COMPARATIVE EXPERIMENTAL INVESTIGATION ON DI DIESEL ENGINE USING ALTERNATIVE FUEL

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ABSTRACT

The present investigation is aimed to examine the performance and emissions of a single cylinder, four stroke, direct injection diesel engine developing 4.4 kW at a rated speed of 1500 rpm, at optimum injection timing and optimum injection pressure of 230 bar, with lower compression ratios of 17:1 and 16:1, fueled with UTO by varying the clearance volume. At lower compression ratio, the engine exhibits a lower thermal efficiency and more smoke level. The engine behavior was also tested at higher compression ratio of 18.5:1. The effect of compression ratio on emission parameters of the engine fueled with UTO in comparison with diesel fuel operation is obtained. The optimum compression ratio was found to be 18.5:1. The results indicated that at higher compression ratio, there was an increase in brake thermal efficiency, NO emission and reduction in smoke. At this CR, the NO emission was found to be increased by about 2.8% and 32.1% respectively than that of the UTO and diesel. The smoke emission was found to decreased by about 3.6% and 10.4% respectively compared to that of UTO and diesel at full load. Further, the engine fueled with UTO was subjected to operate with optimum injection timing and optimum fuel nozzle opening pressure at different compression ratio (two lower and one higher CRs). The results were compared with diesel operation. A comparison was made between the results obtained from these investigations (i.e. operating the engine fueled with UTO at different compression ratio only with optimum injection timing and, optimum injection timing and optimum nozzle opening pressure). With optimum injection timing and optimum nozzle opening pressure, there was 3.25% and 2.14% increase in brake thermal efficiency and NO emission and 6.52% reduction in smoke at higher CR. At CR 17:1 the results are closer to diesel operation with optimum injection timing and optimum nozzle opening pressure.

Key words: *Compression Ratio, Diesel Engine, Emission, Performance, UTO.*

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1. INTRODUCTION

Crude oil is the lifeblood of the modern world, which serves in all the sectors that includes transportation, agricultural, commercial and domestic and power generation. In the year 2009, the world consumed an estimated 84 to 85 million barrels of oil. There is a growing demand and cost of liquid fuel in every country. In the last six decades, India's energy consumption rate increases by 16 times because of fast rate of population growth [1]. At this rate, the fossil fuel will not be available for a long time as the gap between supply and demand increases large. With continuous use of petroleum products world is moving towards technological growth as well as environmental degradation. In order to face the crisis of energy in the future and growing concern with the pollution, the substitutes of petroleum fuels is necessary. A large amount of crude oil is imported from the foreign countries; it is also a reason for the development of alternative fuel. The use of alternative fuels is the only possible solution, which can be obtained either from the renewable sources or non-conventional sources. As we know that, compression ignition (CI) engine is widely used in several applications; the search for alternative fuels for CI engines is very important [2]. Some of the fuel falls under this category are alcohol, vegetable oil, bio mass, bio gas, used oil etc. Few alternative fuels can be directly used without any modification in the fuel or engine, but some of them need little modification to obtain relevant properties like conventional fuel. An important equipment used for transmission and distribution of electrical energy is power or electrical transformer at different power stations and distribution stations. The extent to which the transformer oils are resilient to physico-chemical stresses induced by operating condition depends on the saturation or stability of the constituent substances. High stability leads to minimum sludge formation with time. Higher the compositional purity, higher the cooling function and the heat conduction capacity of the fuel [3]. Due to electric and cyclic thermal stresses transformer oil suffers from continuous deterioration and degradation because of loading and the climate conditions. This will affect the life and as well as performance of the electrical transformer. The expected service life of a transformer is about 40 years which strongly depends on the manufacturer, design, materials used, assembly quality, operating conditions and maintenance [4]. Day by day, more transformers are installed and the old transformer oils have to be scrapped, so it is difficult to estimate the annual disposed quantity of UTO with the statistics. To avoid the hazardousness and deterioration of the oil, it is essential to monitor the characteristics of transformer oil continuously. The preventive maintenance is done according to the utilization of the transformer oil. It is important that transformer oil is less biodegradable so there is some difficulty with disposal, it could be contaminate our soil and waterways. There are many transformers installed in high population areas and malls. By the effective utilization of UTO, the disposal problem in open land and the environmental problem can be solved. Recent literature reveals that in a CI engine, the UTO can be used as an alternative fuel after a proper treatment to obtain a suitable property [5].

2. LITERATURE SURVEY

Several researchers have conducted experiments to study the combustion, performance and emission characteristics of a diesel engine with alternative fuels. Research works on such studies show that different kinds of alternative fuels viz. methyl esters of rapeseed oil methyl ester, palm oil methyl ester, corn oil, olive kernel oil, deccan hemp oil, jojoba oil, paradise oil, eucalyptus oil, poon oil, pongamia pinnata methyl ester, coconut oil have been used as investigated alternative fuels for diesel engine [7].

Wallace, W. et al [8] described the effect of compression ratio on engine behavior, mechanical execution, and development of an automatic, hydraulically-actuated piston that provides a practical method of obtaining a variable compression ratio engine. When applied to compression-ignition engines, an increase in output of 50% (0.5 to 0.75 bhp/cu in.) has been achieved without a corresponding increase in maximum combustion pressure. By providing a high compression ratio for starting and light load conditions, the engine has demonstrated substantial improvements in cold starting ability as well as improved potential for multifuel operation.

Raheman H. et al. [9] investigated the performance of Ricardo E6 engine using biodiesel obtained from mahua oil (B100) and its blend with high speed diesel (HSD) at varying compression ratio (CR), injection timing (IT) and engine loading (L). The brake specific fuel consumption (BSFC) and exhaust gas temperature (EGT) increased, whereas brake thermal efficiency (BTE) decreased with increase in the proportion of biodiesel in the blends at all compression ratios (18:1–20:1) and injection timings (35- 450 before TDC) tested. However, a reverse trend for these parameters was observed with increase in the CR and advancement of injection timing. The BSFC of B100 and its blends with high speed diesel reduced, whereas brake thermal efficiency and exhaust gas temperature increased with the increase in load (L) for the range of compression ratio and injection timing tested. The differences of BTEs between HSD and B100 were also not statistically significant at engine settings of ‘CR20IT40’ and ‘CR20IT45’. Thus, even B100 could be used on the Ricardo engine at these settings without affecting the performance obtained using HSD.

Laguitton et al. [10] studied the effect of CR on the emissions of a diesel engine when CR is reduced from 18.4 to 16, in a single cylinder. This was achieved by modifying the piston bowl while maintaining the production engine squish clearance. Investigations on the effect of injection timing were performed at a number of the key operating points and the corresponding pressure-time curves analysed to help explain the measured results. It was reported that, although there was a small CO and HC penalty, lowering the compression ratio or retarding the injection timing results reduction in NOx and soot emissions.

Cayin and Gumas [11] investigated the influence of the CR, injection timing and injection pressure on the performance and emission of a DI diesel engine using biodiesel blended-diesel fuel. The tests were carriedout at three CRs 17:1, 18:1, and 19:1. It was reported that the BTE increased with increase in the CR while BSFC and BSEC decreased. For the all fuels tested, there was an increase in the NO emission, while the CO, HC emissions and the smoke opacity decrease with increase in the CR.

Deore, Eknath R. et al. [12] conducted experiments on 3.75 kW diesel engine AV1 single cylinder water cooled, Kirloskar make tested for blends of diesel with ethanol. Tests were conducted for three different compression ratios. Engine test setup was developed with moving cylinder head for variation of compression ratio to perform investigations using these blends. The engine performance studies were conducted with rope break dynamometer setup. Parameters like speed of engine, fuel consumption and torque were measured at different loads for pure diesel for blends of diesel with ethanol at different compression ratio. Break Power, BSFC, BTE and heat balance were evaluated. Results were recorded for 5% to 20% ethanol in the blend and three different compression ratios.

The aim of this study is to develop a way to utilize UTO in diesel engine. It is an attempt has been made to assess the combustion, performance and emission characteristics of a CI engine fueled with UTO with different CRs. The experimental results are recorded, analyzed and compared with those of diesel operation and presented in this paper.

3. EXPERIMENTAL SETUP & METHODOLOGY

3.1 Nature of Transformer Oil

Inhibited oil is formulated with hydro treated naphthenic base oil and an oxidation inhibitor to control sludge and deposit formation. It provides an extended service life compared to non-inhibited transformer oils. It has an excellent low-temperature properties and is noncorrosive to copper and copper alloys. This oil does not contain any polychlorinated biphenyls. Mineral oil, synthetic esters and silicon oils are traditionally used as transformer oils. Mineral transformer oil is composed of hydrocarbons of paraffinic, aromatic or naphthenic structure that are obtained by fractional distillation of crude petroleum. Synthetic oils are produced by substituting a chlorine atom for a hydrogen atom in hydrocarbon molecules. Apart from these, transformer oil is also produced from various vegetable oils, such as coconut oil, sunflower oil, soybean oil and castor oil [15, 16].

3.2 Physical properties and chemical composition of UTO

A comparison of the chemical composition between of UTO and diesel is shown in Table 3.1. The chemical composition of the UTO indicates that the fuel has carbon close to that of diesel fuel. The hydrogen present in the UTO is 1.5 times lesser than that of diesel. It is evident from the table that the UTO has considerable oxygen present in it. Due to the oxidized nature, the fuel may be helpful in better combustion of the fuel air mixture. The kinematic viscosity of the UTO is approximately 6 folds higher than that of UTO. Table 3.2 gives the comparison of the physical properties between UTO and diesel.

Table 3.1 Chemical composition of UTO and diesel [5]

Description	Diesel	UTO
C (%)	86.5	89.95
H (%)	13.2	9.19
N (%)	0.18	0.03
S (%)	0.3	0.35
O by difference (%)	0	0.44
C/H ratio	5.437	19.302
Carbon residue (%)	0.02	0.02

Table 3.2 Properties of UTO and diesel

Property	Test method	Diesel	UTO
Kinematic viscosity (cSt@ 27°C)	D-0445	2.4	13
Flash point (°C)	D0093-02A	76	150
Fire point (°C)		56	172
Pour point (°C)	D0097-05A	-16	-16.7
Density (kg/m3)	D-1298	860	890
Lower calorific value (kJ/kg)	D-4809	44800	39270
Sulfur content (%)	D129-00R05	0.05	0.02
Cetane number		40-55	43.6*[67]
Carbon residue (%)	D2500-05	0.01	0.02
T10 (°C)		210	320
T50 (°C)		230	340
T90 (°C)		260	370
T100 (°C)		350	360

3.3 Method to change compression ratio

The compression ratio of the engine can be varied by five methods that are described below;

3.3.1 Cylinder swept volume

The swept volume of the cylinder indicates how much air the piston displaces as it moves from BDC to TDC. Increasing the cylinder volume without making any other changes will increase the compression ratio because it enlarges the cylinder volume without increasing the combustion chamber volume. In other words, the piston will have to suck more air into the same amount of space.

3.3.2 Clearance volume

Clearance volume is determined by the distance from the cylinder block deck to the top of the piston flat (not counting any dishes or domes) when the piston is at TDC. In engines, the pistons don't come all the way up to the height of the deck. They can be anywhere from 0.003 to 0.020 inch below it. This amount is known as the piston deck height, and it affects compression ratio because it affects the volume of air in the combustion area when the piston is at TDC. If the piston deck height is inreased, then clearance volume is increased and the compression ratio is reduced. If the piston is closer to the deck, then the clearance volume is reduced and compression ratio is increased.

3.3.3 Piston dome or dish

It is associated with the piston geometry. Instead of a flat top surface it has a dome which looks like a dome stadium. Clearance volume does not take into account any pop-up domes or sunken-in dishes on the head of the piston. These configurations also increase or decrease volume in the combustion chamber and affect the compression ratio. For the purpose of calculating compression, the dish is preferred as a positive quantity because the dish adds volume to the cylinder hence reduces the compression ratio. Dome is just reverse of the dish, it subtracts volume from the cylinder and increases the compression ratio.

3.3.4 Combustion chamber

The volume of the combustion chambers is the final factor in determining the compression ratio. The larger the chamber, the more volume is added to the cylinder and the lower the compression ratio; smaller chambers reduce the volume and increase the compression ratio.

3.3.5 Head-gasket volume

Head-gasket volume is determined by the compressed thickness of the gasket. A thicker gasket adds volume and reduces compression; a thinner gasket reduces volume and increases compression.

In this experiment, the change in compression ratio was done by change in head gasket volume. Here is how to calculate the volume of the head gasket;

$$\text{Head Gasket Volume} = 0.786 \times (\text{Cylinder Bore})^2 \times \text{Compressed Thickness}$$

Table 3.3 Clearance volume and gasket volume for different compression ratios

Compression ratio	Total clearance volume (cm^3)	Head gasket volume (cm^3)
16:1	44.12	9.67
17:1	41.32	6.92
17.5:1	40.01	5.6
18.5:1	38.21	4.21

3.4 Engine specification

The test engine used in this investigation was a Kirolaskar TAF-1 single cylinder, four-stroke, air cooled, constant speed, direct injection diesel engine. The specifications of the engine are given in Table 3.4.

Table 3.4 Specifications of the test engine

Parameter	Value/dimension
Speed (rpm)	1500
Bore (mm)	87.5
Stroke (mm)	110
Rated brake power (kW@1500 rpm)	4.4
Compression ratio	16:1, 17:1, 17.5:1, 18.5:1
Nozzle opening pressure (bar)	230
Injection timing (°CA bTDC)	20

3.5 Description of the test engine

Fig. 3.1 shows the schematic diagram of the experimental setup. A control panel (1) to provide electrical load to the engine (2) was fitted with the electrical resistance dynamometer (3) known as alternator. A fuel tank (4) was connected to the engine for continuous fuel supply. There was included a burette (5) and a fuel sensor (6) with the fuel circuit to measure the fuel consumption and give input to the computer (7) through data acquisition system (8). A pressure transducer (9) was mounted on the engine head to measure the cylinder pressure with loads. The model of the Kistler pressure transducer was 6613A, which has an advantage of a good frequency response and linear operating range. A continuous circulation of air was maintained for cooling the transducer, by using fins to maintain the required temperature. A crank position sensor (10) was connected to the output shaft to record the crank angle.

Combustion parameters such as the peak pressure, time of occurrence of peak pressure, heat release rate and ignition delay were obtained with the help of software provided by legion brothers. Atmospheric air enters the intake manifold of the engine through an air filter and an air box (11). An air flow sensor (12) fitted with the air box gave the input for the air consumption to the data acquisition system. All the inputs such as air and fuel consumption, engine brake power, cylinder pressure and crank angle were recorded by the data acquisition system, stored in the computer and displayed in the monitor. A thermocouple in conjunction with a temperature indicator (13) was connected at the exhaust pipe to measure the temperature of the exhaust gas. An AVL DiGas444 exhaust gas analyzer (14) was used to examine the engine exhaust components like CO2, CO, O2, HC, and NO in percent. The smoke density of the exhaust was measured by the help of an AVL437 (15) diesel smoke meter.

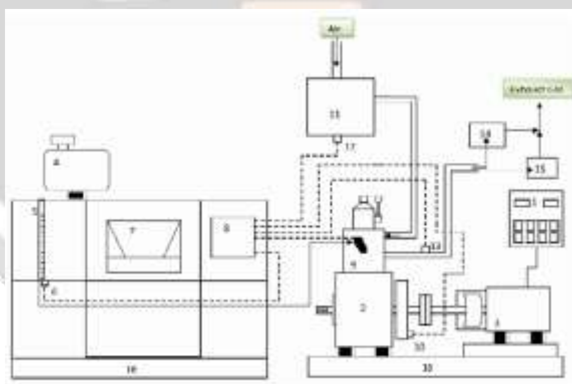

Fig. 3.1 Schematic diagram of experimental setup

Table 3.5 Components of experimental setup

1	Electrical Load	9	Pressure transducer
2	Engine	10	Speed sensor
3	Alternator	11	Air box
4	Fuel tank	12	Air sensor
5	Burette	13	Temperature indicator
6	Fuel sensor	14	Exhaust gas analyzer
7	Computer	15	Smoke meter
8	DAS	16	Bed

By changing the volume of the cylinder head gasket we can change the compression ratio up to a limit.

4. RESULTS AND DISCUSSIONS

Phase I: Preliminary investigation on performance and emission parameters for DI diesel engine only with optimum injection timing of 20° bTDC and standard nozzle opening pressure of 200 bar.

4.1 Performance parameters

4.1.1 Brake thermal efficiency

Brake thermal efficiency give an idea of the output generated by the engine with respect to the heat supplied in the form of fuel. Generally, increasing the compression ratio improved the efficiency of the engine. This improvement in performance of the engine at higher CR is due to the reduced ignition delay. Fig. 4.8 shows the variation of the BTE with brake power for the UTO and standard diesel.

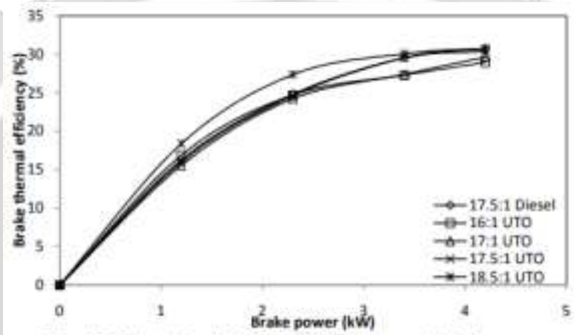

Fig. 4.8 Variation in brake thermal efficiency with brake power

The values of the BTE for diesel and the UTO are found to be about 30.7% and 30.4% at standard CR and maximum brake power. The brake thermal efficiency of UTO is 28.9%, 29.6% and 30.7% at compression ratio of 16:1, 17:1 and 18.5:1. With increase in CR results in improvement of the BTE [18]. This can be attributed to better combustion and better intermixing of air and fuel inside the combustion chamber.

4.1.2 Brake specific energy consumption

An important parameter to measure the engine performance is the specific energy consumption. It is the product of brake specific fuel consumption and lower heating value.

Fig. 4.9 shows the variation between brake specific energy consumption and brake power. As the fact, the BSEC decreases with increase in engine load.

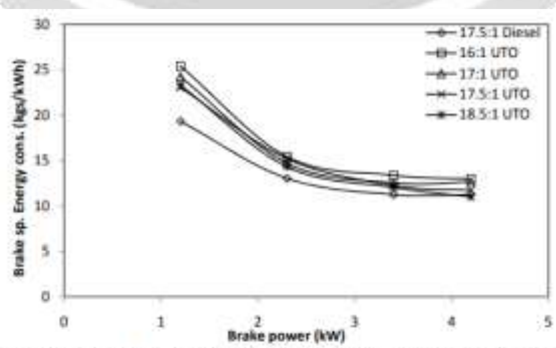

Fig. 4.9 Variation in brake specific energy consumption with brake power

Compare to diesel engine at standard situation, UTO shows 15.5%, 12.4% and 4.3% higher consumption at CR`s 16:1, 17:1 and 18.5:1 while CR of 18.5:1 consumes 2.1% lower than diesel. The increase in the fuel consumption is due to fuel density, viscosity and heating value, but with higher compression ratio lesser value of SEC is apparently desirable.

4.1.3 Exhaust gas temperature

During the combustion inside cylinder the temperature is very high, but with expansion there is a great reduction in the exhaust gas temperature. So the exhaust gas temperature is strongly dependent on in-cylinder temperature and expansion process.

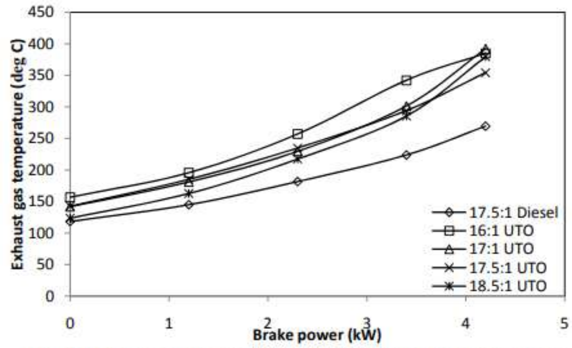

Fig. 4.10 Variation in exhaust gas temperature with brake power

Figure 4.10 portrays the variation of the exhaust gas temperature with brake power for diesel and UTO. The exhaust gas temperature increases with an increase in the engine load for the UTO and diesel [18]. It is higher for the UTO than that of diesel in the entire operation as a result of higher viscosity and density of UTO. The exhaust gas temperature of UTO at compression ratio of 16:1, 17:1, 17.5:1 and 18.5:1 are 385^{o} , 392^{o} , 354^{o} and 350^{o}C respectively whereas in diesel 269^{o}C. The reason for the reduction in the exhaust gas temperature at increased CR is due to improved energy conversion of UTO as compared to that of diesel [19].

4.2 Emission parameters

4.2.1 Carbon monoxide (CO) emission

The amount of CO increases due to less availability of air, poor mixing of air with fuel and rise in temperature in the combustion chamber. A small amount of CO is also occurs due to fuel viscosity and fuel spray quality. Fig. 4.11 portrays the percentage variation of the carbon monoxide emission with brake power for UTO compared to diesel at different compression ratio.

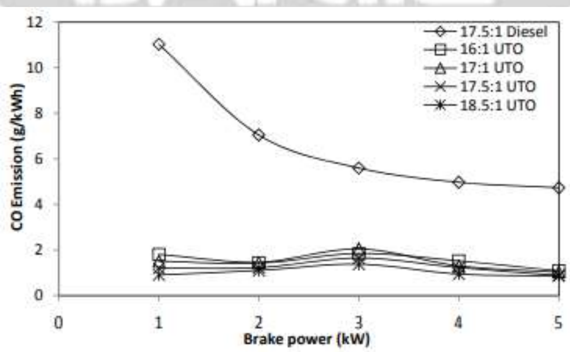

Fig. 4.11 Variation in CO emission with brake power

It can be observed from the figure that the CO emission is higher for the UTO compared to that of diesel at maximum brake power for all compression ratios. Lower CRs of 16:1, 17:1 shows 20%, 18% higher CO than UTO operation at standard compression ratio, whereas 18.5:1 CRs shows20% less CO emission at maximum brake power. The decrease in the CO emission may be due to better combustion and oxygen enrichment of the fuel [11].

4.2.2 Carbon di-oxide (CO2) emission

More amount of CO2 indicates the complete combustion of fuel in the combustion chamber and it is also related to the exhaust gas temperature. The excess amount of CO2 in the atmosphere leads to global warming and environmental problems. These emitted CO2 are absorbed by the plants to maintain constant percentage in atmosphere. CO2 is always less in the UTO fuel compared with diesel as shown in Fig. 4.12.

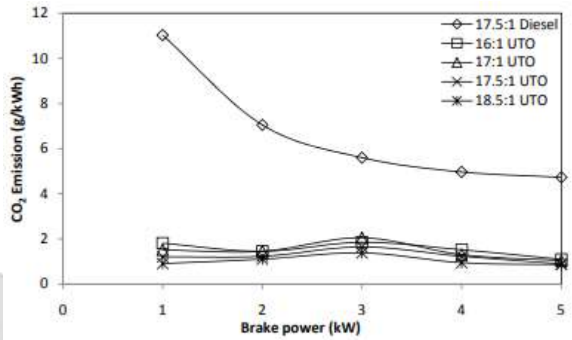

Fig. 4.12 Variation in CO_2 emission with respect to brake power

Compared to standard CR of the UTO, CRs 16:1 and 17:1 shows 2.6% and 1.3% higher whereas 18.5:1 shows lesser CO2 emission by 1.3% at full load. At CR 18.5 the CO2 emission is lesser due to incomplete combustion and lack of oxygen.

4.2.3 Hydrocarbon (HC) emission

The reasons for the HC emission in a CI engine are wall deposit absorption, oil film absorption, crevice volume, incomplete combustion etc. During the combustion process HC particles condenses onto the surface of solid carbon soot. Most of this burned and only a small percentage of carbon soot comes out from the cylinder which contributes to the HC emission of the engine [3]. Fig. 4.13 portrays the percentage variation of the HC emission with brake power for UTO compared to diesel at different compression ratio. The HC emission in UTO is much higher than that of diesel for all CRs. It can be observed from the figure that the HC emission is higher by about 36.6% and 12.7 % for CRs of 16:1 and 17:1 compared to standard CR of UTO while the CR of 18.5:1 shows 30.6% lower at maximum brake power due to delays in ignition, which results to insufficient heat of compression so HC emissions decreases [56]. Fuel viscosity and spray quality is also responsible for increase in HC with UTO.

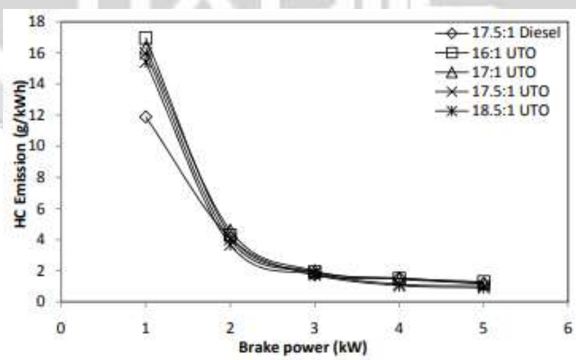

Fig. 4.13 Variation in HC emission with brake power

4.2.4 Nitrogen oxide (NO) emission

An engine can have up to 2000 ppm of oxides of nitrogen in the exhaust gas. With higher compression ratio, the cylinder pressure and high temperature contribute to dissociate diatomic N into monatomic N, thus resulting in more NO formation. The reduction in the NO emission is the prime objective of the engine researcher.

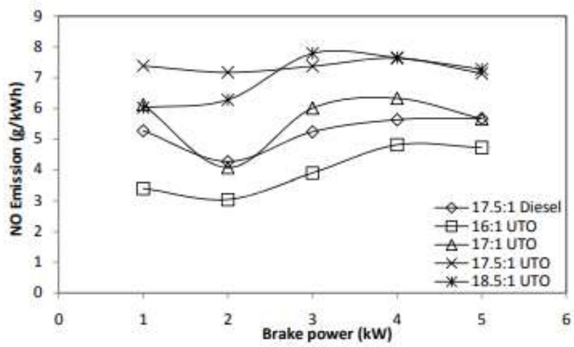

Fig. 4.14 Variation in NO emission with brake power

Figure 4.14 portrays the percentage variation of the NO emission with brake power for UTO compared to diesel at different compression ratio. The NO emission in a CI engine strongly depends on the combustion temperature and oxygen availability. The NO emission for the UTO with CRs of 16:1 shows lower by 6.2% whereas 17:1, 17.5:1 and 18.5:1 are found to be higher by 12.3%, 27.5% and 30.6% to that of diesel at maximum brake power. In comparison with standard CR of UTO, when lower the CR to 16:1 and 17:1 the NO emission is found to be lower by 33.7%. and 15.2% and with higher CR of 18.5:1 higher by 3.1% at maximum brake power. With higher CR, the NO emission for the UTO is increased due to high in-cylinder (peak) temperature [9].

4.2.5 Smoke opacity

Smoke is higher when a fuel's ratio of hydrogen to carbon is less than two [18]. Fig. 4.15 portrays the percentage variation of the smoke opacity with brake power for the UTO compared to diesel at different compression ratios.

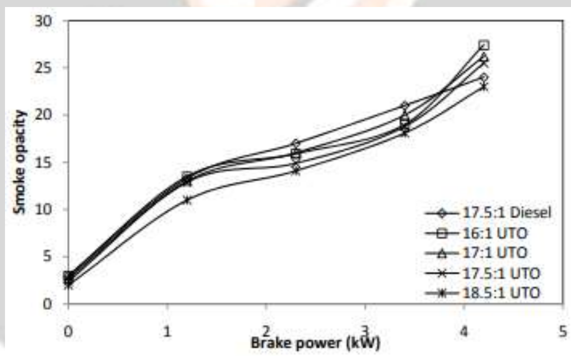

Fig. 4.15 Variation in smoke opacity with brake power

The hydrogen to carbon ratio of the UTO is lower by about 49% than that of diesel. Among all the compression ratios, UTO with CR 18.5 exhibits the lowest smoke opacity by 4.1% compared to diesel. At maximum brake power, the CR of 16:1 and 17.1 UTO shows 14.1% and 9.2% increase while higher CR 18.5:1 shows 8.6% less smoke opacity compared to UTO at standard compression ratio. This may be due to the maximum temperature during the combustion increases and this, in turn, decreases smoke opacity [19].

Phase II: Performance and emission parameters for DI diesel engine operated at optimum injection timing of 20° bTDC and optimum nozzle opening pressure of 230 bar.

4.3 Performance parameters

4.3.1 Brake thermal efficiency

Generally, increase in compression ratio improves the efficiency of the engine because of reduced ignition delay. Fig. 4.23 illustrates the variation of the brake thermal efficiency with brake power for the UTO with different compression ratio and diesel.

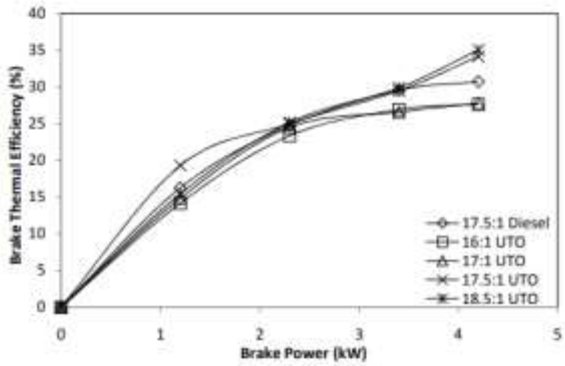

Fig. 4.23 Variation of the brake thermal efficiency with brake power

The values of brake thermal efficiency for diesel and the UTO are 28.6 and 34.1% respectively with standard compression ratio at maximum brake power. The brake thermal efficiency of UTO is 27.6, 27.8 and 35% at compression ratio of 16:1, 17:1 and 18.5:1 respectively. This can be attributed to the better combustion and better intermixing of the fuel and air. The increase in the brake thermal efficiency also seen with increase in load due to lesser losses, due to low calorific value of the UTO. As the fact, the UTO is volatile than the diesel but at higher CR of 18.5 shows relatively more improvement because of higher temperature.

4.3.2 Brake specific energy consumption

An important parameter to measure the engine performance is the specific energy consumption. It is simply described as the product of brake specific fuel consumption (BSFC) and lower heating value (LHV). Fig. 4.24 shows the variation between brake specific fuel consumption and brake power. As the fact, the BSEC decreases with increase in engine load. Compare to diesel engine at standard situation, UTO shows 18.5%, 11.9% and 0.5% higher consumption at CR`s 16:1, 17:1 and 18.5:1 while CR of 18.5:1 consumes 6.7% lower than diesel. The higher CR of 18.5 shows 7.2% lesser BSEC than the standard CR of 17.5 with the UTO. The increase in the fuel consumption is due to fuel density, viscosity and heating value, but with higher compression ratio lesser value of BSEC is apparently desirable because of better atomization which is associated with a marginal delay in admission due to high needle lift pressure during same period, hence less fuel goes inside the combustion chamber [17].

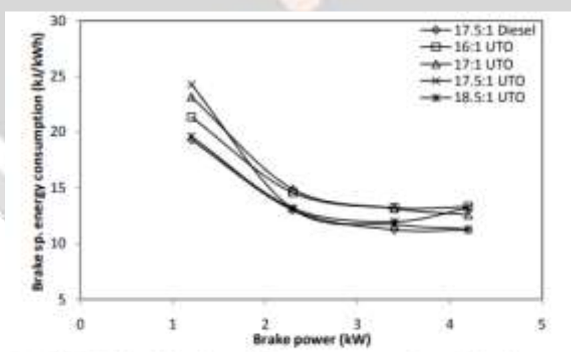

Fig. 4.24 Variation in brake specific energy consumption with brake power

4.3.3 Exhaust gas temperature

Figure 4.25 portrays the variation of the exhaust gas temperature with brake power for diesel and the UTO.

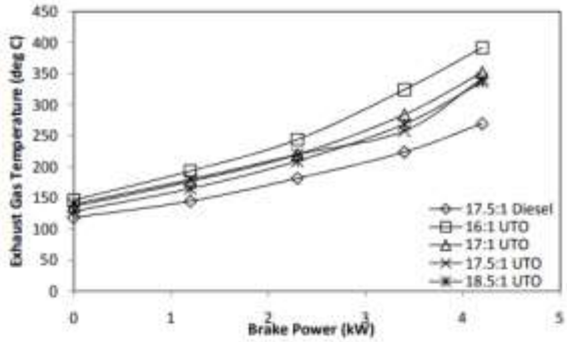

Fig. 4.25 Variation of the exhaust gas temperature with brake power

The exhaust gas temperature increases with an increase in the engine load for the UTO and diesel as expected [18]. It is higher for the UTO than that of diesel in the entire operation as a result of higher viscosity and density of the UTO. The exhaust gas temperature of diesel and the UTO with the CR 16, 17, 17.5 and 18.5:1 are 269, 392, 352, 342 and 336oC respectively. The reduction in the exhaust gas temperature at increased CR may be due to improved energy conversion and lower heating value of the UTO as compared to that of diesel [19]. Another possible reason for the reduction in exhaust gas temperature at higher CR is increase in air temperature inside the cylinder which reduces the ignition lag due to better and more complete burning of the fuel.

4.4 Emission Parameters

4.4.1 Carbon monoxide (CO) emission

Carbon monoxide is an odorless and colorless gas but poisonous in nature. When engine is operated with the rich fuel-air equivalence ratio, it is generated. The CO emission is caused due to less availability of air, poor mixing of air with fuel and rising temperature in the combustion chamber. The variation of the carbon monoxide emission for the UTO and diesel for different engine loads is shown in Fig. 4.26. It shows that the reduction of CO emission by about 75% with the UTO at CR 18.5 than that of the diesel at 17.5:1 CR. It can be observed from the figure that the CO emission is higher for the UTO at 17.5:1 CR compared to that of diesel at full load.

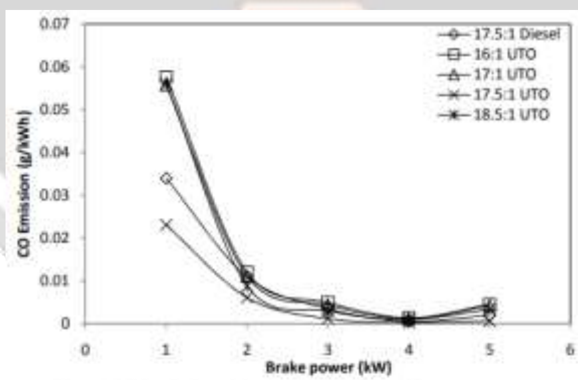

Fig. 4.26 Variation of carbon monoxide with brake power

Higher compression ratio shows a lower CO emission due to better combustion and oxygen enrichment of the fuel [11]. The heat generated is more inside the cylinder due to higher compression. As a result, evaporation rate is increased and results in better fuel air mixing. Thus, the CO emission reduced with increased CR.

4.4.2 Carbon dioxide (CO2) emission

More amount of CO2 indicates the complete combustion of fuel in the combustion chamber and also relates to the exhaust gas temperature. The excess amount of CO2 in the atmosphere leads to global warming and environmental problems. These emitted CO2 are absorbed by the plants to maintain constant percentage in atmosphere. CO2 is always less in UTO fuel compared with diesel as shown in Fig. 4.27.

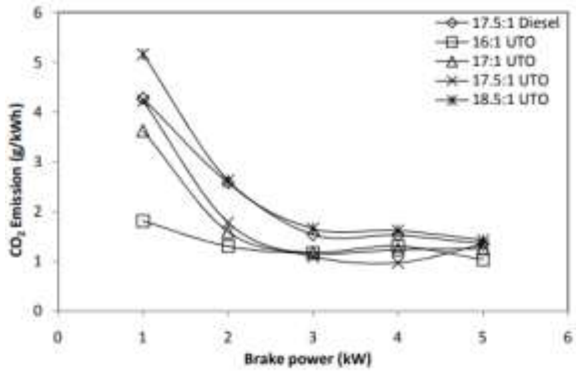

Fig. 4.27 Variation in CO_2 emission with respect to brake power

Compared to standard CR 17.5 of UTO, CRs 16:1 and 17:1 shows 2.6% and 1.3% higher whereas 18.5:1 shows lesser CO2 emission by 1.3% at full load. At CR 18.5 the CO2 emission is lesser due incomplete combustion and insufficient supply of oxygen.

4.4.3 Hydrocarbon (HC) emission

Hydrocarbon emission is caused by the longer ignition delay and accumulation of fuel in the combustion chamber. The variation of the unburned hydrocarbon emission for UTO and diesel for different brake power is shown in Fig. 4.28. It can be observed from the figure that the HC emission is higher by about 11.7% for compression ratio of 16:1 compared to that of diesel at maximum brake power, whereas for 17:1, 17.5:1 and 18.5:1 shows 22, 74 and 54% lower HC emission. As the compression ratio increases it shows a lower HC emission, and lower CR shows higher HC emission due to longer ignition delay. More fuel is accumulated in the delay period and as a result higher HC emission is formed with lower CR than standard CR for UTO [13].

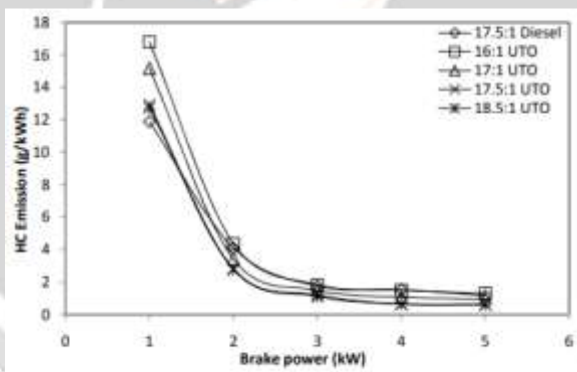

Fig. 4.28 Variation of unburnt hydrocarbon with brake power

4.4.4 Nitric oxide (NO) emission

An engine can have up to 2000 ppm of oxides of nitrogen in the exhaust gas [3]. The variation of the NO emission for UTO with different CR and diesel for different engine loads is shown in Fig. 4.29.

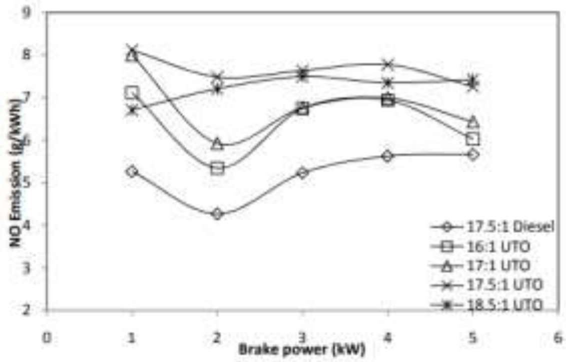

Fig. 4.29 Variation of the NO emission with brake power

The NO emission in a CI engine strongly depends on the combustion temperature and the oxygen availability. At lower compression ratio nitrogen exists as diatomic molecule.

With higher CR there is reduction in ignition delay the cylinder temperature will be more hence the NO emission will be more. The NO emission for diesel and UTO at CR 17.5:1 is 5.6 and 7.2 g/kWh respectively at maximum brake power. The NO emissions for UTO at CR 16, 17 and 18.5:1 are 6, 6.4 and 7.4 g/kWh at maximum brake power. Higher CR shows higher NO emission than that of diesel at maximum brake power. With the higher compression ratio, the NO emission for UTO is increased due to increase in-cylinder temperature [9].

4.4.5 Smoke opacity

Smoke is higher when a fuel's ratio of hydrogen to carbon is less than two. Fig. 4.30 compares the smoke opacity of UTO and diesel at different brake power. It can be observed that the smoke opacity increases with an increase in the brake power as expected, but the smoke opacity of UTO is 7% lesser than that of diesel at maximum brake power. Lower CR of 16:1 shows a higher smoke opacity of 5%, whereas for CR 17:1 and 18.5:1 decreases by 4% and 10% than that of diesel at full load. This may be due to the maximum temperature during the combustion increases and this, in turn, decreases the smoke opacity [19].

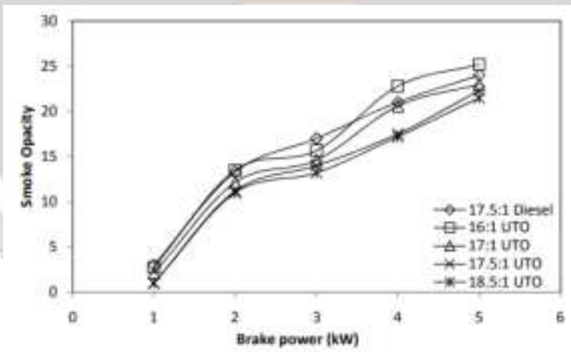

Fig. 4.30 Variation of smoke density with brake power

5. CONCLUSION

The combustion, performance and emission characteristics of a single cylinder, four stroke, air cooled, direct injection diesel engine having a power output of 4.4 kW at a constant speed of 1500 rpm, fueled with UTO, diesel blends and diesel have been analyzed and compared with those of diesel. The following are conclusions;

- The UTO can be used as a fuel in the CI engines as it possesses a heating value. Considering the specific energy consumption, UTO with CR18.5 can be the optimum CR tested.
- The ignition delay for the UTO is shorter by about 1-3°CA compared to that of diesel in the entire range of operation.

- The HC and CO emissions for the all CR of UTO are marginally higher than those of diesel operation at full load.
- The NO emission is higher at optimum CR 18.5 for UTO fuel than diesel at full load.
- Smoke is lower with the UTO than diesel at full load. The smoke value of UTO is lower at CR 18.5 than that of diesel at full load.

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