

Experimental Investigation to Study the Influence of Fuel Additive with Pre-Heated Straight Vegetable Oil (SVO) by Comparing the Injection, Combustion and Emission Characteristics of Diesel Engine Based on IR Diagram

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ABSTRACT

The use of locally available non-edible type SVO (Straight Vegetable Oil) in existing diesel engine without any modification becomes an attractive option in rural and remote areas in decentralized power generation for irrigation and electrification, where grid power is not available, if performance is comparable to diesel fuel. The purpose of the present work is to determine how the commercially available additive influences the injection, combustion and emission characteristics of CI (Compression Ignition) engine fuelled with SVO at elevated temperature. In the present paper attention is concentrated to elucidate how IR (Injection Rate) diagram focuses on the injection characteristics which in turn affects the combustion and emission characteristics of DI (Direct Injection) diesel engine. This paper shows a conventional but novel technique to plot IR diagram by using modified fuel test bench equipped with mechanical stroboscope, which does not require costly pressure sensor in injection system, to measure flow rate and thus it is a cost effective method. A comparative study using diesel fuel and Karanj oil with and without additive at room temperature and at 65°C respectively was carried out on a Cummins two cylinders DI diesel engine by varying brake load. A detailed structural and chemical analysis of additive was done to understand the mechanism of fuel adulteration and interpret the test results. Physical properties of test fuels were also measured at different temperature to justify the engine characteristics. The experimentally obtained results reveal that Karanj oil exhibits advance injection timing, longer injection duration and higher injection rate as compared to diesel at room temperature which in turn increases ignition delay and thus leads to higher NO_x (Oxides of Nitrogen) and lower smoke emissions. However, pre-heating retards injection timing which in turn reduces ignition delay, cylinder pressure and HRR (Heat Release Rate) and thus suppression of NO_x emission. The use of additive at 65°C further improves the injection characteristics and thus simultaneous reduction in NO_x and smoke emissions for Karanj oil is achieved.

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INTRODUCTION

The use of fuel additives for a petroleum fuels is not new, for a long time industries have been using various types of multifunctional chemical additives worldwide. A large variety of fuel additives (metal based, oxygenates, wax dispersant, surfactant, antiknock agents, etc.) are added in small concentrations to diesel fuel to improve the engine efficiency and to reduce emissions [1]. Metal based additives have been reported effective in improving physical properties with biodiesel [2]. These types of additives reduces ignition delay and combustion duration as well as reduces smoke, CO (Carbon Monoxides) and HC (Hydro Carbons) emissions with diesel [3, 4, 5]. Although a wide variety of diesel fuel additives have been developed there has not been developed special additive for straight vegetable oil yet.

The literature on alternative fuels demonstrates that the diesel engine can be made to run on virtually any vegetable oil and trend shows in utilizing non-edible vegetable oil worldwide. The non edible oil bearing species like Neem, Jatropha, Karanj, Mahua, Rubber seed, Mustard seed etc., have a great potential for commercial exploitation and utilize them as fuels in India [6]. Though short term performance of SVO is comparable with diesel, its long term trial shows poor engine performance and leads to power loss, fuel loss, clogging of fuel lines, fuel filters and injectors, fuel dilution, etc. due to its higher viscosity [7]. The higher viscosity and low volatility of straight vegetable oil leads to larger droplet size and thus emits higher exhaust emissions. So it is necessary to reduce droplet size / Sauter Mean Diameter (SMD) [8] and preheating is the easiest way to reduce SMD via viscosity without any engine modification.

Hence utilization of pre-heated SVO as an alternate fuel is a better and attractive option because pre-heating reduces viscosity, density and increases smooth injection into the engine cylinder [9]. Pre-heating of inlet fuel improves the fuel injection rate, reduces the ignition delay, combustion duration and carbon deposit problem which in turn lowers NO_x and smoke emissions whereas increases CO and HC emissions [10-11].

The authors also made a comparative study of the use of fuel additives in straight vegetable oil (SVO) and pre-heated SVO on combustion and emission characteristics of CI engine earlier and showed that there was drastic reduction in injector deposit as well as decreases in cylinder pressure, HRR, NO_x and smoke with additives at pre-heated SVO [12]. The results of this work highlights on simultaneous reductions of smoke and NO_x emissions in CI engine without any modifications.

The use of vegetable oil in unmodified diesel engine greatly depends on the engine sensitivity to fuel injection characteristics and on the combustion characteristics. To achieve an efficient combustion, a timely mixing of fuel and air in the engine cylinder is essential which in turn depends heavily on the fuel injection process. It is possible to predict to some extent the engine characteristics on the basis of injection characteristics (i.e. injection pressure, injection duration, injection timing, fuelling and injection rate history) [13]. This fuel injection characteristic in turn depends on the type of injection system and fuel properties. It is well known that besides the geometry of the injection system, the physical properties of fuel like density, viscosity and surface tension also influences the injection characteristics and atomization characteristics related to ignition quality in CI engine [14-15].

Fuel injection equipment operates on a volume metering system, hence a higher density of vegetable oil results in the delivery of a slightly greater mass of fuel. The density is directly proportional to the bulk modulus [16]. Differences in the physical properties of the fuels can change the fuel injection timing. The properties which have great influence on fuel injection timing are the density, speed of sound, the isentropic bulk modulus and the viscosity [17, 18, 19, 20].

High viscosity can cause early fuel injection due to high line pressure. Szybist et al. [21] attributed to advanced injection timing of biodiesel due to its higher viscosity. However, no mechanism was offered on how higher viscosity advances fuel injection timing. It is known that higher density and bulk modulus of fuel causes advance in injection timing in a mechanically controlled in-line injection system.

Bulk modulus is an important parameter used in fuel injection system modeling, analysis and design. For a given fuel injection system, fuels with higher bulk modulus have a more rapid traversal of the pressure wave from the fuel pump to the injector needle valve and an earlier needle lift causes an advanced or early injection [22]. When the biodiesel is injected, the pressure wave propagates faster and would arrive at the needle valve earlier because of the higher bulk modulus of biodiesel at the same pump speed and rack position. This

reason has been widely used to explain the higher temperature peaks and greater rates of NO_x formation with biodiesel [13,15,17,18,23,24,25].

The effect of viscosity on fuel injection timing relative to the effect of bulk modulus is currently unknown. However, the engine controller itself may shift injection timing due to change in throttle position to meet the required load because of differences in the physical properties of test fuels which may change the engine's combustion delay, thus altering the combustion timing and combustion rate which in turn causes exhaust emissions different from the optimized settings chosen by the engine manufacturer [26]. The NO_x and PM (Particulate Matter) emissions can be reduced by the exact injection rate control according to engine conditions by varying injection pattern and injection timing [27]. The variation in speed of pressure wave propagation changes the start of actual fuel injection and thus increases or decreases injection delay. The injection duration of neat biodiesel is always higher than that of diesel fuel [13, 28].

Rate of fuel injection depends on the density of fuel. Higher injection rate leads to more injection pressure which in turn gives good spray formation and better combustion results in reduction of CO and HC emissions [29]. A higher injection rate shows shorter combustion duration and improved cycle efficiency [30]. Injection timing and injection duration are directly related to fuelling. The higher fuelling of biodiesel is a consequence of its higher bulk modulus and kinematic viscosity [13,17]. There is little publication on the effects of fuel injection rate on combustion and emissions characteristics.

The effects of fuel inlet temperature on the engine in-cylinder combustion are mainly through its fuel injection characteristics. The fuel temperature at the pump affects the timing of fuel pressure wave and finally on the atomization characteristics. The fuel injection delays due to decrease in bulk modulus with increase in fuel temperature. The pre-heated fuel leads to reduction in ignition delay because of faster evaporation of fuel at higher temperature [24]. Higher fuel inlet temperature minimizes the resistance to flow results in improved spray and increases smooth injection into the engine cylinder because of reduced unsaturated bond, viscosity and density [31].

The bulk modulus and density reduces with increase in temperature which in turn reduces the volumetric injection rate and decreases in the fuel density causes the volume of an unchanged amount of fuel injection to increase, both of these prolong the fuel-injection duration [32]. The rise in fuel temperature decreases the fuel viscosity which in turn increases the coefficient of discharge [33]. However, an opposite trend is observed by Chen G. et. al. [34] that the fuel injection rate decreases with increase in fuel temperature due to decrease in injection pressure because of reduction in bulk modulus of elasticity with rise in fuel temperature. It is reasonable to assume that the effect of viscosity on flow rate due to change in fuel temperature is less significant as compared to bulk modulus and density of fuel [35].

The IR (Injection Rate) diagram gives a bird's eye view of an overall injection process because the rate at which the fuel is injected into the combustion chamber determines the performance of the engine. So

far, many fuel injection rate measuring methods have been developed among which Zeuch's and Bosch's are most frequently used today. A Bosch rate of injection meter is used to determine the instantaneous rate of injection by Nehmer et. al. [36] in which they measure the pressure wave to meter the injection rate. The deformational measuring method for measuring the injection rate is based on the calculation of the fuel velocity by the pressure-velocity equation. With Zeuch's measuring method the fuel is injected into a chamber filled with fuel at a certain pressure and through the bulk modulus of elasticity of the fuel, the derivative of the chamber pressure with respect to volume provides the injection rate signal [37].

Thus, it requires costly pressure transducer and other measuring devices in all such methods. In our work, the effort is put to measure instantaneous injection rate per degree of cam angle without measurement of pressure by modifying existing fuel test bench with simple mechanical stroboscope.

The objectives of present research work are as follows:

1. To focus on novel method to plot the IR diagram using modified fuel test bench to elucidate the injection characteristics and predict the combustion and emission characteristics based on IR diagram.
2. To evaluate the influence of fuel additive with Karanj oil at different temperature by conducting experiments on diesel engine and correlate the results of combustion and emission characteristics with predicted one based on IR diagram.
3. To examine the influence of fuel additive on physical properties of fuels with change in temperature.

EXPERIMENTAL SET-UP AND PROCEDURE

The experimental work has been divided into three parts as Part-A, Part-B and Part-C. In the first phase of Part-A, the experiment was carried out on the fuel injection test bench to plot the IR diagram for test fuels. The second phase of Part-B describes in detail about the performance test carried out on multi cylinder diesel engine at constant speed by varying brake load for test fuels and third phase of Part-C describes the instrumentation and characterization of test fuels at different temperature along with structural and chemical analysis of additive.

Part-A. Experimental Set-Up for Fuel Test Bench to Plot IR Diagram

The standard MICO Bosch fuel injection test bench modified with a mechanical stroboscope was used for the determination of instantaneous and integral cyclic injection rates with respect to crank angle. The detail specifications of test bench with injector nozzle are given in [Table 1](#). The speed governor was removed from the injection pump so that fuel control was accomplished by the direct action of the rack, which rotates the pump plunger. It should be noted that the same pump as used in diesel engine under performance test was used on fuel test bench to plot IR diagram whose detail specifications are given in the [Table 2](#). It should also bear in mind that the load was

determined by the rack position and the speed, NOP (Needle Opening Pressure) as well as rack position (set at full load) was kept constant throughout the set of experiments for all test fuels.

Table 1. Specifications of Fuel Test Bench with Injector

Make	MICO BOSCH
Type	03C EFEP 400A
Driving Motor	3 kW 3 Phase 415V 50 cycles 2800 rpm
Transmission Unit	Variable hydraulic transmission in both directions of rotation
Electronic stroke / rpm counter	100 to 1200 strokes in steps of 100 strokes
Speed	0 to 3000 rpm
Temperature Controller Set	Up to 70° C
Nozzle Make	MICO BOSCH
Nozzle Type	Single hole
Orifice diameter	0.46 mm
Part Number	HKBL97S52

Table 2. Specifications of Injection System of Diesel Engine

Fuel Pump	
Make	MICO BOSCH
Model	F 002 A0Z 394
Part Number	PES2A90D320LS2000
Governor	RSV400-750A7C1378R
Type	In-Line pump, A Type
No. of Plunger	02
Plunger Diameter	9 mm
Plunger stroke	8 mm
Injection Timing	11° BTDC
Injector Nozzle	
Make	MICO BOSCH
Part Number	F002C70018 011
Number of holes	05
Hole diameter	0.215 mm
Opening Pressure	210 bar

Construction and Working of Mechanical Stroboscope

The IR diagram can be plotted only with the help of mechanical stroboscope. The standard MICO fuel test bench gives only integral delivery measurements. However, the mechanical stroboscope designed and fabricated at this institute [38] makes it possible to measure instantaneous injection rate per degree crank angle per cycle. i.e. mg/cycle/°CA. This mechanical stroboscope consists of a drum type disc freely rotating on a shaft called as strobo disc enclosed in a casing, Index plate and fix plate as shown in [Figure 1](#) and [Figure 2](#). It is the simplest mechanism in which a rotating disc with evenly spaced holes (Index plate) match in the line with a hole of stationary disc (fixed plate) to make a cyclically moving object (strobo disc) with the same speed. Generally, stroboscope is meant to be an optical instrument used for determining cyclic speed, which would not allow performing gravimetric measurements though in this study the word "Mechanical Stroboscope" is used because it works on the principle of studying the rotating object.

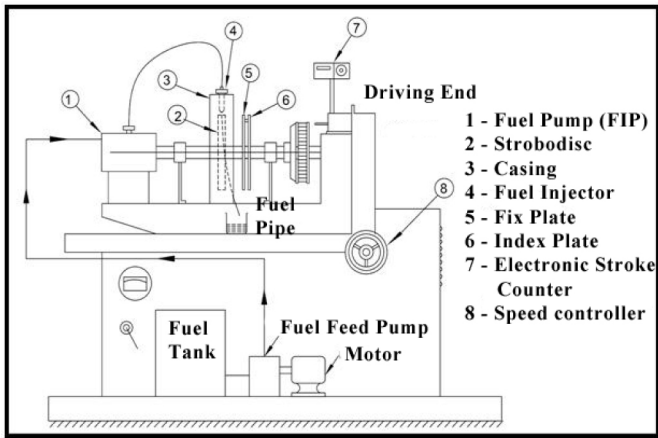


Figure 1. Schematic diagram of fuel test bench set-up

This strobo-disc is coupled to the test bench shaft on one side and the shaft of the test pump on the other side. There is a provision of an axial slit of one degree angular width on the periphery of strobo-disc. This disc is kept enclosed in a casing on which the test injector is so mounted that the spray is directed radially to the slit as shown in Figure 2. The distance between the injector nozzle tip and the slit is kept as 0.5 mm to ensure entire quantity of fuel injects through this axial slit when the slit traverses passed the injector spray during this angular width of one degree. A funnel is provided on the underneath of the slit to receive the fuel passing through the slit. Fix plate having 6 holes of 4mm diameter each spaced equally 60° apart and an indexing plate having 360 holes of 4 mm diameter each spaced equally 1° apart angularly. This index plate rotates freely on the shaft along with strobo-disc and the angular position of the slit can be changed with respect to drive shaft on the bench by locking the index plate with a fixed plate fitted on the drive shaft as shown in Figure 2.

Preliminary Experiments

The same pump as used on diesel engine under load test was used on fuel test bench for preliminary experiments and to plot IR diagram. There are three basic requirements to plot IR diagram namely (i) To use nozzle having single axial hole (ii) To set rack position and (iii) To find delivery period, which can be find by conducting preliminary experiments.

Due to the constructional and design feature of the stroboscope, it was essential to replace the multi hole nozzle of diesel engine by a single axial hole nozzle. To select the appropriate size of nozzle, all single axial hole nozzles available at author's laboratory were tested for integral delivery on fuel test bench and examined to match with fuel consumption of 5-holes nozzle of an engine under test at full load at rated speed conditions. In this way, a single hole nozzle of 0.46 mm orifice diameter having an integral fuel delivery nearest to that of engine's 5-holes nozzle was selected for test.

The fuel consumption rate of two cylinder diesel engine under load test was used as the reference rates for fuel integral delivery and to set rack position on fuel test bench. The rack position can be set by

taking data of fuel consumption of the diesel engine under load test for same quantity of fuel injected by adjusting the micrometer screw gauge (least count 0.01mm) attached to movable jaw which is in contact with the rack as shown in Figure 2. Thus by rotating the screw gauge different rack positions can be set at different load which can be read on the scale. Higher reading on the micrometer indicate higher load. In our study the rack was set at full load condition for all test fuels.

The geometric start of injection can be identified by knowing the delivery period. The delivery period can be determined by spill method in terms of cam angle traversed at set rack position. It was essential to lock the strobe-disc with the angle of the flywheel to determine delivery period. The delivery valve of the fuel pump was replaced with the 'swan neck tube' and the camshaft was rotated slowly till fuel from the swan neck tube was about to stop (1 drop per minute) indicated geometric start of the injection. The angle of geometric start was noted from the angle disc on the flywheel of the test bench as shown in the Figure 2. The camshaft was further rotated till the fuel flow started once again indicated end of delivery period. The difference between two angles denotes the delivery period for that particular rack position.

Procedure to Plot IR Diagram

The mechanical stroboscope was coupled to the main drive shaft of the test bench as seen earlier. First of all, the angle on the flywheel disc adjusted to the start of the geometric delivery angle, which was obtained earlier in preliminary experiment by spill method, was considered as zero degree crank angle in IR diagram. When the axial slit was traversing passed the injector spray, fuel from the injector was injected during angular width of one degree and checked whether the fuel comes out through collecting pipe or not. The angular position of the strobe-disc was then changed with respect to the angle disc on the flywheel by 1 degree through indexing plate and checked again whether the fuel delivered or not till start of fuel delivery obtained. The corresponding angle at the start of fuel delivery was the actual start of injection. The slit passes under the injector tip once in every revolution of strobo-disc allowed the fuel to flow through angular width of one degree which was collected in a beaker through collecting pipe. The fuel injected was collected in beaker for a fixed number of strokes (here 500 strokes) and weighed on precision weigh balance scale having an accuracy of 0.1 mg. Weight of such collected fuel was the rate of fuel injection per cycle corresponding to that angle called as the instantaneous rate of injection. Position of axial slit was varied in relation to the drive shaft in step of one degree by indexing plate and the injection rate data was obtained for different cam angle positions. The angle was further advanced by 1 degree and the process was continued until the end of fuel delivery. Graphical representation of such instantaneous injection rate as a function of cam angle is an Injection Rate (IR) diagram. The area under this plot is the total amount of fuel injected per cycle and is the integral cyclic delivery for a given load. To check for secondary injection, the same procedure was continued further for about 4 to 6 degree cam angle after end of earlier injection. Same procedure was followed for all test fuels and repeatability tests were performed to confirm the reliability of instrument.

Part-B. Experimental Set-Up for Engine Test

The fully automated computerized test bed, twin cylinder, four-stroke, water cooled, Cummins DXP 15-5 series DI Diesel Genset was used for test trial whose detail specifications are given in the [Table 3](#).

Table 3. Engine specifications

Engine Make	Cummins
Model	X 1.7 G1
BHP / kW	20 / 15
No. of cylinders	02
RPM	1500
Bore (mm)	91.44
Stroke (mm)	127
Compression ratio	18.5:1
Displacement (liter)	1.7
Governor	Mechanical

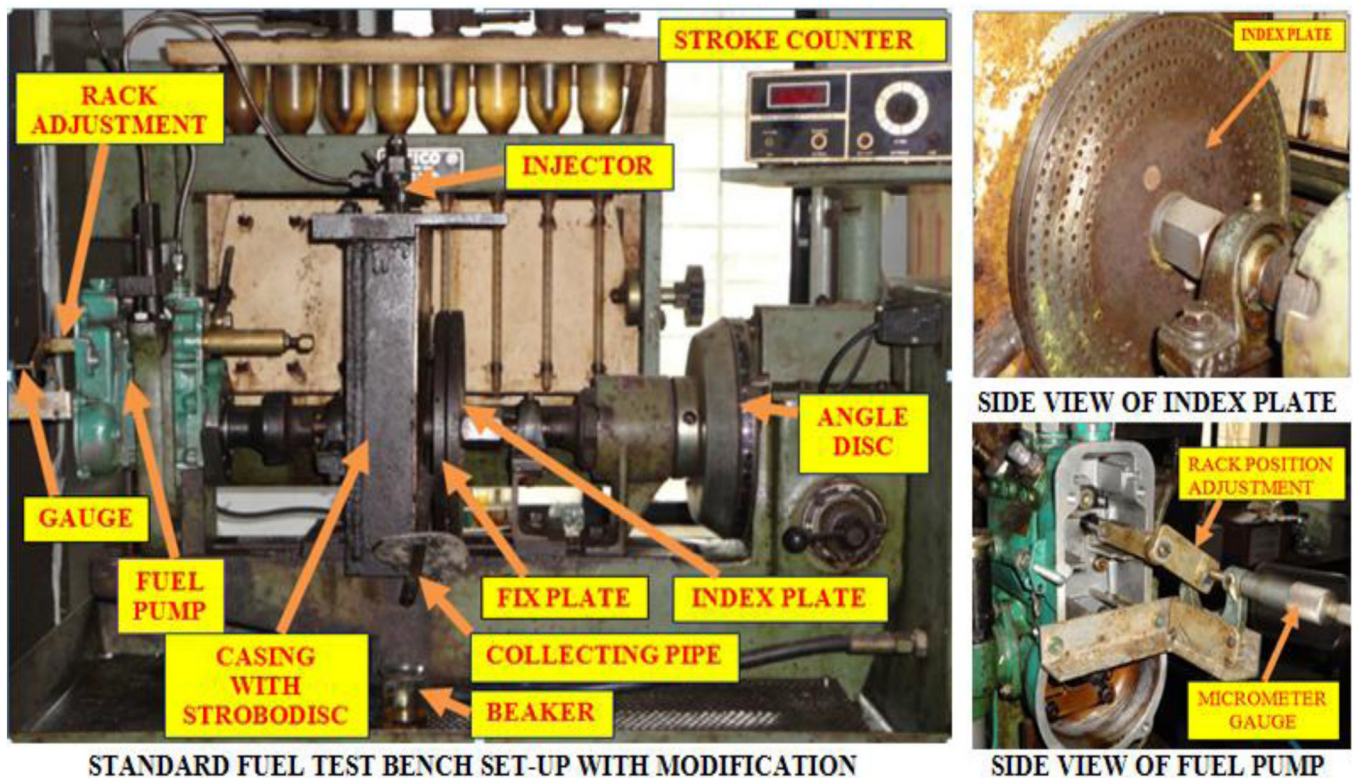


Figure 2. Standard Fuel Test Bench with modification

The schematic diagram of experimental set-up is shown in the [Figure 3](#). The engine was coupled to an alternator to provide the load through connected electric heater load bank. The in cylinder pressure was measured by a water-cooled piezo-electric pressure transducer (Kistler 7061B) and the charge amplifier (Kistler 504E4). The crankshaft angle was obtained from a crank angle encoder mounted on the crankshaft having resolution of 0.5 degree CA. These signals were recorded by a data acquisition system through RS view 32 software cell 610 developed by Cummins Engine Research Centre (CERC), Pune, India. For each operating condition, 100 engine pressure cycles were recorded and the mean cylinder pressure trace was considered and the heat release rate was calculated based on it as

described by Heywood [39] for combustion analysis. The Gas analyzer was used to measure exhaust of CO and NO_x emissions. The smoke opacity was measured by Bosch Smoke unit in BSN (Bosch Smoke Number). The intensity of smoke collected from sampling on filter paper was matched with the standard chart rated in the scale 0-10. The information on uncertainties associated with measuring instruments is shown in [Table A-1](#) in the [Appendix](#). In addition to exhaust emission, the exhaust gas temperature, engine coolant and oil temperatures were also recorded at each test. The system was equipped with PID controller heater for heating and chiller plant with temperature controller for cooling to maintain the constant temperature as shown in the [Figure 3](#).