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Numerical Simulation of Compression Ignition Diesel Injection (CIDI) to investigate Performance parameters.

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Abstract*— *This paper describes the requirement of the Numerical simulation of compression ignition diesel injection by the use of computer language and it also compares the performance parameter for the biodiesel such as jatropha and karanja. The Engine test was carried out in 512 Army base workshops for the experimental validations. It was carried out on SAJ dynamometer which was installed in Engine test house for testing of Engines. Performance parameters such as Brake power, Brake Torque, Mechanical efficiency, Thermal Efficiency, Pressure vs Crank angle and Heat release Rate vs Crank Angle was taken from the Engine test house. The Program code for the Performance parameter of Engine was developed in C++ language. Then the simulation was carried out. The simulation results were compared and analyzed with the experimental results.

The final results was effective for compression ignition diesel injection. As we know that today's world is approaching towards computer simulated results, so the numerically simulated results will save time and money for engine testing house. Likewise we can approach towards numerical simulation for various types of Engines.

Keywords*— *Performance parameter, Compression Ignition Diesel Injection, Karanja biodiesel, and Numerical Simulation.

I. INTRODUCTION

Today world is totally depends on Natural Resources which is limited, so it is necessary to find alternatives to the natural resources. Bio-diesel can we produced from various naturally grown plants which can be alternative to diesel for producing high energy for heavy vehicles and machinery. This paper describes the production of karanja plant and extraction of karanja bio-diesel from it. Karanja plant can be grown on the land which has water scarcity. This plant once cultivated can produce the karanja oil for 15 years. During the latest study of karanja bio-diesel, it was found that the properties of the bio-diesel have similar physical properties to the diesel. Physical properties such as density, calorific value, cetane number and acid value was satisfactory similar to the diesel, when karanja bio-diesel produced by chemical reaction with the methyl ester. Performance parameters of Compression ignition Diesel injection (CIDI) of karanja bio-diesel was compared to the diesel while experimentally using both the diesel used in diesel engine during workshop period.

Mathematical model has been developed for the Numerical simulation of the performance parameter of Compression Ignition Diesel Injection (CIDI). Performance parameters such as Brake Torque, Brake horse power, Specific fuel consumption, Mechanical efficiency and Thermal efficiency. A program was developed in C++ language with the help of Mathematical model. Program developed was then simulated to obtain the result. The result obtained was then compared and analyzed with the experimental result.

The input to the program was Engine geometry, Fuel property and inlet- outlet condition. Outcome of the program will be Torque, Brake power, Specific fuel consumption, and Mechanical and Thermal efficiency.

II. LITERATURE RIVIEW

C McCartan, P McEntee, and R Blair has developed a model for turbocharged direct injection engine to simulate the performance parameters. That software also tested instaneous gas and thermodynamic behavior in the engine cylinder [1].A Pandey, P Sivakumar, M Nandgaonkar, and S Suresh studied the comparison of diesel, karanja biodiesel and jatropha biodiesel. In that they compared the exhaust gases emission and material wear and tear. Engine performance of biodiesel was slightly lower than diesel. And Emissions performance was slightly better than diesel [2]. M Nandgaonkar and A Pandey evaluated the performance parameter of biodiesel where they tested the wear and tear, and Emission gases. They result was good and effective as compare to diesel [3]. J Gupta and A Agarwal has studied the macroscopic and microscopic

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spray characteristics of diesel and karanja biodiesel blends. Effect of pressure on macroscopic spray characteristics such as spray penetration, spray area and cone angle were investigated in a constant volume spray chamber .Microscopic spray characteristics such as velocity distribution of droplets and spray droplet size were measured using PDI (phase Doppler interferometry) [4].

J Lahuerta, and S Samuel developed a simulation model to improve thermal efficiency of IC Engine for operating engine in various range such as cold start etc. This was formulated based on the thermal management strategy in an engine can allow the engine to work at different operating temperature in order to reduce heat transfer loss by ensuring optimum volumetric efficiency, efficient combustion and adequate safety margin durability of mechanical components [5].

L Li, Y Luan, Z Wang and J Deng develop a simulation model for free piston engine alternator integrative power system. the key design parameters such as reciprocating mass of the piston assembly, compression ratio, the ignition timing, the engine fuel consumption rate and power output. It predict the performance parameter using free piston engine [6]. V Jadhav, S kanchan, K Kavathekar and S Thipse optimized the IC Engine by varying the compression ratio, valve lift profile, intake plenium volume, runner length, spark-advance timing, fuel injection location, exhaust pipe length and catalytic converter selection. The simulation was developed and the performance parameter was simulated and compared with experimental [7].W Zottin, P Bacchin and A Gracia developed a numerical simulation model for carbon build –up and oil consumption in a heavy duty diesel engine. It was develop to optimize the engine and components behaviours which are mainly responsible for emission and wear and tear [8]. F Millio , E Pautasso, P Pasero and S Barbero studied and developed a numerical simulation model for Advanced EGR control system for Automotive diesel engine. The result of simulation and experimental was compared and was found to be satisfactory [9]. E Shoukry, S Taylor, N Clarke and P Famouri has developed a simulation model for parametric study of a two stroke Direct Injection Linear engine . Parameters such as rate of combustion, convection heat transferred inside the cylinders, friction forces, external loads, acceleration, velocity profile, compression ratio, and in cylinder pressures were modeled [10]. G Morin, E Nicouleau-Bourles, F Simon and O Prince has developed an methodology that allows to validate and optimize field reliability during development and before start of production. These was developed based on the environmental issues, reliability and developement time reduction [11]. J Arias, E Varela, R Perez and E Navarro has developed simulation model of scavenging process in a two stroke tourbocharged diesel engine [12].

E Benavides, J Perez, R Herrero and E Arroyo developed a numerical simulation model for injection process in a two stroke diesel engine and later validated [13]. Jae Soon Lee, Hee Gag Lee, and Nak Won Sung has developed numerical simulation model for prediction of volumetric efficiency of diesel engine. Simulation model contains factors such as fluid flow in intake and exhaust manifolds[14].T Minagawa, H Kosaka and T Kamimoto has developed a numerical simulation model for Ignition delay of diesel fuel spray. This includes factors such asspray axis and spray periphery [15].

III. SIMULATION APPROACH

Indicated mean effective pressure (IMEP):

Indicated mean effective mean pressure is hypothetical pressure generated inside the cylinder. It is calculated from the indicator's diagram. Indicators diagram is plotted with P-V variation inside the cylinder of Diesel engine throughout the cycle. The average pressure generated in the combustion chamber during the power stroke is considered.

It is work done during power stroke divided by displacement volume.

IMEP= (P2*V2-P1*V1)/Vs;

Friction Mean Effective pressure (FMEP):

FMEP is defined as the power loss in the Engine during Diesel cycle process. All the losses are mention below.

(i) Mean effective pressure (MEP) lost to overcome friction due to gas pressure behind the rings.

(ii) Mean effective pressure absorbed in friction due to wall tension of rings.

(iii) MEP absorbed in friction due to piston and rings.

(iv) Blow-by loss.

(v) MEP lost in overcoming inlet and throttling losses.

(vi) MEP absorbed to overcome friction due to the valve gear.

(vii) MEP lost in pumping.

(viii) MEP absorbed in bearing friction.

(ix) MEP absorbed in overcoming the combustion chamber and wall pumping losses.

Brake Mean effective pressure (BMEP):

Brake Mean effective pressure is the average power output produced due to the pressure imposed uniformly on piston from the top to bottom of each power stroke.

As we know that Indicated mean effective pressure is the sum of Brake mean effective pressure and Friction mean effective pressure.

IMEP=BMEP+FMEP;

So from above relation we can find out the Brake mean effective pressure.

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Brake Mean effective pressure= Indicated Mean effective pressure-friction mean effective pressure.
BMEP=IMEP-FMEP;

Torque (T):

The tendency of a force to cause the rotational motion of a body. It is a twist turning force on an object.
Torque can be calculated from BMEP as follows.
T= (BMEP*A*L*K)/2*3.14;

Mechanical Efficiency (nv);

Mechanical Efficiency measures the performance of a engine in transforming the energy and power that is input to the engine into an output force.
nv=(BMEP)/(IMEP);

Thermal Efficiency (nt);

The efficiency of a engine measured by the ratio of work done by it to the heat supplied to it.
nt=W/Qh;
Qh=mf*CV.
mf=mass fuel flow rate.
CV=calorific value of fuel.

Brake Specific Fuel Consumption:

BSFC is a measure of the fuel efficiency of a engine which burns fuel and produce shaft power.
BSFC is the rate of fuel consumption to the power produced.
BSFC= mf/bp;
mf= fuel mass flow rate;
bp=brake power;

Pressure vs Crank angle (P vs CA):

We have to Measure Pressure inside the cylinder throughout diesel cycle.
Two-zone model views the entire combustion in to burned zone and unburned zone.
After amount of mass fraction burned inside the combustion chamber was found out, the next important parameter to find out is instantaneous Volume of the cylinder. This volume can be obtained by the equation in relation with the crank angle as:
Volume at any instant can be calculated by below given formula.

$$V_\Theta = V_{disp} * \left[\frac{r}{r-1} - \frac{1-\cos\theta}{2} + \frac{L}{S} - \frac{1}{2}\sqrt{\left(\frac{2L}{s}\right)^2 - \sin^2\theta} \right]$$

Mass of burned fuel at any instant is given by:

$$x_b(\theta) = 1 - \exp\left[-a\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{m+1} \right] \qquad 7$$

a. Compression process
During Compression process the pressure developed in cylinder is calculated by relating the instant volume to instant pressure. Same applies for Temperatures. Relation is shown below.

$$P_c(\theta) = P_{ivc}\left(\frac{V_{ivc}}{V(\theta)}\right)^{\gamma_c} \qquad 1$$

$$T_c(\theta) = T_{ivc}\left(\frac{V_{ivc}}{V(\theta)}\right)^{\gamma_c - 1} \qquad 2$$

b. Expansion process
During Expansion process the pressure developed in cylinder is calculated by relating the instant volume to instant pressure. Same applies for Temperatures. Relation is shown below.

$$P_e(\theta) = P_3\left(\frac{V_3}{V(\theta)}\right)^{\gamma_e} \qquad 4$$

$$T_e(\theta) = T_3\left(\frac{V_3}{V(\theta)}\right)^{\gamma_e - 1} \qquad 5$$

c. Combustion process
Pressure during the combustion process is calculated by using interpolating Compression and Expansion process.

$$P_{comb}(\theta) = (1 - x_b(\theta)).P_c(\theta) + x_b(\theta).P_e(\theta) \qquad 8$$

d. And finally the pressure and temp outlet is calculated.

$$P_3 = P_2\left(\frac{T_3}{T_2}\right) \qquad 6$$

IV. EXPERIMENTAL APPROACH

- **a. Dynamometer**
- **b. Pressure Transducer**
- **c. Crank angle encoder**
- **d. DAQ**
- **e. Speed acquisition**

a. Dynamometer:

A dynamometer is a machine used for measuring the brake power and torque required to operate a drive engine.
Hydraulic dynamometer works on the principle of dissipating the power in fluid friction.
Hydraulic dynamometer consists of an impeller or rotating members coupled to the output shaft.

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The impeller in this dynamometer rotates in a casing filled with the fluid. Due to the centrifugal force developed in the outer casing, tends to revolve with the outer casing but it is resisted by a torque arm supporting the balance pressure developed.

The pressure difference developed is used measure the torque developed in engine. Then with the help of torque brake power is calculated.

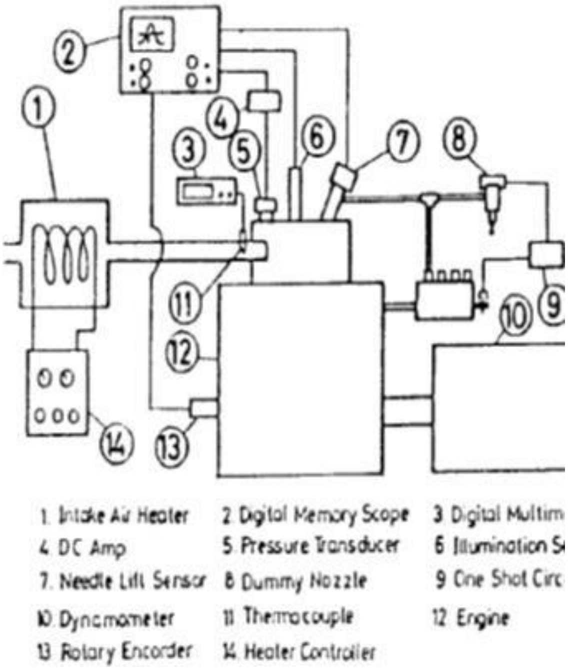

Fig.1: Schematic Diagram of Experimental Apparatus

b. Pressure Transducer:

To measure the pressure inside the cylinder, we have to place the pressure transducer in drilled hole near inlet valve. Then pressure transducer sends electrical signal to the amplifier which is then converted into digital form. The measure pressure is then correlated with the crank angle.

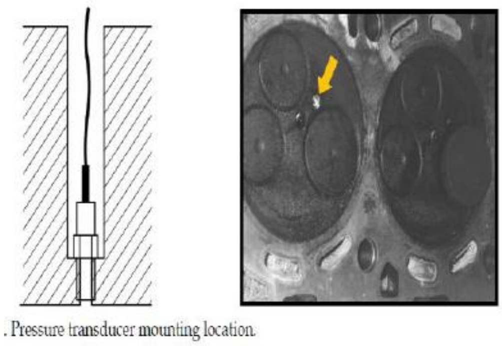

Fig.2: Pressure transducer mounting location

c. Crank angle encoder:

The crankshaft angle was measured with a resolution of 1 degree that is 720 data points per engine cycle. The sensor was mounted at the free end of the crankshaft. The crank angle data was sampled and converted to angle domain. Reference mark is marked on the one of the teeth. The position of crank is measured by passing rays on the incremental mark and the difference between the marked and rays deflected are calculated. Thus the position is found out.

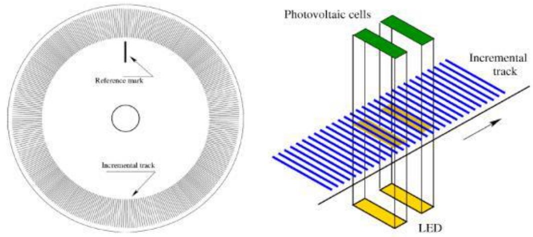

Fig.3: Optical Crank angle encoder.

d. Data Acquisition System:

A data acquisition system could be a device designed to live numerous parameters. the information acquisition system is often natural philosophy primarily based and it's fabricated from hardware and code. The hardware half encompass sensors, signal conditioners and information code.

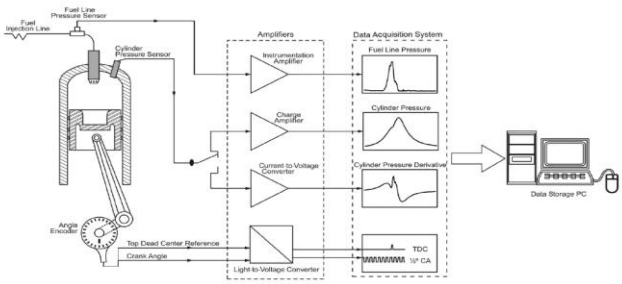

Fig.4: data acquisition system.

e. Speed Acquisition:

The speed is measured digitally at the shaft of the dynamometer by the use of a toothed wheel and a pulse generator. eg Tachometer, Eddy current.

The unit consists of a small permanent magnet with a coil round it. This magnetic pick up is placed near a metallic toothed rotor whose speed is to be measured. As the shaft rotates, the teeth pass in front of the pick-up and produce a change in the reluctance of the magnetic circuit. The field expands or collapses and a voltage is induced in the coil. The frequency of the pulses depends upon the number of teeth on the wheel and its speed of rotation. Since the number of teeth is known, the speed of rotation can be determined by measuring the pulse frequency. To accomplish this task, pulse is amplified and squared, and fed into a counter of frequency measuring unit.

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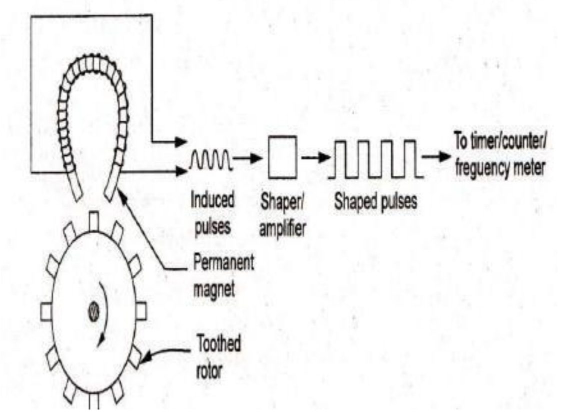

Fig.5: Magnetic RPM measuring instrument.

The following parameters will be determine by the Experimental setup step by step.

I. Measurement of N-(RPM).
II. Measurement of Brake Torque
III. Measurement of Brake Power.
IV. Measurement of Cylinder Pressure.
V. Measurement of Crank angle.
VI. Measurement of Brake specific Fuel Consumption.
VII. Measurement of Thermal Efficiency.
VIII. Measurement of Mechanical efficiency.

KARANJA BIO-DIESEL

An ever increasing demand of fuels has been a challenge for today's scientific employees. The fuel resources square measure dwindling day by day. Biodiesel appear to be an answer for future.

Pongamia pinnata is an exact supply of material as a result of its straightforward convenience in wild. Genus Pongamia pinnata is drought resistant, semi-deciduous, gas fixing herbaceous plant tree. Its height is regarding 16-20 meters with an oversized portion on top of the bottom that spreads equally wide. When trans-esterification of fossil fuel shows wonderful properties like hot worth, iodine range, cetane range and definite quantity etc.

A thick yellow – orange to brown oil is extracted from seed. Regarding pure gold of yield is obtained by mechanical expeller. The oil has bitter check and disagreeable aroma, thus it's thought of as a non-edible one. In our country this oil is used as a fuel for preparation and lamps. Additionally oil is employed as material, chemical and in soap creating industries. The oil has meditative worth within the treatment of rheumatism and in skin diseases.

Table.1: Physical and Chemical properties of karanja oil.

Physico-chemical Properties of Pongamia pinnata -crude oil

Property	Unit	Value
Color	-	Yellowish red
Odor	-	Characteristic odd odor
Density	gm/cc	0.924
Viscosity	mm^2/sec	40.2
Acid Value	mg/KOH	5.40
Iodine Value	-	87
Saponification Value	-	184
Calorific Value	Kcal/KG	8742
Specific Gravity	-	0.925
Unsaponifiable matter	-	2.9
Flash Point	^0C	225
Fire Point	^0C	230
Cloud Point	^0C	3.5
Pour Point	^0C	-3
Boiling Point	^0C	316
Cetane Number	-	42
Copper strip Corrosion	-	No Corrosion observed
Ash Content	in %	0.07

Table.2: Properties of karanja bio-diesel.

Properties of Karanja Methyl Ester-

Property	Unit	ASTM Test Method	Karanja Biodiesel	Diesel
Density	gm/cc	D1498	0.860	0.824
Calorific value	Kcal/KG	D240/ D 4868	3700	4285
Cetane Number	Number	D613	41.7	49
Acid Value	mg/KOH	D664	0.46	0.36
Iodine Value	Number	D1510	91	-
Water and sediments	% vol, max	D2709	0.005	-

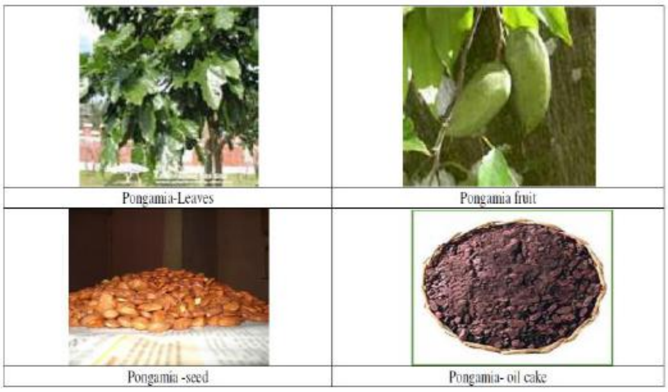

Fig.6: karanja leaves, pongamia fruit, seeds and oil cake.

V. RESULTS AND DISCUSSION

The result of numerical simulation and its validation are discussed below.

In study the performance parameter of CIDI engine is investigated with diesel and karanja bio-diesel. Performance parameter such as Torque, Brake horse power, Specific fuel consumption, Mech effic and Ther effic. Numerically simulated data is validated with the experimental data as shown below.

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a. **Experimental data of CIDI Engine for diesel fuel.**

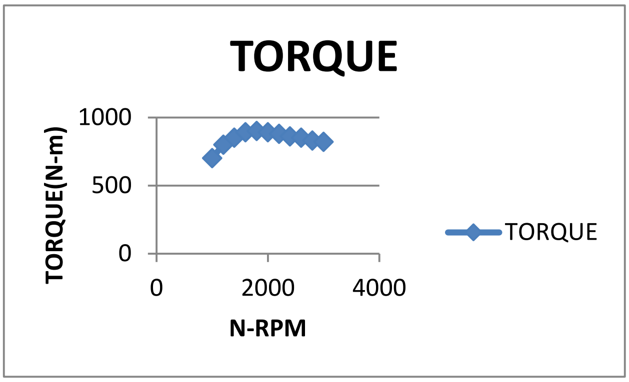

Fig.a:Torque vs RPM.

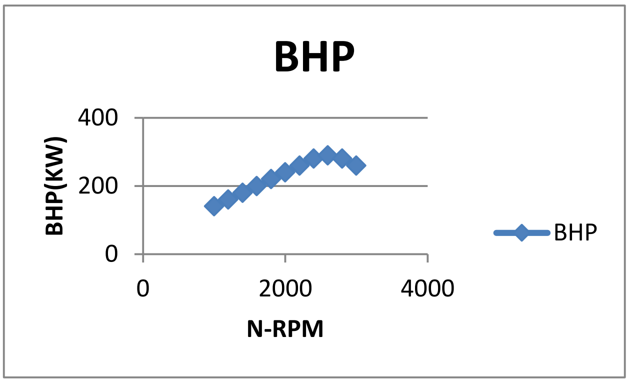

Fig.b: Brake Horse Power vs RPM.

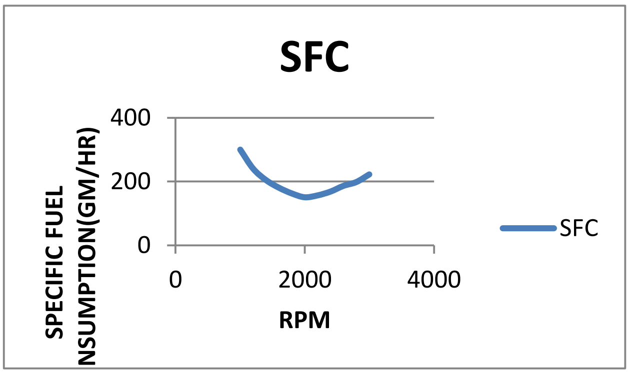

Fig.c: Specific fuel consumption vs RPM.

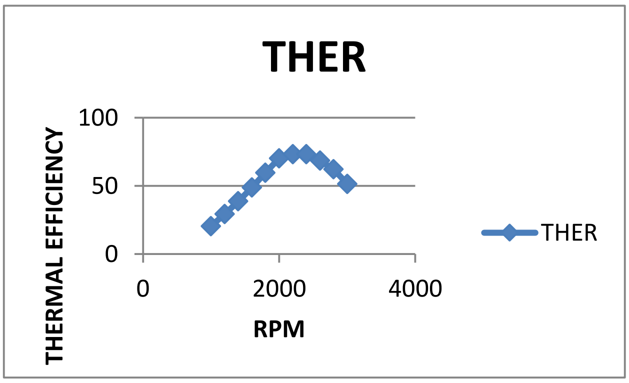

Fig.d:Thermal efficiency vs RPM.

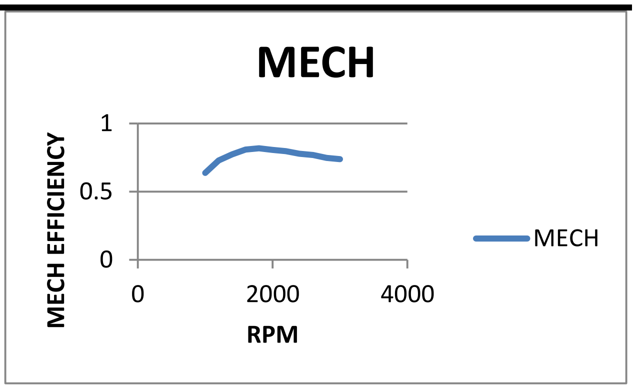

Fig.e:Mechanical efficiency vs RPM.

b. **Comparison of Experimental and Simulated data for Diesel fuel.**

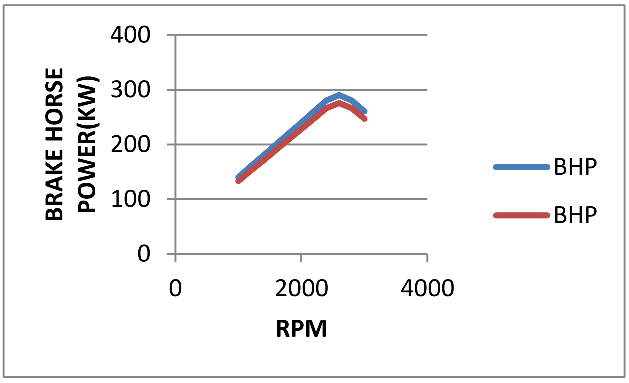

Fig.f: Brake Horse Power vs RPM.

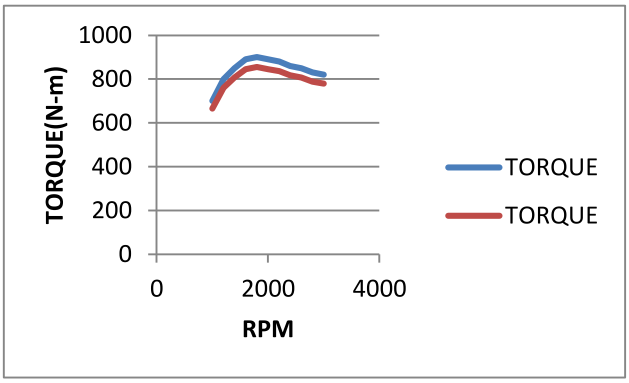

Fig.g:Torque vs RPM.

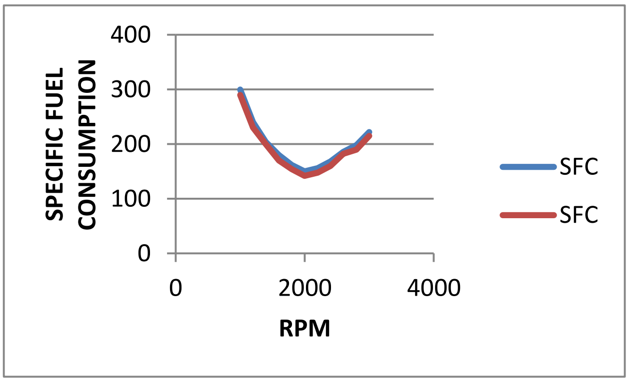

Fig.h: Specific fuel consumption vs RPM.

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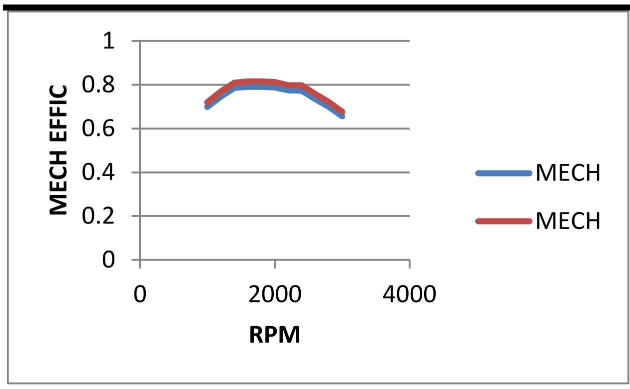

Fig.i:Mechanical efficiency vs RPM.

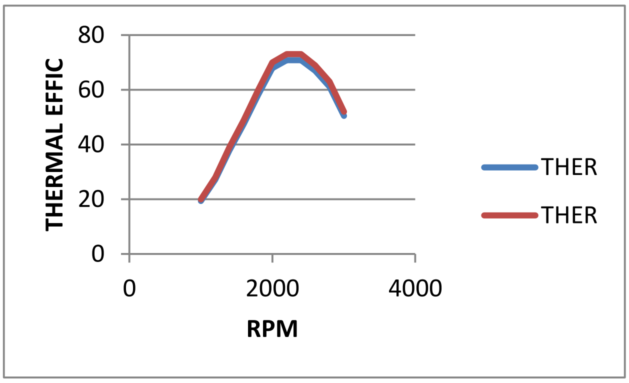

Fig.j:Thermal efficiency vs RPM.

The numerically simulated data is plotted with the experimental data for performance parameter. The graph shows the effective result. The error between them is typically less than 5% which is good indication.

c. Experimental data of CIDI Engine for Karanja Bio-diesel.

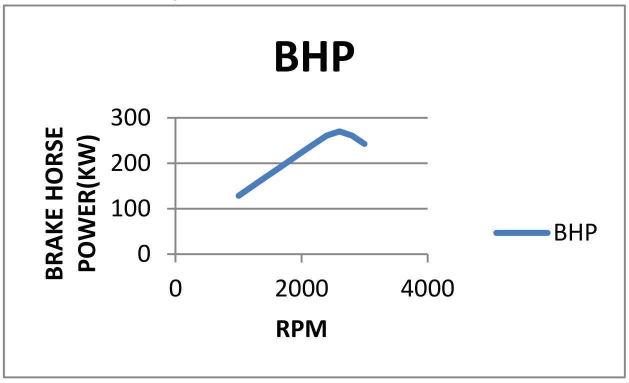

Fig.k: Brake Horse Power vs RPM.

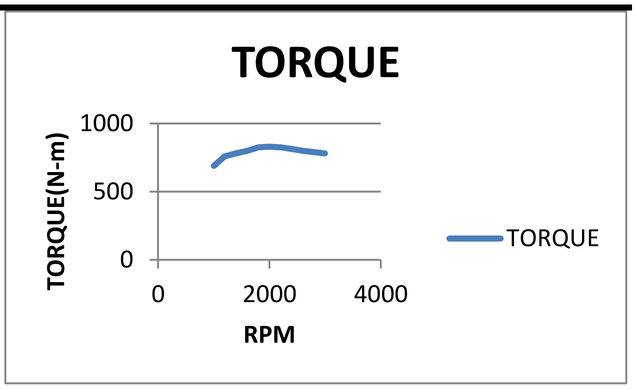

Fig.l:Torque vs RPM.

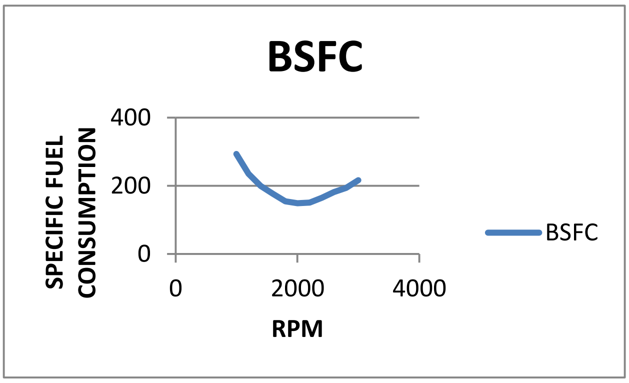

Fig.m: Specific fuel consumption vs RPM.

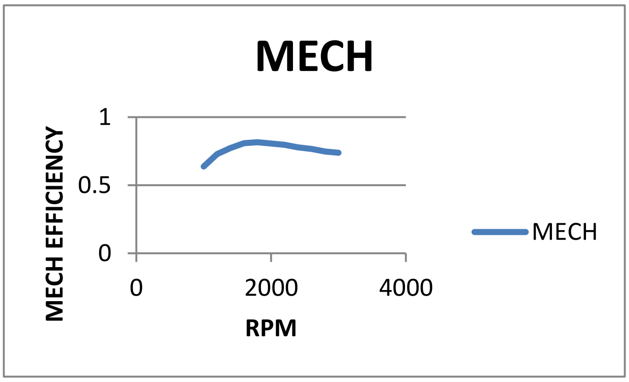

Fig.n:Mechanical efficiency vs RPM.

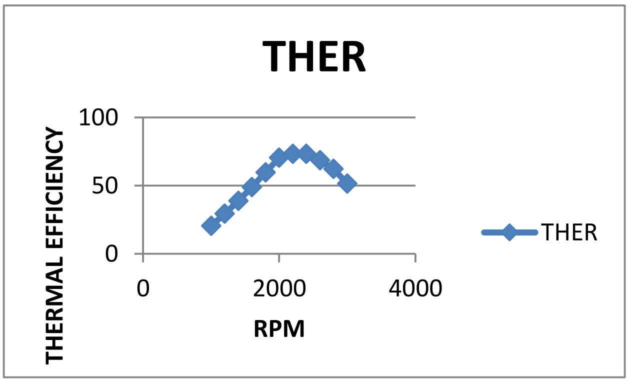

Fig.o:Thermal efficiency vs RPM.

d. Comparison of Experimental and Simulated data

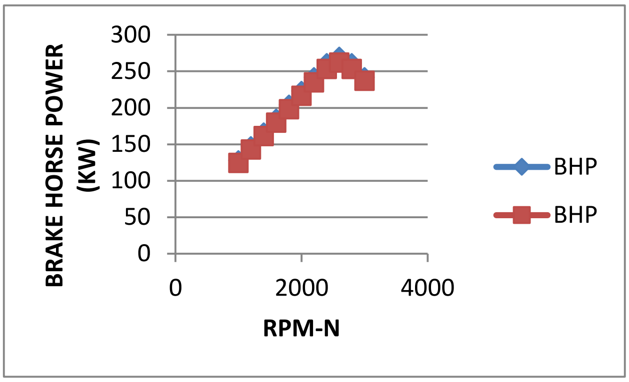

Fig.p: Brake Horse Power vs RPM.

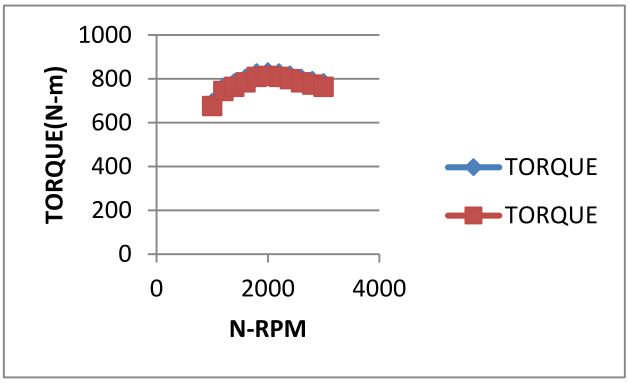

Fig.q:Torque vs RPM.

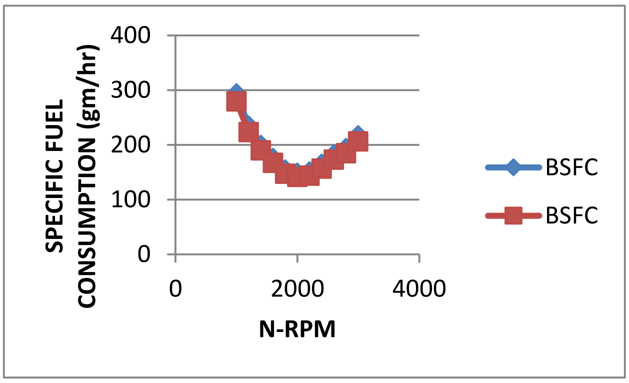

Fig.r: Specific fuel consumption vs RPM.

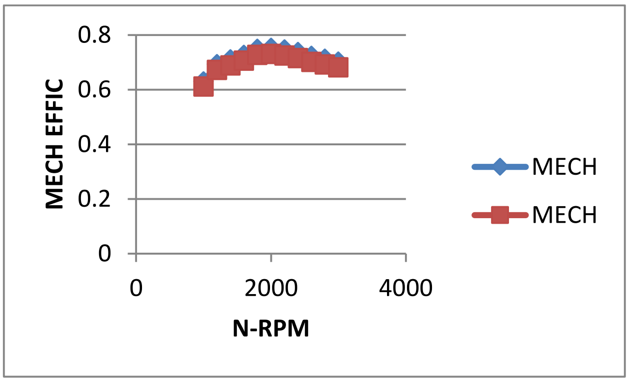

Fig.s:Mechanical efficiency vs RPM.

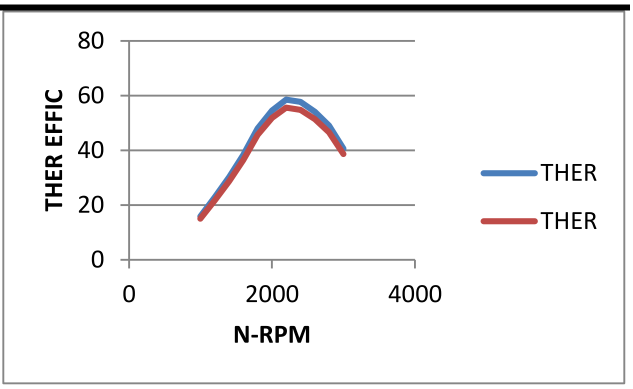

Fig.t:Thermal efficiency vs RPM.

The numerically simulated data is plotted with the experimental data for performance parameter. The graph shows the effective result. The error between them is typically less than 5% which is good indication.

VI. CONCLUSION

A Numerical simulation model was developed to investigate the performance parameter of Compression Ignition Diesel Injection (CIDI). Performance parameter such as Torque, brake horse power, specific fuel consumption, etc is formulated in the form of mathematical model. The model predicts the effective results, which has error percentage less than 5.

Karanja bio-diesel has slightly lower calorific value as compare to diesel.The comparison of biodiesel and karanja bio-diesel shows the comparative similar performance parameter, which shows that karanja bio-diesel can be a best alternative.

REFERENCES

[1] McCartan C., McEntee, P., Fleck, R., Blair, G. et al., "Computer Simulation of the Performance of a 1.9 Litre Direct Injection Diesel Engine," SAE Technical Paper 2002-01-0070, 2002

[2] Pandey, A., Sivakumar, P., Nandgaonkar, M., and Suresh, S., "Comparison and Evaluation of Engine Wear, Combustion and Emissions Performance between Diesel, Karanja and Jatropha Oil Methyl Ester Biodiesel in a 780 hp Military Diesel Engine," SAE Technical Paper 2014-01-1395, 2014

[3] Pandey, A. and Nandgaonkar, M., "Comparison and Evaluation of Wear, Performance and Emission of Diesel, Karanja Oil Biodiesel and JP-8 in a Military 585 kW CIDI Engine," SAE Technical Paper 2013-01-2658, 2013

[4] Gupta, J. and Agarwal, A., "Macroscopic and Microscopic Spray Characteristics of Diesel and Karanja Biodiesel Blends," SAE Technical Paper 2016-01-0869, 2016

[5] Lahuerta, J. and Samuel, S., "Numerical Simulation of Warm-Up Characteristics and Thermal

https://dx.doi.org/10.22161/ijaers.5.4.5 *ISSN: 2349-6495(P) | 2456-1908(O)*

Management of a GDI Engine," SAE Technical Paper 2013-01-0870, 2013

[6] Li, L., Luan, Y., Wang, Z., Deng, J. et al., "Simulations of Key Design Parameters and Performance Optimization for a Free-piston Engine," SAE Technical Paper 2010-01-1105, 2010,

[7] Jadhav, V., Kanchan, S., Thipse, S., Kavathekar, K. et al., "Optimizing and Validating the Engine Performance and Emission Parameters on Engine Dynamometer through 1D Simulation of a Multi-Cylinder CNG Engine," SAE Technical Paper 2016-28-0102, 2016

[8] Zottin, W., Bacchin, P., and Garcia, A., "Numerical Simulation Study of Carbon Build-up and Oil Consumption in a Heavy Duty Diesel Engine," SAE Int. J. Engines 5(3):1477-1486, 2012,

[9] Millo, F., Pautasso, E., Pasero, P., Barbero, S. et al., "An Experimental and Numerical Study of an Advanced EGR Control System for Automotive Diesel Engine," SAE Int. J. Engines 1(1):188-197, 2009

[10] Shoukry, E., Taylor, S., Clark, N., and Famouri, P., "Numerical Simulation for Parametric Study of a Two-Stroke Direct Injection Linear Engine," SAE Technical Paper 2002-01-1739, 2002

[11] Morin, G., Nicouleau-Bourles, E., Simon, F., and Prince, O., "Reliable Diesel Engine Design Based on a New Numerical Method," SAE Technical Paper 2005-01-1762, 2005

[12] Arias, J., Varela, E., Pérez, R., Navarro, E. et al., "Numerical Simulation of the Scavenging Process in a Two Stroke Turbocharged Diesel Engine," SAE Technical Paper 2001

[13] Benavides, E., Pérez, J., Herrero, R., and Arroyo, E., "Numerical Simulation of the Injection Process in a Two Stroke Diesel Engine," SAE Technical Paper 2000-01-0291, 2000,

[14] Jae Soon Lee, Hee Gag Lee, Nak Won Sung Numerical Study on the Prediction of Volumetric Efficiency of Diesel Engine SAE Technical paper

[15] Minagawa, T., Kosaka, H., and Kamimoto, T., "A Study on Ignition Delay of Diesel Fuel Spray via Numerical Simulation," SAE Technical Paper 2000-01-1892,