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Effect of injection timing on combustion and performance of a direct injection diesel engine running on Jatropha methyl ester

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Abstract
The present study aims at evaluation of effect of injection timing on the combustion, performance and emissions of a small power diesel engine, commonly used for agriculture purpose, running on pure biodiesel, prepared from Jatropha (Jatropha curcas) vegetable oil. The effect of varying injection timing was evaluated in terms of thermal efficiency, specific fuel consumption, power and mean effective pressure, exhaust temperature, cylinder pressure, rate of pressure rise and the heat release rate. It was found that retarding the injection timing by 3 degrees enhances the thermal efficiency by about 8 percent.

Keywords: Injection timing, Jatropha methyl ester, Heat release rate, Thermal efficiency.

1. Introduction
For diesel engines, a significant research effort has been directed towards using vegetable oils and their derivatives as fuels. Biodiesel is considered a promising alternative fuel for use in diesel engines, boilers and other combustion equipment.
Although biodiesel has many advantages over diesel fuel, there are several problems that need to be addressed such as its lower calorific value, higher flash point, higher viscosity, poor cold flow properties, poor oxidative stability and sometimes its comparatively higher emission of nitrogen oxides [1]. Biodiesel obtained from some feed stocks might produce slightly more oxides of nitrogen (1-6 %), which is an ozone depressor, than that of fossil origin fuels but can be managed with the utilization of blended fuel of biodiesel and high speed diesel fuel [2]. It is found that the lower concentrations of biodiesel blends improve the thermal efficiency. Reduction in emission and brake specific fuel consumption is also observed while using 10% biodiesel blend (B10) [3].
Since the introduction of petroleum fuels, the development of compression ignition (CI) engines has been done keeping the properties of 'mineral diesel' fuel in front. The present designs and operating parameters of available engines are standardised for this fuel only. For all other fuels, the operating parameters must be re-set in the light of the specific fuel properties. Effect of injection parameters: multiple injection [4], injection system [5], injection timing and compression ratio [6], injection pressure and compression ratio [7], have been studied with various engines and oils. Most of the research studies concluded that in the existing design of engine and parameters at which engines are operating, a 20% blend of biodiesel with diesel works well [3], but they indicated the need of research in the areas of engine modifications so as to

suit to higher blends without severe drop in performance so that the renewability advantages alongwith emission reduction can be harnessed to a greater extent.

It is commonly accepted that there is some advancement of injection time when biodiesel is used in place of diesel because of its bulk density. The higher bulk density and viscosity transfers the pressure wave through fuel pipe lines faster and an earlier needle lift will lead to advanced injection. Due to the difference in cetane number, it is often suggested that injection timing be retarded to attain more complete combustion of vegetable oil based fuels [8]. Late injection of fuel into the combustion chamber helps in reducing the NOx emission of a diesel engine [9].

Biodiesel made from different feed stocks have been tried by many and the effect of feedstock on engine performance and emissions are well documented. One of the major feedstock researched in India is 'Jatropha curcas'. Looking to the availability and its biodiesel potential, this oil is becoming more and more popular in many other countries as well. Evaluation of Jatropha esters [10] indicates its superiority over many other vegetable oils in terms of engine performance, emissions, ease of use and availability.

Jatropha curcas, locally known as ratanjyot, belongs to the family of Euphorbiaceae. It is a quick yielding plant that survives in degraded, barren, forest land and draught prone areas and is cultivated as hedge on the farm boundaries (Figure 1). The de-oiled cake is excellent organic manure which retains soil moisture. This oil is gaining popularity due to its good properties and has been accepted and recommended by National Biodiesel Board of India [11] as a source of alternative fuel for blending in the commercial diesel. The potential of Jatropha oil as a source of fuel for the biodiesel industry is well recognized [12].

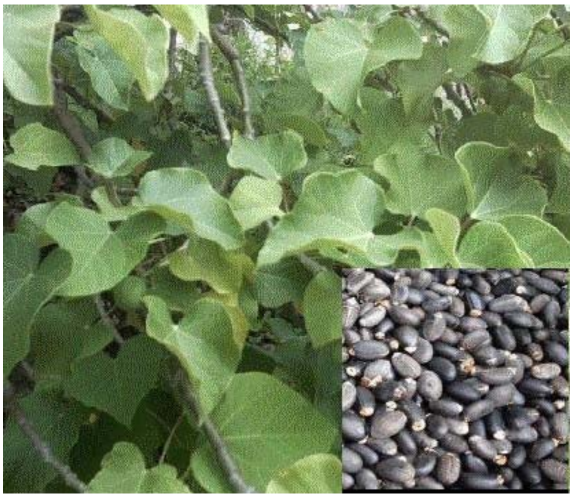

Figure 1. Plant and seed of Jatropha curcas

An effort is made in this study to evaluate the effect of varying the injection timing on the combustion, performance and emissions of a 3.5 kW engine fuelled with pure methyl ester of this oil (B100) for establishing the appropriate injection timing. The aim was to establish the modifications required in small, constant speed, direct injection diesel engines used extensively for agricultural applications so that these can be made to run on pure biodiesel (B100) with better performance and at the same time improve the emissions.

2. Experiment and procedure

In the study, the selected vegetable oil was transesterified and the major properties were evaluated. Further, the evaluation of the methyl ester was done in a compression ignition engine for combustion, performance and emissions at different injection timings.

2.1 Transesterification

The transesterification of the oil sample was carried out in the lab using standard procedures adopted commonly through out the world [13]. As Jatropha oil contains low FFA (less than 5%), methanol with KOH as catalyst was used for transesterification. After separation of glycerol, the ester was water washed to remove un-reacted methoxide. It was then heated to remove the water traces to obtain clear biodiesel.

The properties of so prepared biodiesel were tested in the laboratory using standard test procedures as per ASTM/BIS and are listed in Table 1. The properties tested were relative density (standard RD bottles of 50 ml capacity), calorific value (adiabatic bomb calorimeter), Kinematic viscosity (Redwood No.1 viscometer), flash point (Pensky-Marten closed cup apparatus), cloud and pour points, free fatty acid (FFA) contents (chemical titration method) and Iodine value (using Wij's solution).

Table 1. Evaluated properties of Jatropha oil and its methyl ester

Property	Unit	Biodiesel BIS Std	Diesel	Jatropha	
				Oil	Methyl ester
Relative density	gm/cm^3	0.87-0.90	0.8394	0.9187	0.8838
Calorific value	MJ/kg	-	44.13	40.65	39.40
Kinematic Viscosity					
at 32°C	cSt	3.5-5.0	3.93	46.8	10.8
at 60°C	cSt		2.77	23	5.82
Cloud Point	°C	-	<3	4	0
Pour Point	°C	-	<-5	1	-3
Flash Point	°C	>100	56	210	118
FFA	%	<0.8	-	5.01	0.28
Iodine value		<115	-	98.91	100.01

2.2 Experimental set-up

The study was carried out in the laboratory on an advanced fully computerised experimental engine test rig comprising of a single cylinder, water cooled, four stroke diesel engine (3.5 kW), commonly used in agriculture sector for minor irrigation needs, connected to eddy current type dynamometer for loading.

The setup (Figure 2) includes necessary instruments for online measurement of cylinder pressure, injection pressure and crank-angle. One Piezo sensor is mounted on engine head through a sleeve and other mounted on fuel line near injector for measurement of pressures. The set up has transmitters for air and fuel flow measurements, process indicator and engine indicator. Provision is also made for online measurement of temperature of -exhaust, -cooling water and -calorimeter water inlet and outlet and load on the engine. These signals are interfaced to computer through data acquisition system and the software displays the P-Θ and P-V diagrams. The specifications of the engine and instrumentation used are given in Table 2.

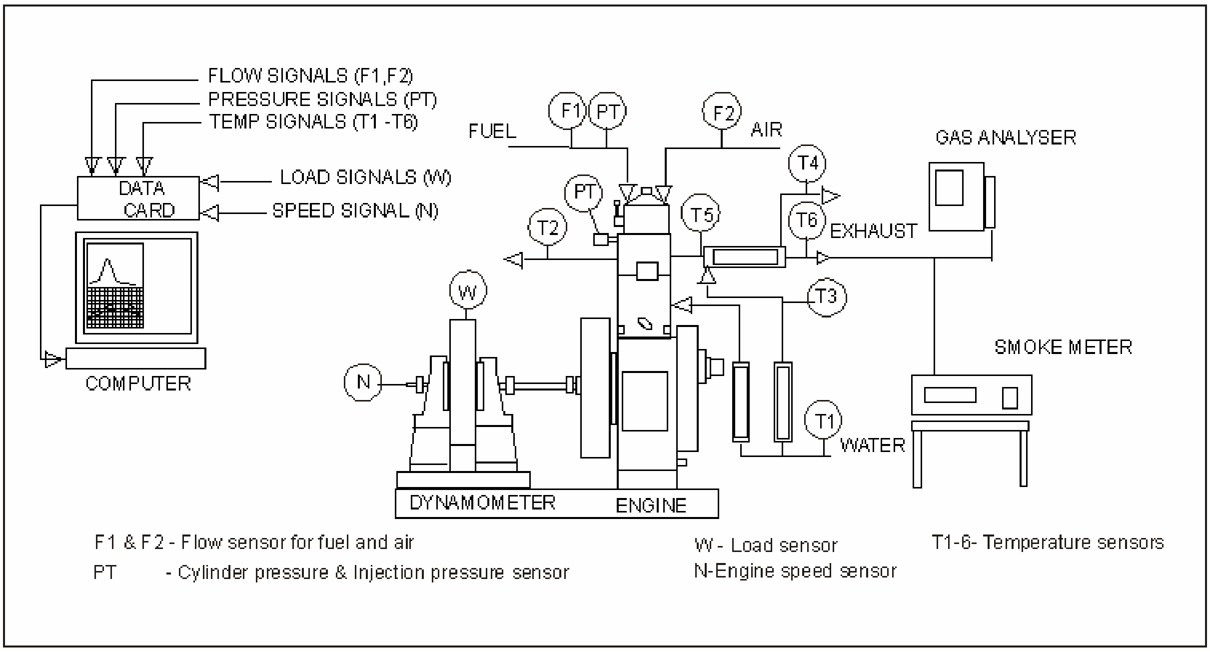

Figure 2. Engine test setup

Table 2. Test engine details

ITEM	Make/Model/Specs
Engine	
Make & Type	Kirloskar (TV1) - Single cylinder, DI, Four stroke, Water cooled
Bore and stroke	87.5mm × 110 mm
Cubic capacity	0.661 liters
Compression ratio	17.5:1
Rated power	3.5 kW at 1500 rpm (Load at rated power- 12 kg)
Injector opening pressure	210 bar
Injection timing	23 ° BTDC static (diesel)
Instrumentation	
Dynamometer	Eddy Current Type- Model AG10 of Saj Test Plant Pvt Ltd
Cylinder pressure sensor	Piezo sensor of PCB Piezotronics Inc, Model- M111A22; Resolution- 0.1 psi; sensitivity- 1 mV/psi
Fuel pressure sensor	Piezo sensor of PCB Piezotronics Inc, Model- M108A02; Resolution- 0.4 psi; sensitivity- 0.5 mV/psi
Load measurement	Load Cell -Sensortronics make, model 60001 with Digital indicator, Range 0-50 Kg, Supply 230VAC
Fuel flow measurement	Differential pressure transmitter, make- Yokogawa; Model- EJA110A-DMS5A-92NN
Air Flow Transmitter	Make- Wika; Model- SL1
Temperature sensor	Type RTD, PT100 and Thermocouple, Type K
Crank angle sensor	Digital encoder- Resolution 1 Deg, Speed 5500 RPM with TDC pulse
Engine indicator	Input: Piezo sensor(cylinder pressure and injection pressure), crank angle sensor, No of channels 2, Communication RS232
Software	"Enginesoft LV" Engine performance analysis software (on NI platform)

2.3 Experimental procedure

The performance test of the engine was conducted as per IS: 10000 [P: 5]:1980. Initially the engine was run on no load condition and its speed was adjusted to 1600 ±10 rpm. The engine was then tested at no load and at 25, 50, 75, 100 and 125 percent loads. For each load condition, the engine was run for at least three minutes after which data were collected. The experiment was replicated three times. For all settings, the emission values were recorded thrice and a mean of these was taken for comparison. The performance of the engine at different loads and settings was evaluated in terms of brake thermal efficiency (BTHE), brake specific fuel consumption (BSFC), indicated power (IP) and brake power (BP), exhaust temperature, indicated mean effective pressure (IMEP), cylinder pressure (P_c), rate of pressure rise ($dP/d\theta$), net heat release rate ($dQn/d\theta$) and emissions of carbon monoxide (CO), carbon dioxide (CO_2), un-burnt hydrocarbon (HC) and oxides of nitrogen (NOx) with exhaust gas opacity. The software enables evaluation of performance from the acquired data using standard relationships. The BTHE is evaluated using the expression BTHE = (brake power x 3600 x 100 / volumetric fuel flow in one hour x fuel density x calorific value of fuel). Similarly, BSFC is evaluated on the basis of fuel flow and brake power developed by the engine using the expression BSFC = (volumetric fuel flow in one hour x fuel density / brake power). The indicated work done per cylinder per cycle (Area of indicator diagram x scale factor x 10^5) and the indicated power (indicated work done per cycle x speed/2 x 10^{-3}) are computed from the area of indicator diagram.

The indicated mean effective pressure is a measure of the indicated work output per unit swept volume, in a form independent of the size and number of cylinders in the engine and engine speed and is computed as IMEP = indicated work output per cylinder per cycle / swept volume per cylinder. The rate of pressure rise ($\Delta p_c^* = \Delta p_c \times V_i/V_c$) and net heat release rate ($dQn/d\theta = (\gamma/\gamma\text{-}1) \times p \times (dV/d\theta) + (1/\gamma\text{-}1) \times V \times dp/d\theta$) are computed using the cylinder pressure history (p-θ) [14].

As injection timing plays crucial role in start of combustion and quality of combustion, the effects of varying the timing were studied for both –advancement and –retardation. For changing the injection timing in a jerk-pulse pump, the pump is fitted with different number of shims under the pump body. The standard setting of the engine used for test is with three shims to give standard injection timing of 23°

before top dead center (BTDC). Every single shim, of thickness 0.21 mm, deviate the injection timing by about 3°. The study was done with 3° advancement, normal and 3° and 6° retarded timings.

3. Results and discussion

The effect of transesterification on major properties of the oil is given in Table 1. The relationships between independent variables (load and injection timing) and dependent variables are shown in the Figures and the overall effects of injection timing on combustion, engine performance and emissions are discussed in this section.

3.1 Effect on brake thermal efficiency

The effect of injection timing on engine performance is significant. It can be seen from the Figure 3 that retarding the injection timing by 3° increases the thermal efficiency remarkably. Further retardation is not so beneficial, whereas advancement of injection is not desirable as it leads to drop in thermal efficiency of the engine. With JME as fuel, the thermal efficiency at full load increases from 22.96% to 24.90% on retarding by 3° and to 23.38% on retarding by 6°. On advancing the injection by 3°, the thermal efficiency drops to 22.58%. About 8% improvement in thermal efficiency is obtained by retarding the injection timing by 3°. At advanced injection timing, more of the fuel is injected and injection starts early in the cycle leading to earlier pressure rise before the piston reaches TDC position. Greater pressure rise in the compression stroke increases the negative work and consumes the momentum of flywheel. With reduction in net work output and increased fuel consumption, the thermal efficiency has to drop. Zeng et al. [15] also reported an increase in thermal efficiency and corresponding decrease in fuel consumption on retarding the injection timing upto a point after which the trend reversed. They stated that, there is an optimum timing at which engine delivers best efficiency which depends on the combustion properties of the fuel.

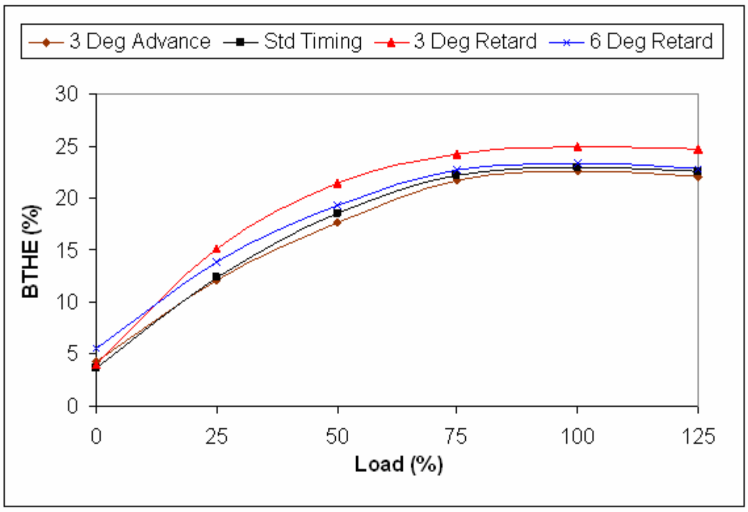

Figure 3. Effect of injection timing on brake thermal efficiency

3.2 Effect on brake specific fuel consumption

The brake specific fuel consumption is also affected by changes in the injection timing corresponding to the changes in thermal efficiency. With the advancement of the injection timing, the specific fuel consumption increases whereas retarding leads to improvement (Figure 4). With JME as fuel, the BSFC value increases to 0.40 from 0.39 Kg/kW-hr on advancing the injection by 3° degrees and it decreases to 0.36 and 0.37 Kg/kW-hr on retarding the injection by 3° and 6° respectively. On retarding the injection, the delay period increases but fuel delivery to cylinder reduces with a higher mean effective pressure in the cycle maintaining the power, thereby reducing the specific fuel consumption. On further delay in injection, unburned fuel gets exhausted, whereas on advancing the injection, shorter delay with sharp rise in pressure reduces the mean effective pressure. Nwafor [16] observed increased fuel consumption on advancing the injection timing in a natural gas engine and recommended not to advance the injection under high loading conditions. Parlak et al. [17] observed reduction in BSFC by 6% on retarding the engine by 4 degrees.

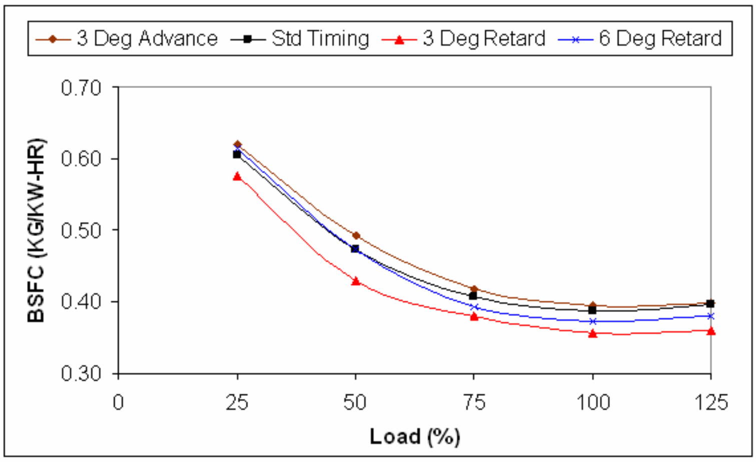

Figure 4. Effect of injection timing on brake specific fuel consumption

3.3 Effect on exhaust temperature

The trend line for exhaust temperature with different injection timings indicates increase in temperature of exhaust gases with retarded injection (Figure 5). As the combustion is delayed and more of the heat is released in mixing controlled combustion regime, greater amount of heat goes with exhaust gases. With advanced injection, wall heat transfer is more due to earlier combustion in the cycle leading to lower exhaust temperature.

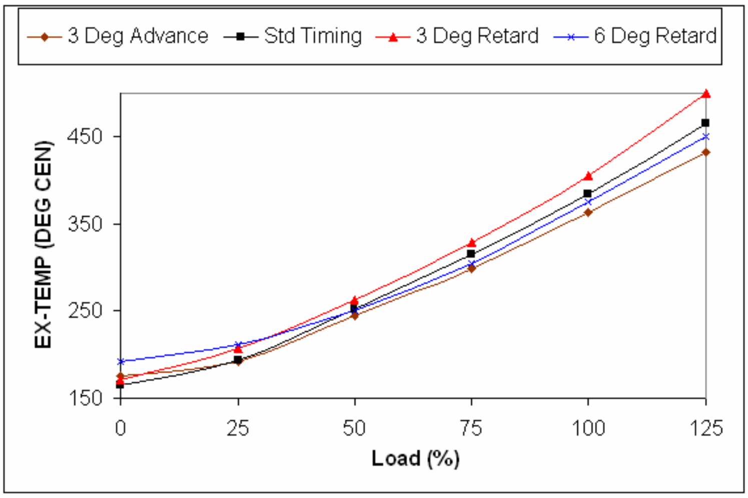

Figure 5. Effect of injection timing on exhaust temperature

3.4 Effect on power and mean effective pressure

The effect of injection timing on indicated mean effective pressure, indicated power and brake power is shown in Figure 6. On advancing the injection, the mean effective pressure in the cycle drops. This results in lower indicated power. The indicated power of the engine increases slightly on retarding the injection by 3^o whereas; it decreases on further retarding to 6^o or advancing by 3^o. When the injection is retarded by 3^o, better mean pressure is obtained and engine runs smoother. The indicated power and mean effective pressure with JME increases to 5.25 kW and 6.60 bar from 4.96 kW and 6.09 bar respectively on retarding the injection by 3^o. The indicated power changes to 5.04 kW and 4.80 kW; and mean effective pressure changes to 6.30 bar and 5.84 bar at 6^o retard and 3^o advance respectively. Similar increase in brake mean effective pressure on retarding the injection was reported by Zeng et al. [15] while using natural gas as fuel. Brake power of the engine is little affected by change in injection timing and remains almost same at all selected timings accept at 6 degree retarded.

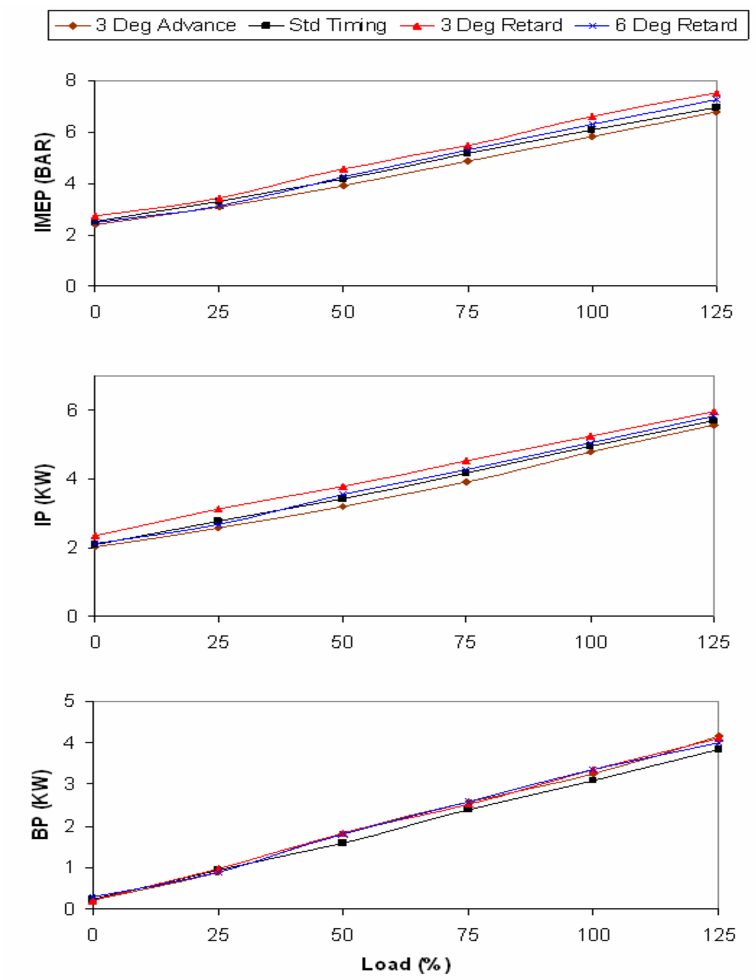

Figure 6. Effect of injection timing on indicated mean effective pressure, indicated power and brake power

3.5 Effect on cylinder pressure, rate of pressure rise and net heat release rate

Figure 7 represents the effect of injection timing on cylinder pressure, rate of pressure rise and net heat release rate at full load conditions. With changes in injection timing, as expected, the in-cylinder pressure, rate of pressure rise and net rate of heat release also changes. On advancing the injection, the pressure in the cylinder reaches to higher value as compared to the retarded injection scheme. This is mainly due to the fact that, on advancing the injection, larger amount of fuel is injected (injection starting earlier and stopping later). Higher pressure is also found before TDC with advancement due to early start of combustion.

With advancement of injection timing, the peak rate of pressure rise increases but it shifts before TDC (358 degree crank angle) with shorter delay period. On retarding the injection, the rate of pressure rise decreases slightly with a shift away from TDC and the ignition delay also increases.

Similar effects are seen on the rate of heat release. With advancement of injection by 3 degrees, the peak rate of heat release is at 357 degree crank angle and on retarding by 3 and 6 degrees, the peak heat release rate is found at 362 and 363 degrees against 361 degree with standard timing. With retardation, larger amount of heat is released in mixing controlled combustion regime resulting in higher mean pressure in the cycle. Zeng et al. [15] also reported similar results of shifting pressure rise rates and heat release locations with natural gas.

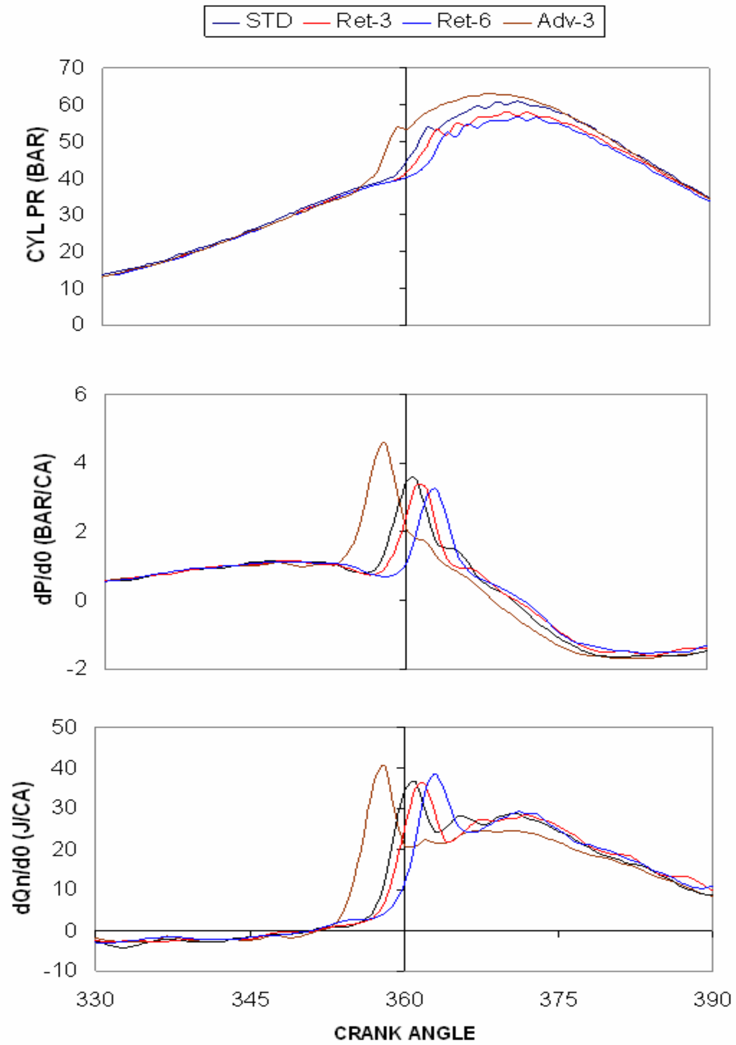

Figure 7. Effect of injection timing on cylinder pressure, rate of pressure rise and net heat release rate at full load

4. Conclusion

The fuel properties of biodiesel are comparable with that of diesel and lower blends with diesel are found suitable even for long term uses. Higher blends are still away from acceptance due to poor performance, mainly due to the reason that, the present age engines are the result of extensive research keeping petro-diesel only as fuel in mind. Biodiesel being a fuel of different origin and quality, the engine design needs revision and different settings for optimum performance.

As the combustion advances with biodiesel due to early entry, retarding the injection timing by 3^o is found to increase the thermal efficiency by 8% and reduce the specific fuel consumption by 9% when Jatropha methyl ester is used as fuel. Highest exhaust temperature and indicated power are obtained on 3^o retarded injection. By retarding the injection, the fuel delivery is also reduced resulting in slightly lower pressure rise with peak shifting towards outward stroke reducing the negative work.

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