Simulation of CI (Combustion Ignition) Engine

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Abstract: The development engages in creating a model for predicting the power characteristics of a CI engine. The goal is to formulate a program that is computationally economical and can forecast the power characteristics of CI engine with higher precision. This has been achieved through coding a program in the C-Language. Upon examination of the model with the experimental data, the results are validated. Because of its simplicity, the model can also be utilized for an extensive range of alternate fuels to optimize the design of CI engines.

Chapter 1: Introduction

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. Diesel engine simulation models can be used to understand the combustion performance; these models can reduce the number of experiments. It has been observed that the performance characteristics of I.C Engines strongly depends on the design of the engine which in turn can be successfully achieved by making the engine prone to work under the controlled set of variables. In daily life, a vehicle is frequently said to be “a necessary evil.” This is because although it provides great convenience in our daily activity, there are also problems associated with it, such as air pollution. It is so natural, therefore, that a more efficient and less pollutant emitting engine is being sought by the automotive industry. With its high thermal efficiency and lean operation capability, the Diesel engine is a viable candidate, but inherent emission problems must be solved. The high thermal efficiency of the Diesel engine means less greenhouse gas emission, such as CO₂, compared to the gasoline engine, which is currently becoming the most important and critical issue to be solved. Moreover, emission problems have also been significantly improved owing to advanced technologies such as the common rail system, which is in recent production vehicles to comply with strict pollutant and noise standards. However, upcoming emission standards in 2013 are going to be very difficult to meet. As a result of this trend, the penetration of Diesel engines into the Indian passenger car market has kept increasing, from 10% in the 1990’s to 20% in 2001 and approaching 40% in 2010. This trend has generated the need for a simple and fast simulation model which can predict the power and emission characteristics of a CI engine with reasonable accuracy.

The reason why modelling is conducted in almost every research is that it can predict the result of not-yet-conducted experiment and modify design variables by analyzing the estimated data without substantial testing time, and cost. 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. Diesel engine simulation models can be used to understand the combustion performance; these models can reduce the number of experiments.

The modelling of reciprocating and rotary internal combustion (IC) engines is a multidisciplinary subject that involves thermodynamics, fluid mechanics, turbulence, heat transfer, combustion, chemical reactions, mathematical analysis, and numerical methods. Historically, different levels of approximation have been used to predict the performance of IC engines; from simple air standard cycles to complex 3D models including turbulence, chemical reactions, spray dynamics, etc. IC engine simulation can be classified into four categories, namely zero-dimensional single zone, 0D/1D single zone models, quasi-dimensional multi-zone models and multidimensional models. In 0D/1D models, the engine is represented as a network of pipes (intake and exhaust manifolds) interconnected among them with “devices” that simulate different parts of the machine (valves, cylinders, pipe junctions, etc.). One-dimensional CFD models are used for pipes and thermodynamic (or zero-dimensional) models for the above-mentioned “devices”.

For 0D models, most properties are averaged over the total volume and no spatial information is
available. These models rely on some understanding of the physics involved and try to capture the main features of the processes. By including the description of the most important aspects, the models have performed surprisingly well and are ideally suited for parametric studies. A zero-dimensional single-zone model is capable of predicting engine performance and fuel economy accurately with a high computational efficiency. The major drawback of single-zone models is their inability to simulate the wave propagation into pipes and manifolds that strongly influence on volumetric efficiency. Also, these models are unable to account for fuel spray evolution and the spatial variation in mixture composition and temperature, both of which are essential in predicting harmful species formed during the combustion process.

On the other hand, multi-dimensional models resolve the cylinder space into fine grids, thus providing a considerable amount of spatial information. However, multidimensional models still employ phenomenological sub-models describing fuel spray processes, and their simulation results may vary with assumed initial or boundary conditions. Consequently, the accuracy of the results cannot always be guaranteed. Furthermore, computational time and storage constraints still preclude these models from routine use for design purposes. Currently, an intermediate step between zero-dimensional and multi-dimensional models has arisen, called quasi-dimensional. Multi-zone models can be effectively used to simulate new technology engine combustion systems, by combining the advantages of zero-dimensional models and multi-dimensional models. These models are able to provide the spatial information required to predict emission products with significantly less requirement on computing resources than for multidimensional models.

In this work, we develop 0D/1D models in order to obtain a computational tool that can predict with sufficient precision the performance of a CI engine at a relatively low computational cost.

Chapter 2: Literature Review

The reason why modelling is conducted in almost every research is that it can predict the result of not-yet-conducted experiment and modify design variables by analyzing the estimated data without substantial testing time, and cost. Thus, combustion modelling for the automotive diesel engine is always adopted in engine research and demonstrates its usefulness by providing combustion mechanism and design insights.

In 1963, for the first time, Lyn[1] phenomenological explained the heat release mechanism of diesel combustion and his work has been considered as a stepping stone towards describing the diesel combustion phenomenon. Henein [2, 3] may be the first researcher to attempt to model the Diesel engine combustion. He phenomenological analyzed diesel spray combustion including evaporation and ignition. Thus, he provided a basic starting point for succeeding researchers. Then, Shahed et al. [4] made a thermodynamic cycle simulation with an NO formation model assuming local stoichiometric combustion. They assumed spray combustion is negligible and that there is no interaction between combustion gas packages. Hiroyasu et al. [5] devised the first mathematical multi-zone approach in droplet evaporation assuming stoichiometric combustion. They provided a basis for a multi-zone approach to diesel fuel injection. In their study, the burned gas temperature, which can be calculated by considering the combustion a polytrophic process, is used for NOx modelling [6].

There are two categories of combustion model, the thermodynamic model, and the dimensional model. One of the major advantages of the thermodynamic model is its computational efficiency, while the dimensional model considers spatial distribution of all parameters, and thus requires substantial computational time. The thermodynamic models include single and multi-zone models and mean value models. A mean value model (MVM) has been adopted by some researchers for control and diagnostic purposes [8]. [26]. But the purpose of MVM is not for analyzing the Diesel combustion itself, but for control purposes so that modelling can be oriented on the result only. When adopting a single zone approach for the Diesel engine combustion, the cylinder charge is assumed to be a homogeneous mixture of an ideal gas at all times. The instantaneous state of the mixture can be described by the mixture pressure, temperature, and equivalence ratio. Fuel burns instantaneously as it is added to the cylinder [14]. On the contrary to this, in a multi-zone model [5]. [6] the mixture in the combustion chamber is divided into more than two regions: the unburned, the burned zone or the quenching zone near the wall. In this model, a single computational zone is created at each fuel injection and has its own separate thermodynamic history throughout the calculation. Each zone is treated with the single zone approach, in which all the properties are uniform in its zone, and is assumed to be composed of fuel droplets of the same size. Although multi-dimensional model (CFD) using such as KIVA, STAR-CD etc. cannot prove the preciseness in modelling the Diesel engine combustion because of complicated combustion mechanisms, some researchers have
used commercial package due to its relatively high predictability over other models. [9], [10], [11], [12], [13]

As can be inferred from the discussion so far, due to its efficient computational capability and simplicity of application, a single-zone approach for CIDI engine combustion is popular, such as in Ramos [14], Reitz [9], and Assanis [3], [12]. Even with CFD model such as the KIVA code [24], getting a satisfactory prediction for the pilot injection combustion and emissions is difficult, so a single zone approach seems not a bad choice. From now on, the discussion will be concentrated on a single zone approach.

In a single-zone model approach, there are some heat release rate estimation models for diesel engines [7], [27], [28]. The famous Wiebe function is useful and frequently applied in combustion-ignition engines. Since there are two distinctive modes in diesel engines combustion such as premixed and diffusion, heat release rate model utilizing double Wiebe functions is also applied in diesel engine combustion model. Two combustion modes are assumed to be the same form of Wiebe function except the shape factors involved.

Chmela and Orthaber developed a diesel engine heat release model by considering the fuel injection kinetic energy called mixing controlled combustion (MCC) [28]. According to the model, the heat release rate is described as functions of the injected fuel quantity and inlet turbulence strength. As the model name implies, they consider the diffusion combustion mode only in diesel engines, which is not a reasonable assumption. The importance of premixed combustion is significant both in emission and performance aspect, and thus should not be neglected in diesel engine combustion.

Kreiger and Borman proposed a heat release rate model using a gamma function describing the asymptotic formula of the shape factor, which depends on the combustion chamber configuration [27]. But, there is no consideration about auto-ignition delay in this heat release model. The shape factors need to be estimated with test data.

Assanis et al. [10] used the Arrhenius type of combustion model, which was prepared by Hiroyasu et al. [15] and frequently used in the multi-zone approach. In the single-zone approach, the heat release rate may account for both premixed and diffusion combustion by means of the Wiebe function, so it can be expressed by one or more algebraic formulas. But model coefficients in these formulas, which may vary with engine design details and operating conditions, are determined empirically by fitting with data [17].

Ignition is a phenomenon in which heat is released rapidly as a result of chemical reaction of the fuel. Actually, ignition is initiated spontaneously when there are enough chain carriers to accelerate the chemical reaction. To identify this chain reaction step in the reaction system is a key point in auto-ignition mechanism. For example, the ignition models such as Henein [16], Heywood [17], and Hardenberg [18] could be applied due to their simplicity. When a more detailed chemical reaction is involved, some models like Westbrook [19], Minetti [20] using the CHEMKIN code need to be examined. There is a study in which the single-zone model using double Wiebe functions is used, and ignition delay is neglected [25].

For the heat transfer model, the Woschni correlation [21] modified by Hohenberg [22] can predict fairly well result in automotive engines. Due to its low computation cost and efficiency, a closed cycle simulation is often adopted in engine modelling by several researchers such as Assanis [12], Reitz [9], [23], and Hiroyasu [5]. A closed cycle is defined as the engine period from the intake valve closing (IVC) to the exhaust opening.

Chapter 3: CI Engine Modelling

Table 1. Nomenclature

<table>
<thead>
<tr>
<th>Subscripts</th>
<th>Nomenclature</th>
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</thead>
<tbody>
<tr>
<td>B</td>
<td>Bore, m</td>
</tr>
<tr>
<td>B</td>
<td>Wiebe’s law form factor</td>
</tr>
<tr>
<td>C&lt;sub&gt;p&lt;/sub&gt;</td>
<td>Specific heat at constant pressure, J kg&lt;sup&gt;-1&lt;/sup&gt; K&lt;sup&gt;-1&lt;/sup&gt;</td>
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<tr>
<td>HLV</td>
<td>Lower heating value J kg&lt;sup&gt;-1&lt;/sup&gt;</td>
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<tr>
<td>H</td>
<td>Heat transfer coefficient, W m&lt;sup&gt;-2&lt;/sup&gt; K&lt;sup&gt;-1&lt;/sup&gt;</td>
</tr>
<tr>
<td>k&lt;sub&gt;hoh&lt;/sub&gt;</td>
<td>Hohenberg constant</td>
</tr>
<tr>
<td>L</td>
<td>Length of connecting rod, m</td>
</tr>
<tr>
<td>P</td>
<td>Pressure, Pa</td>
</tr>
<tr>
<td>Q</td>
<td>Heat, J</td>
</tr>
<tr>
<td>R</td>
<td>Ideal gas constant</td>
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We intend to develop a model for stationary engine applications, essentially Combined Heat and Power engine. For the CHP ignition compression engine a “one zone” model is used hereafter, i.e., the state of the cylinder charge is defined in terms of average spatial properties. Pressure, temperature, and composition of cylinder charge are assumed to be uniform at each time step that is to say that no distinction is made between burned and unburned gas during the combustion phase inside the cylinder.

The assumptions are taken for development of model are:

1. The composition of the cylinder is homogenous at all times.
2. The rate of fuel injection is constant.
3. Fuel is instantaneously vaporized after injection.
4. Fuel is instantaneously combusted after injection.
If the ratio of connected rod length to crank radius, 
\[ R = \frac{2L}{S} \], then the instantaneous cylinder volume 
\[ V(\theta) = \frac{\pi B^2}{4} \left(L(1 + R(1 - \cos(\theta))) - R^2(\sin(\theta))^2\right) + V_{\text{clear}} \]  
(1)

Where \( V_{\text{clear}} = \frac{V_g(\theta)}{\varepsilon} \), with \( \varepsilon \) the compression ratio.

The piston speed is:
\[ v_p = \frac{4}{\pi B^2} \frac{dV_g(\theta)}{d\theta} \omega \]
With \( \theta = \omega t \); \( \omega \) is the engine speed in rad s\(^{-1}\).

After the closing of the intake valve, \( m_g \) is the gas mass enclosed in the cylinder. Assuming cylinder gas is an ideal gas, and calorific capacities are independent of the temperature. The entropy corresponding to \( m_g \) is obtained according to:
\[ dS_g = m_g C_p \frac{dT_g}{T_g} - m_g r \frac{dP_g}{P_g} \]
(2)

Knowing that:
\[ \frac{dP_g}{P_g} = -\frac{dV_g}{V_g} + \frac{dT_g}{T_g} \]
Eq. (2) becomes:
\[ dS_g = \frac{1}{m_g r} \frac{dT_g}{\gamma - 1} + \frac{dV_g}{V_g} \]
(3)

In the cylinder, the gas temperature \( T_g \) is considered uniform in the control volume. Then the entropy is expressed as a ratio between heat and a reference temperature; the gas furnishes heat to the wall (heat transfer) \( Q_{\text{gw}} \) and receives heat from combustion \( Q_{\text{comb}} \). Consequently, we obtain:
\[ dS_g = \frac{\partial Q_{\text{gw}}}{T_g} + \frac{\partial Q_{\text{comb}}}{T_g} = \frac{h_g S_{g_w}(T_w - T_g)}{T_g} + \frac{\partial Q_{\text{comb}}}{T_g} \]  
(4)

Then Eq. (3) implies:
\[ \frac{\partial Q_{\text{gw}}}{m_g r T_g} + \frac{\partial Q_{\text{comb}}}{V_g} = \frac{1}{m_g r} \frac{dT_g}{\gamma - 1} + \frac{dV_g}{V_g} \]
(5)

During the combustion phase, the term \( Q_{\text{comb}} \) is equal to zero apart from this phase.

Also, the state of the gas at the point of start of compression is derived using a perfect mixing model for the induced fresh charge and residuals from the previous cycle. Reactions are neglected during compression due to their insignificance at the pressure and temperature pertaining, and a perfect gas mixture is assumed.

The first law of thermodynamics then becomes:
\[ \frac{dQ}{d\theta} = mC_v \frac{dT}{d\theta} + p \frac{dV}{d\theta} \]  
(6)

The equation of state is:
\[ pV = mRT \]  
(7)

Combining equations (6) and (7) gives:
\[ \frac{dP}{d\theta} = \left[ r \left( \frac{dQ}{d\theta} - p \frac{dV}{d\theta} \left( \frac{r}{C_v} + 1 \right) \right) \right] \frac{1}{V} \]  
(8)

Chapter 4: Details of the model

4.1. Heat Transfer

Heat transfer at cylinder walls are represented by the Woschni correlation modified by Hohenberg\(^{[31]}\), where \( k_{\text{hoh}} \) is a constant which characterise the engine.
\[ h_g(\theta) = k_{\text{hoh}} \left( 10^{-5} P_g(\theta) \right)^{0.8} \left( \frac{v_{\text{plis}} + 1.4}{T_g(\theta)^{0.4} V_g(\theta)^{0.8}} \right) \]
(9)

With the ideal gas, the correlation of global heat transfer coefficient \( h_g \) becomes:
\[ h_g(\theta) = k_{hob} \left(10^{-5} m_g r \right)^{0.8} T_g(\theta)^{0.4} \left( \frac{v_{pis} + 1.4}{V_g(\theta)^{0.86}} \right) \]  
(10)

\( h_g(\theta) \) represents the value of the cited coefficient at a given \( \theta \) corresponding to a given piston position. The heat loss by convective heat transfer is:

\[ \frac{dQ_{gP} (\theta)}{d\theta} = h_g(\theta) S_{gw}(\theta) \left( \frac{T_w}{T_g(\theta)} - T_g(\theta) \right) \]

\( \omega \)

(11)

\( \langle T_w \rangle \) is spatially and temporally averaged wall temperature. The temperature’s variations of inner cylinder surface during the thermodynamic cycle are weak (about 10 K) compared to the temperature’s variations of the combustion gases (about 1000 K). The Temperature of the wall can be considered as constant according to the results of Rakapoulos et al. \[32\]. Hereafter for simplification \( \langle T_w \rangle \) will be noted \( T_w \).

### 4.2. Combustion Model

According to the experimental results of Caika et al. \[29\] on a diesel engine of new generation, the release of heat can be modelled by using a simple Wiebe law, which describes the combustion process:

\[ \frac{dQ_{comb}}{d\theta} = m_f HLV \frac{d\chi(\theta)}{d\theta} \]

\( \omega \)

(12)

Where HLV denotes heat low values, or the mass of fuel to be burned, \( \frac{d\chi(\theta)}{d\theta} \) is the combustion rate obtained by the Wiebe function:

\[ \frac{d\chi(\theta)}{d\theta} = 6.908 b + 1 \left( \frac{\theta - \theta_0}{\Delta \theta_d} \right)^b \exp \left[-6.908 \left( \frac{\theta - \theta_0}{\Delta \theta_d} \right)^{b+1} \right] \]

(13)

If, \( \theta_0 \geq \theta \geq \theta_0 + \Delta \theta_d \)

\( \theta_0 \) is the crankshaft angle marking the outbreak of the combustion, \( \Delta \theta_d \) is the combustion duration and \( b \) the Wiebe’s law form factor, constant parameter to be identified. Arque’s \[33\] proposes the values of these parameters, and the fuel injected inside the cylinder is directly connected to the fuel/air equivalence ratio

\[ \phi^{-1} = \frac{(A / F)_{actual}}{(A / F)_{stoichiometric}} \]

Then,

\[ m_f = \frac{\phi m_g}{(A / F)_{stoichiometric}} \]

(14)

### 4.3. Friction Losses and Pumping

Heywood\[17\] indicates, for total friction losses,

\[ P_f = C + 1.44 \frac{v_{pis}}{B} + 0.4 v_{pis} \]

(15)

where \( v_{pis} \) the mean piston speed in \( ms^{-1} \) and \( B \) the bore in \( m \). \( C \) is a constant, which depends on the engine type, \( C = 75 \) kPa for direct injection engine. The proposed model differs particularly from Chen’s paper \[34\] where the friction is modeled, as proportional only to the square of speed. Heywood\[17\] indicates that the constant term \( C \) of Eq. (15) represents a constant base pressure which is to be overcome first.

Combining Eq. (5) with (11) and (12) gives:

\[ \frac{h_g S_{gw}(T_w - T_g)}{T_g} + m_f HLV d\chi = \frac{1}{\gamma - 1} \frac{dP}{P} + \frac{dV}{V} \]

(16)

According to variation of \( \theta \),

\[ \frac{1}{m_f r_j \omega} \left[ \frac{h_g S_{gw}(T_w - T_g)}{T_g} \right] \frac{d\chi}{d\theta} + \right] \frac{1}{m_f r_j \omega} \frac{dP}{P} = \frac{1}{m_f r_j \omega} \frac{dV}{V} \]

(17)

Thus the gas temperature variation is:

\[ \frac{dT_g(\theta)}{d\theta} = \frac{1 - \gamma}{m_f r_j \omega} \left[ \frac{h_g S_{gw}(T_w - T_g)}{T_g} \right] \frac{d\chi}{d\theta} + \right] \frac{1 - \gamma - 1}{m_f r_j \omega} \frac{dP}{P} \]

(18)

Fourth order Runge-Kutta method is used to solve the differential equations of pressure and temperature.
For a differential equation of the form $y' = f(t, y)$, the fourth order Runge-Kutta method is given by the following equations:

$$y_{n+1} = y_n + \frac{1}{6}(k_1 + 2k_2 + 2k_3 + k_4)$$

$$t_{n+1} = t_n + h$$

Where $y_{n+1}$ is the Runge-Kutta approximation of $y(t_{n+1})$, and,

$$k_1 = hf(t_n, y_n),$$

$$k_2 = hf(t_n + \frac{1}{2}h, y_n + \frac{1}{2}k_1),$$

$$k_3 = hf(t_n + \frac{1}{2}h, y_n + \frac{1}{2}k_2),$$

And,

$$k_4 = hf(t_n + h, y_n + k_3)$$

### Chapter 5: Structure of the Program

Based on above model, a program is developed in “C” programming language. Engine parameters and fuel properties are required as the input. The computational part of the program consists of three loops, namely, compression, combustion and expansion loops. There are four different functions, “runge”, “runge1”, “runge2” and “runge3”. These functions are used to solve the governing differential equations using Runge-Kutta method. After obtaining the input, the compression loop is started. From this loop, the value of crank angle and pressure is passed on to the runge functions. In runge function, the equation is solved and the value of $dP/d\theta$ is returned back to the loop, which becomes the Runge-Kutta coefficient in the main program. The same process is repeated till we get the four coefficients of the fourth order Runge-Kutta method, and from these, the new value of pressure is calculated. The same process is repeated with the runge2 function to get the value of temperature.

Combustion and Expansion loops also use the similar process. Since the governing equations are same for compression and expansion, same functions (runge and runge2) are used for both of them.

The output of the program consists of Pressure and Temperature values for corresponding crank angle and the values of work output and work done against friction for one cycle. From this, we can calculate the value of Indicated Power, Friction Power, and Brake Power.
Figure 2. Structure of the Program
Chapter 6: Model Validation

To validate the model, the experimental results of Descieux et al. [30] are used. The specifications of the engine used are given below in the table.

<table>
<thead>
<tr>
<th>Table 2. Specifications of Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type of Engine</td>
</tr>
<tr>
<td>Number of cylinders</td>
</tr>
<tr>
<td>Bore Diameter</td>
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<td>Stroke</td>
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<td>Length of Connecting Rod</td>
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<td>Displacement</td>
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<td>IVC</td>
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<td>Start of Injection</td>
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<td>End of Injection</td>
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<tr>
<td>EVO</td>
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<tr>
<td>Wall Temperature</td>
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<tr>
<td>Compression Ratio</td>
</tr>
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<td>Equivalence Ratio</td>
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<tr>
<td>Type of Fuel</td>
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<td>Lower Heating Value of fuel</td>
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</tbody>
</table>

The comparison of experimental and simulation results are given below:

![Figure 3. Comparison of P-θ Diagram with Experimental Results for 3000 RPM](image1)

Figure 3 indicates that the simulation result closely matches the experimental results. The peak deviation between the simulation and experimental results is about 4%.

![Figure 4. Comparison of T-θ Diagram with Experimental Results for 3000 RPM](image2)

Figure 4 indicates that simulation results are in close agreement with the experimental results. The peak deviation between simulation and experimental results is about 6%. The variation in the crank angle at which peak temperature is obtained is because of the assumption that fuel is instantaneously burned upon injection.

Both the pressure and temperature curves closely matches the experimental results. Based on this, we can conclude that our model is reasonably accurate and can be used to predict the performance characteristics of CI engines.

Chapter 7: Results and Discussions

7.1. Effect of Injection Timing on Pressure:
Table 3. Simulation Data for Figure 5

<table>
<thead>
<tr>
<th>θ</th>
<th>P (5 deg bTDC)</th>
<th>P (10 deg bTDC)</th>
<th>P (15 deg bTDC)</th>
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Figure 5. Variation of Pressure with Injection Timing for Compression Ratio=16 and Equivalence Ratio=1

Table 4. Simulation Data for Figure 6

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Figure 6. Variation of Pressure with Injection Timing for Compression Ratio=17 and Equivalence Ratio=0.7
Figure 7. Variation of Pressure with Injection Timing for Compression Ratio=18 and Equivalence Ratio=0.7

Figures 5, 6 and 7 indicate the variation of pressure with injection timing for compression ratios 16, 17 and 18. As injection timing is advanced away from TDC, we see a rise in peak pressure as well as higher pressure gradient until the optimum injection timing is achieved. If we further advance the injection timing, indicated mean effective pressure (imep) will decrease. We can also see that as the compression ratio is increased, the general trend of the curve remains the same but peak values of pressure increases which indicate an increased imep. This indicates a higher work output per cycle and thus higher thermal efficiency.

7.2. Effect of Injection Timing on Temperature:

Figures 8, 9 and 10 shows the variation of temperature with injection timing for compression ratio 16, 17, and 18. We can see that the peak temperature is obtained at the end of fuel injection. Peak temperature also increases up to the optimum point when injection timing is advanced. We can further see that the injection timing also has an effect on exhaust temperature. When injection timing is changed from 5 degrees bTDC to 10 degrees, an increase in exhaust temperature of approximately 14% is observed in each case but when the injection timing is changed to 15 degrees bTDC, an increase in exhaust temperature is only 3%.

![Figure 8. Variation of Temperature with Injection Timing for Compression Ratio=16 and Equivalence Ratio=0.7](image)

Table 5. Simulation Data for Figure 8

<table>
<thead>
<tr>
<th>β</th>
<th>T(5 deg bTDC)</th>
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Figure 9. Variation of Temperature with Injection Timing for Compression Ratio=17 and Equivalence Ratio=0.7

Table 6. Simulation Data for Figure 9

<table>
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<tr>
<th>θ</th>
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Figure 10. Variation of Temperature with Injection Timing for Compression Ratio=18 and Equivalence Ratio=0.7

7.3. Effect of RPM on Friction Power:
Figure 11 shows the variation of Friction Power with RPM. Since the friction power is only a function of mean piston speed, it remains virtually unchanged with injection timing and increases with RPM.

**Chapter 8: Conclusion**

The aim of the project to develop a model for predicting the power characteristics of a CI engine has been achieved. The results were validated and compared with the experimental data for a single cylinder direct injection Diesel engine for pressure and temperature. A good agreement between the predicted and experimental values ensures the accuracy of the numerical predictions of the present work. The simulation model developed successfully captures the single cylinder compression ignition engine operating characteristics. The program developed is computationally cost effective and can predict the power characteristics of CI engine with high accuracy.

The operational range of the model is wide and computational run time is short, thus making the simulation model suitable for use with thermodynamically based cycle simulations in compression ignition engines.

Due to its simplicity, the model can also be used for wide range of alternate fuels to optimize the design of CI engines.

Further conclusions from the analysis are:

1. Peak pressure and temperature increases when the injection timing is advanced up to the optimum point.

2. Work output, as well as Brake Power, increases when the injection timing is advanced up to the optimum point, after which it starts decreasing.

3. The optimum injection timing for the test engine is found to be 15 degrees bTDC.

4. There is no effect of compression ratio on optimum injection timing.

**Bibliography**


# APPENDIX

List of Variables:

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>r₀</td>
<td>Compression Ratio</td>
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<tr>
<td>n</td>
<td>RPM</td>
</tr>
<tr>
<td>d</td>
<td>Bore Diameter</td>
</tr>
<tr>
<td>b</td>
<td>Crank Radius</td>
</tr>
<tr>
<td>l</td>
<td>Length of Connecting Rod</td>
</tr>
<tr>
<td>a₁</td>
<td>IVC</td>
</tr>
<tr>
<td>a₂</td>
<td>Start of Injection</td>
</tr>
<tr>
<td>a₃</td>
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</tr>
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</tr>
<tr>
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</tr>
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<tr>
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<td>$dP / dθ$</td>
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<td>dtdx</td>
<td>$dT / dθ$</td>
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</table>
Program:

```c
#include<stdio.h>
#include<conio.h>
#include<math.h>
double runge(double x,double y);
double runge1(double x,double y);
double runge2(double x,double y,double z);
double runge3(double x,double y,double z);
main()
{
    clrscr();
    //int n;
    double p_i,p,a,k1,k2,k3,k4,k,a1,a2,t1,t2,pi,T,T_i;
    double r0,d,b,l,R,t,x,e,v_s,v,n,vm_pis,p_f,f_p,dV,power,IP;
    double j1,j2,j3,j4,j;
    FILE *fp;
    fp = fopen("result.txt","w");
    pi=3.141592654;
    r0=18;
    n=3000;
    d=.111;     //bore diameter
    b=.0635;   //crank radius
    l=.251;   //length of crank rod
    R=l/b;
    p_i=1.01e5;
    p=p_i;
    a1=225;//IVC
    a2=350;//start of injection
    t1=(a1/180)*pi;
    t2=(a2/180)*pi;
    T_i=300;
    T=T_i;
    power=0;
    //printf("%f",t1);
    for(a=t1;a<=t2;a=a+.01) //compression loop
    {
        k1=0;
        k2=0;
        k3=0;
        k4=0;
        k1=0.01*runge(a,p);
        k2=0.01*runge(a+0.005,p+(k1/2));
        k3=0.01*runge(a+0.005,p+(k2/2));
        k4=0.01*runge(a+0.01,p+(k3/2));
        k=(k1+2*k2+2*k3+k4)/6;
        p=p+k;
        t=a;
        x=sin(t);
        e=pow((R*R-(x)*x),0.5);
        v_s=(pi*(d*d)*b)/2;
        v=v_s/(r0-1) + (pi/4)*(d*d)*(l+b-b*cos(t)-b*e);
        dV=v_s*0.5*sin(t)*(1+(cos(t))/e);
        power=power+p*dV;
        j1=0;
        j2=0;
        j3=0;
        j4=0;
        j1=0.01*runge2(a,T,p);
    }
}
```
\[ j_2 = 0.01 \times \text{runge2}(a + 0.005, T + (j_1/2), p); \]
\[ j_3 = 0.01 \times \text{runge2}(a + 0.005, T + (j_2/2), p); \]
\[ j_4 = 0.01 \times \text{runge2}(a + 0.01, T + (j_3/2), p); \]
\[ j = (j_1 + 2j_2 + 2j_3 + j_4)/6; \]
\[ T = T + j; \]

\[
\text{printf}("%f\t", a \times 180/\pi); \\
\text{fprintf}(fp, "%f\t", a \times 180/\pi); \\
\text{printf}("%f\t", p/100000); \\
\text{fprintf}(fp, "%f\t", p/100000); \\
\text{fprintf}(fp, "%f\t", v); \\
\text{printf}("%f\n", T); \\
\text{fprintf}(fp, "%f\n", T); \\
\}
\]
\[ a_1 = 350; \text{start of injection} \]
\[ a_2 = 390; \text{end of injection} \]
\[ t_1 = (a_1/180) \times \pi; \]
\[ t_2 = (a_2/180) \times \pi; \]
\[ \text{for}(a = t_1; a \leq t_2; a = a + 0.01) \quad \text{//combustion loop} \]
\[ \}
\[ k_1 = 0; \]
\[ k_2 = 0; \]
\[ k_3 = 0; \]
\[ k_4 = 0; \]
\[ k_1 = 0.01 \times \text{runge1}(a, p); \]
\[ k_2 = 0.01 \times \text{runge1}(a + 0.005, p + (k_1/2)); \]
\[ k_3 = 0.01 \times \text{runge1}(a + 0.005, p + (k_2/2)); \]
\[ k_4 = 0.01 \times \text{runge1}(a + 0.01, p + (k_3/2)); \]
\[ k = (k_1 + 2k_2 + 2k_3 + k_4)/6; \]
\[ p = p + k; \]
\[ t = a; \\
\]
\[ x = \sin(t); \]
\[ e = \text{pow}((R \times R - (x) \times x), 0.5); \]
\[ v_s = (\pi \times (d \times d) \times b)/2; \]
\[ v = v_s/(r_0 - 1) + (\pi/4) \times (d \times d) \times (1 + b \times \cos(t) - b \times e); \]
\[ \text{dV} = v_s \times 0.5 \times \sin(t) \times (1 + \cos(t))/e; \]
\[ \text{power} = \text{power} + p \times \text{dV}; \]
\[ j_1 = 0; \]
\[ j_2 = 0; \]
\[ j_3 = 0; \]
\[ j_4 = 0; \]
\[ j_1 = 0.01 \times \text{runge3}(a, T, p); \]
\[ j_2 = 0.01 \times \text{runge3}(a + 0.005, T + (j_1/2), p); \]
\[ j_3 = 0.01 \times \text{runge3}(a + 0.005, T + (j_2/2), p); \]
\[ j_4 = 0.01 \times \text{runge3}(a + 0.01, T + (j_3/2), p); \]
\[ j = (j_1 + 2j_2 + 2j_3 + j_4)/6; \]
\[ T = T + j; \]

\[
\text{printf}("%f\t", a \times 180/\pi); \\
\text{fprintf}(fp, "%f\t", a \times 180/\pi); \\
\text{printf}("%f\t", p/100000); \\
\text{fprintf}(fp, "%f\t", p/100000); \\
\text{fprintf}(fp, "%f\t", v); \\
\text{printf}("%f\n", T); \\
\text{fprintf}(fp, "%f\n", T); \\
\}
\]
\[ a_1 = 390; \text{end of injection} \]
\[ a_2 = 490; \]  
1: \[ t_1 = \left(\frac{a_1}{180}\right) \pi; \]  
2: \[ t_2 = \left(\frac{a_2}{180}\right) \pi; \]  

for (a = t_1; a <= t_2; a = a + 0.01) // expansion loop

{ 
    k1 = 0;
    k2 = 0;
    k3 = 0;
    k4 = 0;
    k1 = 0.01 * runge(a, p);
    k2 = 0.01 * runge(a + 0.005, p + (k1 / 2));
    k3 = 0.01 * runge(a + 0.005, p + (k2 / 2));
    k4 = 0.01 * runge(a + 0.01, p + (k3 / 2));
    k = (k1 + 2 * k2 + 2 * k3 + k4) / 6;
    p = p + k;

    t = a;
    x = sin(t);
    e = pow((R * R - (x) * x), 0.5);
    v_s = (pi * (d * d) * b) / 2;
    v = v_s / (r0 - 1) + (pi / 4) * (d * d) * (l + b - b * cos(t) - b * e);
    dV = v_s * 0.5 * sin(t) * (1 + cos(t)) / e;
    power = power + p * dV;

    j1 = 0;
    j2 = 0;
    j3 = 0;
    j4 = 0;
    j1 = 0.01 * runge2(a, T, p);
    j2 = 0.01 * runge2(a + 0.005, T + (j1 / 2), p);
    j3 = 0.01 * runge2(a + 0.005, T + (j2 / 2), p);
    j4 = 0.01 * runge2(a + 0.01, T + (j3 / 2), p);
    j = (j1 + 2 * j2 + 2 * j3 + j4) / 6;
    T = T + j;

    printf("%f	", a * 180 / pi);
    fprintf(fp, "%f	", a * 180 / pi);
    printf("%f	", p / 100000);
    fprintf(fp, "%f	", p / 100000);
    fprintf(fp, "%f", v);
    printf("%f", T);
    fprintf(fp, "%f", T);

}  

// Indicated Power  
IP = power * (n / 120);
printf("Indicated power is %f Watt\n", IP);
fprintf(fp, "Indicated power is %f Watt\n", IP);

// Friction power  
vm_pis = d * b * n / 60;
printf("%f\n", vm_pis);
// printf("%f\n", vm_pis);
p_f = 75000 + (1.44 * vm_pis / d) + 0.4 * vm_pis * vm_pis;
printf("Pressure loss due to friction is %f Pascal\n", p_f);
fprintf(fp, "Pressure loss due to friction is %f Pascal\n", p_f);
f_p = 2 * v * s * p_f * (n / 120);
printf("Friction power is %f Watt\n", f_p);
fprintf(fp, "Friction power is %f Watt\n", f_p);
fclose(fp);
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ISSN: 2454-1362, http://www.onlinejournal.in

getchar();
}

double runge(double a, double p)
{
    double r0,p,i,d,b,l,r1,R,t,x,e,v_s,v,dV,dpdx;

    r0=18;
pi=3.141592654;
d=.111;
b=.0635;
l=.251;
r1=287;
R=l/b;
t=a;
x=sin(t);
e=pow((R*R-(x)*x),0.5);
v_s=(pi*(d*d)*b)/2;
v=v_s/(r0-1) + (pi/4)*(d*d)*(l+b-b*cos(t)-b*e);
dV=v_s*0.5*sin(t)*(1+(cos(t))/e);
    //printf("%f",dV);

    dpdx=(p/v)*(-dV)*((r1/896.5)+1);
    //fprintf(fp,"%f","v);
    return(dpdx);
}

double runge1(double a, double p)
{
    double r0,p,i,d,b,l,r1,R,t,x,e,v_s,v,dV,dpdx;

    double m_p,t0,hv,t_p,w_fp,dQ,u,g;

    r0=18;
pi=3.141592654;
d=.111;
b=.0635;
l=.251;
r1=287;
R=l/b;
t=a;
m_p=1.5;
t0=1.9166*pi;
hv=4.2e7;
t_p=0.2222*pi;
w_fp=.000035;

g=pow(((t-t0)/t_p),(m_p+1));

dQ=hv*w_fp*(6.908*(m_p+1)/t_p)*(pow(((t-t0)/t_p),m_p))*exp(-6.908*g);

    x=sin(t);
e=pow((R*R-(x)*x),0.5);
v_s=(pi*(d*d)*b)/2;
v=v_s/(r0-1) + (pi/4)*(d*d)*(l+b-b*cos(t)-b*e);
dV=v_s*0.5*sin(t)*(1+(cos(t))/e);
    //printf(fp,"%f",v);

    dpdx=(1/v)*(((r1/896.5)*dQ)-(p*dV)*((r1/896.5)+1));
    return(dpdx);
}
double runge2(double a, double T, double p)
{
    double r0, pi, d, b, l, r1, R, t, x, e, v_s, v, dV, dpdx;
    r0 = 18;
    pi = 3.141592654;
    d = 111;
    b = 0.0635;
    l = 0.251;
    r1 = 287;
    R = l / b;
    t = a;
    x = sin(t);
    e = pow((R * R - (x) * x), 0.5);
    v_s = (pi * (d * d) * b) / 2;
    v = v_s / (r0 - 1) + (pi / 4) * (d * d) * (l + b * cos(t) - b * e);
    dV = v_s * 0.5 * sin(t) * (1 + (cos(t) / e));
    double w, n, vpis, hg, kh, mg, s_s, sg, tw, dt; dx;
    n = 3000;
    kh = 1;
    tw = 300;
    mg = 1e-3;
    w = n * 2 * pi / 60;
    vpis = 4 / (pi * d * d) * dV * w;
    //printf("%f", vpis);
    hg = kh * (pow((p / 1e5), 0.8)) * ((vpis) + 1.4) / ((pow(T, 0.4)) * pow(v, 0.06));
    //printf("%f\n", hg);
    s_s = pi * d * b;
    sg = s_s / (r0 - 1) + (pi) * (d) * (l + b * cos(t) - b * e);
    //printf("%f\n", dV / v);
    dt = (0.2 / (mg * r1)) * ((hg * sg * (tw - T)) / w) - (0.2 * T * dV / v);
    //printf("%f\n", dV);
    return(dt); dx;
}

double runge3(double a, double T, double p)
{
    double r0, pi, d, b, l, r1, R, t, x, e, v_s, v, dV, dpdx;
    double m_p, t0, hv, t_p, w_fp, dQ, u, g;
    r0 = 18;
    pi = 3.141592654;
    d = 111;
    b = 0.0635;
    l = 0.251;
    r1 = 287;
    R = l / b;
    t = a;
    m_p = 1.5;
    t0 = 1.9166 * pi;
    hv = 4.2e7;
    t_p = 0.2222 * pi;
    w_fp = 0.00035;
    g = pow(((t - t0) / t_p), (m_p + 1));
    x = sin(t);
    e = pow((R * R - (x) * x), 0.5);
v_s=(π*(d*d)*b)/2;
v=v_s/(r_0-1) + (π/4)*(d*d)*(l+b-b*cos(t)-b*e);
dV=v_s*0.5*sin(t)*(1+(cos(t))/e);

double w,n,vpis,hg,kh,mg,s_s,sg,tw,dtdx;
n=3000;
kh=1;
tw=300;
mg=.00154;
w=n*2*pi/60;
vpis=(4/(π*d*d))*dV*w;
//printf("%f",vpis);

hg=kh*(pow((p/1e5),0.8))*((vpis)+1.4)/((pow(T,0.4))*pow(v,0.06));
//printf("%f",hg);
s_s=π*d*b;
sg=s_s/(r_0-1) + (π)*(d)*(l+b-b*cos(t)-b*e);
//printf("%f",dV/v);
dtdx=(0.2/(mg*r_1))*(hg*sg*(tw-T)/w)-(0.2*T*dV/v)+dQ;
//printf("%f",dV);
return(dtdx);