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Computer Simulation of Compression Ignition Engine through MATLAB

K. Hareesh[1*], Rohith Teja. N[2], B. Konda Reddy[3]
[1,2] Students in Bachelor of Technology, [3]Asst. Professor in Department of Mechanical Engineering.
[1,2,3] *Rajiv Gandhi University of Knowledge Technology, RK Valley, Andhra Pradesh – 516329.*

Abstract: In the present work, a computer simulation has been developed using MATLAB to determine the performance of a four stroke Compression Ignition internal combustion (IC) engine. The modeling of this process begins with the simulation of one cylinder of the four stroke IC engine which is assumed to have an ideal pressure-volume (p-V) relationship allowing for computation of peak performance. The computer simulation is modeled for Ideal Cycle System with encryption of thermodynamic laws of heat transfer and then it is also modeled for the prediction of emissions. The second phase of the model focuses on fuel cycle system where all the real factors are to be considered for the prediction of performance parameters and emissions. Along with the thermodynamic model to compute heat release some standard models like Woschni and Annand models are also used to predict the heat release. Performance parameters computed include brake power and brake specific fuel consumption for an engine's entire operating range.

Keywords: Computer simulation, IC engines, Ideal cycle system, Heat release models, Performance parameters, Specific fuel consumption, MATLAB simulation.

I. INTRODUCTION

One of the major polluting contributors to our environment today is the internal combustion engine, either in the form of spark ignition (Otto) or Diesel versions. In parallel to this serious environmental threat, the main source of fuel for these engines, crude oil, is being depleted at high rates, so that the development of less polluting and more efficient engines is today of extreme importance for engine manufacturers. Also, to this end, the fact of the increasing threat posed by the rivals of the internal combustion engine, for smaller size engines, such as the electric motors, the hybrid engines, the fuel cells and the like corroborates the importance [1].

Experimental work aimed at fuel economy and low pollutants emissions from Diesel and Otto engines includes successive changes of each of the many parameters involved, which is very demanding in terms of money and time. Today, the development of powerful digital computers leads to the obvious alternative of simulation of the engine performance by a mathematical model. In these models, the effects of various design and operation changes can be estimated in a fast and non-expensive way, provided that the main mechanisms are recognized and correctly modeled [2, 3].

The process of combustion in a Diesel engine is inherently very complex due to its transient and heterogeneous character, controlled mainly by turbulent mixing of fuel and air in the fuel jets issuing from the nozzle holes. High speed photography studies and in-cylinder sampling techniques have revealed some interesting features of combustion [4]. The first attempts to simulate the Diesel engine cycle substituted the ‘‘internal combustion’’ by ‘‘external heat addition’’. Apparent heat release rates were empirically correlated to fuel injection rates and eventually used in a thermodynamic cycle calculation to obtain the cylinder pressure in a uniform mixture [5]. Models based on droplet evaporation and combustion, while still in a mono-zone mixture, can only partially take into account the heterogeneous character of Diesel combustion [4].

The need for accurate predictions of exhaust emissions pollutants forced the researchers to attempt developing two zone combustion models [1–4]. Eventually, some multi-zone combustion models have appeared, carrying the expected drawbacks of the first attempts, where the detailed analysis of fuel-air distribution permits calculation of the exhaust gas composition with reasonable accuracy [3-6]. However, this happens under the rising computing time cost when compared to lower zones Diesel combustion models. At this point, it is mentioned that multi-dimensional models have proved useful in examining problems characterized by the need for detailed spatial information and complex interactions of many phenomena simultaneously [6, 7]. However, these are limited by the relative inadequacy of sub-models for turbulence, combustion chemistry and by computer size and cost of operation to crude approximations to the real flow and combustion problems.

Therefore, it is felt that a reasonable choice seems to be a two zone model, which includes the effects of changes in engine design and operation on the details of the combustion process through a phenomenological model where the geometric details are fairly well approximated by detailed modelling of the various mechanisms involved [1, 8]. This is going to have the advantage of relative simplicity and very reasonable computer time cost.

Thus, the object of the present work concerns a comprehensive two zone model, applied to a direct injection (DI) Diesel engine, similar in broad outline to others, but with several differences that one must expect from an independent research source. The model contains upgraded jet mixing, heat transfer and chemistry sub-models, using as simply as possible the numerical analysis treatment of the governing differential and algebraic equations, thus leading to good solution convergence with reduced computer time cost [9, 10].

Extending that work further, the present paper, after exposing a rather short description of the model, as a first step, verifies its validity by using data from a vast experimental investigation. This data is taken at the authors laboratory on a fully automated test bed, four stroke, water cooled, standard ‘‘Hydra’’, direct injection, high speed, Diesel engine. Plots of pressure, temperatures in the two zones, nitric oxide (NO) concentration, soot density, efficiency and other interesting quantities are presented as a function of crank angle, for various loads and injection timings, providing insight into the physical mechanisms governing the combustion and pollutants formation.

After gaining confidence in the predictive capabilities of the model, the second step follows with an extensive investigation of the sensitivity of the model to variation of the constants used in the fuel preparation and reaction sub-models, which are proved critical to the model predictions. For this purpose, the coincident experimental and predicted points are used as baseline values around which changes to these constants are effected. As a feedback, this leads to a better understanding of the physical mechanisms governed by these constants, explaining the behavior of the combustion and pollutants formation for various fuels and conditions used, as reported in the literature. At the same time, this analysis paves the way for the construction of a reliable and relatively simple multi-zone model, which incorporates in each zone (packet) the philosophy of the present two zone model, while it may also be useful for the construction of a combustion model during transient engine operation [8,11,12].

II. THERMODYNAMIC ANALYSIS

Two-zone model views the entire combustion in to burned zone and unburned zone as shown in the Figure 1.

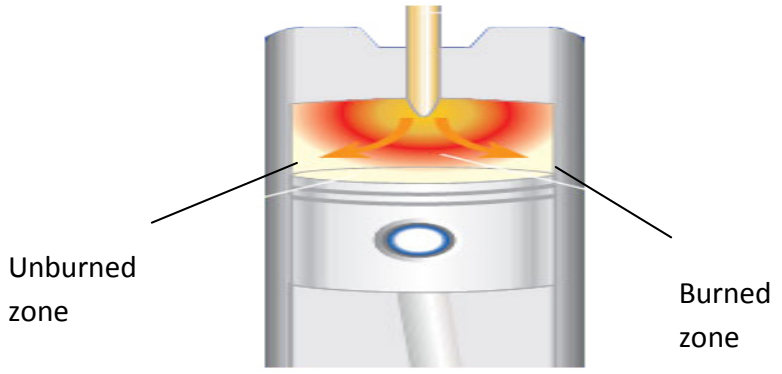

Figure 1: Two Zone Combustion Model

$$m_t = m_a + m_f$$

$$m_b = m_i + \left(m_t \times \chi\left(\theta_i\right)\right)$$

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:

However, the two-zone model recognizes a burned and unburned zone, thus predicting heat transfer and emissions more accurately. The construction of two zone model begins with weibe function to identify burned and unburned regions.

$$\chi(\theta) = 1 - \exp(-6.908\left(\frac{\theta - \theta_i}{\theta_d}\right)^{(b+1)})$$

The mass fraction profile follows the 'S'. So, that's why it is called as S-function. The burn profile is engine-specific.

So, from the mass fraction profile the total amount of mass burned inside the chamber can be find out as:

$$m_a = \frac{P_{\text{int}} \times V_{\text{int}}}{R_g \times T_{\text{int}}}$$

m_f = mass of fuel

$$V = Vcl + \left(\frac{\pi D^2}{4}\right) r\left[1 + \lambda^{-1} - \cos(\theta) - \left(\lambda^{-2} - \sin(\theta)^2\right)^{(1/2)}\right]$$

The peculiarity of this process among all other processes of combustion analysis is, here the total combustion range is divided into six regions such as suction, Compression, Combustion, expansion, constant volume heat rejection and exhaust. All the processes in the above listed will follow the ideal relations show in the below Figure 2.

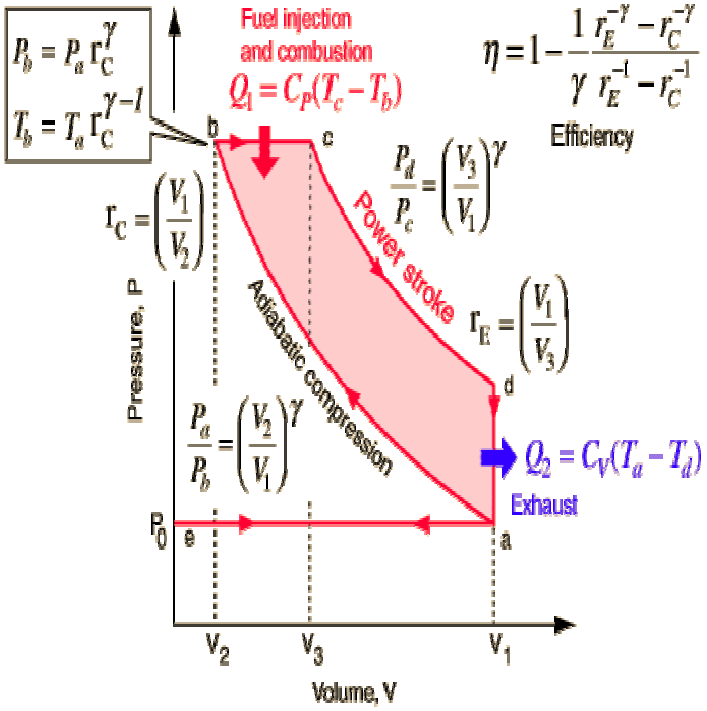

Figure 2: Diesel Cycle

But, the prediction of performance parameters in combustion zone is bit complicated and doesn't follow the ideal relations. Generally in the combustion region, Volume and Temperature does not remains constant which leads to the increasing of pressure as the combustion proceeds. Inorder to find the pressure the temperature has to be obtained by using heat equations shown below:

$$Q_{in} = m_b \times LHV \times \chi(\theta) = m_u \times C_p \times (T_{new}(\theta) - T)$$

From the above relation new temperature can be obtained as

$$T_{new}(\theta) = T + \frac{m_b \times LHV \times \chi(\theta)}{m_u \times C_p}$$

Temperature increases rapidly during the combustion and then decreases due to heat convection by cooling water. The temperature in the total cycle changes as show in below Figure 3.

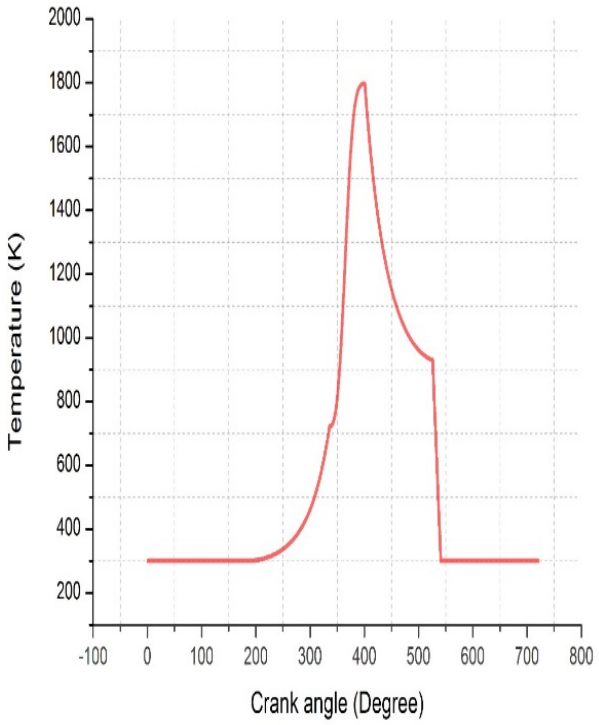

Figure 3: Variation of Temperature in combustion zone

Now the new pressure can be obtained as

$$P_{new}(\theta) = \frac{m_t \times R_g \times T_{new}(\theta)}{V(\theta)}$$

In actual combustion process unlike the idea case the pressure doesn't remain constant at heat addition process, rather it reaches a maximum value at maximum heat addition point and then decreases. This is because that there was no assumption that the pressure remains constant in the region of combustion. The following Figure 4a, 4b justify the above information and also gives an idea of PV-Diagram:

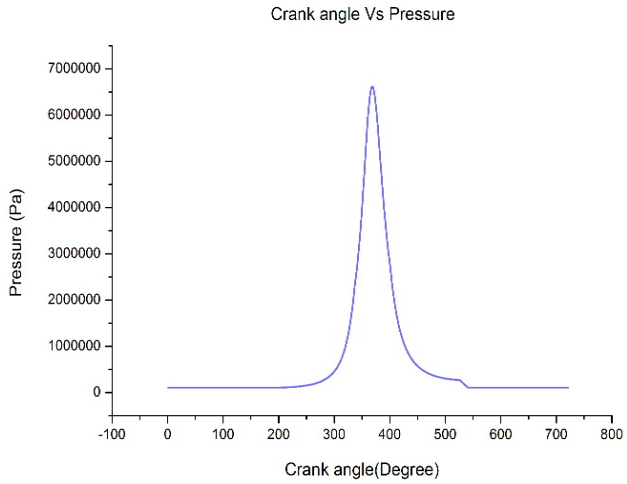

Figure 4(a): Pressure Vs Theta

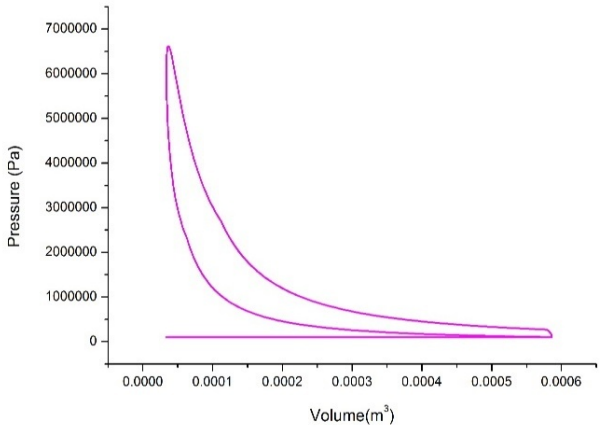

Figure 4(b): PV Diagram

A. Rated Powers and Efficiencies

This section starts with the calculation of total work done in the cycle from where mean effective pressure can be calculated which is equal to the Indicated mean effective pressure when the friction is neglected [5]. The work output can be obtained from

$$W = P_{max}\left(V_3 - V_2\right) + \frac{P_3V_3 - P_4V_4}{\gamma - 1} - \frac{P_{max}V_2 - P_1V_1}{\gamma - 1}$$

Then the Indicated power is

$$I_p = W \times \left(\frac{n}{60}\right)$$ (KW), where n = N/2 for four stroke engine

From the I_p mean effective pressure was obtained from,

$$imep = \frac{60000 \times I_p}{LAnK} \text{ (Pa)}$$

Friction mean effective pressure for direct injection diesel engine is given [6] as

$$fmep = C + 48 \times \left(\frac{RPM}{1000} \right) \times 0.4 \times U_p{}^2$$

Where C is constant which is equal to 75 Kpa and Up is the mean piston speed.

Then friction power is,

$$F_p = \frac{fmep \times LAnK}{60000}$$

Now, the brake Power and brake mean effective pressure are obtained from the following relations

$$B_p = I_p - F_p$$

$$bmep = \frac{60000 \times B_p}{LAnK}$$

Once the all rates powers were obtained calculation of their efficiencies becomes easy. The brake thermal efficiency is obtained [4] as

$$BTH = \frac{B_p}{MFVL \times CV}$$

Where $MFVL$ is the fuel in terms of Kg/s

Then the brake specific fuel consumption is determined as

$$BSFC = \frac{MFVL \times 3600}{B_p}$$

The brake specific fuel consumption varies with the load, fuel and engine model. It is high at full load condition where maximum power output is generated in the form of brake power [4].

B. Heat Release Models

This section starts with the first law of thermodynamics for the four stroke engine energy balance [3] is

$$\Delta U = Q - W$$

Where Q the total energy is transferred into the system, W is the work transferred out of the system, and ΔU is the change in internal energy within the system. In differentiating equation the above equation with respect to $d\theta$

$$\frac{dU}{d\theta} = \frac{dQ}{d\theta} - \frac{dW}{d\theta} = mC_v\left(\frac{dT}{d\theta}\right)$$

Where C_v is the specific heat of the combustion chamber gas. Upon dividing the specific heat by the universal gas constant [9],

$$\frac{C_v}{R} = \frac{C_v}{C_p - C_v} = \frac{1}{\gamma - 1}$$

Where γ is the specific heat ratio.

The above equations can be used to describe the formation of work and the net heat input:

$$\frac{dW}{d\theta} = P\left(\frac{dV}{d\theta}\right)$$

$$\frac{dQ_n}{d\theta} = \frac{dQ_{in}}{d\theta} - \frac{dQ_{loss}}{d\theta}$$

Which can be expanded as,

$$\frac{dQ_n}{d\theta} = m_f \times LHV \times \left(\frac{d\chi}{d\theta}\right) - \frac{dQ_{loss}}{d\theta}$$

If we further implements the heat model to get net heat with respect to crank angle [3], the equation becomes:

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1}P\frac{dV}{d\theta} + \frac{1}{\gamma - 1}V\frac{dP}{d\theta}$$

However, it is also necessary to continue discussion to get more accuracy for more complex heat release models. Lastly the change in pressure [6] is defined by:

$$\frac{dP}{d\theta} = \frac{\gamma - 1}{V}\left(\frac{dQ_{in}}{d\theta} - \frac{dQ_{loss}}{d\theta}\right) - \gamma\frac{P}{V}\frac{dV}{d\theta}$$

The rate of heat loss can be expressed as

$$\frac{dQ_{loss}}{d\theta} = h(\theta)A(\theta)(T(\theta) - T_w)\left(\frac{1}{\omega}\right)$$

This can be modelled by two models namely woschni's heat transfer model and Annand's heat transfer model.

C. Woschni's Heat Release Model

Woschni's method is a set of empirical equations that predicts the heat transfer coefficient between in-cylinder gasses and walls [5]. The convective losses between in-cylinder gasses and walls can be predicted using Newton's law of cooling:

$$\frac{dQ_{ht}}{d\theta} = h(\theta)A(\theta)(T_\theta - T_w)\left(\frac{1}{\omega}\right) + 5.76\left(\frac{T_\theta^{\,4}}{100} - \frac{T_w^{\,4}}{100}\right)$$

$h(\theta)$ is the instantaneous heat transfer coefficient, $A(\theta)$ is the instantaneous heat transfer area, T_θ is the instantaneous bulk gas temperature, and T_w is the cylinder wall temperature. With the heat loss modelled angularly, the convective heat transfer coefficient is now defined as:

$$h(\theta) = 3.26D^{-0.2}P^{0.8}T^{-0.55}w^{0.8}$$

Where that w denotes the woschni's factor which also changes with the crank angle and it is expressed [11] [12] [4] as,

$$w(\theta) = 2.28U_p + C_1 \frac{V_\theta T_r}{P_r V_r}(P(\theta) - P_m)$$

Where C_1 the constant is varies with the process. For combustion process C_1=0.00324, U_p is mean piston speed, V_θ is the instantaneous volume, T_r is the reference temperature, P_r is the reference pressure, V_r is the reference volume, $P(\theta)$ instantaneous pressure at the combustion period and P_m is the motored cylinder pressure.

In compression and expansion processes, Watson and Janota [6] suggested modelling the motored cylinder pressure as a polytrophic process:

$$P_m = P_r\left(\frac{V_r}{V_\theta}\right)^\gamma$$

With all of these variables previously expressed, the convective heat transfer coefficient and corresponding heat loss could then be calculated.

The change in convective heat transfer coefficient with the crank angle during combustion according woschni's has represented in the Figure 5.

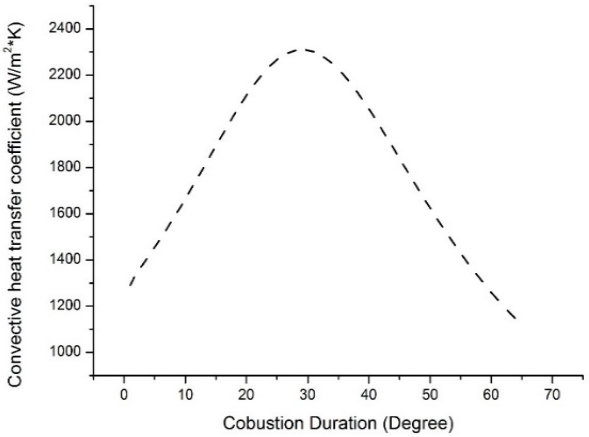

Figure 5: Rate of change of convective heat transfer coefficient with crank angle

D. Annand's Heat Release Model

Annand's and Woschni's heat transfer models differed in the fact that Annand's approach separated the convective and radiation terms. Annand's method solved for the heat transfer coefficient by assuming pipe-like fluid dynamics, and using the in-cylinder density, and Reynolds and Nusselt numbers as functions of time.

Using Annand's method, Newton's law of cooling can be broken into convective and radiation terms [3] as follows:

$$\frac{dQ}{d\theta} = (h_c(\theta) + h_r(\theta))A(\theta)(T_\theta - T_w)\left(\frac{1}{\omega}\right)$$

Where $h_c(\theta)$ is the convective heat transfer coefficient and $h_r(\theta)$ is the radiation heat transfer coefficient. The convective heat transfer coefficient can be extracted from the relationship between the Nusselt number and fluid properties [7] as

$$h_c = \frac{k_{gas} Nu}{D}$$

Where k_{gas} is the gas thermal conductivity, Nu is the Nusselt number, and D is the cylinder bore. With an iterative solver, the thermal conductivity of the cylinder gas can be modelled using a polynomial curve-fitting of experimental data. Heywood [2] suggests using the curve fitted equation:

$$k_{gas} = 6.1944 \times 10^{-3} + 7.3814 \times 10^{-5} T_\theta - 1.2491 \times 10^{-8} T_\theta^{\,2}, \text{ units are } \left(\frac{W}{m * K} \right)$$

The Nusselt number can be described relative to the Reynolds number and the type of engine:

$$Nu = a \,\mathrm{Re}^{0.7}$$

Where a is a constant having a value of 0.26 for a two-stroke engine and 0.49 for a four stroke engine [2], and is the instantaneous Reynolds number. The Reynolds number is expressed as:

$$\mathrm{Re} = \frac{\rho_{gas} U_p D}{\mu_{gas}}$$

Where ρ_{gas} is the instantaneous cylinder gas density, U_p is the mean piston velocity, and μ_{gas} is the instantaneous gas viscosity. Since the model assumes ideal gas behavior, the cylinder gas density can be found by rearranging the ideal gas law:

$$\rho_{gas} = \frac{P}{R_{gas} T}$$

Where R_{gas} is the fluid-specific gas constant, and an assumed value of 287[$\frac{J}{KgK}$] was used for this variable.

As with the thermal conductivity, the cylinder gas viscosity was modelled using empirical equations. According to Heywood [2], the cylinder gas viscosity can be expressed as:

$$\mu_{gas} \left(\frac{kg}{m * s} \right) = 7.457 \times 10.^{-6} + 4.1547 \times 10^{-8} T_\theta - 7.4793 \times 10^{-12} T_\theta^{\,2}$$

Although the radiative heat transfer coefficient is small [2], it was decided that radiation should be included in considering overall heat losses in the model. The radiative heat transfer coefficient is defined as [5]:

$$h_r \left(\frac{w}{m^2 * K} \right) = 4.25 \times 10^{-9} \left(\frac{T_\theta^{\,4} - T_w}{T_\theta - T_w} \right)$$

Where the instantaneous cylinder temperature and wall temperature must be provided in units of [K]. With known pressure and temperature traces from the calculations, Annand's method could then be used to calculate heat losses. A comparison of predicted heat loss rates of both Woishni and Annand model would be shown in the Figure 6.

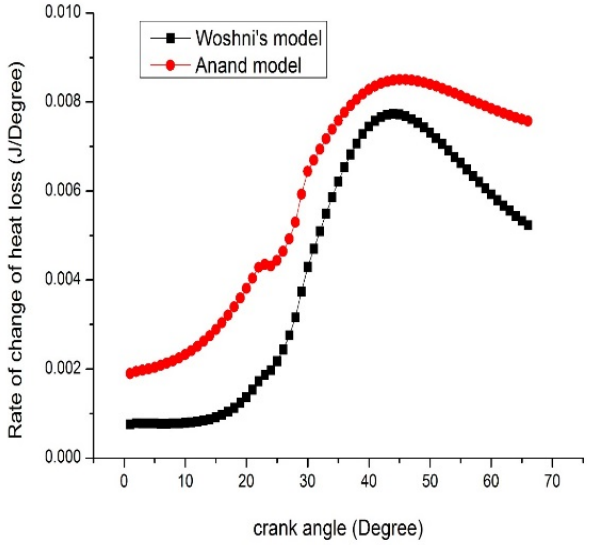

Figure 6: ROHL with respect to crank angle

Total heat losses other than convective, radiative heat transfer are accounted in the below Figure 7.

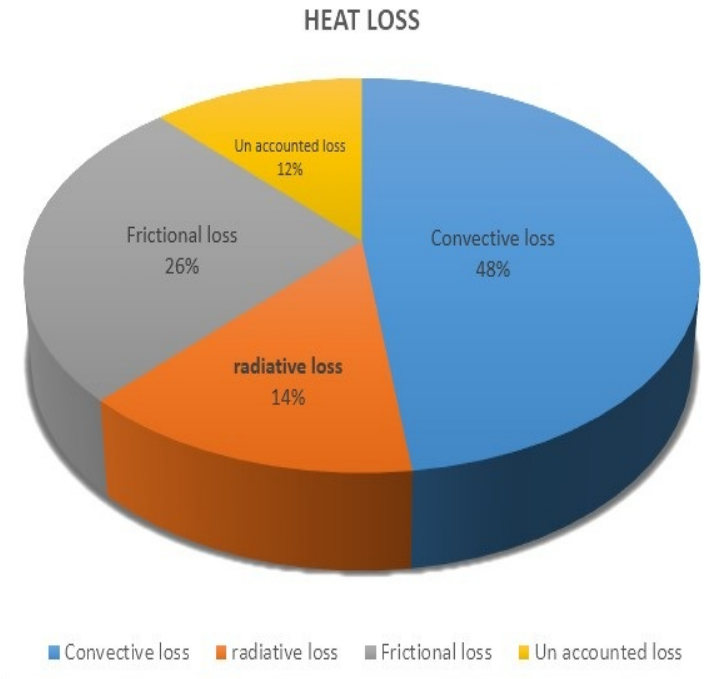

Figure 7: Energy losses

E. NOx Model

While nitric oxide (NO) and nitrogen dioxide (NO_2) are usually grouped together as NOx emissions, nitric oxide is the predominant oxide of nitrogen produced inside the engine cylinder. The principle source of NO is the oxidation of atmospheric nitrogen. However, if the fuel contains significant nitrogen, the oxidation of the fuel nitrogen-containing compounds is an additional source of NO.

The mechanism of the formation of NO has been revised and the principle equations governing the formation of NO has formulated as

O + N2 = NO + N (3.1.1)

N + O2 = NO + O (3.1.2)

N + OH = NO + H (3.1.3)

The forward rate constant for reaction (3.1.1) and the reverse rate constants for reactions (3.1.2) and (3.1.3) have large activation energies which results in a strong temperature dependence of NO formation. And the NO formation rate is given as

$$\frac{d[NO]}{dt} = 2k_1^{+}[O][N_2]\frac{1-[NO]^2/(K[O_2][N_2])}{1+k_1^{-}[NO]/(k_2^{+}[O_2]+k_3^{+}[OH])}$$

Where K = $(k_1^{+}/k_1^{-})(k_2^{+}/k_2^{-})$.

To introduce the equilibrium assumption it is convenient to use the notations

$$R_1 = k_1^{+}[O]_e[N_2]_e = k_1^{-}[NO]_e[N]_e \text{ , for (3.1.1)}$$

$$R_2 = k_2^{+}[N]_e[O_2]_e = k_2^{-}[NO]_e[O]_e \text{ , for (3.1.2)}$$

$$R_3 = k_3^{+}[N]_e[OH]_e = k_3^{-}[NO]_e[H]_e \text{ ,for (3.1.3)}$$

Then the NO formation rate equation becomes

$$\frac{d[NO]}{dt} = \frac{2R_1\{1-([NO]/[NO]_e)^2\}}{1+([NO]/[NO]_e)R_1/(R_2+R_3)}$$

The strong temperature dependence of the NO formation rate makes the formation rate very simple and depends on the equilibrium concentration of Oxygen and Nitrogen and it is given as

$$\frac{d[NO]}{dt} = \frac{6\times10^{16}}{T^{\frac{1}{2}}}\exp\left(\frac{-69090}{T}\right)[O_2]_e^{1/2}[N_2]_e$$

The strong dependence of $\frac{d[NO]}{dt}$ on temperature in the exponential term is evident. High temperatures and high oxygen concentrations result in high NO formation rates. The characteristic time for the NO formation process is

$$\tau_{NO} = \frac{8\times10^{-16}T\exp(58300/T)}{P^{1/2}}$$

By multiplying this time with NOx formation rate we can get the NOx in to PPM as

$$\text{NOx (PPM)} = \left(\frac{d[NO]}{dt}\right)\times\tau_{NO}\times10^6\times\left(\frac{1000}{3600}\right)$$

The change in NOx emission in PPM with respect to temperature is shown in the Figure 8.

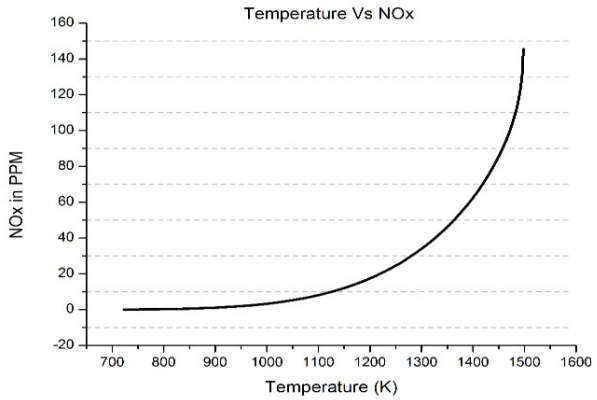

Figure 8: NOx emissions with respect to Temperature.

F. HC Model

Hydrocarbons, or more appropriately organic emissions, are the consequence of incomplete combustion of the hydrocarbon fuel. The level of unburned hydrocarbons (HC) in the exhaust gases is generally specified in terms of the total hydrocarbons concentration expressed in parts per million carbon atoms.

A reasonable fit to the experimental data on unburned HC burn up is the rate expression [9] as

$$\frac{d[HC]}{dt} = -6.7 \times 10^{15} \exp\left(\frac{-18735}{T}\right) x_{HC} x_{O_2} \left(\frac{P}{RT}\right)^2$$

Where x_{HC}, x_{O_2} are the mole fraction of HC and mole fraction of Oxygen respectively.

The main contribution of unburned Hydro Carbons are depicted in the Figure 9.

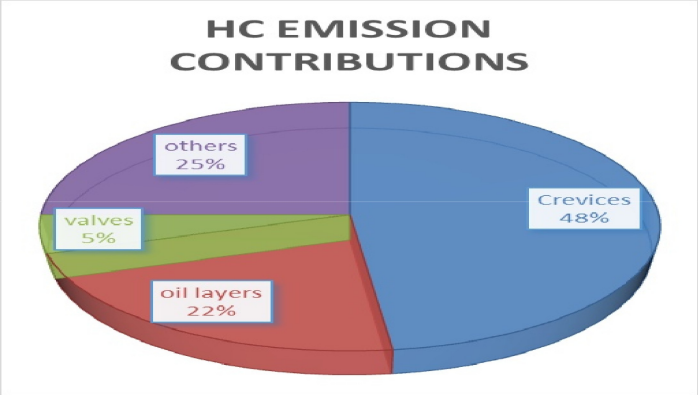

Figure 9: % of factors contributing to HC emissions

G. CO Emissions

Carbon monoxide is an intermediate species in the oxidation of hydrocarbon fuels to CO_2 and H_2O. In fuel-rich regions of a flame, the CO levels are necessarily high since there is insufficient oxygen for complete combustion. Only if sufficient air is mixed with such gases at sufficiently high temperature can the CO be oxidized. Thus, imperfect mixing can allow carbon monoxide to escape from combustors that are operated fuel-lean overall. Even in premixed combustion systems, carbon monoxide levels can be relatively high

due to the high equilibrium concentrations at the flame temperature, particularly in internal combustion engines where the gases are hot prior to ignition due to compression.

As the combustion products are cooled by heat or work transfer, the equilibrium CO level decreases. If equilibrium were maintained as the temperature decreased, carbon monoxide emissions from automobiles and other well-mixed Combustors would be very low in fuel-lean operation. The extent to which CO is actually oxidized, however, depends on the kinetics of the oxidation reactions and the manner of cooling. In this section we explore the kinetics of CO oxidation and the mechanisms that allow CO to escape oxidation in locally fuel-lean combustion.

The predominant reaction leading to carbon monoxide oxidation in hydrocarbon combustion is

$$CO + OH \Leftrightarrow CO_2 + H$$

Where,

$$k_{+1} = 4.4T^{1.5} \exp(372/T) \quad m^3 mol^{-1} S^{-1}$$

The rate of carbon monoxide production for the reaction (4.9.2.1) is given by,

$$R_{+1} = k_{+1}[CO]_e[OH]_e$$

The speed of the reaction is expressed in terms of the characteristic reaction time

$$\tau_{CO} = \frac{[CO]}{R_{+1}}$$

The proportions of the emissions of NOx, HC and CO emissions are shown in the Figure 10.

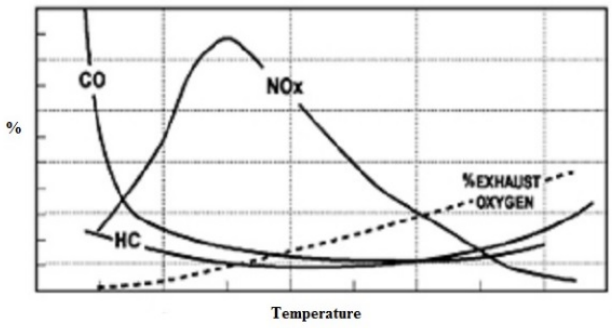

Figure 10: % of emissions with respect to Temperature

III. SIMULATION MODEL

The thermodynamic simulation model through MATLAB is developed to validate the experimental results for a compression ignition engine to reduce the time consumption in manually computing for comparison of actual data with experimental data. The validation of a mathematical model of a structural dynamic system entails the comparison of predictions from the model to measured results from experiments. There are some more numerous, related reasons for performing validations of mathematical models. For example:

1. There may be a need or desire to replace experimentation with model predictions, and, of course, a corresponding requirement that the model predictions have some degree of accuracy. The need for a model may arise from the fact that it is impossible to test system behavior or survivability in some regimes of operation.

2. Alternately, the need for validation may arise from a necessity to prove the reliability of a structure under a broad range of operating conditions or environments. Because it may be expensive to simulate the conditions in the laboratory or realize them in the field, an accurate model is required.

3. Another reason for model validation is that a system may be undergoing changes in design that require analyses to assure that the design modifications yield acceptable system behaviour. A validated model can be used to assess the system behaviour. In all these situations it is useful to confirm analysts' abilities to produce accurate models.

In this work, the task of developing the MATLAB simulation model for compression ignition engine is for testing different fuels at different conditions to select the best to have high efficiency and less emissions.

IV. MATLAB SCRIPT PROCEDURE

In selecting a computer program to execute the demands of a two-zone model, Matrix Laboratory (MATLAB) was considered. It was considered as because of its ease of simulation and speed of validation. With keeping all constraints in mind total script was developed in MATLAB for the future use.

The bulk of MATLAB code was set up through the use of script and the total script is divided in to some sub sections.the purpose of these sub-sections and the organization of the MATLAB model will be elucidated in subsequent sections.

A. Engine Geometry And Atmospheric Inputs

The MATLAB script began with known engine inputs. The bore, stroke, connecting rod length, number of cylinders, compression ratio, and operating characteristics has to mentioned to the program as inputs for the validation of an experimental details. Based on the inputs script would calculate the area of the cylinder, clearance volume of the cylinder and surface area of the piston head. And the atmospheric conditions were chosen like, the initial inlet Temperature 300K (room temperature) and pressure as 1 atm.

B. Pre-allocation of Arrays

Through experimentation, it was found that pre-allocating arrays and matrices drastically improved the efficiency of the program. This prevented MATLAB from having to re-size arrays or matrices between iterations, thus decreasing the overall computation time. Pre-allocated arrays and matrices were also used as a means of setting appropriate properties for the recursion of the program to compare between the graphs of multiple fuel tests.

C. Fuel inputs

Fuel inputs such as mass of the fuel, Calorific value, Lower Heat Values and air-fuel ratio were taken as per the experiment model of the engine.

D. Instantaneous Engine and Fluid Properties

In order to that the main program has divided in to two sub loops where in the first loop with a specified index (i=0:720) calculated instantaneous engine properties discussed in the above section. In addition to the properties of instantaneous properties like volume, pressure and temperature it has also scripted for to calculate work done during the total range of the cycle, Indicated power, friction power, brake power, correction factor for the calculation of brake power and brake specific fuel function.

The second loop with specified index in the range of combustion, MATLAB script has statements to cope up with heat release and heat loss model in Woschni and Annand model.

E. Plot statements

Each plot was sized based on the minimum and maximum variable values, and each plot was given a title appropriate to the variable being plotted. MATLAB script was so developed to have a plots between all the performance parameters as a function of crank angle,

PV diagrams and for the relation between pressure and temperature. Plots were modelled for mass-fraction, heat release and heat loss as a function of crank angle.

F. Emission Predictions

The NO prediction model was included in the MATLAB script, predicted the quantitative fraction of NO particles. The residence time for NO formation was calculated and the integrated amount of NO was calculated in PPM.

V. RESULTS AND DISCUSSION

As the ideal cycle system for four stroke Direct Injection Diesel Engine starts with the calculation of mass fraction profile, it was calibrated and a graph drawn for mass fraction verses crank angle during combustion zone and it is shown in figure 11.

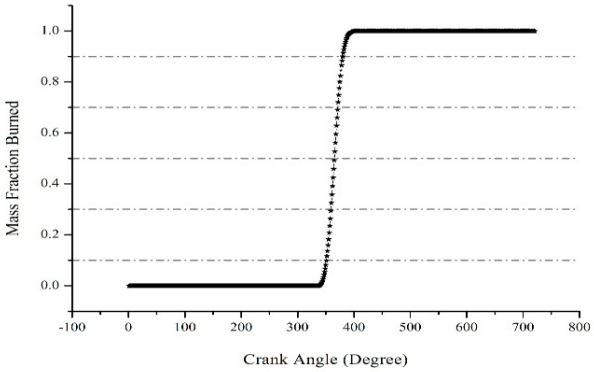

Figure 11: Weib's Mass Fraction profile

Figure 12 shows the Volume profile for the total range of four stroke engine as a function of crank angle and found that it never changes with fuel and it's properties. It is only depends upon the specifications of the engine.

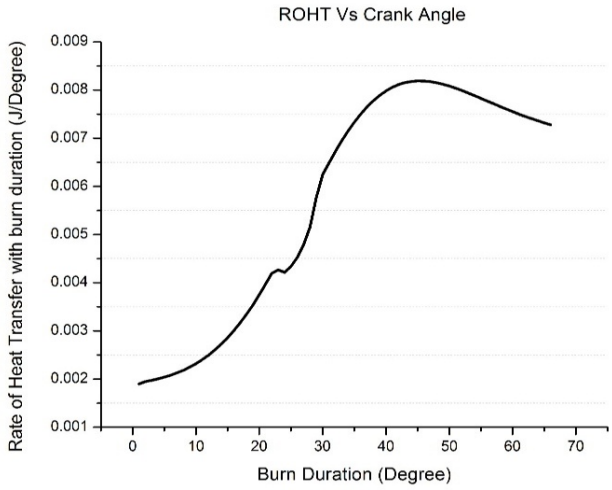

Figure 12: Volume Vs Crank angle

Figure 13 shows the rate of heat transfer during the combustion period according to Thermodynamic analysis.

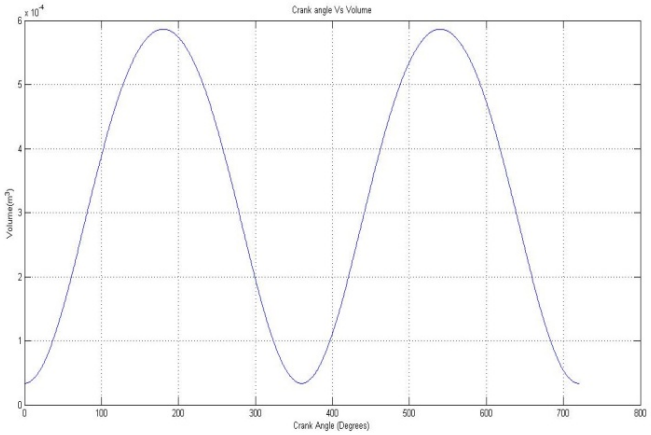

Figure 13: ROHT Vs Crank angle during Combustion

Figure 14 show the difference between the heat transfer coefficient for both Woschni's and Annand heat release models and found they follow same trend as they follow for standard cycles systems.

Using the burned-zone temperature, sub-functions were created to calculate NOx and HC emissions and graphs were generated as shown in Figure 10.

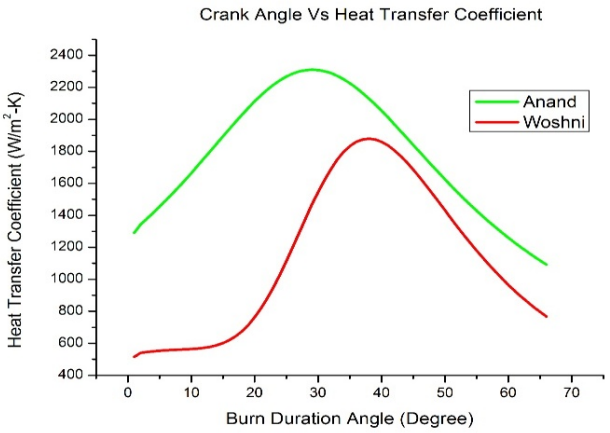

Figure 14: Comparison of Heat transfer Coefficients of both Woschni's and Annand Heat Release Models

VI. CONCLUSION

It was found that the model could be used in simulating any diesel engine. This could save an enormous amount of time in tuning an engine, especially when little is known about the engine. With an air-fuel ratio and volumetric efficiency map, Injection-timing could be optimized, thus minimizing wear-and-tear on the engine and dynamometer equipment. Much research could be directed towards refining the model and using it for the improvement of engine performance and reducing the NOx emissions by testing different fuels.

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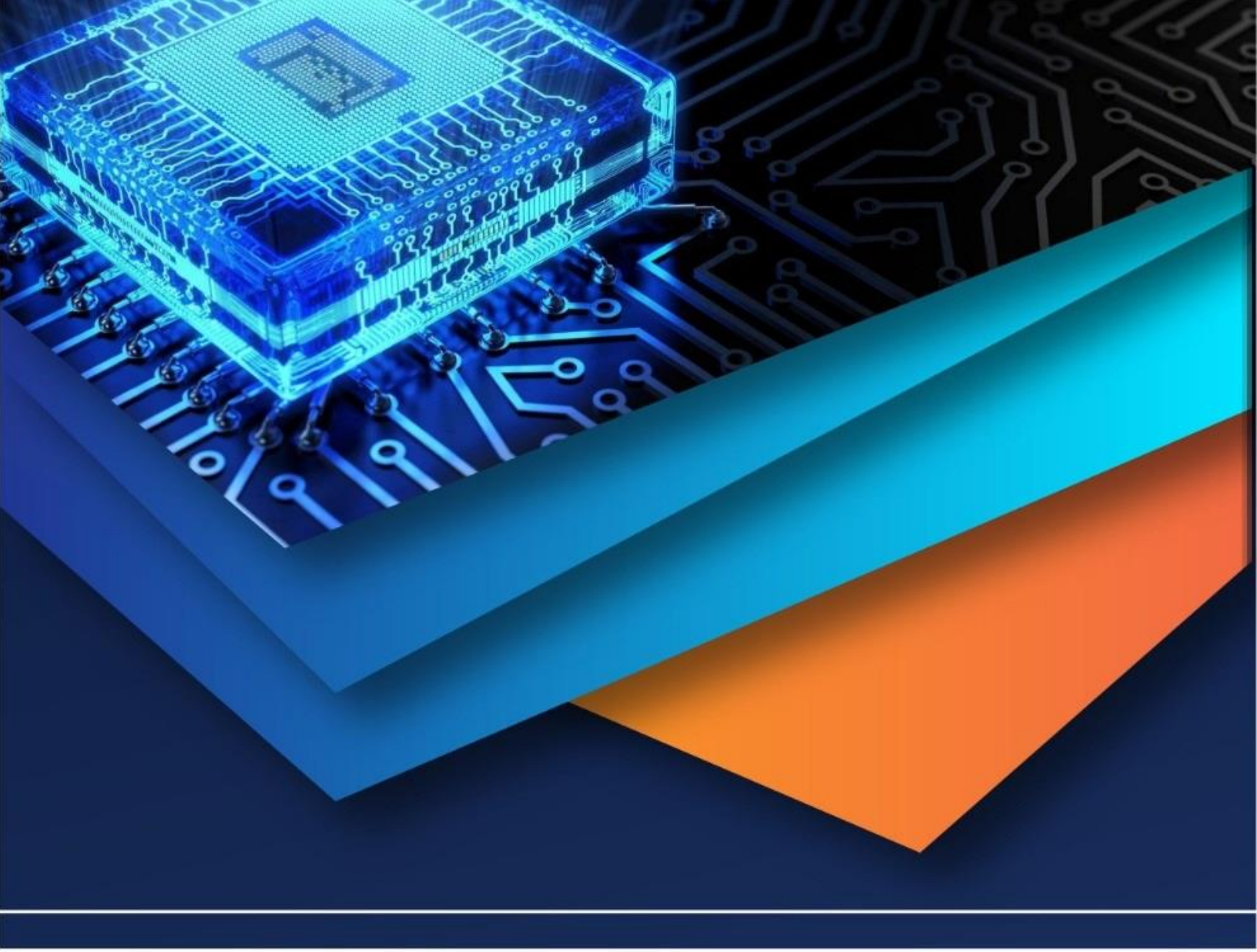

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