Phenomenological three zone model prediction for fuel spray penetration and air entrainment of 4 stroke CI engine

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Abstract: *Current and future legislation on emissions and power development efficiently will require internal combustion engine developers to produce more efficient and cleaner power generation's systems. The increased cost and depletion of natural resources of the hydrocarbon fuel have made the study of the IC engines more attractive and complexities in the process study made the job challenging. Modelling process has come to mean developing stage and using the appropriate combination of assumptions and equations that permit critical features of the process to be analyzed in engineering applications.*

Though multidimensional/fluid dynamic models are expected to give more accurate predictions, it involves huge computational requirements in terms of speed and memory and they are still in the process of development and could not be validated due to experimentation difficulties. Under such conditions, it is better to depend on the well established phenomenological models and try to overcome the difficiences of them and bring them closer to the actual phenomena by refining them as all the processes in this will be represented by ordinary differential equations.

The present paper deals with the three-zone model where the fuel jet is divided into several elements. The combustion process in each element is analysed by considering mixing between the fuel jet and surrounding air and air entrainment into jet as three zones considered are fuel spray zone, fuel air entrainment zone and combustion zone. Treating the process as an open system, models are developed based on contreol volume approach. Mass conservation equation and energy conservation equations are formulated to this open system and solved seperately for these three zones. Experimentally variation of fuel injection timing, fuel jet radius and average jet velocity etc. are compared with the predictions and predictions of KIVA code in this paper and found to almost closer. Hence use of such models can save the time and money and pave the way for further rapid development and optimization of many engine design parameters.

Keywords*: fuel spray penetration, air entrainment, fuel spray zone, fuel air entrainment zone, combustion zone*

INTRODUCTION

Technological advancement in the country and the awareness of the pollution effects on mankind has made the job of the engine modelling highly professional and rewarding. In addition to this, the increased cost and the depletion of natural resources of the hydrocarbon fuel have made the study of the internal combustion engines more attractive and complexities in the process study made the job challenging. Modelling process has come to mean developing stage and using the appropriate combination of assumptions and equations that permit critical features of the process to be analysed in engineering applications.

The main intention of the present investigation is to formulate a mathematical model for the fuel spray, fuel penetration, air entertainment, combustion characteristics and exhaust emission taking into account of three zone model viz. fuel spray zone, fuel air entertainment zone and combustion zone. The validation of the model is done by the predicted in-cylinder pressure variation with the crank angles and pollutant emissions with the available experimental results.

LITERATURE REVIEW:

1. BASIC MODELS IN I.C. ENGINES

In order to understand the processes that govern the engine performance and emissions, two basic models have been developed. They are;

1. Thermodynamic models and
2. Fluid dynamic models

Thermodynamic models are once again divided as zero dimensional models, phenomendogical models, quasidimensional models. Fluid dynamic models are of two dimensional modeling and three dimensional modeling.The modelling in Diesel engine study can be also classified broadly as

a) Zonal modelling
b) Process modelling in closed period and
c) Process modelling in open period

Zonal models can be further classified in to

i) Single zone models
ii) Multi zone models and
iii) Two Dimensional models
iv) Three Dimensional models
v) Multi Dimensional models

2. THERMODYNAMIC MODELS

Basic governing equations that give the predominant structure of thermodynamic models are based on the law of energy conservation [2].

Further classification of the thermodynamic models are

- Zero dimensional models
- Phenomenological models
- Quasidemensional models

2.1 Quasi dimensional Models :

Quasi dimensional models are further divided into two zone and three zone models. The two zone models [7] divide the combustion chamber into burn and unburned zone. With in the burn zone the fuel composition is assumed to be uniform. This is an unrealistic assumption. In the three zone models, fuel jet is divided in to several elements. Combustion process in each element is analysed by considering mixing between the fuel jet and surrounding air and entrainment in to the jet. (Three zone model developments is the main aim of the present work). In this three zone model combustion process in each element is analyzed as a process of mixing between the jet and surrounding air, entrainment in to jet and subsequent combustion. Jet penetration, vaporization and deflection of jet by surrounding air are also considered in the present three zone model investigation.

3. FLUID DYNAMIC BASED MODELS

The principal component of these models is the set of equations that can describe the flow processes. The fluid dynamic based models are often called multi dimensional models due to their inherent ability to provide detailed geometric information. Turbulence model which describes the small scale features of the flow is the important one.

PROBLEM FORMULATION:

1. FORMULATION OF THE THEORETICAL MODEL

The present modelling commences with a consideration of the engine cylinder with its contents as a thermodynamic system. Next, this system is divided into three zones viz, fuel spray zone, air fuel entrainment zone and combustion zone. With in the spray zone, nozzle flow, atomization of fuel, turbulent dispersion of droplets, droplets breakup, droplet collisions and coalescence are to be modelled. The effects of spray evaporation, fuel air mixing are to be introduced in the fuel air entrainment zone. The ignition, combustion and heat release rate formation, pollutants and soot are to be accounted in the combustion zone.

The mathematical equations and empirical relations are formulated with the above concepts in the proposed three zone model and solutions are worked out for the determination temperature and pressure at different levels with the variation of crank angles with in the cylinder Using this open period analysis applicable to the thermodynamic system it is easy to determine different properties like pressure, temperature and volume at different crank angles. Along with these parameters, it is easy to obtain the mass of the fuel and gases within the cylinder. In fact this open period analysis will enable to evaluate the mass of air in the cylinder before the compression stroke is started.

Determination of pressure and volume at each crank angle along with the engine friction enables in the calculation of engine performance for different operating conditions. Engine friction is also modelled by taking various operating parameters and mechanical constraints into account.

2. THREE ZONE MODEL – ENERGY BALANCE EQUATION

In order to explain the concept of three zone model, the cylind volume, which is being in focus is divided into three zones namely; Fuel spray zone, Air fuel entrainment zone and Combustion zone are presented below.

Energy conservation equations for combustion process are formulated and solved separately so that determination of temperatures in different zones with crank angles and heat release rate calculations with the crank angle are possible.

Energy balance equation for the fuel spray zone as well as the combustion zone is as follows [11].

$$\frac{d}{d\theta}(m_c u_c) - \frac{d}{d\theta}(m_s u_s) = \frac{d}{d\theta}(m_{fs}\, h_f) - p\,\frac{dV_s}{d\theta} - \frac{dq_s}{d\theta} + h_n\,\frac{dm_e}{d\theta} \qquad \ldots \quad \text{Eq 1}$$

where,

$\frac{d}{d\theta}(m_c u_c)$ is the rate of change of internal energy of the combustion zone having m_c, the mass of charge involved in combustion.

$\frac{d}{d\theta}(m_s u_s)$ is the rate of change of total internal energy of the fuel spray zone having mass of m_s with the change in the crank angle.

$\frac{d}{d\theta}(m_{fs}\, h_f)$ is the rate of heat release during combustion .

$\frac{dq_s}{d\theta}$ is the rate of heat transfer from the combustion zone.

$h_n \frac{dm_e}{d\theta}$ is the enthalpy transfer into the combustion zone due to fuel air. entrainment

Similarly energy balance for fuel air entrainment zone with different variables is as follows [11]

$$\frac{d}{d\theta}(m_a u_a) = -p\frac{dV_a}{d\theta} - h_a\frac{dm_e}{d\theta} \qquad \text{... Eq 2}$$

and for the closed cycle, energy equation is

$$\frac{d(mu)}{d\theta} = \frac{dQ_c}{d\theta} - \frac{dQ_n}{d\theta} - \frac{d_w}{d\theta} \qquad \text{... Eq 3}$$

Where,

$\frac{d}{d\theta}(m_a u_a)$ is the rate of change of internal energy during the air fuel entrainment .

$h_a \frac{dm_e}{d\theta}$ is the enthalpy change after taking into account of other energies in

to consideration and Q_c,Q_n and W are the heat release due to combustion, net heat transfer and work transfer.

Using gas laws, energy balance equations and heat transfer correlations, computation of different temperatures, pressures and volumes is possible by iterative procedure. In order to start the calculations, guessing of pressure is needed which can be done from experimental knowledge.

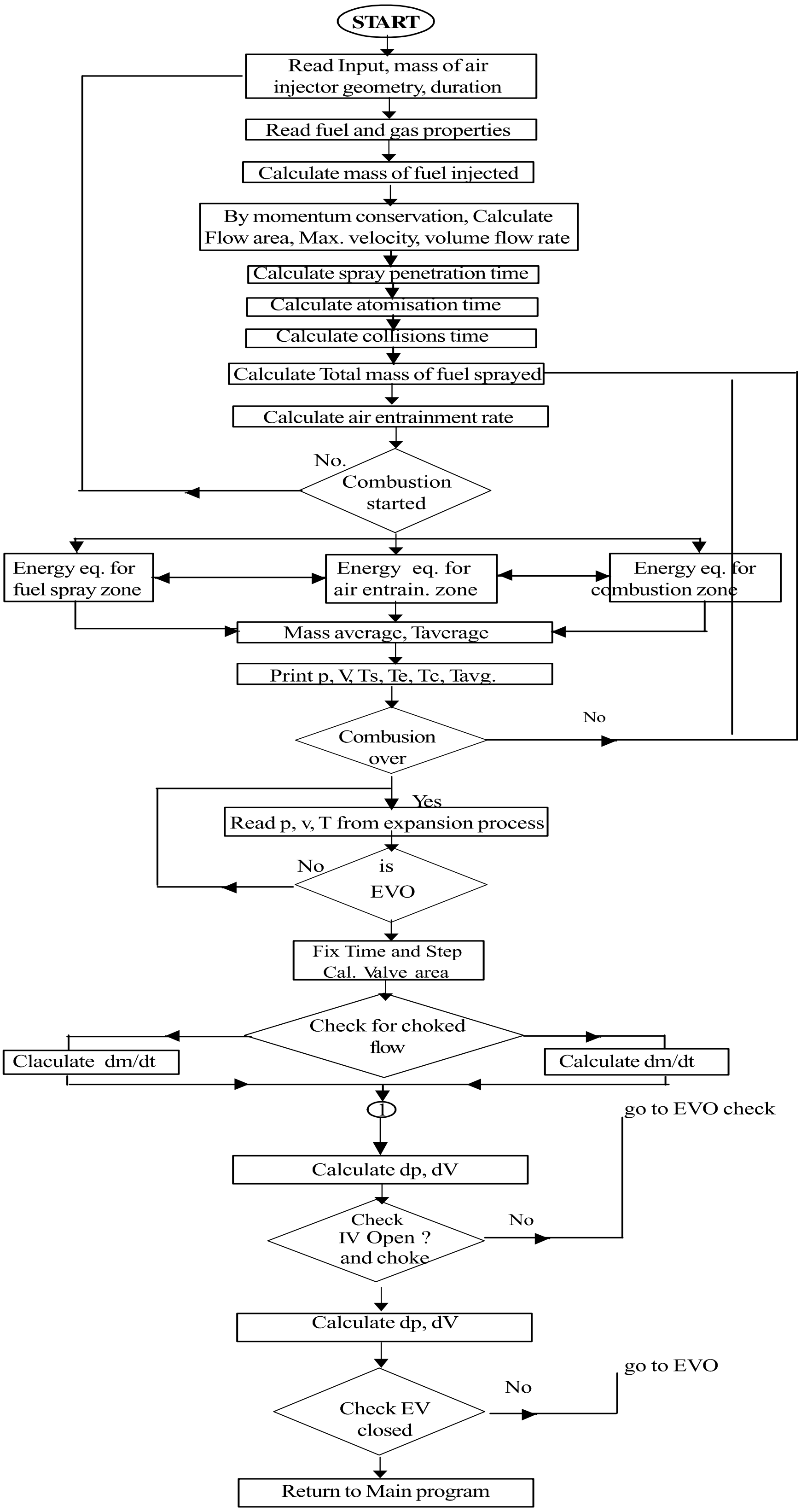
5. EXPERIMENTAL WORK:

The important components of the set up are

i) The engine
ii) Dynamometer
iii) Fuel injection pump
iv) Device for changing / starting of fuel injection
v) Indicator for dynamic injection
vi) Data acquisition equipment
vii) Exhaust gas analyzer
viii) Smoke meter

Different load tests are conducted at the different operating conditions during the experimentation. The cooling water outlet temperature is maintained at 70^0 C separately for both the cylinder liner and the cylinder head. The lubrication oil temperature is maintained at 60^0 C for all the experiments. In the experimentation, load, fuel injection timing, compression ratio and the speed are the four important parameters selected. First, the load is going to vary by keeping other three parameters as constant. Like wise, next FIT is varied, compression ratio is varied and speed is varied respectively after keeping remaining three parameters as constant. At each operating condition, dynamometer load, speed, air flow rate, fuel flow rate, exhaust temperature, manifold pressure, cooling water flow rate, cylinder, head and cylinder liner temperatures, pressure time signal, TDC marker signal, dynamic injection timing, HC, CO and smoke readings are noted and recorded after allowing sufficient time for the engine to stabilize.

The exhaust gas analyzer is switched on quite early so that all its systems will get stabilized before the commencement of the experiment. After the starting of the engine and stabilizing it, air flow, fuel flow, inlet manifold vacuum, temperatures of ambient air, temperature of exhaust gases etc., are noted. The dynamometer readings such as load and speed are also noted. The pressure and TDC signals are recorded on the mini floppy disc, averaged for 100 consecutive cycles.



Flow chart for intake and exhaust processes

Fig 1. Flow chart

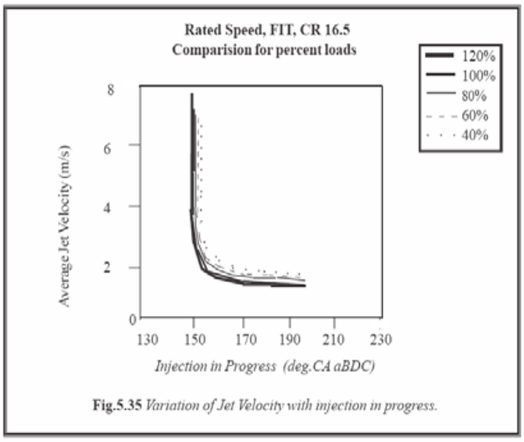


Fig. 2 Variation of jet velocity with injection pressure

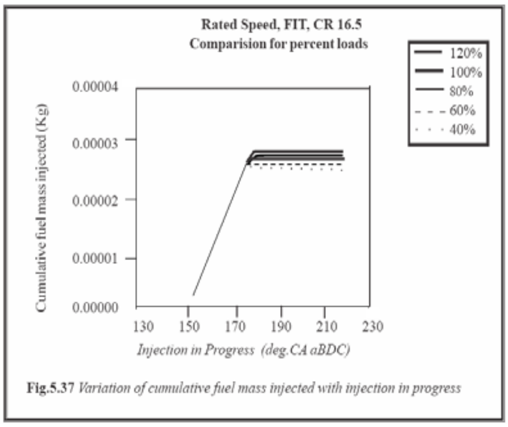


Fig 3. Variation of cumulative fuel mass injected with injection in progress

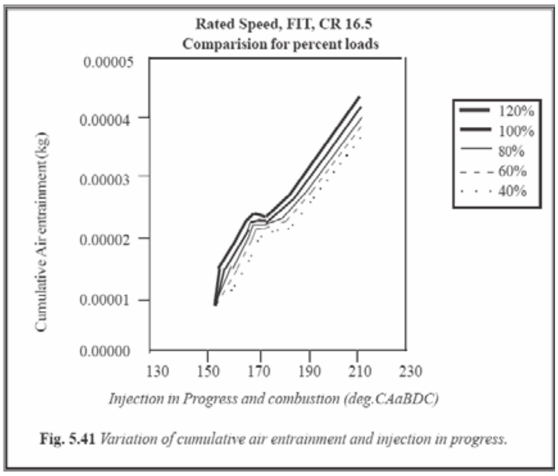


Fig 4. Variation of cumulative air entrainment and in progress

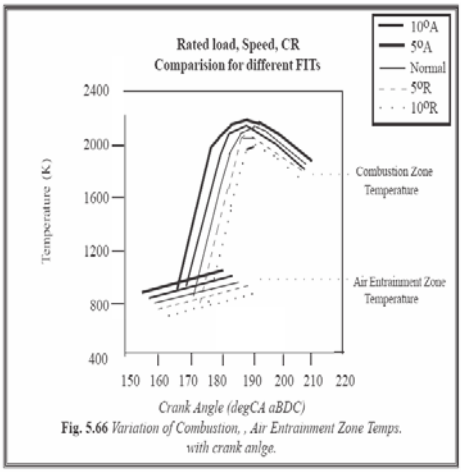


Fig 5.Variation of combustion, air injection entrainment zone temp, with crank angle

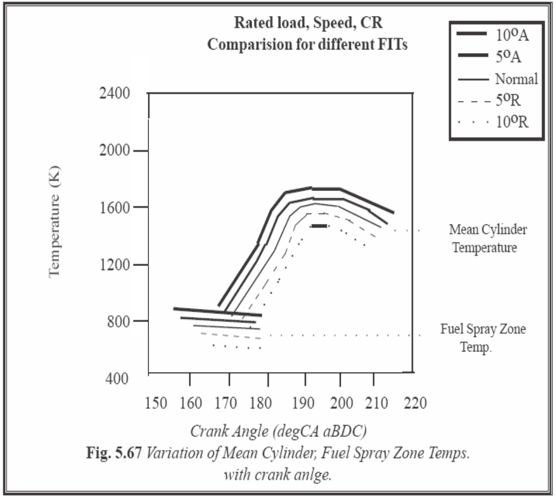


Fig 6. Variation of mean cylinder, fuel spray zone temp, With crank angle

RESULTS AND DISCUSSION:

Figure 2 represents the variation of average jet velocity with the change in the percentage load. As the load is increasing, maximum average jet velocity is also slightly increasing and it is very fastly approaching the minimum velocity. As the load is decreasing, average jet velocity is also decreasing in the predictions of three zone model and KIVA predictions. However, at different loads at random, certain variations are shown to compare with the KIVA predictions and in general compared at different loads.

Figure 3 indicates the variations of the cumulative fuel mass injected with the percentage change of the load. It is observed from the above figure that as the load is increased the amount of cumulative fuel mass injected is proportionately increasing. Similarly for the part load conditions the same proportion is followed in decreasing the amount of cumulative fuel mass injected. But this variation with percent variation in the load shown in the graph is very slight, because the graph is drawn for the cumulative fuel mass injected in Kgs.

Figure 4 indicating the variations observed in the cumulative air entrainment with the duration of injection and combustion at all percent load operations. From observations, it is clear that injection duration and mass of fuel injected increase in the ϕ values results in inadequate oxygen supply and consequently increasing the smoke and carbon monoxide levels (Because in the ϕ value numerator corresponding to fuel injected and denominator corresponding to air entrainment).

Figure 5 is indicating variation of temperatures of combustion zone and air entrainment zone temperatures for different fuel injection timings

Figure 6 is indicating mean cylinder temperature and fuel spray zone temperatures with different fuel injection timings and from all the figures it is clear that the temperatures are increasing with advance in the fuel injection and decreasing with retard of FIT.

CONCLUSION:

Fuel injection end velocity is going to be increased in the part load conditions where as the velocity at the start of injection is almost remaining same from the predictions as well as the KIVA predictions. For every 20 percent decrease from the rated load, there is an increase of 1.5 percent in the end fuel injection velocity is observed. For 20 percent over load conditions, the end velocity is decreased by 2 percent.

- Fuelling rate increasing proportionately with the overload is observed from the predictions but at the same time engine smokes heavily at this condition.

- Cumulative fuel mass injected is increasing with the load increasing. For 20 percent increase, nearly 0.5 grams per cycle is increased with a similar trend observed in the cumulative air entertainment also.

- Exhaust emissions including smoke number are increased with the advance of fuel injection timings and also with the retard conditions. This predicted trend is observed closely with the experimental as well as the KIVA model. A 10 degree advance resulted in the increase by 8 percent and also 10 degree retard resulted in 9 percent increase

- By advancing the fuel injection timing, injection velocity increase is observed. Every 5 degree CA advance is causing 2 percent hike in the mean injection velocity and similar trend of nearly 2.5 percent decrease with the every 5 degree retard from the rated conditions.

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