Journal of Basic and Applied Engineering Research
p-ISSN: 2350-0077; e-ISSN: 2350-0255; Volume 2, Number 17; July-September, 2015, pp. 1496-1501

http://www.krishisanskriti.org/jbaer.html

# Heat Release Analysis of Small Single Cylinder Diesel Engine Fueled with Diesel and Biodiesel

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**Abstract—***With the continuous depletion of petroleum fuel and the contribution of these fuels to pollute the environment, use of petroleum-based fuels is now widely recognized as unsustainable. Biodiesel is one of the promising renewable, alternative and environmentally friendly biofuels that can be used in diesel engine with little or no modification in the engine. Heat release analysis is powerful tool to investigate the combustion process and engine performance. The objective of this study is to develop the simulation model which is capable to predict the influence of load and speed on the heat release of diesel engine. To investigate this experiments are performed on a small single-cylinder, four- stroke, water cooled, direct injection diesel engine. The engine control parameters varied during the experiments are, load and speed. The various output parameters measured and computed for analysis are cylinder pressure, peak cylinder pressure (Pmax), net heat release rate, maximum rate of heat release (HRRmax).*
*Using the First Law of Thermodynamics as a foundation, a simulation model is constructed for the calculation of several in-cylinder engine parameters (cylinder pressure, peak cylinder pressure ($P_{max}$), net heat release rate, maximum rate of heat release ($HRR_{max}$). Calculation of the speed of combustion (reaction rate) from the pressure measurements give the reaction co-ordinates (RCOs), which tells how much fuel that is burned at any time. In present investigation, the influences of load and speed on performance and combination characteristics of diesel engine using diesel and biodiesel. In addition empirical woshni model is applied. The reliability of the model has been verified by using a 4-stroke single cylinder direct injection diesel engine to simulate in Matlab/Simulink and the result of the simulation model is coincident with the actual operating condition of the engine.*

**Keywords***: Heat release, first law of thermodynamics, diesel engine, biodiesel,*

## 1. INTRODUCTION

Due to continuous depletion of finite resources of fossil fuels world is going to face the energy crisis in near future. India meets nearly 75-80% of its total petroleum requirements through imports. Moreover burning of fossil fuels us he global sources of $CO_2$ emissions all these problems lead to global search for alternative fuel. The troubles of using fossil fuels have made researchers to think of finding the alternative fuel. Biodiesel is defined as mono-alkyl esters of long chain fatty acids derived from vegetable oils or animal fats which conform to ASTM D6751 specifications for use in diesel engines. Biodiesel refers to the pure fuel before blending with diesel fuel. Biodiesel blends are denoted as, "BXX" with "XX" representing the percentage of biodiesel contained in the blend (ie: B20 is 20% biodiesel, 80% petroleum diesel). The performance of engine can be measured by the heat release analysis of the combustion process.

Heat release analysis in diesel engine provides an efficient way to acquire the combustion information. The HRR is defined as the rate at which the chemical energy of fuel is released by the combustion process [1]. The previous investigators carry out heat release analysis on engine using diesel as fuel. However, heat release analysis of diesel engine run by biodiesel derived from jatropha are very rare. Knowing the heat release, the in-cylinder process can be simulated basis of the thermodynamics laws. In present study, the in-cylinder pressure is measured from the engine test bed therefore the simulation model can be used to reconstruct the temperature and heat release rate. The main aim of this paper is to present the method to calculate the heat release rate during the combustion in a diesel engine and discuss the MATLAB/SIMULINK simulation results for the diesel and biodiesel as fuel. The influence of the speed and load on the heat release has also investigated. Diesel engine performance for both the diesel and biodiesel fuels have also discussed.

## 2. METHODOLOGY AND EXPERIMENT SETUP

***Test engine:*** The present research work was carried out on a small single cylinder 4-stroke water cooled direct injection diesel engine having the main technical features presented in Table 2.1. This engine has been widely used in small automotive vehicle applications.. For conducting the desired set of experiments and to gather required data from the engine, it is essential to get the various instruments mounted at the appropriate location on the experimental setup. A schematic of test rig used in the investigation is presented in Fig. 2.1

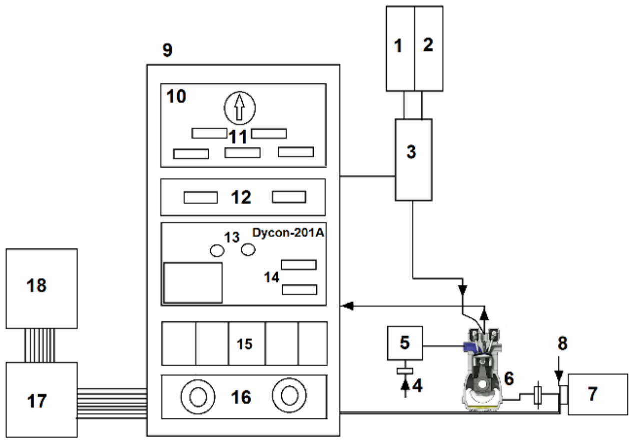


**Fig. 2.1: Schematic diagram of experimental set-up**

**1.** Diesel Tank **2.** Jatropha biodiesel tank **3.** AVL Fuel consumption **4.** Intake air manometer **5.** Intake air surge tank **6.** Test engine **7.** AVL Eddy current dynamometer **8.** TDC, crank angle, and speed pick-ups **9.** AVL Control panel **10.** Lubricant oil pressure **11.** Lubricating oil, cooling water, and exhaust gas temperature indicators **12.** Fuel consumption meter (cubic centimeter, time indicator) **13.** Dynamometer controller-201A **14.** Torque and Speed indicators **15.** Signal conditioning rack **16.** Throttle actuator **17.** Data Acquisition System **18.** Personal Computer

**Table 2.1: Technical specifications of the test engine**

| Parameter | Specification |
|---|---|
| Engine type | DI, water cooled |
| Number of cylinders | 1 |
| Bore (mm) | 92 |
| Stroke (mm) | 92 |
| Displacement (cm3) | 611 |
| Compression ratio | 18 |
| Rated rpm | 3000 |
| Maximum torque (Nm)@1600-1800rpm | 32 |
| S.F.C. gm./bhp/hr | 210 (Rated) |
| Dry weight (kg) | 75 kg (approx.) |

The main tasks for the models are not only researching the conditions of the combustion but also knowing the influence of the combustion process to the significant parameters in a diesel engine. Then the research of heat release opens up an area both in theory and experiment so that the combustion process in the cylinder can be known better.

### 2.1 Rate of heat release and combustion characteristics

Thermodynamic analysis of measured cylinder pressure data is a very powerful tool used for quantifying combustion parameters. Heat release analysis is most commonly used for diesel engine combustion studies and produces absolute energy with units of Joules or Joules/crank angle degree. The details about combustion stages and events can often be determined by analyzing the heat release rates. The trend of heat release (Instantaneous heat release rate and maximum heat release rate) can be obtained by processing in-cylinder pressure data. The analysis for the heat release rate is based on the application of the first law of thermodynamics for an open system. It is assumed that the cylinder contents are a homogeneous mixture of air and combustion products and is at uniform temperature and pressure at each instant in time during the combustion process. The first law for such a system can be written as

$$\frac{dU}{d\theta} = \frac{dQ}{d\theta} + \sum m_i\, h_i - p\frac{dV}{d\theta} \quad (1)$$

where, dQ/dθ is the heat transfer rate across the system boundary, p (dV/ dθ is the rate of work transfer done by the system due to system boundary displacement, and $m_i$ and $h_i$ are the mass and enthalpy of the flow into the system respectively. p is pressure of gas in the cylinder. V is volume of the cylinder and U is the internal energy of the cylinder contents. If the crevice effect is neglected, the above equation can be reduced to

$$\frac{dU}{d\theta} = \frac{dQ}{d\theta} + \sum m_f\, h_f - p\frac{dV}{d\theta} \quad (2)$$

where $m_f$ and $h_f$ the fuel mass flow rate and enthalpy respectively of the fuel entering the cylinder.

### 2.2 Structure of the simulation model

Heat release rate can be defined in many different ways. The three main definitions used in this research are:

(a) Net Apparent Heat Release Rate (*NAHRR*) [3]

$$\text{NAHRR}= \dot{Q}_{comb} - \dot{Q}_{loss} + \dot{E}_f = m.c_v.\frac{dT}{dt} + p.\frac{dV}{dt} \quad (b)$$

Gross Apparent Heat Release Rate (GAHRR)v[4]

$$\text{GAHRR}=\dot{Q}_{comb} + \dot{E}_f = m.c_v.\frac{dT}{dt} + p.\frac{dV}{dt} + \dot{Q}_{loss} \quad [\text{J/s}]$$

(c) Combustion Reaction Rate (CRR):

$$\text{CRR}=\xi=\frac{\text{m.c}_\text{v}.\frac{\text{dT}}{\text{dt}} + \text{p}.\frac{\text{dV}}{\text{dt}} + \dot{\text{Q}}_{\text{loss}}}{\text{u}_{\text{comb}} + \text{e}_\text{f}} \quad [\frac{kg}{s}] \quad (5)$$

NAHRR can directly be calculated form the pressure and temperature in the cylinder.

*GAHRR* includes heat loss to the walls and indicates the heat produced by combustion correctly, but it is not possible to calculate it directly from measurements. Therefore the accuracy of *GAHRR* relies on the appropriate estimation model of the heat loss. For the *CRR* the effective combustion value ($u_{eff} + e_f$) must be known including its dependency on temperature. Also the effect of 'energy of fuel' is included, for which it is assumed that the injection rate is equal to the evaporation rate of the fuel entering the system.

The heat release model is based on the combustion reaction rate ($\xi$, fuel burn rate, i.e. mass of fuel burnt per unit time).

When multiplied with – a temperature dependent – heat of combustion

($u_{comb}$ ) this acts as heat source ($\dot{Q}_{comb}$ ) in the first law. The reaction rate after integration also causes a (slight) mass addition in the mass balance. The in-cylinder volume is calculated using the crank angle ($\alpha$), cylinder geometry and dimensions, assuming a constant engine speed to calculate the crank angle in previous equation.

$$\frac{dT}{dt} = \frac{\dot{Q}_{comb} - \dot{Q}_{loss} - p.\frac{dV}{dt} + \dot{E}_f}{m.c_v} \quad [6]$$

$$V(\alpha) = A_b.L_s.\left[\frac{1}{\varepsilon-1} + \frac{1}{2}.\left\{(1+\cos\alpha) + 1\lambda CR1-1-\lambda CR2.sin2\alpha \right.\right. \quad [7]$$

Both the mass and volume, together with a feedback of the in-cylinder temperature and gas constant, are used in the gas law to calculate in-cylinder pressure. When calculating work and internal energy terms, volume is differentiated: this can be done straightforward in a simulation environment. Woschni's model is used to evaluate the heat transfer coefficient [5] to calculate heat loss to the walls. All the properties of air, stoichiometric gas and fuel (e.g. gas constant, internal energy, specific heat) are calculated in the 'Properties library'. The energy of fuel ($E_f$ ) indicates the fuel carrying energy into cylinder upon entry.

The volume at trapped condition is:

$$V_1 = V_{IC}$$

The mass at trapped condition can be calculated using the gas law:

$$m_1 = \frac{p_1 V_1}{R_1 T_1} \quad (8)$$

### 2.3 Mass balance and composition

The purpose in this sub-model is to acquire the mass in the cylinder and more importantly the composition parameter '*x*' which is used in the 'Properties library' to calculate specific heat, internal energy, and enthalpy etc.

In the 'Heat release model', the Combustion Reaction Rate (*CRR* or $\xi$) is modelled with a double Vibe model that will be discussed later:

$$\xi = \frac{dm_f}{dt} \quad (9)$$

Adding the mass of fuel to the trapped mass in *IC* (Inlet valve Closed) condition, the total mass in the cylinder is known according to the mass balance:

$$m = m1 + \int_{IC}^{EO} \xi.dt \quad (10)$$

Last but not least, the composition of gas in terms of the air fraction, which is an important variable in the 'Properties library' (presented in the next section), can be calculated using $\sigma$ (air/fuel ratio). The '$x_1$' in equation [2.11] is the air fraction of the trapped condition, which is strictly speaking not equal to unity, due to the residual exhaust gas remaining in the cylinder after scavenging.

$$x = \frac{m_1.x_1 - \sigma.\int_{IC}^{EO} \xi.dt}{m_1 + \int_{IC}^{EO} \xi.dt} \quad (11)$$

### 2.4 Properties library

In order to obtain the in-cylinder gas properties (e.g. $c_v, c_p$, $u$ and $h$), the in-cylinder gas is modelled as a mixture of two well defined basic mixtures – air and stoichiometric gas, which are in turn also considered to be mixtures of several species. Finally the in-cylinder properties can be calculated as a function of temperature using a single parameter, in this case the air mass fraction, which is calculated in equation [11].

First of all, the properties of each species in air, stoichiometric gas and fuel are calculated:

$$u_j = \Delta u_j + u_j^{ref} = h_j - R_j.T \quad (12)$$

$$c_{v,j} = c_{p,j} - R_j = \sum_{k=1}^{m} a_{k,j}.\theta^{k-1} - R_j \quad (13)$$

$$\Delta h_j = h_j - h_j^{ref} = \sum_{k=1}^{m} \frac{a_{k,j}}{k}.T_{norm}.\theta^k - \sum_{k=1}^{m} \frac{a_{k,j}}{k}.T_{norm}.\theta_{ref}^k \quad \mathbf{2.5}$$

### Heat of combustion

The quantity $Q_{comb}$ is defined as 'combustion heat', for a closed system based on internal energy, and is calculated using the properties of fuel, air and stoichiometric gas, which have been presented in Appendix II. The combustion heat rate is:

$$\dot{Q}_{comb} = \xi.(u_{comb})$$

$$\dot{Q}_{comb} = \xi.\left(u_f + \sigma.u_a - (1+\sigma).u_{sg}\right) \quad (14)$$

In most research studies, $u_{comb}$ is set to be constant (e.g. 4.27*104 kJ/kg for normal diesel fuel), often also neglecting the small difference between heat of combustion based on enthalpy and based on internal energy. In this study all terms in principle are temperature dependent and heat of combustion therefore is not a constant. The composition of diesel fuel is rather complex

### 2.6 Energy of fuel

The quantity $E_f$ is defined as 'energy of fuel' and for a closed system there are two constituents: one is the difference caused by fuel phase changing from liquid to gas; the other is the difference between fuel injection pressure and in-cylinder pressure.The energy fuel rate is:

$$\dot{E_f} = \dot{m_{f,in}}.e_f = \dot{m_{f,in}}.(h_{f.liquid}^{in+} - u_{f,gas}) \quad [15]$$

The $h_{f,liquid}^{in}$ in equation [2.13] is the inflow fuel enthalpy in *liquid* phase.

### 2.7 Heat loss to the walls

During combustion, there is a large temperature difference between the combustion gas and the cylinder wall. The heat loss to the surrounding metal should be taken into account

$$\dot{Q_{loss}} = \sum_{i=1}^{3}\{\alpha_{g\rightarrow w}.(T - T_{wall,i}).A_{wall,i}\} \quad [16]$$

With: i=1, cylinder wall;i=2, cylinder cover; i=3, piston crown.

Since it is difficult to measure the surface temperature in these three places and these temperatures do not play a significant role in the heat loss value, these three temperatures are estimated and kept constant. The heat transfer coefficient can be estimated using semi-empirical formulae. In this model, the Woschni method is used.

$$\alpha = C_1 \frac{1}{D_B^{0.214}}.\frac{p^{0.786}}{T^{0.525}}.(C_3.c_m + C_4.\frac{p_0}{p_1}.\frac{V_s}{V_1}.T_1)^{0.786} \quad [17]$$

Note that the p0 and T0 here are the pressure and temperature in the ‘no fuel injection’ condition instead of ambient condition. The constants are valid for a heat transfer coefficient in [W/m2K], while pressure must be in [bar], bore in [meter] and temperature in [Kelvin].

Since it was impossible to measure wt , the ratio wt /cm is unknown. As C1 will be considered a variable, C3 cannot very well be determined from measurements as well. According to literature wt /cm can vary within the range of 5 to 50, for this engine, it is set to 10 because limited swirl is expected for this engine.

The parameter C4 is related to the shape of the combustion chamber. Woschni advised:

Direct injection:

$$C_4 = 0.00324 \quad \left[\frac{m}{s.K}\right] \quad [18]$$

Here, we can write the formula in a more simpler way ; The heat transfer coefficient given by Woschni is:

$$h = 3.26b^{-0.2}\,P^{0.8}T^{-0.55}v^{0.8}$$

Where, h= heat transfer coefficient

A=exposed combustion chamber surface area

$T_g$ = temperature of the cylinder gas

$T_w$ = cylinder wall temperature

ω = engine speed [rad/s]

$h_c$ = heat transfer coefficient

b = bore

k = thermal conductivity = 0.15 W/mK

$\overline{S_p}$ = mean piston speed = 2s□

a = varies from 0.35 to 0.8 for normal combustion

b = 0.7

## 3. RESULTS AND ANALYSIS: EFFECT OF VARIATION OF LOAD AND SPEED

The effect of variation of speed on cylinder pressure at normal injection (rated) timing at 5Nm, 10Nm, and 15Nm for different speeds are shown in Figs 3.1 3.2 and 3.3 respectively. These Figures reveal that as the speed increases, the peak cylinder pressure reduces and occurs late (8-20 CAD after TDC) in the expansion stroke for a given load, 5Nm, 10Nm or 15Nm and for a given fuel, diesel or Jatropha biodiesel. loads.

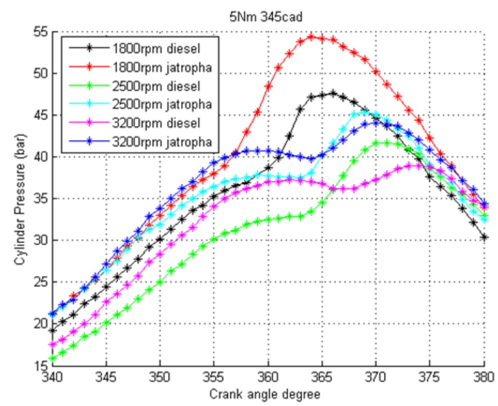


**Fig. 3.1: Effect of variation of speed on cylinder pressure (5Nm, normal injection timing)**

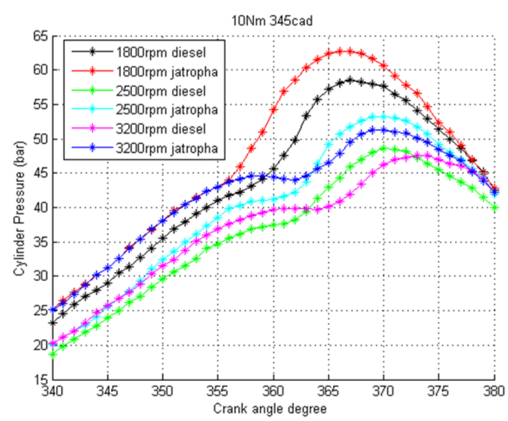


**Fig. 3.2: Effect of variation of speed on cylinder pressure (10 Nm, normal injection timing)**

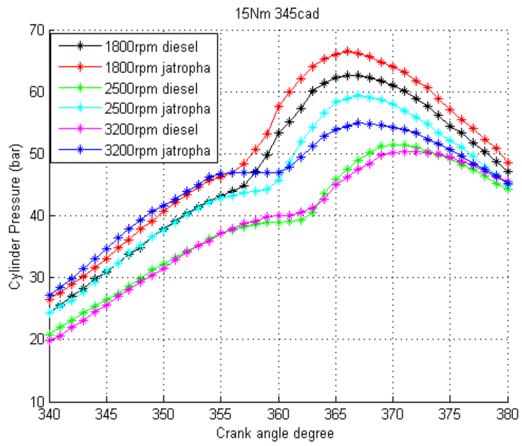


**Fig. 3.3: Effect of variation of speed on cylinder pressure (15 Nm, normal injection timing)**

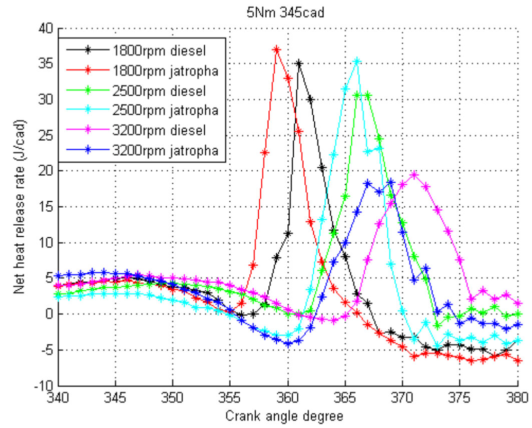


**Fig. 3.5: Effect of variation of speed on net heat release rate (5Nm)**

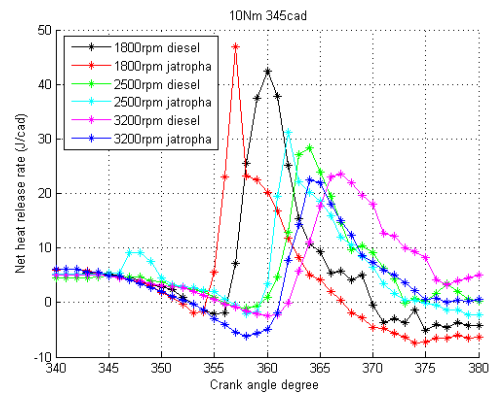


**Fig. 3.6: Effect of variation of speed on net heat release rate (10Nm)**

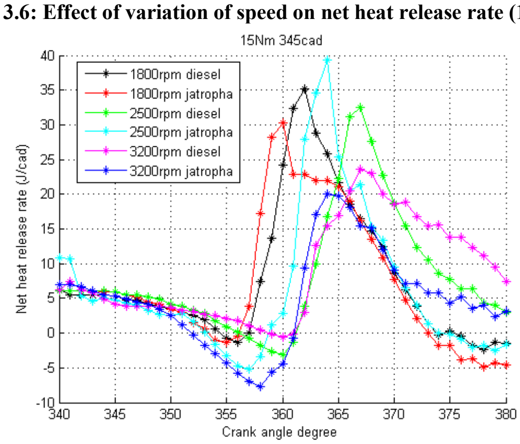


**Fig. 3.7: Effect of variation of speed on net heat release rate (15Nm)**

It can be observed that the cylinder pressure characteristics at higher speeds are notably different from those at lower speeds. Two peaks are observed in cylinder pressure histories at 2500 rpm and 3200 rpm at any given load and injection timing for both cases of diesel and Jatropha biodiesel. This is because. as the speed increases, the delay period increases and the actual burning time decreases, which results in slow and late combustion that lasts even after the start of expansion stroke. Also it can be observed that at advanced injection timing, the peak pressure of Jatropha biodiesel is lower than that of diesel for a given load (5 Nm, 10 Nm and 15 Nm) and speed (1800rpm, 2500 rpm and 3200 rpm) condition.

### 3.2 Net heat release rate

The effects of variation of speed on net heat release rate for different loads with normal injection tinting is given in Figs. 3.5-3.7. It can be observed that increasing the speed, increases the delay period. Consequently the maximum rate of net heat release is decreased, for both fuels. The heat release peaks of Jatropha biodiesel are higher than that of diesel at 5Nm, and 10Nm. However at 15 Nm, they are lower than the diesel except at 2500rpm (Fig. 3.7).This is due to the relatively earlier start of combustion of Jatropha biodiesel at 2500 rpm. to the expansion stroke.

Hence the late lower peaked (more flat) heat release characteristic results at higher speeds.

**Effect of load on net heat release: fig. 3.8, 3.9 and 3.10 show the effect of load on the net heat release.**

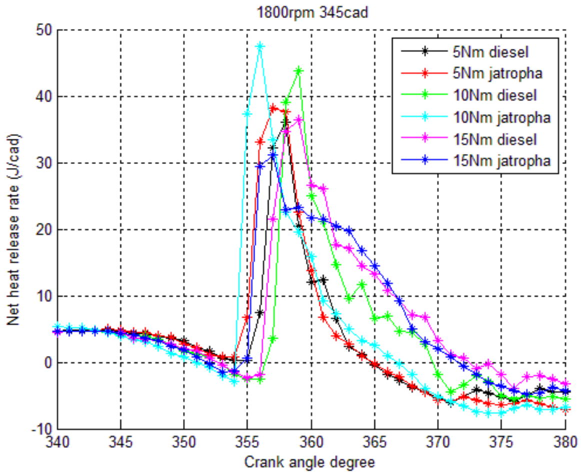


**Fig. 3.8: Effect of variation of load on the net heat relase 1800rpm**

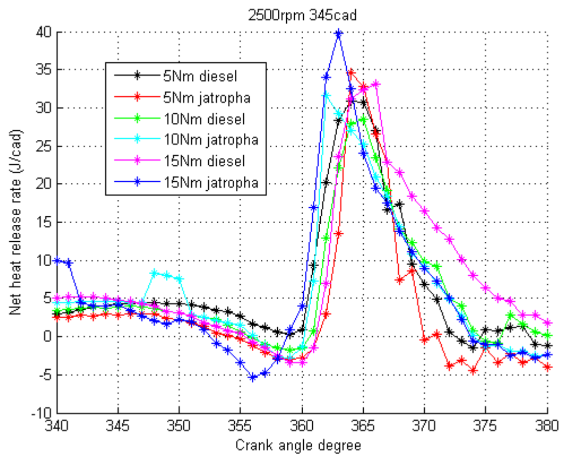


**Fig. 3.9: Effect of variation of load on the et heat release 2500 rpm**

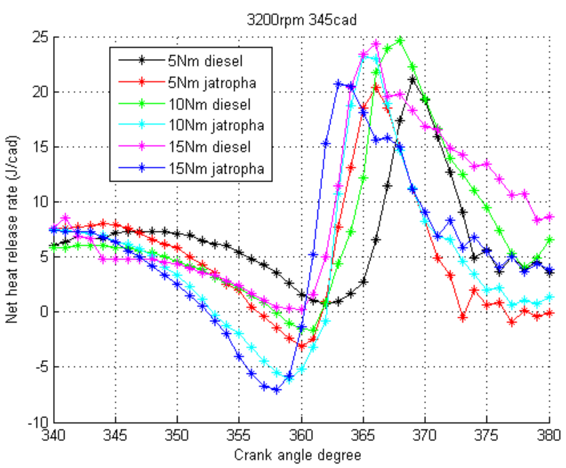


**Fig. 3.10: Effect of variation of load on the et heat release 3200 rpm**

## 4. CONCLUSION

In this present investigation, the effects of engine operating, injection system on a Jatropha biodiesel fueled small, single cylinder, water cooled, vertical DI diesel engine used in three wheeler automotive vehicles, and stand-by power generating sets applications have been studied. Accordingly, it focused on changes in combustion parameters, cylinder pressure, net heat release rate, peak cylinder pressure.

1. Jatropha is having more HRR and so is more appropriate to use at moderate rpm.
2. Calorific value is less, but it is taken care by the huge ignition quality of jatropha.

## 5. ACKNOWLEDGMENT

This work was supported in part by a grant from the National Science Foundation.

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