

# International Journal of Innovative Research in Science, Engineering and Technology

(A High Impact Factor, Monthly, Peer Reviewed Journal)

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Vol. 9, Issue 3, March 2020

## Study of Thermodynamic Model of CI Engine

Krishna Shrivastava

Ph.D. Student, Dept. of Mechanical Engineering, SSBT's College of Engineering and Technology, Jalgaon, India

**ABSTRACT:** In order to reduce the uses of hydrocarbon and its depletion of environment, bio-diesel and Ethanol is used as a substitute with diesel or with its blend. The nonedible oil biodiesel can be a big substitute to diesel engine. This paper focuses to develop a zero dimensional single zone thermodynamic model for compression ignition engine to compute the performance of the CI engine and emission parameter. The effect of compression ratio, Injection angle, fuel blends with varying load on the engine performance and emission is not considered in this study. The basic concept of thermodynamic model is developed by considering single cylinder Kirloskar diesel engine research test rig available at Apex innovation Ltd, Sangli, Maharashtra.

**KEY WORDS:** Thermodynamic modeling, Compression ignition engine, Diesel engine

### I. INTRODUCTION

Energy is big requirement of present society. IC engines are used for generating power for various applications from more than 100 years. The huge increasing demand, depletion and increasing price of the petroleum has promoted research worldwide on alternative energy sources for IC engines. Direct use of biodiesel in CI engine affects the engine performance and because of higher viscosity it affects the nozzles etc.. Non edible seed karanja and Jatropha biodiesel, etc. are popular substitutes for blending with diesel [1]. But to control Nox the ethanol may be added to the blend. In the present energy scenario efforts are being focused on use of bio diesel and Ethanol along with diesel in compression ignition engine, but there are many issues related to performance and emission. More research is required with the blends of Diesel, Biodiesel and Ethanol. The optimum operating parameters can be determined using experimental techniques but the experimentation is time consuming and expensive. Computer simulation [2] may serve as a tool for a better understanding of the variables involved. The simulation technique allows examining the effects of various parameters and reduces the need for complex experimental analysis of the engine. A validated simulation model is very useful tool to study engines with any new type of fuels. The theoretical model (thermodynamic model) of internal combustion engines based on the first law of thermodynamics and is used to analyze the performance characteristics of compression ignition engines. The pressure, temperature, Enthalpy and other required properties are evaluated with respect to crank angle are applied the simulation of process. A zero-dimensional single-zone model as compared with multi-zone models is much simpler and easier to run and it is capable of predicting engine performance and fuel consumption accurately [3]. Therefore zero-dimensional single-zone model is developed which is based on first law of thermodynamics. It can be easily programmed in MATLAB to predict the engine performance for varying blend. This paper focuses on thermodynamic model. The thermodynamic model & heat transfer models [4] to predict the emission & combustion characteristics of a DI Diesel engine runs at improved blend by classical two zones methods. By First law of thermodynamics they find temperatures and pressure of cylinder in each zone. The model validation done by experimentation on a single cylinder DI diesel engine fuelled with 2-ethoxy ethanol blend with diesel. It decreases the smoke exhaust and the thermal efficiency for the power outputs increases. It is observed that experimental results and simulated results of the proposed model has good agreement. Thermodynamic models are usually based on the 1st law of thermodynamics [3] and it may be used to analyze the performance characteristics of CI engines. A single-zone, zero-dimensional model as compared with multizone models is much simpler, easier and quicker to run, [5, 6] to predict engine performance and SFC accurately with a high computational efficiency. The Internal Combustion Engines Fundamentals [7] are discussed prominently to develop analytical model based on crank angle, its simulation and

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optimization [ 8 ] is discussed along with detail study of thermodynamic processes. The Engine Crank Mechanism and Load variation Formulation by Kinematics [10] in crank angle domain helpful for analysis. Therefore zero dimensional single zone model may be developed based on first law of thermodynamics and it may be programmed in MATLAB to analyze and predict the engine performance for varying blend, compression ratio, injection angle and relative air-fuel ratio.

### Nomenclature

#### Symbol

$r$	= compression ratio
$l$	= length of connecting rod (m).
$B$	= bore diameter (m)
$L$	= Stroke Length (m)
$V_{disp}$	= displacement volume (m <sup>3</sup> ).
$\theta$	= angular displacement in degrees with respect to bottom dead center (BDC).
$\theta_s$	= Crank angle for start of combustion
$\gamma$	= Specific Heat ratio
$P$	= Pressure ratio
$P_m$	= mean effective pressure
$V$	= volume (m <sup>3</sup> ).
$m_c$	= number of moles of carbon in fuel one mole.
$hc$	= heat transfer coefficient due to convection (W/m <sup>2</sup> .K).
$A$	= interior surface area of cylinder (m <sup>2</sup> )
$C_{pex}$	= Specific heat of exhaust gas kJ/kgK
$C_w$	= Specific heat of water kJ/kgK
$m_f$	= mass of fuel consumption kg/hr
$m_a$	= mass of air consumption kg/hr
$m_w$	= Calorimeter water flow kg/hr
$T_3$	= Calorimeter water inlet temperature (K)
$T_4$	= Calorimeter water outlet temperature (K)
$T_5$	= Exhaust gas temperature at inlet to calorimeter (K)
$T_6$	= Exhaust gas temperature at outlet to calorimeter (K)
$T_{amb}$	= Temp. ambient (K)
$T$	= Instantaneous gas temperature (Kelvin )
$T_w$	= Cylinder wall temperature (Kelvin )
$R$	= universal gas constant(8.314 kJ/kmole.kelvin)
$C_m$	= Piston mean speed(m/s)
$U$	= Internal energy ( KJ)
$H$	= Enthalpy (KJ)
$C_p$	= specific heat at constant pressure (kJ/kg.kelvin)
$C_v$	= specific heat at constant volume (kJ/kg.kelvin )
$N$	= Rotational speed of engine (RPM )
$N_i$	= Number of Moles
$ID$	= Ignition Delay
$\Delta\theta$	= combustion duration in crank angle ( deg )
$Q$	= heat release per cycle ( KJ)
$dQ_r/d\theta$	= rate of heat released during combustion (kJ/deg A)

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- $dQh/d\theta$  = rate of heat transfer by convection (kJ/ deg A)
- $dw/d\theta$  = rate of work done(kJ/ deg A)
- $du/d\theta$  = rate of change of Internal Energy(kJ/ deg A)
- $du/d\theta$  = rate of work done(kJ/ deg A)
- $dV/d\theta$  = incremental change in cylinder volume ( $m^3/ deg A$ )
- $dT/d\theta$  = rate of Temperature change(Kelvin/ deg A)

## II. DESCRIPTION OF MATHEMATICAL MODELING

### 2.1 Energy balance equation

According to the first law of thermodynamics, the energy balance equation [3] for the closed cyclic system is

$$m \frac{du}{d\theta} = \frac{dQr}{d\theta} - \frac{dw}{d\theta} \quad - (1)$$

The rate of heat transfer is due to combustion of the fuel and from cylinder walls of the cylinder walls to gases. The equation (1) can be written as

$$m \frac{du}{d\theta} = \frac{dQr}{d\theta} - \frac{dQh}{d\theta} - \frac{dw}{d\theta} \quad - (2)$$

on simplification

$$m \frac{du}{d\theta} = \frac{dQr}{d\theta} - \frac{hc.A dT}{d\theta} - \frac{R.T.dV}{d\theta} \quad - (3)$$

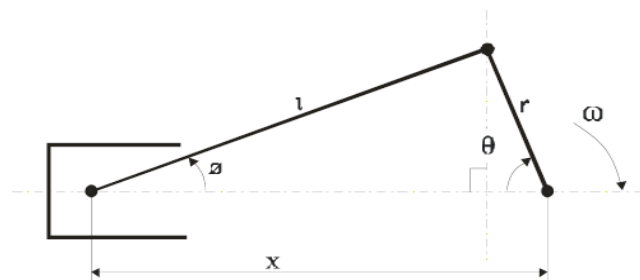


Fig. 1 : slider crank Mechanism

### 2.2 Cylinder volume at any crank angle

The figure1 shows crank-slider mechanism[ 3,7]. The piston pin position is given by

$$x = r \cos\theta + l \cos\phi$$

From figure2.  $1r \sin\theta = l \sin\phi$  and recalling  $\cos\phi = \sqrt{1 - \sin^2\phi}$

$$x = r \left\{ \cos\theta + \frac{l}{r} \left[ 1 - \frac{1}{2} \left( \frac{r}{l} \right)^2 \sin^2\theta - \frac{1}{8} \left( \frac{r}{l} \right)^4 \sin^4\theta + \dots \right] \right\} \quad - (4)$$

The  $\sin^2\theta$  can be written as

$$\sin^2\theta = (1 - \cos 2\theta)/2 \quad \& \quad \sin^4\theta = (3/8) - (1/2)\cos 2\theta - (1/8)\cos 4\theta$$

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Substituting the results from equation 4 into equation 5 gives

$$x = r \{ \cos\theta + l/r [ 1 - 1/2 (r/l)^2 (1 - \cos 2\theta) / 2 - 1/8 (r/l)^4 ((3/8) - (1/2)\cos 2\theta - (1/8)\cos 4\theta) + \dots ] \} \quad (5)$$

As per geometry of the engine  $(r/l)^2$  is less than 0.1 and  $(r/l)^4$  terms is comparatively less may be neglected. The tentative position of piston is expressed as

$$x = r \{ \cos\theta + l/r [ 1 - 1/2 (r/l)^2 (1/2 - 1/2 \cos 2\theta) ] \} \quad (6)$$

Where  $r$  = radius ( Length ) of crankshaft and  $l$  = connecting rod length. Find the value of  $x$  by equation 6 ; then  $(l + r - x)$  shall give distance traversed by piston from its top ( TDC ) position at any angle  $\theta$ . The slider crank angle formula may be to get the cylinder volume at any position of crank

$$V(\theta) = V_{dis} \left[ \frac{r^3}{r-1} - \frac{(1-\cos\theta)}{2} - \frac{1}{2} \sqrt{(2l/s)^2 - \sin^2\theta} \right] \quad (7)$$

By differentiating find change of cylinder volume with respect to change of crank angle

$$\frac{dV(\theta)}{d\theta} = \frac{V_{dis}}{2} \left[ \frac{1}{\sqrt{(2l/s)^2 - \sin^2\theta}} - \sin\theta \right] \quad (8)$$

### 2.3 Compression and Expansion strokes

During compression and expansion processes the temperature and pressure at each step are calculated using ideal gas equation and an isentropic process [4].

$$P_{i+1} = P_i \left( \frac{V_i}{V_{i+1}} \right)^\gamma \quad (9)$$

$$T_{i+1} = T_i \left( \frac{V_i}{V_{i+1}} \right)^{\gamma-1} \quad (10)$$

$$\gamma = \frac{C_p}{C_v} \quad (11)$$

$$C_p = \frac{\sum Ni C_{pi}}{\sum Ni} \quad (12)$$

$$C_p - C_v = R \quad (13)$$

### 2.4 Combustion Process

Heat release due to combustion is calculated by using single Weibe's heat release correlation [1]

$$\frac{dQ_r}{d\theta} = a(n+1) \left( \frac{Q_r}{\theta t} \right)^n \left( \frac{\theta}{\theta t} \right)^n \exp \left[ -a \left( \frac{\theta}{\theta t} \right)^{n+1} \right] \quad (14)$$

Where  $a=6.908$  and  $n=2$  are wiebe's constants characterizing the rate of combustion for naturally aspirated normal burning engines.  $\theta t$  = the combustion duration of crank angle.  $\theta$  = Crank angle at the start of combustion. These constants can determine by fitting the Weibe's heat release correlation to experimental profile.

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### 2.5 Heat Transfer

The convective heat transfer in engine is considerable which directly affects the engine performance.

The convection heat transfer is given by

$$\frac{dQh}{d\theta} = hcA(T-T_w) \quad - (15)$$

hc is given by Hohenberg equation [7,8].

$$hc = \frac{130P^{0.8}(Cm+1.48)^{0.8}}{V^{0.06}T^{0.4}} \quad -(16)$$

### 2.6 Ignition Delay

An empirical formula, developed by Hardenberg and Hase [1] is used for predicting Ignition delay in crank angle degrees.

$$\text{Ignition Delay (ID)} = (0.36+0.22Cm) \exp\left[E_A\left(\frac{1}{RT} - \frac{1}{1719.99}\right)\left(\frac{21.2}{P-12.4}\right)^{0.63}\right] \quad -(17)$$

Where  $E_A$  is apparent activation Energy

### 2.7 Gas Properties Calculation

A hydrocarbon fuel can be represented by  $C_xH_yO_z$ . The amount of oxygen required for combustion per mole of fuel is given [10] by:

$$Y_{cc} = m_c + 0.25m_h - 0.5m_o \quad - (18)$$

The minimum amount of oxygen required for combustion per mole of fuel is [10]

$$Y_{min} = Y_{cc} - 0.5 m_c \quad - (19)$$

Calculation of Internal energy, Enthalpy, Specific heat, temperature can be calculated using following equations:

$$U(T) = A + (B-R).T + C.\ln(T) \quad - (20)$$

$$H(T) = A + B.T + C.\ln(T) \quad - (21)$$

$$C_p(T) = B + \frac{C}{T} \quad - (22)$$

$$C_v(T) = (B-R) + \frac{C}{T} \quad - (23)$$

The A, B and C are the coefficients in the above given equation.

### 2.8 Performance calculation:

The performance calculation as per [7,8,9,10] mentioned below:

- Thermal efficiency (Indicated): The Indicated thermal efficiency is the ratio of energy used in the indicated power to fuel heating value.

$$\eta_t = \text{IndicatedPower} / \text{fuel heating value} \quad -(24)$$

- Thermal efficiency (Brake) : Brake thermal efficiency is the ratio of brake power to the fuel heating value

$$\eta_{bth} = \text{BrakePower} / \text{fuel heating value} \quad -(25)$$

- Mech. Efficiency : Mech. efficiency is the ratio of brake power (delivered power) and indicated power (power available at the piston).

$$\eta_m = \text{BrakePower} / \text{IndicatedPower} \quad - (26)$$

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- Volumetric efficiency : The engine power output is limited dependent on available air during suction stroke in cylinder to burned fuel effectively. It is defined as the ratio of Air enters into the cylinder to the engine swept volume. So volumetric efficiency is defined as:

$\eta_v$  (%) = Mass of air consumed / mass of flow of air to fill swept volume at atmospheric conditions

$$\eta_v (\%) = \frac{\text{AirFlow} \left( \frac{\text{kg}}{\text{hr}} \right)}{\frac{\pi}{4} D^2 \cdot \frac{N}{n} (\text{No. of Cylinder}) \cdot 60 \cdot \rho_{\text{air}}} \cdot 100 \quad - (27)$$

Take  $n = 1 - 2$  stroke engine,  $n = 2 - 4$  stroke engine.

$$\text{AirFlow} = C_d \cdot \frac{\pi}{4} \cdot D^2 \sqrt{2gh_w \cdot \rho_{\text{water}} / \rho_{\text{air}}} \cdot \rho_{\text{air}} \cdot 3600 \quad - (28)$$

- Fuel consumption is also known as Specific Fuel consumption (SFC): Brake specific fuel consumption is known as BSFC on the basis of Brake power and indicated specific fuel consumption is known as ISFC on the basis of Indicated power.

- A/F ratio: Air-fuel (A/F) ratio is the ratio of fuel and air in the engine available for combustion .

It is denoted as  $A/F = m_a / m_f$  - (29)

- Heating value or Calorific value: It is defined as the energy released by the combustion of per unit quantity of the fuel and the products of combustion get cooled back to the initial temperature. It is called the higher calorific value when water vapors in the products of combustion remains in vapor form, The calorific value obtained is called lower calorific value.

- Power =  $2\pi NT/60$  in Watt - (30)

torque  $T = WR$  Nm

$W$  = Net weight (N),  $R$  = Radius (m),  $N$  in RPM

- Average Pressure ( Mean effective pressure,  $P_m$ ) : It is as a hypothetical pressure, which about to be acts on the piston during the power stroke.

$$\text{Power} = (P_m \cdot L \cdot A \cdot N/n) / 60 \quad - (31)$$

where

$P_m$  in pascals

$$A = \frac{\pi}{4} \cdot D^2$$

Brake power :

$$BP = 2\pi NT / 60 \quad - (32)$$

Brake mean effective pressure (bar):

$$BMEP = BP \cdot 60 / \left( \frac{\pi}{4} \cdot D^2 \cdot L \cdot \frac{N}{n} (\text{No. of cylinders}) \right) \quad - (33)$$

$$IP (\text{Nm}) = (\text{Area of PV diagram}) \quad - (34)$$

Indicated mean effective pressure :

$$IMEP = IP \cdot 60 / \left( \frac{\pi}{4} \cdot D^2 \cdot L \cdot \frac{N}{n} (\text{No. of Cylinder}) \right) \quad - (35)$$

The frictional means effective pressure (FMEP) can be find as

$$FMEP = IMEP - BMEP \quad - (36)$$

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- Frictional power :  
 $FP = IP - BP$  - (37)  
 $FHP = IHP - BHP$  - (38)  
 $BHP = IHP - FHP$
- Heat Balance :  
 i) Heat supplied by fuel =  $m_f \times C.V.$  - (39)  
 ii) Heat equivalent to useful work =  $BP$   
 $\% \text{ Heat equivalent to useful work} = \frac{\text{Heat equivalent to useful work}}{m_f \times C.V.} \cdot 100$  - (40)  
 iii) Heat in cooling water Jacket =  $m_w \cdot C_w \cdot (T_2 - T_1)$  - (41)  
 Where  $m_w$  = Jacket cooling water Kg/hr,  $C_w$  = specific heat of water in kJ/kg.K,  $T_2$  = water jacket outlet temperature and  $T_1$  = water jacket inlet temperature in K.  
 $\% \text{ Heat in cooling water Jacket} = \frac{m_w \cdot C_w \cdot (T_2 - T_1)}{m_f \times C.V.} \cdot 100$  - (42)
- Heat in Exhaust :  
 $\text{Specific heat of exhaust gas } C_{pex} = \frac{m_w \times C_w \times (T_4 - T_3)}{(m_f + m_a)(T_5 - T_6)}$  - (43)  
 $\text{Heat in exhaust gas } Q_{ex} = (m_f + m_a) \cdot C_{pex} (T_5 - T_{amb})$  - (44)  
 $\% Q_{ex} = \frac{Q_{ex}}{m_f \times C.V.} \times 100$  - (45)
- Heat to radiation and unaccounted  
 $\% Q_{rad \& unacc.} = \% \text{ Heat supplied by fuel} - \{ \% \text{ Heat equivalent to useful work} + \% \text{ Heat in cooling water Jacket} + \% \text{ Heat in exhaust gas} \}$  - (46)

### III. METHODOLOGY

#### 3.1 Simulation

A thermodynamic model can be used to simulate the compression, combustion and expansion processes as closed cycle period with the control volume approach. Wiebe function is used to fit the experimental mass burned profile. The molecular formula of diesel fuel is taken as  $C_{10}H_{22}$ , biodiesel  $C_{19}H_{34}O_2$  and ethanol  $C_2H_5OH$ . A computer program in matlab may be developed for the numerical solution of the equations used in the thermodynamic model described above. The main parameters

required from the engine were selected and the observed parameters are Engine load, Brake thermal efficiency, Fuel consumption rate, In-cylinder pressure, Emissions of CO,  $CO_2$ , NOx, HC, Exhaust temperature. Fuels considered for analysis are diesel and blends of diesel, biodiesel & ethanol. The properties taken under consideration are mentioned in table no.1. At certain Injection pressure variation ( $21^\circ, 23^\circ, 25^\circ, 27^\circ, 29^\circ$ ) the Pressure ratio variation (16,17,18) are to be considered in the model for simulation as future work.

#### 3.2 Engine Specification

A stationary kirloskar single cylinder engine's technical specifications as given in Table 1. are considered for mathematical model development which can be used for simulation purpose.

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Table 1: Engine specifications

Sr. No.	Parameter	Specification
1	Type	Research Engine test setup 1 cylinder, 4 stroke, Multifuel, VCR, 661cc
2	Bore	87.5
3	Stroke	110mm
4	Petrol mode	4.5 KW@ 1800 rpm, Speed range 1200-1800 rpm, CR range 6-10,
5	Diesel mode	3.5 KW, 1500 rpm, CR range 12-18 ( Std.17.5).Injection angle variation:0- 25 <sup>0</sup> BTDC
6	Dynamometer	Type eddy current, water cooled, with Dynamometer Arm Length 185mm,
7	Propeller shaft	With universal joints
8	Air box	M S fabricated with orifice meter and manometer
9	Fuel tank	Capacity 15 lit, Type: Duel compartment, with fuel metering pipe of glass. Fuel Pipe dia 12.40mm, loading unit Orifice Diameter 20.00mm , Orifice Coeff. of Discharge: 0.60
10	Calorimeter	Type Pipe in pipe
11	ECU	PE-3 Series, Model PE-3, 8400P. Includes pe-Monitor &pe-Viewer software.
12	Peizo sensor	Combustion: Range 350Bar, Diesel line: Range 350 Bar, with low noise cable
13	Crank angle sensor	Resolution 1 Deg, Speed 5500 RPM with TDC pulse.
14	Data acquisition device	NI USB-6210, 16-bit, 250ks/s.
15	Temperature sensor	Type RTD, PT100 and Thermocouple, Type K
16	Temperature transmitter	Input RTD PT100,Range 0–100 °C, 4–20 mA Output and Type two wire, Input-Thermocouple
17	Load sensor	Load cell, type strain gauge, range 0-50 Kg
18	Fuel flow transmitter	DP transmitter, Range 0-500 mm WC
19	Air flow transmitter	Pressure transmitter, Range (-) 250 mm WC
20	Software	“Enginesoft” Engine performance analysis software
21	Rotameter	Engine cooling 40-400 LPH; Calorimeter 25-250 LPH,
22	Pump	Type Monoblock
23	Overall dimensions	W 2000 x D 2500 x H 1500 mm



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### 3.3 Instrumentation

Product is supplied with best quality instruments. The eddycurrent dynamometer is SAJ,Pune make. The components like Open ECU (PE USA), Combustion pressure sensor (PCB Piezotronics, USA), Crankangle sensor(Kubler, Germany), Fuel flow transmitter(Yokogawa, Japan), Pressure transmitter (Wika, Germany), High speed data acquisition device (National instruments, USA) are of MNC grades.

The following parameters are assumed as constant

Sp. Gas Constant (kJ/kgK) : 1.00, Density of Air (kg/m<sup>3</sup>) : 1.17, Adiabatic process Index : 1.41, Polytrophic process Index : 1.14, Number Of Cycles : 10, Cylinder Pressure : 5, Smoothing: 2.

### 3.4 Fuel properties

The types of fuel used in the experimentation along with the properties in detail are described in table2. The percentage variation of diesel (fossils fuel) is decreased as the presence of biofuel (biodiesel and Ethanol) should increase. As per the engine operation, it is being observed that at due to viscosity, calorific value and cetane number higher percentage of biofuel is limitation. At higher percentage of biofuel the engine operation was not smooth. The maximum readings are taken in account to observe the performance of Engine. The blends prepared in percentage by volume are used in experimentation.

Table2: Fuel Properties

Sr. No.	Type of Blend	Lubricity WSD, $\mu\text{m}$	Density	Cetane Number	Lower Heating value KJ/Kg	Flash point	Cold filter plugging point (CFPP), °C
1	Diesel	324	85.21	52	42600	61	-9
2	Biodiesel	218	76.97	55.5	39760	126	-14
3	Ethanol	0	52.14	8	26805	13	-
4	B1- 75%D20%B5%E	243	849.6	51.68	41269	17	17
5	B2- 70%D20%B10%E	264	846.8	49.41	40524	16	7
6	B3 - 80%D15%B5%E	276	847.2	51.52	41414	16	13
5	B4- 65%D20%B5%E10%K	272	844.7	49.24	40668	15.5	4
7	Std. as per EN590	460 (max.)	820-845	51 (Min)	--	55 (min.)	Climate cond.

B- Biodiesel, D- Diesel, E- Ethanol, K- Kerosene

The diesel was purchased from local market of Sangli (M.S.).Karanja Seed biodiesel was purchased from GKVK (Gandhi Krishi, Vignana Kendra), Bengaluru, Karnataka. The Ethanol (Anhydrous 99% ) was purchased from Jalgaon (M.S.).

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The Diesel, Biodiesel & Ethanol properties are taken as standard not tested in lab.. The fuel property viscosity ( Saybolt viscometer ), Density and flash point are measured at labs of SSBT's COET ,Jalgaon (M.S.). The Cetane number and Calorific value are measured at Biodiesel Corporation of India, Baramati (M.S.).

### 3.5 Experimental Setup

In the present experimentation fuel blends B1,B2,B3,B4 and B5 is used. By using trans- esterification process, Karanja biodiesel was produced. All the fuels blends are properly mixed with continuous stirring using magnetic stirrer for about 10 to 15mins to obtain homogeneous mixture. The blends were tested on Research Engine Test Setup (Computerized) ...Code 240PE at Engine research lab, Apex Innovations Pvt. Ltd. Sangli( M.S.).The engine performance was measured using blends as mentioned below fig,2 Experimental setup.

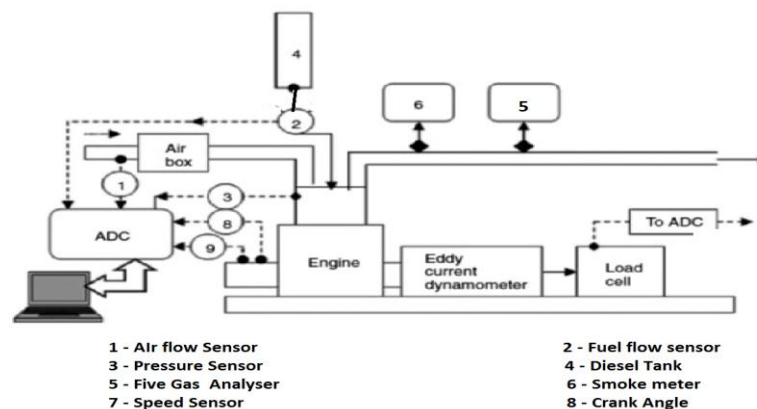


Fig.2 Experimental setup

### IV. CONCLUSION

A thermodynamic model is developed for analyzing the performance of the CI engine. This model is can be used for characterizing any CI engines, as diesel, biodiesel, ethanol or their blends. The model required to develop for varying pressure ratio and injection timing. This model will predict the engine performance characteristics closer to experimental results.

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