

Prediction and Validation of Heat Release Direct Injection Diesel Engine Using Multi-Zone Model

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Abstract. The objective of this study is to develop simulation model which capable to predict heat release of diesel combustion accurately in efficient computation time. A multi-zone packet model has been applied to solve the combustion phenomena inside diesel cylinder. The model formulations are presented first and then the numerical results are validated on a single cylinder direct injection diesel engine at various engine speed and timing injections. The model were found to be promising to fulfill the objective above.

Keywords: *diesel engine, heat release, multi-zone model, working fluids, step size.*

1. Introduction

A valuable information regarding with combustion study is heat release data. Heat release data expresses the rate of fuel mass burned during combustion process. Base on this data, various combustion phenomena can be explained. In the field of combustion modelling, the accurate prediction of heat release provides accurate performance prediction and as an initial stage for further prediction like emission and abnormal combustion.

Several model frameworks are used for the simulation of the diesel combustion, can be classified as zero, multi-zone and multi-dimensional models. While zero model also termed single-zone models cannot be used for heat release prediction and multi-dimensional models still need long computation time, therefore multi zone model also termed phenomenology models is promising framework to be used for heat release prediction. Multi-zone model is considered as an intermediate step between zero-dimensional and multi-dimensional models. It means that the multi-zone model combine efficient computation time almost like zero-models and good accuracy almost like multi-dimensional models.

Among of various multi-zone model, the model proposed by Hiroyasu and his coworkers [2] has been proven as the most comprehensive model and well documented up to date. The model dividing the spray into zones in the radial and penetrating directions and tracking the evolution of the zones over time. Correlations based on the constant volume vessel experiments to prescribe the spray cone angle, the spray break-up length and droplet size distribution, and the spray tip penetration has been implemented successfully. Hiroyashu model can also capture the effects of swirl and spray impingement on walls.



Jung [3] stated that within multi-zone models, combustion models that can simulate both the premixed and diffusion-controlled phases are relatively scarce. It because of the model assumed that the combustion rate was only related to the total amount of air entrainment during premixed combustion and was independent of the fuel-air mixing process. It must be noticed that the simple stoichiometric combustion concept is so sensitive to air entrainment that ad-hoc calibration coefficients are often applied to air entrainment rates in order to match the heat release rate with experimental data. The values of those empirical coefficients vary very widely among different references [3].

The model has been developed for similar engine by Rakopoulos et al [9] and give well results compare with experiment at engine speed 2500 rpm for three variation loads 40%, 60% and 80% of full load and two static injection timings of 15 and 20 deg CA BTDC. This paper adopts a similar principle but using several different empirical equations and validated at three various engine speed and timing injection.

2. Model Description

2.1. Hiroyashu Multi-Zone Model Concept

The spray injected into the combustion chamber from the injection nozzle is divided into many small packets of equal fuel mass as shown in figure 1. No intermixing among the packages is assumed. During the duration of injection, one new spray packets are generated (axial - i direction) is created for every time step, whereby individual packet content five ring layer (radial - j direction).

The fuel remains at liquid condition until time break up is reached. After the time break up, fuel is atomized into small drops. The small drop with same size will be arranged to fill the same packet. Parameter to represent the size of drop is called Sauter Mean Diameter or SMD. The entrainment of gasses from surrounding fresh air to the packets begins during droplet process. The amount of fresh gas fill the packet is assumed mixing stoichiometric., base on the assumption This process will continue during ignition delay period. After the end of ignition delay, the mixtures begin to burn.

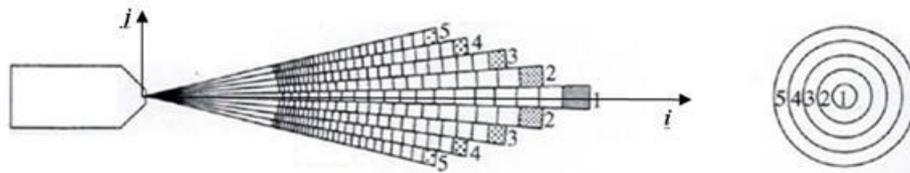


Figure 1. Packet Model Hiroyasu [4]

2.2. Model Formulation

Refer to the description of Hiroyasu model, the only effect was not counted at this model is impingement effect. The multi-zone model will solve the problem above base on continuity mass and energy. The complicated problem is solved by availability of empirical equations which it follows the correlation described by Kom [4] and Merker et. al [7]. Combination of the equations were simplified to the ordinary differential equation (ODE). Euler's method then can be applied to solve the ODE equations. At this sub section, several sub model are described more detail to distinguish the current approach with other approaches.

2.2.1 Nozzle Injection

Considering the injection pressure and nozzle open area is constant during the injection period, mass of the fuel injected, m_{inj} , for each time step is calculated using the relation as given at equation (1), where C_d is discharge coefficient of nozzle, ρ_f (kg/m^3) is density of fuel at liquid condition, A_{noz} is area of nozzle hole (m^2) and Δp_{noz} (Pa) is the difference between injection and cylinder pressure at any time of analysis. At this model, the value of 0.6 has been used for C_d . Payri et. al [8] stated that

injection profile is important to the heat release prediction, the application of Δp_{noz} was assumed adequate to represent injection profile.

$$\frac{dm_{inj}}{dt} = C_d A_{noz} (2\Delta p_{noz} \rho_{fl})^{0.5} \tag{1}$$

2.2.2 Ignition Delay

Ignition delay time is the difference time between combustion starts and the time at which injection starts. In general, ignition delay is a complicated function of mixture temperature, pressure, equivalence ratio, and fuel properties. Refer to this definition, the correct data of start of injection is important. Refer to Seykens [10], the lack of injection which is caused by injection system must be involved to the model. While the only information of start of injection only from setting the injection pump, then it was considered as uncorrected start of injection. At this model, the correction value of 20 deg CA was used.

During injection time, the average mean cylinder pressure, \bar{p}_{cyl} , (Pa) and average mean cylinder temperature, \bar{T}_{cyl} , (K) has been calculated. Base on information of Cetane number of fuel (CN) which is 40 and average piston speed, \bar{S}_p , the ignition delay can be estimated by using classical Hardenberg and Hase model as given at equation (2). However this model cannot predict well compare with experiments as also described by Lakshminarayanan [6]. To match with experiment, the exponential fitting regression has been applied as shown at figure 2.

$$\tau_{id}(CA) = (0.36 + 0.22\bar{S}_p) \cdot \exp \left[\left(\frac{618840}{CN+25} \right) \cdot \left(\frac{1}{\bar{R}\bar{T}_{cyl}} - \frac{1}{17190} \right) \cdot \left(\frac{21.2}{\bar{p}_{cyl}-12.4} \right)^{0.63} \right] \tag{2}$$

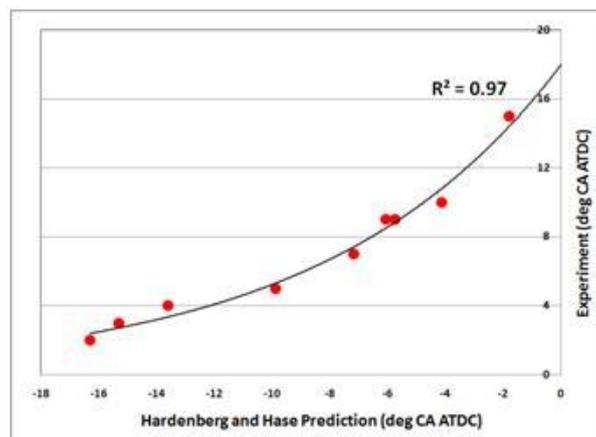


Figure 2. Start of Combustion

The next task is to find out the portion of packets was ready to burn. As described previously, during spray progression the fuel would interact with fresh air therefore the total equivalent ratio, Φ_{packet} can be calculated beside pressure, p_{cyl} and T_{packet} . The second criteria were applied to each packet zone as given at equation (3) and (4) [4]. The ignition of each packet occurs when the ignition delay integral reaches the value of 1.

$$\tau_{id} = 4 \cdot 10^{-3} \cdot (p_{cyl} \cdot 10^5)^{-2.5} \cdot (\Phi_{packet})^{-1.04} \cdot \exp \left(\frac{6000}{T_{packet}} \right) \tag{3}$$

$$\int_0^t \frac{1}{\tau_{id}} \cdot dt \geq 1 \tag{4}$$

2.2.3 Heat Release

Following Kom [4] and Merker et. al [7], there are three aspects that limited rate of mass fuel burned, dm_{fb}/dt , those are:

- If the mixture is stoichiometric or lean, it is assumed that all fuel available will be burned all.
- If the mixture is rich, the amount of fuel burned depend on the amount of fresh air is limited by the amount of air through into the packet.
- The third limitation is given by maximum rate of reaction of the fuel. By knowing the density of mixture, ρ_{mix} , the mass fraction of fuel (vapour), x_{fv} , and the fraction of oxygen, x_{O_2} , the third limitation for each volume packet, V_{packet} , can be calculate by using equation (5).

$$\frac{dm_{fb}}{dt} \leq 5 \cdot 10^{10} \cdot \rho_{mix}^2 \cdot x_{fv}^1 \cdot x_{O_2}^5 \cdot \exp\left[\frac{-12000}{T_{packet}}\right] \cdot V_{packet} \quad (5)$$

Additional empirical correction for combustion after half of total injection fuel burned was also included for the current model. The assumption behind this approach was the heat release tends to decline after half of injection fuel burned. The incline of heat release was estimated because of the ultra rich mixture. Therefore after half of total injection fuel burned and if the heat release higher than previous step condition, the rate of mass fuel burned needs to be corrected. The correlation as function of equivalent ratio, Φ , was originally used by Khrisnan et. al [5] for correcting ultra lean combustion mixture has been extended for correcting ultra rich mixture by using exponential factor as shown at equation (6). The equivalence ratio exponent C_Φ is set equal to 0.25 for the current study. As comparison, Jung et. al [3] proposed several model which differ between premixed combustion and controlled combustion

$$\frac{dm_{fb}}{dt} = \frac{dm_{fb}}{dt} \left\{ \exp\left(1 - |1 - \Phi_{packet}|^{C_\Phi}\right) \right\} \quad (6)$$

3. Model Validation

Reconstruction of multi zone model has been done by using software Matlab. Single step size crank-angle has been used to get stability [3]. Using minicomputer with 1.86 Ghz processor and 32 bits capacity, the model need less than five minutes to solve the problem with error tolerance 0.1% . It is considered to meet the objective of efficient computation time.

Experimental based on single cylinder DI diesel engine is used in this study for validation purpose. Specifications and operating conditions of the engine are shown in table 1 and table 2. A schematic diagram of system is shown in figure 3. Base on cylinder pressure and volume data each step time, the rate dQ_{hr}/dt , and total Q_{hr} , of net heat release can be calculated based on equation (7) and (8) [1], where ratio heat specific γ is equal to 1.3. The results for measurement and prediction are given at figure 4 to 6.

$$\frac{dQ_{hr}}{dt} = \frac{\gamma}{\gamma-1} p_{cyl} \frac{dV_{cyl}}{dt} + \frac{1}{\gamma-1} V_{cyl} \frac{dp_{cyl}}{dt} \quad (7)$$

$$Q_{hr} = \int_t^{\Delta t} \frac{dQ_{hr}}{dt} dt \quad (8)$$

Refer to figure 4 to 6, the graph shows that total heat release predictions give similar trend with experiments with slightly over estimate at the end of process. The error comes from two sources, the first error can be explained from graphic of heat release rate, where after half combustion process there are increasing of heat release rate at several points and it is appears more clearly at figure 6. The second error comes from the difference when combustion ends. Both errors can be solved easily by improvement of definition of when the equation (5) can be applied and the proper value of C_Φ .

However in general, the validation results from three different engine speed and three different timing injection for each engine speed shows acceptable accuracy of multi zone model predictions

Table 1. Engine specification

Subjects	Description
Engine type	DI diesel
Number of cylinder	1
Number of valve	2
Swept volume (ltrs)	0.45
Bore (mm) x Stroke (mm)	80.26 x 88.90
Compression ratio	20.3 : 1
Combustion chamber shape	Bowl in piston
Swirl ratio	3.57
Injector operation pressure (bar)	200
Nozzle Diameter (mm)	0.25
No of Nozzle Hole	4
Valve timing	IVO 8 BTDC, IVC 42 ATDC, EVO 60 BBDC, EVC 12 ATDC

Table 2. Engine Operating Set-Up

Case	Uncorrected Injection Timing	Diesel Fuel Mass Flow [g/h]	Air Mass Flow [kg/s]	Temperature of Diesel Fuel [deg C]	Inlet Manifold Pressure [mbar]	Air Inlet Temperature [deg C]	Engine Speed [rpm]
1	32.19	1844.33	0.01006	31.60	996.70	28.00	3000.00
2	28.10	1731.48	0.01035	31.60	996.00	28.00	3000.00
3	24.12	1917.49	0.01580	31.50	998.20	28.00	3000.00
4	38.01	2217.44	0.01127	32.30	992.70	27.00	3600.00
5	36.17	2239.40	0.01261	32.30	993.40	26.00	3601.00
6	28.78	2152.69	0.01146	32.20	992.10	27.00	3600.00
7	39.40	1328.49	0.01252	32.80	988.70	26.00	4200.00
8	30.28	1359.05	0.01229	32.60	988.80	27.00	4200.00
9	27.23	1379.56	0.01250	32.70	988.90	27.00	4200.00

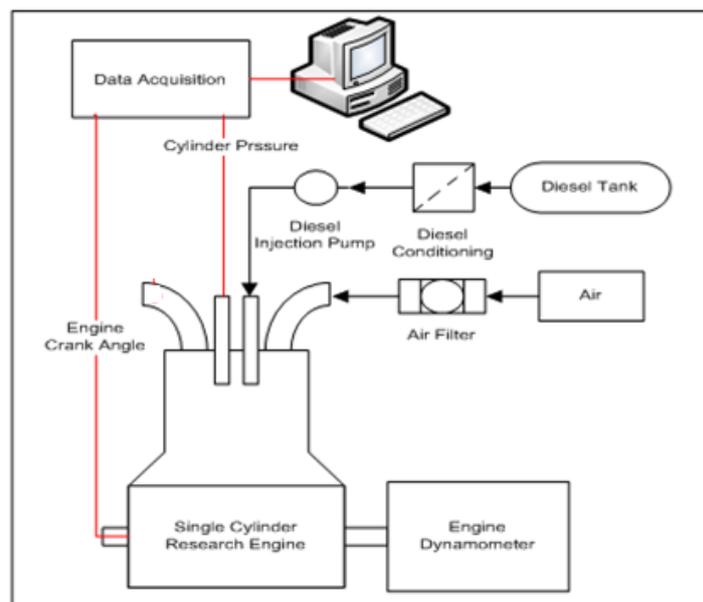


Figure 3. Schematic of experimental setup

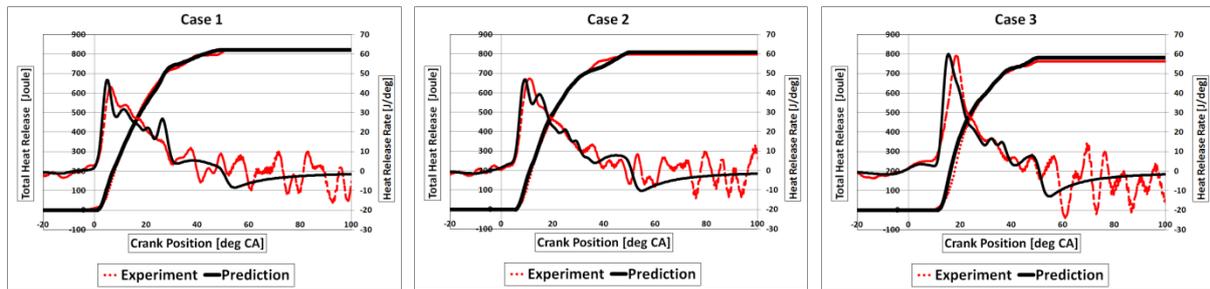


Figure 4. Heat Release at 3000 rpm

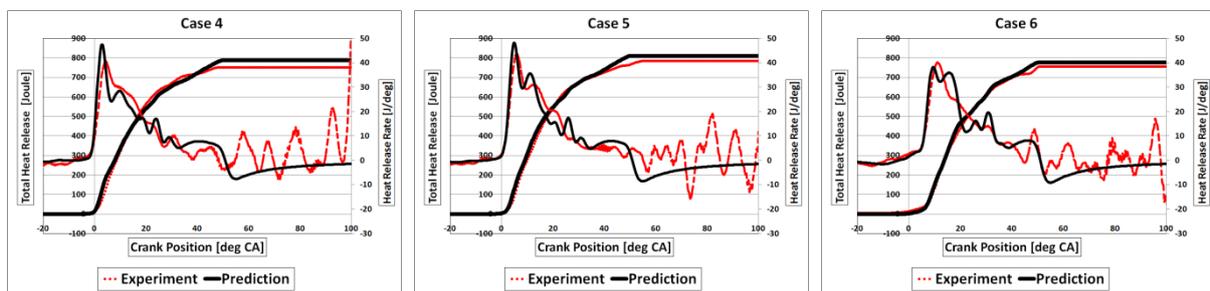


Figure 5. Heat Release at 3600 rpm

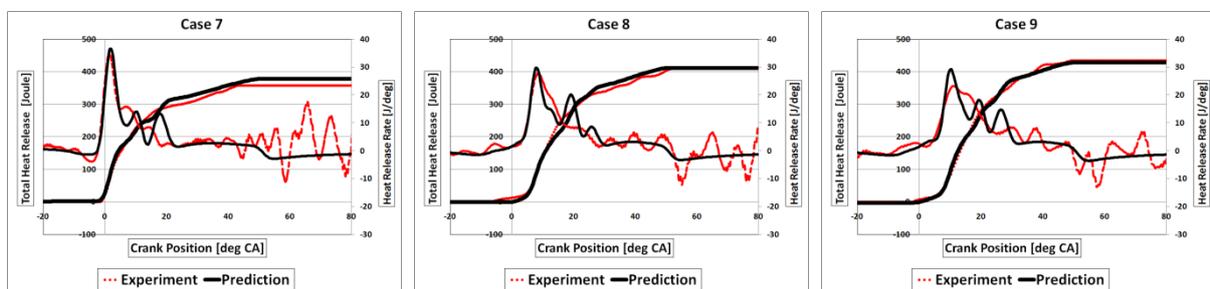


Figure 6. Heat Release at 4200 rpm

4. Conclusions

In the present study the multi-zone modeling of diesel combustion was implemented for heat release prediction. The documentation of several corrections and simplification approach was also presented. It is also reported that efficient computation time can be achieved by this model. The validation of the multi zone model by using heat release data from three various engine speed and three various injection timing for each engine speed shows acceptable accuracy compared to experimental data. However, further improvement still needed especially on start of combustion prediction, while at this study the model still depend on experimental data input. More extensive validation is also needed to find out the also the sensitivity of the current multi-zone model.

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