$$P_2 = \left(\frac{V_1}{V_2}\right) * \left(\frac{T_2}{T_1}\right) * P_1 \quad (5)$$

and

$$T_2 = T_1 * \left(\frac{V_1}{V_2}\right)^{\frac{R}{C_v(T_1)}} \quad (6)$$

4.4 Heat Transfer Model

The gas-wall heat transfer is found out using following formula [3]:

$$\frac{dQ_{ht}}{dt} = h_c A_c (T_g - T_w) \quad (7)$$

Where,
hc - Heat transfer coefficient (w/m2K)
Ac - Convection heat transfer area (m2)
Tg & Tw are gas and wall temperature respectively (K)

Heat transfer coefficient correlation given by Hohenberg has been used to calculate convective heat transfer [16]:

$$h = 130.V^{-0.006}.p^{0.8}.T^{-0.4}(\bar{v}_p + 1.4)^{0.8} \quad (8)$$

Where,
P - Pressure
T - Temperature
V - Volume of the cylinder
V p - Mean piston speed, V p = 2LN/60

4.5 Combustion Model

The combustion of fuel and air is a very complex process, and would require extensive modeling to fully capture. In this work Wiebe model is used which some time is spelled Wiebe function to simulate the combustion process. [12]
Rate of heat release can be predict using following weibe formula

$$\frac{dQ}{d\theta} = na\frac{Q_{in}}{\theta_d}\exp\left[-a\left(\frac{\theta-\theta_s}{\theta_d}\right)^n\right]\times\left(\frac{\theta-\theta_s}{\theta_d}\right)^{n-1} \quad [9]$$

Where
$\theta =$ Crank angle
$\theta_s =$ Start of combustion
$\theta_s =$ X+ID
X= fuel injection angle
ID= ignition delay in deg
$\theta_d =$ Heat release duration
$n =$ Weibe form factor
$a =$ Weibe efficiency factor
$Qin = m_f \times L.H.V.$
Pressure prediction from predicted heat release rate

$$\frac{dP}{d\theta} = \frac{\lambda-1}{\gamma}\left[\frac{dQ}{d\theta} - \frac{dQ_w}{d\theta}\right] - \gamma\frac{P}{V}\frac{dV}{d\theta}$$

4.6 Ignition Delay

Ignition delay in direct injection diesel engines is of great interest to researchers and engineers because of its direct impact on the intensity of heat release rate. The delay period is composed of a physical and chemical delay. An empirical formula developed by Hardenberg and Hase for predicting the duration of the ignition delay in diesel engine. It is given in terms of crank angle [16].

$$\tau = (0.36 + 0.22V_p)Exp\left[E_a\left(\frac{1}{RT} - \frac{1}{17190}\right)\left(\frac{21.1}{P-12.4}\right)^{0.63}\right] \quad (11)$$

Where,
V p - Mean piston speed (m/s)
R - Universal gas constant (8.3143 J/mol-K)
Ea - Apparent activation energy
Ea = 618840/ (CN+25)
CN - Cetane number.

5. Results and Discussions

In this study combustion parameters like cylinder pressure, heat release rate, ignition delay and combustion zone temperature are discussed. Simulated curve has been tuned with experimental curve for various load conditions to find out the shape parameters of the functions. Correlation for these shape parameters are modified with adjusting coefficients using the least square curve fitting method.

5.1 Cylinder Pressure

Experimental data for 1500 rpm and different blend condition ranging 10% and 20% of 50% load has been used to fit the simulated heat release rate curve and coefficients are adjusted. Resulting model is then validated with the experimental pressure data. Figure 2 & 3 show the cylinder pressure and crank angle. it can be seen that model showing good agreement between experimental versus simulated data, for all loads at constant speed particularly for maximum pressure and respective crank angle.

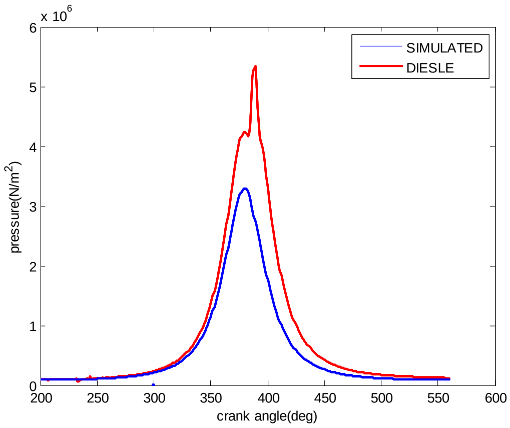

Figure 2: In-cylinder pressure vs. crank angle

However during post combustion pressure is over predicted by the model. This is the main drawback of this model because for the diesel engine combustion as the ideal heat release rate is having two curves one for premixed and one for diffusion the duel wiebe heat release approximation should have been used. It should be clear that data used for crating simulation cannot be used for validating the model because this data will show high approximation with the simulation and hence data for validation is obtained independently from the data used for simulation. And a correctly correlated (simulated) model will reproduce the accurate result for all the variations of speed and load.

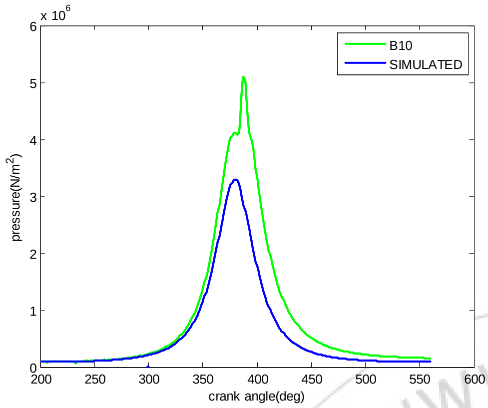

Figure 3: Comparison of experimental and predicted in-cylinder pressure for B10

5.2 Heat Release Rate

The heat release rate for different blends is shown in figure 4. Heat release rate during pre-mixed combustion is responsible for high peak pressure and high peak pressure means high rate of heat release. Superior volatility of diesel fuel ensures better air fuel mixing relatively higher heat release takes place when piston is at TDC. At lower engine loads, the heat release pattern is similar for diesel and biodiesel. However at higher loads, biodiesel blends showed quite well heat release rate during mixing controlled combustion this is due to the lower volatility and higher viscosity. Fig 4 show that Heat Release rate is peak for a small crank angle rotations.(12-25deg) at different Loads. As the load on engine reduced the heat release rate get decreases as shown in graph

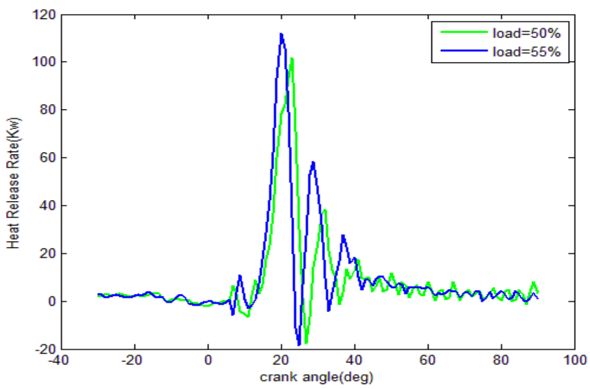

Figure 4: Heat release rate vs. crank angle

5.3 Heat transfer

Heat loss through cylinder wall occurred via radiation and convection. Hohenberge’s correlation is applied to calculate heat loss through cylinder wall. The variation of engine heat transfer with crank angle is shown in figure 6. It has been observed that Heat losses increase with temperature since heat loss is proportion to temperature difference inside the cylinder. It is observed that the heat loss rate is maximum at 70% load and lower in 50% load.

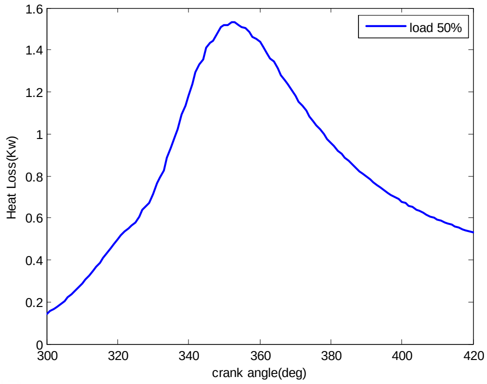

Figure 6: Heat loss vs. crank angle

5.4 In-cylinder temperature

The temperature distribution inside the engine cylinder at different crank angle position is shown in figure 6. High pressure results in high temperature. High peak pressure is obtained for pure diesel as compared to biodiesel.

1. Temperature Variations with various crank angle is obtained by the above characteristic gas equations.
2. It is observed that the temp is maximum when the piston reaches TDC and after that piston comes towards BDC that is during expansion stroke.
3. During the expansion of gases the temperature decrease Different curve at different load is plotted, it is observed that as load on engine increases the temperature increases. Higher temperature limit is obtained at 20% blend.

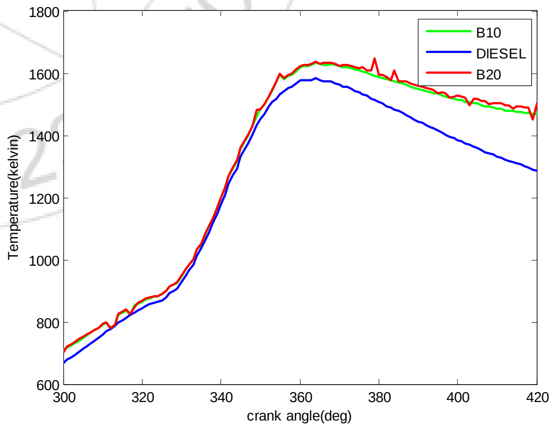

Figure 6: Temperature vs. crank angle

6. Conclusion

In a modeling and energy analysis, zero dimensional single zone combustion model simulation has been carried out to predict the single cylinder constant speed diesel engine performance. present study deals with experimental calculation and simulation of Rate of heat release and pressure for the diesel engine fuelled with biodiesel (10% & 20%). Following are the analysis done:

1. The engine performance improved with low quantity blends of biodiesel to diesel, this indicated by the higher maximum combustion temperature and pressure and are shown in graph.
2. Obtained the pressure and temperature variations inside the cylinder using the combustion correlations which give similar simulated results with experimental results
3. Heat loss can effectively studied with this simulations
4. Modifying the equations if necessary so that it could be applied over a much wider range of speed and load.
5. Simulation on model by using MATLAB software is easier and has wide scope for any micro analysis of engine performance.
6. The results of the present models are well in agreement with experimental result.

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Author Profile

Sagar Potdukhe is PG Student, Department of Mechanical Engineering, Govt. College of Engineering, Amravati (M. S.), India.