knock.
3. Knocking in a DF mode could be suppressed by induction of water diluent through the inlet manifold.
4. In comparison with the normal DF engine, the knock limited power increases.
5. Brake thermal efficiency of DF modes decreases with the induction of water.
6. Ignition delay increases due to cooling of the charge at the end of the compression.
7. EGT and NO_x are found lower than that of diesel baseline at all loads.
8. Approximately 35% of higher CO emission was observed at full load of dual fuel engine.
9. 48% of higher UBHC emission and 45% reduced smoke were observed at full load of dual fuel engine.
10. A maximum of 6% drop in volumetric efficiency was reported in DF engine at full load.
11. Most of the research papers submitted on the same topic earlier have reported that the DF operation at light loads are less efficient than its diesel counterpart; but, this investigation proves that the brake thermal efficiency at lighter loads is very close to that of diesel baseline.

From the above experiment, it is concluded that approximately 80% displacement of diesel with turpentine is quite possible. Also, the occurrence of knock due to admission of octane fuel in C.I engine can be suppressed fully by the addition of water diluents. Except increased CO and UBHC emissions, all other emission parameters like smoke, EGT and NO_x are superior to that

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Table IV. List of instruments and their range, accuracy, measurement technique and uncertainities.

Instruments	Range	Accuracy	Measurement techniques	Percentage uncertainties
1. Gas analyser	CO 0–10%	± 0.02%	NDIR principle (non-depressive infra-red sensor)	± 0.2%
	CO_2 0–20%	± 0.03%		± 0.15%
	UBHC 0–10 000 ppm	± 20 ppm		± 0.2%
	NO_x 0–5000 ppm	± 10 ppm		± 0.2%
2. Smoke level measuring instrument	BSN 0–10	± 0.1	Electro chemical sensor	± 1%
3. EGT indicator	0–900°C	± 1°C	k-type (Cr Al) thermocouple	± 0.15%
4. Speed measuring unit	0–10 000 rpm	± 10 rpm	Magnetic pickup type	± 0.1%
5. Load indicator	0–100 kg	± 0.1 kg	Strain gauge-type load cell	± 0.2%
6. Burette for fuel measurement		± 0.1 cc		± 1%
7. Digital stop watch		± 0.6 s		± 0.2%
8. Manometer		± 1 mm		± 1%
9. Pressure pickup	0–110 bar	± 0.1 kg		± 0.1%
10. Crank angle encoder		± 1°	Magnetic pickup type	± 0.2%

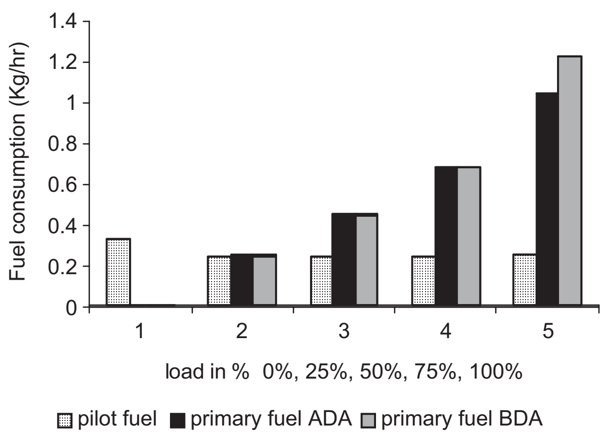

Figure 11. Comparison of fuel consumption of pilot fuel (diesel) and primary fuel (turpentine) before and after diluent admission with brake power output.

of diesel baseline. Performance parameter like brake thermal efficiency is also found closer to that of diesel baseline within 75% load.

NOMENCLATURE

ADA = after diluent admission
BDA = before diluent admission

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C.I	= compression ignition
CO	= carbon monoxide
DBL	= diesel baseline
DF	= dual fuel
DFADA	= dual fuel mode after diluent admission
DFBDA	= dual fuel mode before diluent admission
D.I	= direct injection
EGT	= exhaust gas temperature
NO_x	= nitric oxide
UBHC	= unburned hydrocarbon

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