International Journal of Innovative Technology and Exploring Engineering (IJITEE)
ISSN: 2278-3075 (Online), Volume-3 Issue-2, July 2013

Computer Simulation of CI Engine for Diesel and Biodiesel Blends

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Abstract-Among the alternative fuels, biodiesel and its blends are considered suitable and the most promising fuel for diesel engine. The properties of biodiesel are found similar to that of diesel. Many researchers have experimentally evaluated the performance characteristics of conventional diesel engines fuelled by biodiesel and its blends. However, experiments require enormous effort, money and time. Hence, a cycle simulation model incorporating a thermodynamic based single zone combustion model is developed to predict the performance of diesel engine. A comprehensive computer code using "C" language was developed for compression ignition (C.I) engine. Combustion characteristics such as cylinder pressure, heat release, heat transfer and performance characteristics such as work done, brake power and brake thermal efficiency (BTE) were analyzed. On the basis of first law of thermodynamics the properties at each degree crank angle was calculated. The simulated combustion and performance characteristics are found satisfactory with the experimental results.

Keywords: - Biodiesel, Numerical modeling, simulation.

I. INTRODUCTION

Modeling compression ignition engine depends on characteristics of fuel. It is a process of designing a model of real system and conducting experiment with it for the purpose of understanding the behavior of the system. The Numerical Model of a diesel engine can be regarded as an explanation of real engine operation, which combines mathematical relation between the relative components, can be used to simulate the dynamic process of diesel engine. A clear overview of engine operation is helpful to understand the modeling of a real diesel engine. It serves as a tool for better understanding of combustion and its effect on engine, so as to build up more strong real systems. Computer simulation has contributed enormously towards new evaluation in the field of internal combustion engines. Mathematical tools have become very popular in recent years owing to the continuously increasing improvement in computational power. Diesel engines occupy a prominent role in the present transportation and power generation sectors. There have been many methods tried and are in use to reduce pollutant emissions from a diesel engine. The main options to reduce pollutants are the usage of bio-fuels and adopting some modifications to the combustion process. Diesel engine simulation models can be used to understand the combustion performance; these models can reduce the number of experiments.

From the point of view of protecting the global environment and the concern for long-term supplies of conventional diesel fuel, it becomes necessary to develop alternative fuels that give engine performance at par with diesel. Among the alternative fuels, biodiesel holds good promises as an eco-friendly alternative fuel [1]. Vegetable oil obtained from non edible sources are considered promising alternate fuel for compression ignition (CI) engine compared to their edible counterpart due to the food vs. fuel controversy. Engine performances using various sources of biodiesel viz., (a) salmon oil [2]; (b) rapeseed oil [3–5] ; (c) rubber seed oil [6]; (d) tobacco seed oil [7] ; (e) sunflower seed oil [8]; and; (f) soybean oil [9]; (g) jatropha curcus oil [10]; (h) karanja oil [11] were studied. As stated above, researchers have experimentally evaluated the performance characteristics of conventional diesel engines fuelled by biodiesel and its blends. However, experiments require enormous effort, money and time. A realistic numerical simulation model could reduce such effort. Numerical simulation based on mathematical modeling of diesel engine processes have long been used as an aid by design engineers to develop new design concepts. The present study describes a cycle simulation model. This thermodynamic based model follows the changing thermodynamic state of the working fluid through the engine intake, compression, combustion, expansion and exhaust processes for predicting the performance of a diesel engine fuelled by diesel and also the different blends of diesel and biodiesel. The model predicts the performance of a CI engine in terms of brake power and brake thermal efficiency for all the fuels considered for the present study. Fuel properties [11] and the engine design and operating parameters are specified as inputs to the model.

The purpose of this project i.e. Numerical Modeling of CI Engine is to determine the effects of fuelling a diesel engine with diesel and bio-diesel fuel blends. The investigation has been done on 100% diesel fuel and 20% bio-diesel blend with diesel. The results are the compared with the results get from experimentations.

Some specific objects are to evaluate the performance of,

1. Engine output.
2. Numerical Modeling results.
3. To predict the net heat release for B20.
4. To investigate the output parameters such as temperature, pressure, heat release etc.

II. BIODIESEL CHARACTERISTICS

2.1 Transesterification of Vegetable Oil

The vegetable oil is transesterified using methanol in the presence of sodium hydroxide (NaOH) as a catalyst (Figure 1 and 2). The parameter involved in the processing such as catalyst amount, molar ratio of alcohol to oil, reaction temperature and reaction time are optimized.

Manuscript published on 30 July 2013.
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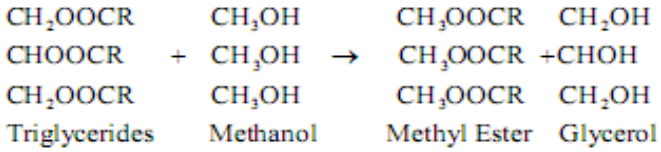

Figure 1: - Transesterification chemistry of vegetable oil

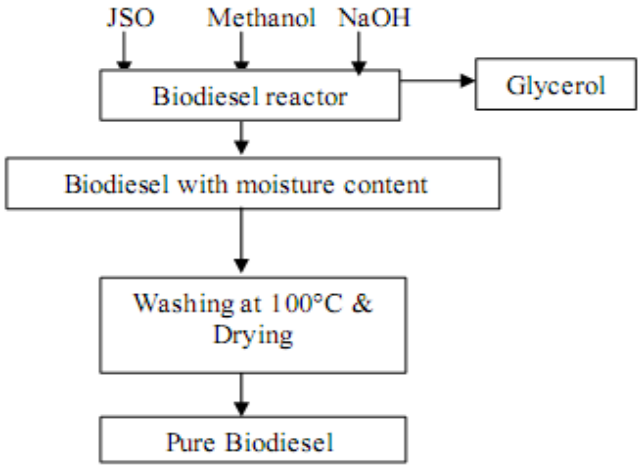

Figure 2: - Transesterification process of Jatropha seed oil

Known quantity of vegetable oil is taken in a biodiesel reactor. A water-cooled condenser and a thermometer with cork are connected to the side openings. The required amount of catalyst (NaOH) is weighed and dissolved completely in the required amount of methanol by using a stirrer to form sodium methoxide solution. The oil is then warmed by placing the reactor in water bath maintained at the selected temperature. The sodium methoxide solution then added into the oil and stirred vigorously by means of a mechanical stirrer. The required temperature is maintained throughout the reaction time and the reacted mixture is kept in the separating drum. The mixture is then allowed to separate and settled down by gravity settling into a clear, golden liquid biodiesel on the top with the light brown glycerol at the bottom. The glycerol was drained off from the separating drum leaving the biodiesel at the top. This pure biodiesel was measured on weight basis and the important fuel and chemical properties were determined (Table 1).

In this study, diesel and biodiesel was used as a fuel for conventional engine. Biodiesel and diesel with 20% and 80% by volume was mixed thoroughly and thus a stable mixture (hereafter referred as biodiesel) was prepared.

Table1: Properties of diesel fuel, B100 and B20

Properties	**Diesel Fuel (DF)**	**Biodiesel (B100)**	**20%DF/80 %B100 (B20)**
Density @15°C (kg/m^3)	830	880	840
Viscosity @40°C (cSt)	2.8	4.6	3.15
Flash point (°C)	55	170	80
Cetane number	45	50	46
Lower Heating Value	42	36	40.5

III. EXECUTION OF PROGRAM AND VARIOUS EXPRESSIONS USED FOR MODEL

3.1 Basic Input Data

The program developed here predicts the combustion characteristics like pressure, temperature, heat release and performance characteristics such as brake thermal efficiency and brake power. Thus the basic input for the program is the value of initial pressure, temperature corresponding to the crank angle, for a selected range of combustion cycle.

The range chosen here is from 25^0 BTDC to 25^{O}ATDC. Provision is made in the program, to read pressure-crank angle data, at an interval of 1^0 crank angles. Other data supplied initially in the program, is regarding engine dimensions, properties of fuel and properties fluid in the combustion chamber. These values are as follows:

B Cylinder bore = 0.0875 m
S Stroke = 0.110 m
CR Compression ratio = 18
RPM Rotations per minute = 1500 rpm
L Connecting rod length = 0.2 m
HV heating value of fuel = 10,500 Kcal/kg
= 44100 kJ//Kg.

An estimate of temperature of piston, cylinder-wall and cylinder head is made. These values are given in the program as below:

Ti: - Initial temperature = 311K
Tw: - Cylinder wall temperature = 900K

3.2 Calculation of Cylinder Volume at Various Crank Angle

Four stroke internal combustion engines are currently produced in two configurations reciprocating piston and rotary.

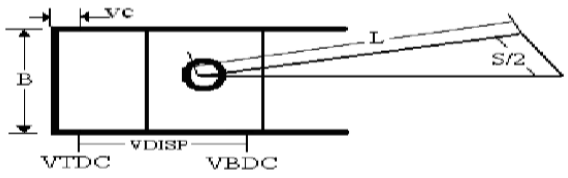

Figure 3: - Geometry of reciprocating piston engine

The basic geometry of reciprocating piston engine is shown in figure 3. It is describe in terms of cylinder bore B, length of stroke S, length of connecting rod L and compression ratio r,. The displacement volume V_{disp} is swept out as the piston move from bottom dead centre to top dead centre.

Now,

$$V_{disp} = V_{bdc} - V_{tdc} = \frac{\Pi}{4} B^2 S$$

Since

$$r = \frac{V_{bdc}}{V_{tdc}}$$

we may write

$$V_{bdc} = \left[\frac{r}{r-1}\right] * V_{disp}$$

$$V_{tdc} = \left[\frac{1}{r-1}\right] * V_{disp}$$

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With θ denoting the angular displacement of the crank from BDC, the volume V(θ) at any crank angle is represented by

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

3.3 Wiebe Heat Release Model

The Wiebe heat release pattern is based on the exponential rate of the chemical reactions. In this model; it is assumed that all the fuel is injected before the end of ignition delay period itself.

The fraction of heat-released pattern is expressed by non-dimensional equations

$$X = 1 - \exp\left[-a\left(\frac{\theta-\theta i}{\Delta\theta c}\right)^{m+1}\right]$$

By differentiating above equation, the rate of heat release can be expressed as

$$Q = a(m+1)\left(\frac{\theta-\theta i}{\Delta\theta}\right)^{m}\exp\left[-a\left(\frac{\theta-\theta i}{\Delta\theta c}\right)^{m+1}\right]$$

The above equation can be expressed in terms of KJ/$^{\theta}$CA as follows

$$\frac{dQc}{d\theta} = a(m+1)\left(\frac{Qav}{\Delta\theta c}\right)^{m}\left(\frac{\theta-\theta i}{\Delta\theta}\right)^{m}\exp\left[-a\left(\frac{\theta-\theta i}{\Delta\theta c}\right)^{m+1}\right]$$

Where

$\Delta\theta_c$ duration of combustion in $^{\theta}$CA.
θ crank angle at any instant.
θ_i crank angle at the start of combustion.
Q_{av} heat release per cycle in KJ.
a This is the parameter which characterizes the completeness of combustion. Wiebe assume Xmax=0.990 and hence, a = 6.908m

It is also a parameter characterizing the rate of combustion. The small value of m means a high rate at the beginning of the combustion, while large value of m m means high rate by the end of the combustion,

3.4 Heat Transfer Process

Heat transfer is must in IC engine to maintain cylinder walls, cylinder heads and piston faces at safe operating temperature. Heat is transfer from or to the working fluid during every part of each cycle, and the net work done by the working fluid in one complete cycle is given by

$$W_{net} = \oint\left(p + \frac{\Delta P}{2}\right)\Delta V$$

Where Δp is the pressure change inside the cylinder as a result of piston motion, combustion, flow into or out of the cylinder and heat transfer.

The pressure change Δp due to heat transfer is given by

$$\frac{\Delta p}{p} = \frac{h_c A(T_w - T)}{MC_v T}\Delta T$$

Where

h_c Heat transfer coefficient.
A interior surface area of engine volume.
T_w interior surface temperature.
M mass of working fluid.
C_V working fluid specific heat.
T working fluid temperature

The heat transfer between the working fluid and the interior surface is by forced convection, and the value of h_c depends on the gas velocity at the surfaces. As little is known about the gas motion inside; it is common to use some empirical formulae for calculating h_c. The formula of Pflaum is widely used.

Pflaum equation

$$h_c = \sqrt{p_g}T_g f(C_m)\ Kcal/m^2 h^o C$$

Where

$$f(C_m) = 3 \pm 2.57\left[1 - e^{-(1.5-0.416C_m)}\right]$$

The positive sign is used if

$$C_m \geq 3.6\ m/s$$

And the negative sign is used if,

$$C_m \leq 3.6\ m/s$$

3.5 Friction Calculations

Frictional losses affect the maximum brake torque and the minimum brake specific fuel consumption directly and are often a criterion of good engine design. These losses not only reduce the power but also influences the size of the coolant systems. The mean effective losses of power due to friction in different moving parts are calculated by using the following empirical relations.

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

$$F_{mep1} = 0.42\ x\ (p_a - p_{imf})\ x\ \frac{S}{B^2}\ x\ (0.0888\ C_r + 0.182\ C_r^{1.33-0.394C_m/100})\ x\ 10$$

Where,

p_a atmospheric pressure (bar),
p_{imf} manifold vaccume (bar),
S Stroke (mm),
B Bore (mm),
r compression ratio,
C_m mean piston speed (m/min)

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

$$F_{mep2} = 10\ x\ \frac{0.377\ S\ n_{pr}}{b^2}$$

where

n_{pr} number of rings

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

$$F_{mep3} = 12.85\ x\ \frac{P_{sl}}{BS}\ x\ \frac{100\ C_m}{1000}$$

where

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P_{sl} Piston skirt length (mm)

(iv) Blow-by loss

$$F_{mep4} = \sqrt{p_a - p_{imf}} \text{ x } \left[0.212\, r^{0.4} - (0.0345 + 0.0001055r) \text{ x } \left(\frac{N}{1000}\right)^{1.185}\right]$$

where

N piston speed in rpm

(v) MEP lost in overcoming inlet and throttling losses

$$F_{mep5} = \frac{P_e}{2.75} + p_{imf}$$

where,

P_e exhaust gas pressure

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

$$F_{mep6} = 0.226 \text{ x } \left(30 - \frac{4N}{1000}\right) \text{ x } \frac{GH^{1.75}}{B^2S}$$

where

G number of intake valve/cylinder

H intake valve diameter

(vii) MEP lost in pumping

$$F_{mep7} = 0.0275 \text{ x } \left(\frac{N}{1000}\right)^{1.5}$$

(viii) MEP absorbed in bearing friction

$$F_{mep8} = 0.0564 \text{ x } \left(\frac{B}{S}\right) \text{ x } \left(\frac{N}{1000}\right)$$

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

$$F_{mep9} \sqrt{\frac{p_{imep}}{11.45}} \text{ x } 0.0915 \text{ x } \left(\frac{N}{1000}\right)^{1.7}$$

Total MEP lost in friction,

$$F_{mep} = F_{mep1} + \ldots\ldots\ldots\ldots\ldots\ldots + F_{mep9}$$

Net brake MEP = Indicated MEP – Friction MEP

$$\text{Mechanical Efficiency} = \frac{b_{mep}}{imep}$$

3.6 Theoretical Considerations

In this analysis the molecular formula for diesel and biodiesel are approximated, as $C_{10}H_{22}$ and $C19H34O_2$. The combustion model is developed for the C.I engine and suitable for any hydrocarbon fuel and their blends.

(i) Calculation of Number of Moles of Reactants and Products

In this simulation during the start of combustion, the moles of different species are considered includes O2, N2 from intake air and CO2, H2O, N2 and O_2 from the residual gases. The overall combustion equation considered for the fuel with C-H-O-N is

$$C_xH_yO_z + \lambda^* y_{ac}{}^* (O_2 + 3.773N_2) \longrightarrow xCO_2 + (y/2)H_2O + (\lambda - 1)y_{ac}O_2 + \lambda y_{ac}{}^* 3.773N_2$$

Stochiometric AFR $y_{ac} = x + (y/4) - (z/2)$

Total number of reactants and products during the start of combustion as well every degree crank angle was calculated from the equations.

$$tmr = 1 + \lambda^* y_{cc}{}^* 4.773$$

$$tmp = x + (y/4) + 3.773^* \lambda^* y_{cc} + (\lambda - 1)^* y_{cc}$$

(ii) Calculation of Specific Heat

Specific heat at constant volume and constant pressure for each species is calculated using the expression given below.

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

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

where A, B and C are the coefficients of the polynomial equation.

(iii) Initial Pressure and Temperature at the Start of Compression

Initial pressure and temperature at the beginning of the compression process is calculated as follows

$$P_2 = \left(\frac{V_1}{V_2}\right) * \left(\frac{T_2}{T_1}\right) * P_1$$

and

$$T_2 = T_1 * \left(\frac{V_1}{V_2}\right)^{\frac{R}{C_v(T_1)}}$$

(iv) Calculation of Enthalpy and Internal Energy

Enthalpy of each species is calculated from the expression given below which is used to calculate the peak flame temperature of the cyclic process.

$$H(T) = A + B(T) + C * \ln(T)$$

The internal energy for each species and overall internal energy are calculated from the expressions given below

$$U(T) = A + (B - R) * T + C \ln(T)$$

$$U(T) = \sum x_i U_i(T)$$

where A, B and C are the coefficients of the polynomial equation.

(v) Work Done

Work done in each crank angle is calculated from

$$dW = \left(\frac{P_1 + P_2}{2}\right)(V_2 - V_1)$$

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IV. RESULTS AND DISCUSSION

In this study combustion parameters like cylinder pressure, peak cylinder pressure and peak temperature are discussed. Performance parameters like brake power is also discussed. The results are compared with the experimental results.

4.1. Cylinder pressure

In a CI engine the cylinder pressure is depends on the fuel-burning rate during the premixed burning phase. The high cylinder pressure ensures the better combustion and heat release.

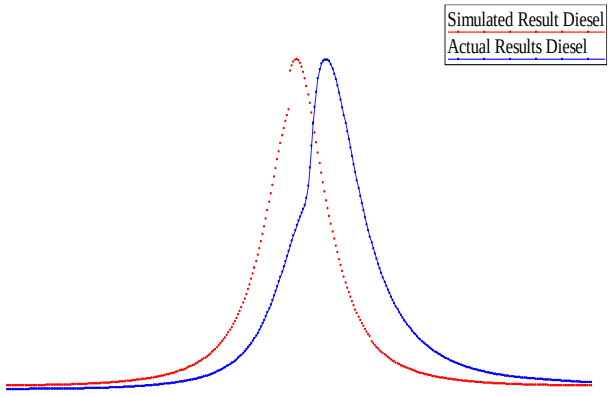

Figure 4: - Comparison of simulated and experimental pressure with crank angle

The Figure 4 shows the typical pressure variation with respect to crank angle. It has been observed that the peak pressure obtained through experimentation is 64.30 bar and the pressure obtained through simulation is 63.65 bar.

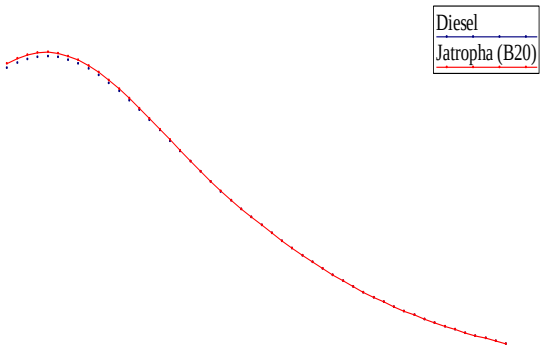

Figure 5: - Pressure variation with crank angle for diesel and biodiesel blend during combustion

Figure 5 shows the pressure variation with crank angle during combustion for diesel and bio-diesel blends. The peak pressure observed for jatropha B20 and diesel is 64.39 bar and 63.65 bar respectively. Hence we say that the internal pressure increases which ultimately increases the stress on the piston and cylinder walls.

4.2. Cylinder temperature

High pressure of compressed mixture increases its burning rate. This increases the peak pressure inside the combustion chamber. The comparisons of peak temperatures inside the cylinder for diesel and bio-diesel is shown in figure 6. The presence of oxygen in the biodiesel makes complete combustion of fuel thereby producing more CO and hence more heat is released from the gases. Thus, the peak temperature of biodiesel-fueled engine is higher than that of diesel fueled engine. The peak temperature is observed for B20 and diesel is 2780 K and 2610 K respectively.

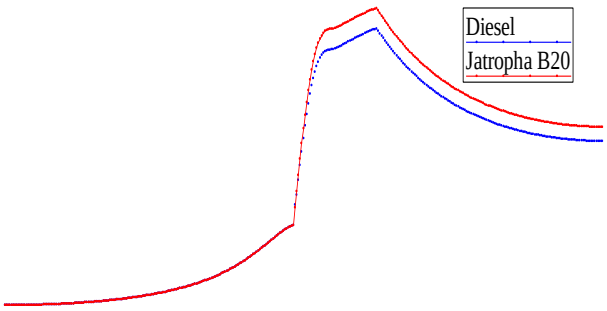

Figure 6: - Temperature variation with crank angle for diesel and biodiesel blend

4.3 Effect of CR on Brake Power

Figure 7 summarize the predicted effect of CR on engine brake power at 1500 rpm. With increasing CR, the brake power increases for all the fuels. With the change in CR, engine processes that influence its performance and efficiency, namely, combustion rate, heat transfer and friction, also vary. As the CR is increased, the heat loss to the combustion chamber wall and frictional losses decrease; hence, there is an improved performance at higher CR. However, there is a limit at which further increase in CR would not be beneficial as it may lead to increasing surface to volume ratio and slower combustion; because at higher CR, the height of the combustion chamber becomes very small. The brake power results predicted by the present model also show an increasing trend with CR for all the fuels.

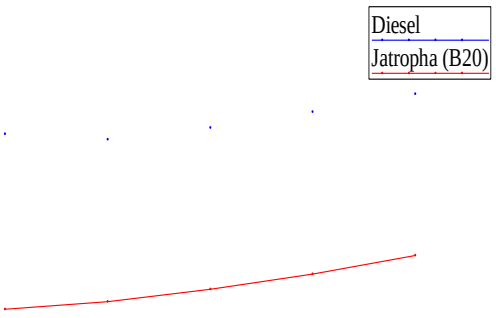

Figure 7: - Variation of Brake power with compression ratio

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At 1500 rpm as shown in figure 7, the brake power values for diesel fuel varied from a minimum of 12.89 kW (at CR 12) to a maximum of 13.47 kW (at CR 20). The corresponding minimum and maximum values for B20 is 10.34kW and 11.12 kW respectively.

4.4 Effect of Peak Pressure and Temperature on Engine Components

4.4.1 Stresses Acting Due to Peak Pressure

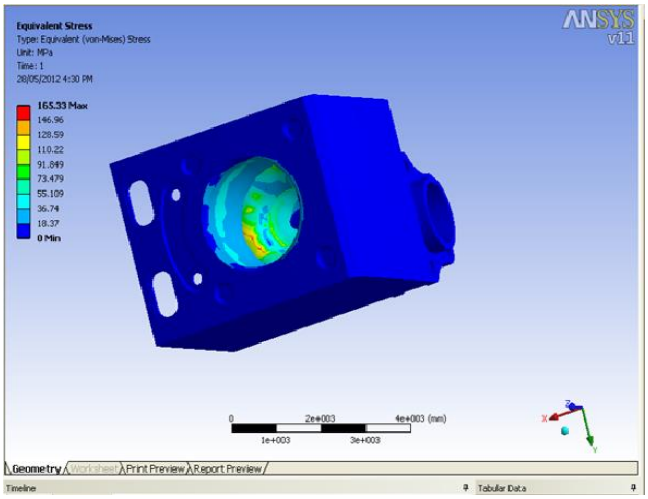

Figure 8: - Stresses acting due to application of peak pressure

From figure 8 we can say that the maximum stress acting on the piston head and it is 165.55 MPa which is less than the ultimate tensile strength (245 MPa) of the material.

4.4.2 Temperature Distribution

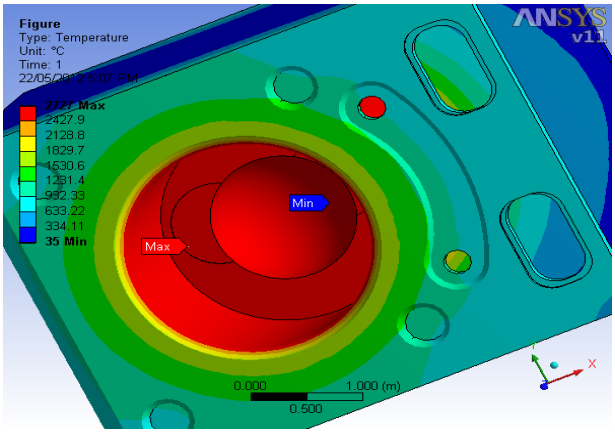

Figure 9:- Heat distribution along the assembly

Figure 9 shows the heat distribution along the assembly. The maximum temperature (2727°C) is inside the combustion chamber and at the cylinder block wall it is found 35 °C.

V. CONCLUSION

A diesel engine cycle simulation model is developed for predicting the performance of a single cylinder four stroke diesel engine fuelled by diesel and various blends of diesel and biodiesel. The model has been developed in such way that it can be used for characterizing any hydrocarbon fuels and their blends.

- The model predicts a higher rate of pressure and temperature rise for the blend during combustion as compared to diesel.
- It is observed that with the blend B20 the peak pressure and temperature increases and hence we can say that the resultant stresses will also increase.
- It is also observed that with the increase in compression ratio the brake power also increases.

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