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Effect of Thermal Barrier Coating on Performance, Combustion and Exhaust Emission in Bio-Fuel Powered Diesel Engine

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Abstract

The aim of this study is to investigate the suitability of turpentine oil fuel as an alternate fuel for the diesel engine with both thermal barrier coated engine and uncoated engine, and to compare the results obtained pertaining engine performance, combustion and exhaust emission. To provide thermal barrier in the combustion chamber, functionally graded material with its varying properties and a layer of aluminum oxide coating are used. Pure turpentine oil fuel and its diesel blends in four different proportions - 20-80%, 40-60%, 60-40% and 80-20% are prepared in volume basis and tested in constant speed direct injection diesel engine at different load conditions. The work has been carried out in both the engines and the results are compared, having diesel fuel as base line. The results reveal that in the coated engine there is an increase in brake thermal efficiency proportional to turpentine oil fuel and peak cylinder pressure and decrease in specific fuel consumption, oxides of nitrogen, carbonmonoxide and heat release rate, as compared to uncoated engine. At the same time, smoke density and unburned hydrocarbon emissions are slightly increased compared to uncoated engine. The performance of both the engine indicated that the properties and characteristics of turpentine oil fuel make it a viable additive or alternative to conventional fossil diesel fuel.

Keywords: Biofuel, Turpentine oil, Zirconia, Thermal barrier, Combustion, Emission.

1. Introduction

Among the problems, which require the unity and integration of the nations to fight against and find a solution, Global warming is a major issue. Though the reasons for this handicap are many, the one which aggravates the situation worser is the increasing air pollution. The use of fossil fuel for transportation, contributes a lot to air pollution .Therefore it is high time to design an alternative fuel engine and fuel system that should provide adequate power without violating the emission limits. Great deals of researches are being conducted in this respect. Most of the researches recommend biofuel as an alternative fuel for diesel engines, since the proprieties of biofuel are very closer to diesel fuel [1-3]. Biofuel is suitable for diesel engine with certain modifications.Biofuel, which is renewable and has low sulphur, is generally obtained specific plant sources like resin and plant seeds. The factors which restrained the widespread use of biofuel in I.C.engines are high cost, high viscosity and low volatility [4, 5]. The high viscosity and low volatility create adverse effect on fuel injection system and heavy carbon deposits in the engine combustion chamber [6, 7]. In this study turpentine oil is used as alternative fuel to conventional diesel fuel.

2. Biofuel production

Since 1700, Turpentine oil had widely been used as commercial fuel. It is obtained from pine tree – by resin fraction. The distillation of pinus resin yields two products – turpentine and rosin. Turpentine is a yellowish, opaque, sticky volatile, combustible mixture of hydrocarbon isomers obtained either from pine gum or pine wood. Turpentine oil consists of 58-65% gamma – pinene, beta – pinene and other isometric terpenses. The turpentine oil absorbs active oxygen at 100°C and its properties can be retained for years if placed in dark place. The analysis properties of turpentine oil made it suitable for diesel engine operations. It was used in early engines without any modifications. But abundant availability of diesel fuel had stopped the use of turpentine oil in I.C.engines.As turpentine produces low polluting exhaust emission, it will be alternative fuel. The cost of the turpentine is slightly higher than the petro-fuels, but it would be least when compared to global emission management cost.

3. Thermal barrier coating with yttria-stabilized zirconia

The reduction of polluting vehicle emission can be achieved by providing combustion of the fuel .Some researches assure that coating the combustion chambers with ceramic material reduce the polluting vehicle emission. The engine which has this inbuilt facility is called adiabatic (or) low heat loss engine. Coating of some parts of combustion chamber with low thermal conducting material increases the combustion temperature and reduces in the polluting emission [7]. The partially adiabatic engine minimizes the heat loss during normal running of the engines, decreases the load on the cooling system and also the power consumed by it. As a result, the efficiency of the engine is increased [8].The combustion chamber temperature of ceramic coated engine is higher than the uncoated engine and this permit to use even low quality fuels. Since the heat loss which goes to the cooling system is reduced, the temperature of gas after combustion is increased. It enables the engine to start even at cold weather conditions and ensures noise reduction at engine operations because of the controlled combustion [9]. From various studies, it is noticed that improved engine performance and low polluting emission of vehicles are possible when some parts of the combustion chamber are coated with ceramic material [10, 11].

The ceramic coating material suggested in this study is zirconia ceramics. By controlling phase transition of the fabrication process, the desired Zirconia property can be achieved. Normally zirconia becomes very strong and tough at room temperature. Understanding of the phase transitions is crucial to determine the required properties of zirconia ceramics. The zirconium dioxide (ZrO_2) has a monoclinic crystallographic structure at ambient temperatures. Upon raising the temperature the oxide undergoes the phase transitions - from monoclinic to tetragonal at the phase transitions temperature of 1170^oC, from tetragonal to cubic at the transition temperature of 2370^oC and from cubic to liquid at transition temperature of 2680^oC . The transformation from tetragonal to monoclinic phase, when temperature decreases approximately at 1170^oC , is quite disruptive and renders pure ZrO_2 unusable as it is a high temperature structural ceramic. This disruption is caused by 6.5% volume expansion upon the transformation from tetragonal to monoclinic phase change which could cause structural failure of any ceramic coating.

However, ZrO_2 forms solid solutions with aliovalent oxides such as CaO, MgO, Y_2O_3 and other rare oxides of earth. This is obtained by through mixing of the powders - ZrO_2 and Y_2O_3 - and is pressed to form a solid body and sintering at the temperature adequately high to promote the inter-diffusion of the coatings. This solid solution (yttria-doped zirconia) behaves quite differently. The pure $ZrO_{2,}$ as the temperature phases, tetragonal and cubic, tend to be stabilized at temperatures lower than 1170^oC and 2370^oC, respectively. Hence, doping of the aforementioned aliovalent oxides serves as the stabilizing agent for the zirconia. With the addition of sufficient fraction of stabilizer, the cubic phase could be stabilized at the ambient temperature. The addition of 9% mole fraction of yttria (Y_2O_3) or more to ZrO_2 will result in fully stabilized zirconia (FSZ) which has the cubic structure at all temperature from ambient upwards. Addition of 6% mole fraction of Y_2O_3 or less generates partially stabilized zirconia (PSZ) which consists of the cubic matrix with dispersed tetragonal or monoclinic precipitates, or both, depending on the temperature in the process.

3.1 Functionally graded material

Based on the considerations of phase stability of zirconia and thermal / mechanical compatibility between the coated (the thermal barrier coat) and uncoated (the underlying aluminum alloy) materials of the piston crown, cylinder head and valves, the functionally graded material (FGM) properties and the thickness of the coat are varied. In this study, the FGM is actually a layer of ceramic consisting of zirconia with different proportions of yttria doping along its thickness. Functionally graded materials (FGMs) display continuously either varying compositions or microstructures over definable geometric orientations and distances. The grades can be continuous on a microscopic level or they can be laminated comprised of gradients of metals, ceramics, polymers or variations in porosity/density. Several processing techniques have been exploited for the fabrication of FGM for structural application: e.g., powder metallurgy, plasma spraying, in situ synthesis, self-propagating high temperature synthesis, reactive infiltration, etc. Physical and chemical vapor deposition techniques are also being explored to process FGM films with nanometer level composition gradients. In this study, plasma spray is and the FGM is laminated by four sub-layers of zirconia-yttria with varying compositions from a pure zirconia ceramic to 25% zirconia-75%-yttria ceramic (See table1).

Table 1 Properties of FGM coating.

Description	**Density(g.m $^{-3}$)**	**Porosity (%)**	**Elastic modulus(GPa)**
100%ZrO_2	6.0374	12.16	53
75% ZrO_2/25%Y_2O_3	6.2076	11.20	105
50% ZrO_2/50% Y_2O_3	6.6264	10.01	158
25% ZrO_2/75% Y_2O_3	6.9599	8.91	187

Plasma spray process offers a flexible and relatively economic means for producing FGM. It has been used for many years to apply layered and graded deposits (bond coats) to enhance the survivability of engines. These graded coatings are applied to reduce discontinuities in thermal expansion coefficients in order to avoid mismatch- related failure in service.

In this study, the effect of ceramic coating on cylinder head, valves and piston crown face are analyzed with respect to performance, emission and combustion characteristic of diesel engine. Before applying the thermal barrier coating in a manner of "functionally graded" onto the crown, cylinder head and valve a thickness of 300 microns of a new set of piston crown are to be machined off. Ceramic layers are made with partially stabilized zirconia with aluminum oxide (Al_2 O_3 ZrO_2), using plasma spray method, of thickness about 300 micron. The thickness of coating is selected within the optimum range of thickness 0.1–1.5 mm [12]. The Cylinder liner is not coated as it is very small area. The Combustion chamber geometry was maintained by machining the components before ceramic coating. The table 2 shows the chemical properties of turpentine oil, diesel and Indian requirements as per IS 1460-1974 and table 3 shows fuel composition, variation of calorific value and viscosity for various properties of turpentine with diesel fuel.

Table 2 Chemical properties of Turpentine oil, diesel and Indian requirements as per IS 1460-1974

Properties	**Turpentine**	**Diesel**	**Indian requirements as per IS 1460-1974**
Density @ 40°C in gm/cc	0.92	0.827	Nil
Kinematic viscosity @ 40°C in CST	3.89	3-4	2.0 – 7.5
Flash point in °C	38	74	38
Pour point in °C	-	-23	6 max.
Heating value in kJ/kg	37280	42700	-
Calculated cetane number	38	40-55	42

Table 3 Fuel composition, Variation of calorific value and viscosity for various properties of Turpentine with diesel fuel

Sl. No.	**Fuels**	**Fuels blended (% volume)**	**Viscosity (cst)**	**Calorific value (kJ/kg)**	**Density**
1.	DF	100% Diesel fuel	3	42700	0.827
2.	T20	80% Diesel fuel + 20% Turpentine oil fuel	3.178	41616	0.846
3.	T40	60% Diesel fuel + 40% Turpentine oil fuel	3.356	40532	0.8642
4.	T60	40% Diesel fuel + 60% Turpentine oil fuel	3.534	39448	0.8828
5.	T80	20% Diesel fuel + 80% Turpentine oil fuel	3.713	38364	0.9014
6.	T100	100 % Turpentine oil fuel	3.89	37280	0.92

3. Materials and Methods

A four stroke single cylinder, direct injection, water cooled, constant speed, vertical cylinder direct injection Kirloskar TV-1 were used in the test. Technical specification of the engine is given in the table 4 and the experimental set up as shown in the figure 1. The engine was coupled with an eddy current dynamometer to apply different loads.

The AVL make transducer with the sensitivity 16:11 pc / bar water cooled piezo-electric pressure transducer into the cylinder head is used to measure the combustion chamber pressure. An AVL 3057 charge amplifier converts the charge yield by the piezo-electric transducer into the proportional electric signals. A personal computer (PC) interfaced with an AVL 619 indimeter hardware and medium – software version 2.2 data acquisition system to collect combustion parameter data such as heat release rate, incylinder pressure and cyclic variations. Engine speed was measured by inbuilt magnetic pick-up sensor connected to a frequency meter. An AVL 444 Di gas analyzer was used to measure the oxides of Nitrogen (NOx), Carbon monoxide (CO) and Hydrocarbon (HC) emissions. The smoke intensity was measured by an AVL 413 smoke meter. The exhaust gas temperature was measured with k-type thermocouple.

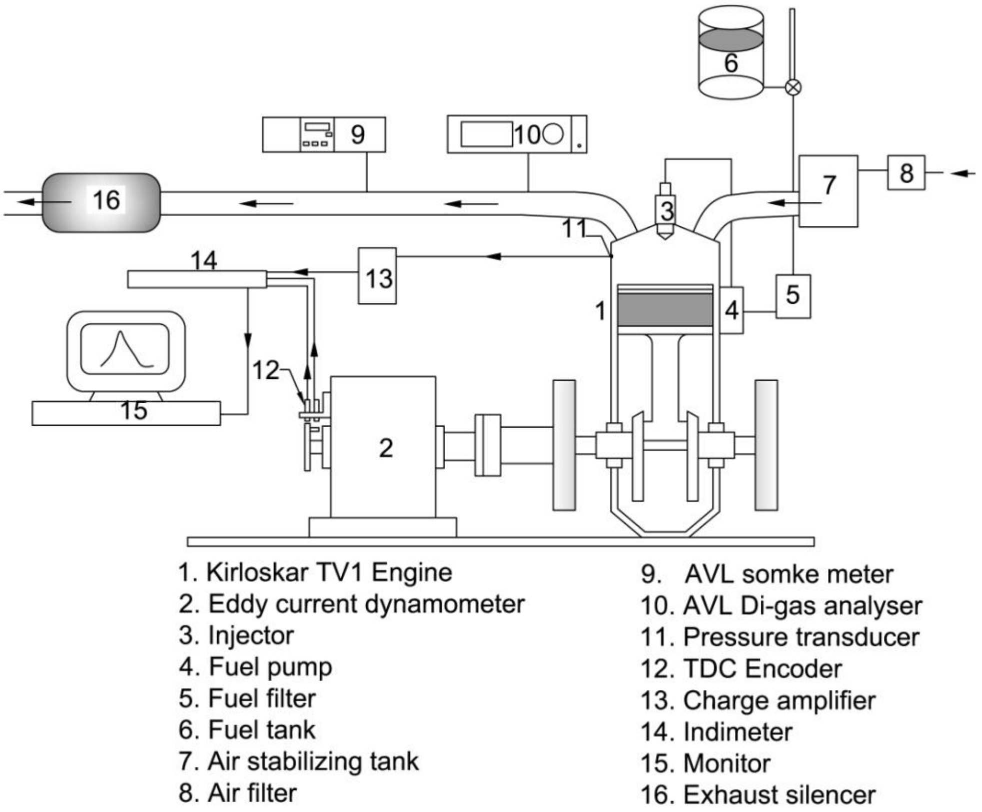


Figure 1. Experimental setup

4. Engine test

The engine was run for 30 min with diesel fuel to attain a normal working temperature. In first phase the test was conducted in the uncoated engine (UCE) and the results were obtained. In second phase of work, the test was repeated under the same conditions in the coated engine (CE). The engine was maintained at the constant speed and all the measurements were repeated for a minimum of three times. Finally the average value of the three readings was taken for the calculation.

4.1. Error Analysis

The Errors and uncertainties in the experiments can arise normally from selection of the instruments, condition, calibration, environment, observation, reading and test planning. The uncertainty analysis is needed to show the accuracy of the experiments. The various parameters like total fuel consumption, brake power, specific fuel consumption and brake thermal efficiency were calculated using the percentage uncertainties of various instruments shown in the table 5.

Total percentage uncertainty of this experiment is

= Square root of [(Uncertainty of TFC)^2 + (Uncertainty of brake power)^2 + (Uncertainty of specific fuel consumption)^2 + (Uncertainty of brake thermal efficiency)^2 + (Uncertainty of HC)^2 + (Uncertainty of NOx)^2 + (Uncertainty of EGT indicator)^2 + (Uncertainty of hartidge smoke meter)^2 + (Uncertainty of pressure pickup)^2]

= Square root of [$(1)^2 + (0.2)^2 + (1)^2 + (1)^2 + (1.03)^2 + (0.014)^2 + (0.15)^2 + (1.0)^2 + (0.1)^2$]

= 2.26 %.

Using the above calculation procedure the total uncertainty of the whole experiment was obtained and it is ± 2.26 %.

Tabe-4 Error Analysis

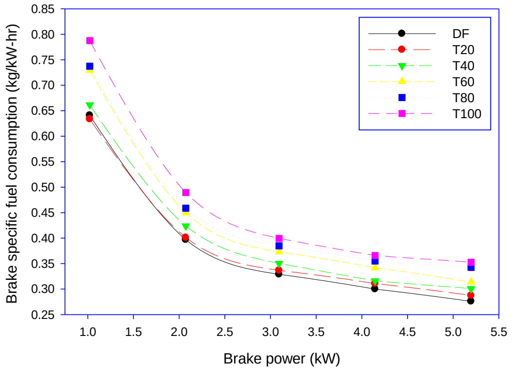
Instruments	**Range**	**Accuracy**	**Percentage uncertainties**
Gas analyzer	CO 0-10%	± 0.02%	±0.2%
	CO$_2$ 0-20%	± 0.03%	± 0.09 %
	UBHC 0-100000 ppm	± 20 ppm	± 1.03 %
	NOx 0-5000 ppm	± 10 ppm	± 0.014 %
Smoke meter	HSU	± 0.1	± 1%
EGT indicator	0-900 °C	± 1 °C	± 0.15%
Load indicator	0-100 kg	± 0.1 kg	± 0.2%
Pressure pickup	0-110 bar	± 0.1 kg	± 0.1%
Crank angle encoder	-------	± 1 °	± 0.2%

5. Results and Discussion

5.1 Performance parameters

5.1.1. Brake Specific fuel consumption

Figure 2 a and b shows the brake specific fuel consumption (BSFC) of diesel and TPOF blends at varying brake loads in an uncoated and coated engine respectively. The BSFC of diesel and TPOF blends is less for coated engine (CE) compared to uncoated engine (UCE). The decrease in brake specific fuel consumption is 1- 6 % for coated engine. At all blends of TPOF in the coated engine the BSFC is lesser. At full load,BSFC of the CE for T40 blend is 3.89% lesser as compared to UCE. Due to the ceramic coating, there is higher operating combustion temperature in the combustion chamber as compared to uncoated engine which improves the combustion characteristics and hence fuel combustion is effective. And also, the heat loss through radiation and coolant is less for the CE. However the fuel consumption is found to increase with higher proposition of TPOF blends as compared to diesel fuel (DF) in both UCE and CE. This trend was obtained because of higher fuel density and viscosity [5-7].The higher density of blends contains higher percentage of TPOF which led to more discharge of fuel for the same displacement of the plunger in the fuel injection pump. The BSFC increases in both the engines are also attributed due to 10% lower calorific value of TPOF blends as compared to DF. Similar results are obtained by researchers [9-11].

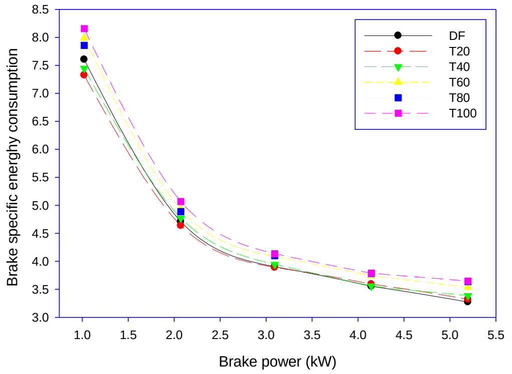


(a) (b)

Figure 2: Comparison of BSFC of test fuels at different brake power in UCE (a) and CE (b)

5.1.2 Brake specific energy consumption

In an Internal combustion engines, the chemical energy is converted into mechanical energy. Thus the brake specific energy consumption (BSEC) is used for comparison rather than specific fuel consumption. The BSEC is an important parameter which measures the amount of energy given as input to develop one – kilowatt power [13, 14]. Figure 3 a and b shows the brake specific energy consumption against the brake power. The BSEC of coated engines is less as compared to uncoated engine at all loads. However, the specific energy consumption increases for higher concentration of TPOF in the blend as compared to DF in both uncoated and coated engines at all the loads. This due to the combined effect of lower heating value of fuel and high density of TPOF blends and hence larger amount of TPOF is supplied to maintain constant brake power output [15].

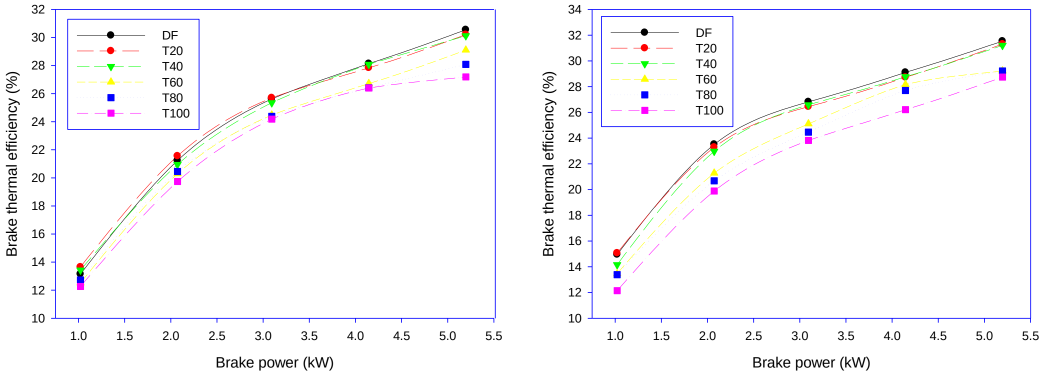


(a) (b)

Figure 3 Comparison of BSEC of test fuels at different brake power in UCE (a) and CE(b)

5.1.3 Brake thermal efficiency

The Brake thermal efficiency of the uncoated and coated engine against brake power is shown in the figure 4 a and b. In coated engine, the brake thermal efficiency of TPOF blends and diesel fuel is higher as compared to uncoated engine. Because of the effect of ceramic coating lower heat rejection from combustion chamber through thermally insulated components, there is an increase in available energy which leads to greater adiabatic of energy in combustion chamber and hence complete combustion is possible. Thus the thermal efficiency is higher than the conventional engine [16]. However in both engines, it is observed that when the concentration of TPOF increases in the blends, there is reduction in brake thermal efficiency as compared to DF. The brake thermal efficiency of an engine depends upon many factors. The most important property is calorific value and specific gravity of the fuels. The energy input or consumption to the engine in case of higher concentration of TPOF is higher as compared to DF, this leads to reduction in brake thermal efficiency at all loads. On comparing test results of the engines,T40 blend brake thermal efficiency is almost similar to the DF .This is observed because of larger fuel droplets and oxygen content in TPOF blend which leads to better atomization and gives complete combustion.



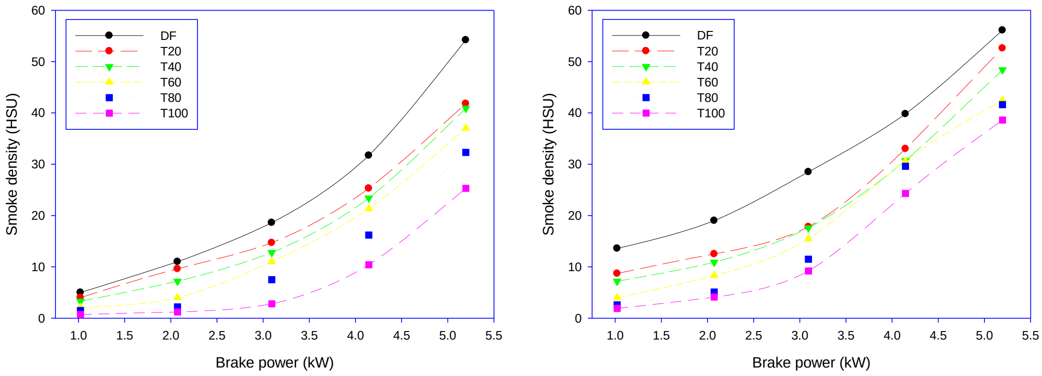
(a) (b)

Figure 4 Comparison of Brake thermal efficiency of test fuels at different brake power in UCE (a) &CE (b)

5.2 Emission parameters

5.2.1 Smoke density

The figure 5 a and b shows the variation of smoke density for diesel fuel and various blends of TPOF over the entire range of brake power in an uncoated and coated engine. From the results, in both the engines smoke density is reduced with respect to increase in turpentine fraction in the blends as compared to DF. The reason for reduced smoke is due to higher heat release rate during diffusive combustion phase with increase in turpentine fraction. The higher oxygen content in TPOF blends leads to an improvement in diffusive combustion phase. At full load about 25% reduction of smoke for T40 blend compared to DF in UCE and CE. However the smoke density for the CE is higher as compared to UCE. The increase in the smoke density for coated engine may be due to long combustion duration.

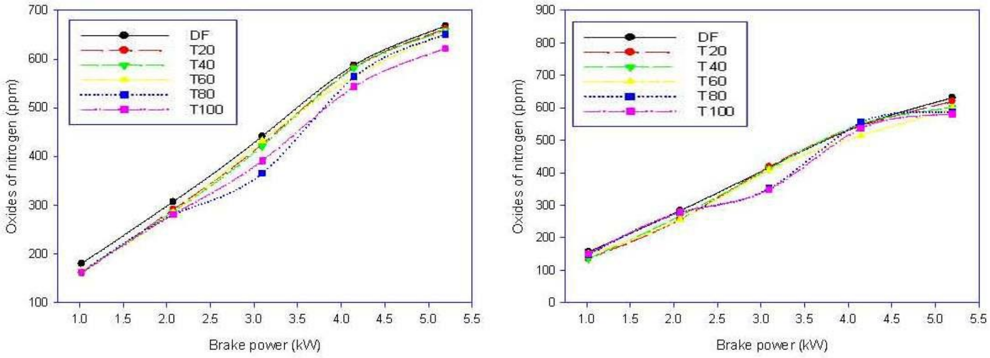


a b

Figure 5 Comparison of smoke density of test fuels at different brake power in UCE (a) and CE (b)

5.2.2 Oxides of nitrogen

The figure 6 a and b shows the variation in Oxides of nitrogen **(**NOx) emission in UCE and CE for various TPOF blends as reference to diesel fuel. It is seen that there is decreasing tendency in NOx emission with the use of TPOF blends in UCE and CE as compared to the DF.The same trend is observed in both the engines. The decrease in NOx emissions is usually proportional to the TPOF in the blends. It is known that NOx emissions are caused by higher combustion temperature and higher oxygen concentration in the cylinder .TPOF produce lower combustion temperature due to their lower heating value and oxygen content. Both the lowering combustion temperature and extending ignition delay as well as low heating value constitute a combined effect for reduction of NOx emissions with the use of TPOF blends. It is seen that about 10.02% of NOx emission with 19.2 % reduction in exhaust temperature is reduced for T40 blend in coated engine. Low temperature combustion of TPOF blend is the main reason to reduce NOx emission and exhaust temperature.

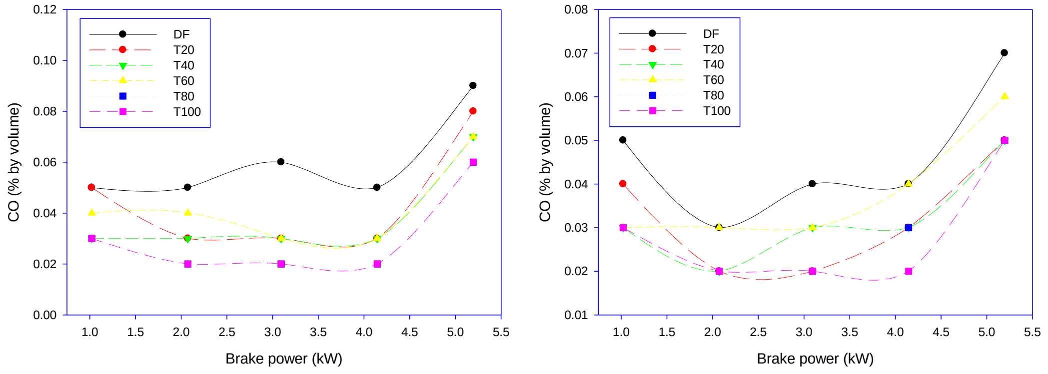


a b

Figure 6 Comparison of NOx of test fuels at different brake power in UCE (a) and CE (b)

5.2.3 Carbon monoxide (CO) emission

Incomplete combustion leads to formation of carbon monoxide. The insufficiency of oxygen during the combustion is the main reason for the formation of CO. The figure 7 a and b shows that the CO emission of various TPOF blends in uncoated and coated engine against brake power. In both the engines, CO emission is lesser for TPOF blends as compared to diesel fuel. The CO emission of CE is less as compared to UCE. The improved spray performance and combustion behavior may be the reason for this reduction [12,17]. From the results, it is observed that at full load the T 40 blend provide lower CO emissions compared to diesel baseline in both the engines. There is approximately 22 – 40 % of CO emission reduced at full load for T40 blend as compared to DF in UCE and CE.Moreover, another reason for reducing CO emission is because of lower cetane number of TPOF in the blends.

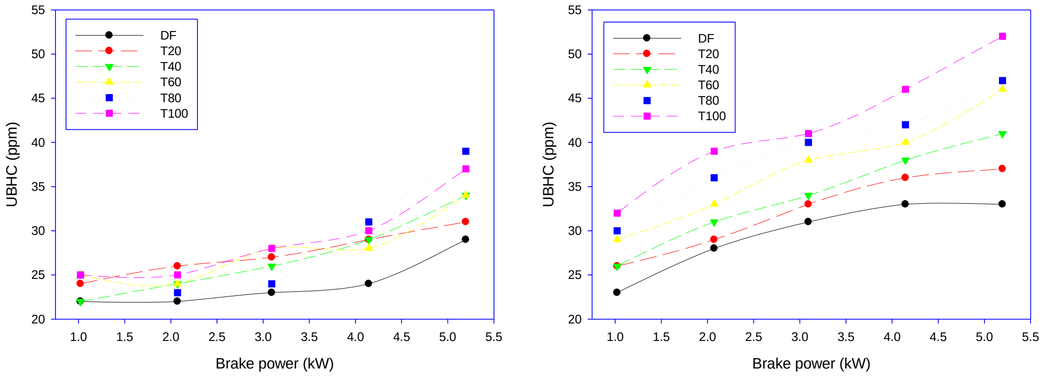


a b

Figure 7 Comparison of CO of test fuels at different brake power in UCE (a) and CE (b)

5.2.4 Unburned hydrocarbon (UBHC) emission

The figure 8 a and b shows the UBHC emission against brake power in uncoated and coated engine. The UBHC emission is slightly increased both in UCE and CE. From the results, the average increase in UBHC emissions compared to base line DF in both UCE and CE is between 11.8 and 16.7% with blends of TPOF .The low cetane number of the TPOF is the main reason for increase in UBHC emissions with blends. The low cetane number deteriorates self – ignition characteristics of the blends and promotes quenching effect in the leaner mixture zone of the cylinder. It leads to higher fumigation rate and the dominance of oil cracking. Fuel escapes combustion by means of wall quenching of the flame, bulk quenching, valve overlap and lean mixing during the ignition delay period [18]. Therefore, UBHC emission is increases for increase in concentration of TPOF in the blends.



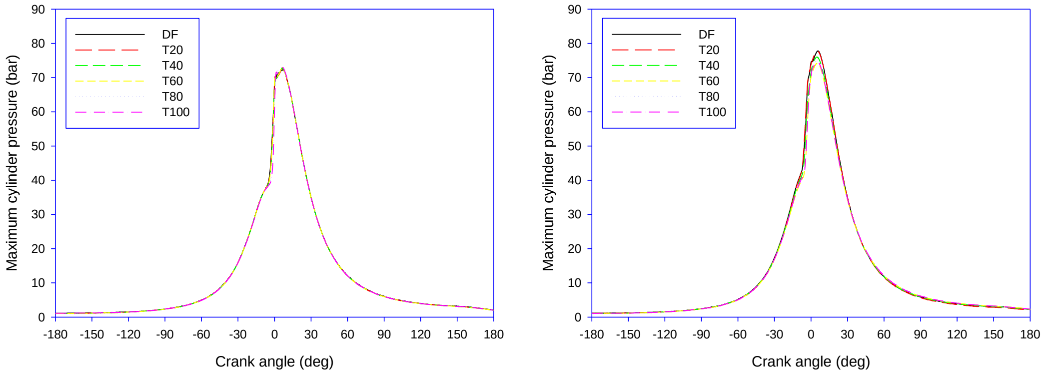
a b

Figure 8 Comparison of UBHC of test fuels at different brake power in UCE (a) and CE (b)

5.3 Combustion analysis

5.3.1 Cylinder pressure

The figure 9 a and b shows the P-θ diagram for a diesel fuel and various TPOF blends in uncoated and coated engine. The various factors that affect the peak pressure are compression ratio and influence of combustion chamber design and combustion duration, fuel specific heat, energy content and quality. During the combustion of fuel, the energy release is used to generate internal pressure developed in the net calorific value of the fuel [12]. From the figure it has been observed that the TPOF blends produces similar in-cylinder pressure patterns as compared to diesel fuel. The maximum cylinder pressure for T40 blend in coated engine is 76.019 bar and 72.71 bar for the uncoated engine. It is found that the TPOF and diesel fuel produce peak pressure within 5 – 15 crank angle after TDC, due to its high burning rate at the initial stage . Thus it is concluded that the engine operation with these TPOF blends did not pose any problem related to knock, combustion (or) partial burn.

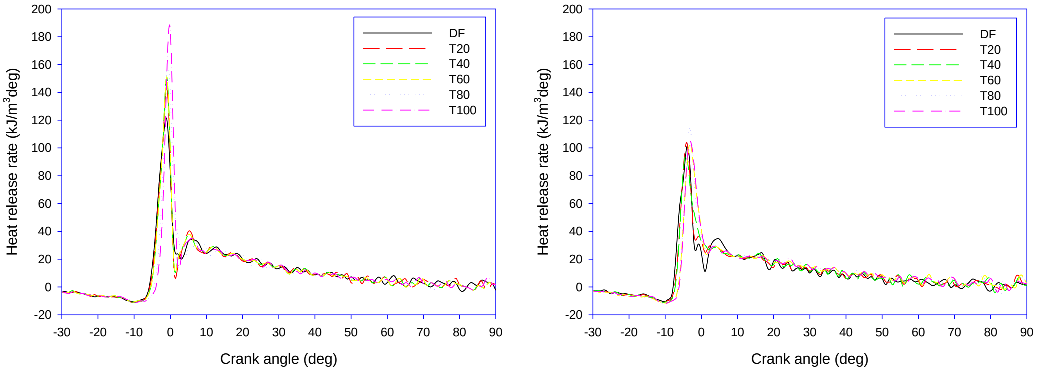


a b

Figure 9 Comparison of Cylinder pressure of test fuels at different brake power in UCE (a) and CE (b)

5.3.2 Heat release rate

The figure 10 a and b shows the heat release rate (HRR) of TPOF blend and diesel fuel at varying crank angles from -30° to 90° in uncoated and coated engine. The heat release rate traces are similar for the various TPOF blends and TPOF alone but it will be differ slightly from the diesel fuel. It has been observed that the heat release rate for diesel fuel in uncoated engine and coated engine is 121.650 kJ/m^3 deg & 100.9810 kJ/m^3 deg respectively. For T40 blend the heat release rate is 151.057 kJ/m^3 deg and 94.052kJ/m^{3}deg for UCE and CE at full load. This shows that the HRR of TPOF blend in coated engine is lower as compared to uncoated engine. And also the occurrence of peak heat release rate for T40 blend is advanced approx 2 degree in case of uncoated engine and 4-5 degrees in coated engine. The rate of heat release of ceramic coated engine slightly shifted from the top dead center due to reduced premixed combustion. Due to this advance, it increases the heat losses, friction losses and peak cylinder pressure. The other blends of TPOF also show the same trend in CE. This could be attributed to lower internal energy and slow burning, because of lower calorific value and slight increase in density of TPOF blends. The same trend is observed by the other researcher R.Kama.et.al. [19].

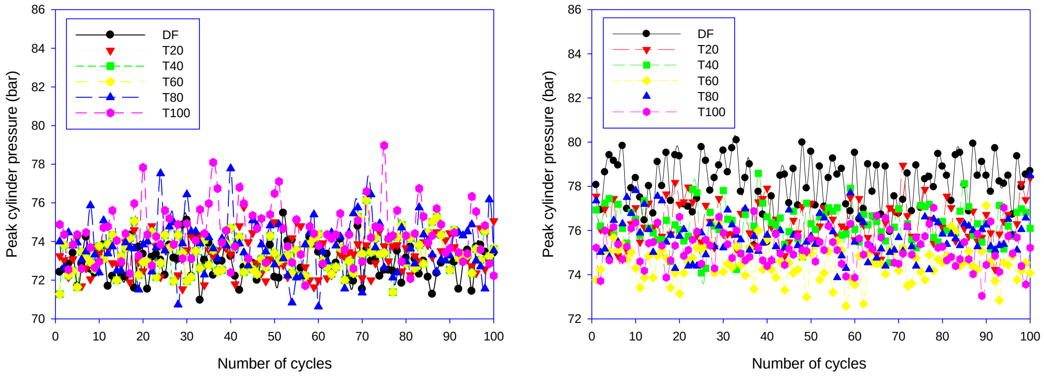


a b

Figure 10 Comparison of Heat release rate of test fuels at different brake power in UCE (a) and CE (b)

5.3.3 Peak cylinder pressure

The figure 11 a and b shows that plot of Peak cylinder pressure for the uncoated and coated engine at maximum brake power. The cylinder pressure is an important parameter of the cyclic variations.100 continuous cycles of incylinder pressure are measured and analyzed for peak pressure variation. It has been observed that cylinder pressure for CE increase in the range of 2.62 bar at low loads and 5.05 bar increases at full load as compared to UCE for T40 blend. The same trend is observed for all the TPOF blends in CE. Effective combustion is due to high average cycle temperature. In both the engines, it is clearly shown that the DF, TPOF in isolation and TPOF blends are almost similar, but a greater magnitude deviation is observed between TPOF in isolation and diesel fuel.



a b

Figure 11 Comparison of Peak cylinder pressure of test fuels at different brake power in UCE (a) & CE (b)

6. Conclusion

The test is conducted in both uncoated and coated engine. From the result, it is concluded that

1. The brake thermal efficiency of DF and TPOF blends increases up to 6 % in the coated engine as compared to the uncoated engine.
2. The BSEC of coated engines is less as compared to uncoated engine at all loads.
3. Exhaust emission reduces in the coated engine compared to uncoated engine except smoke and UBHC emission.
4. The Heat release rate reduces about 27.97% for coated engine when compared to uncoated engine
5. Peak cylinder pressure slightly increases for the thermal barrier coated engine

6. There is no significant carbon deposit found in the injector nozzle tips for all the blends of TPOF and DF.

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