A Developed Quasi-Dimensional Combustion Model in Spark-Ignition Engines

Seyedmohsen Hosseini, Rushed Abdolah, and Amir Khani

Abstract - Computer simulations of internal combustion engine cycles are desirable because of the aid that they provide in design studies, in predicting trends, in serving as investigations tools, in giving more data than are normally accessible from experiments, and in helping to understand the complex process that occur inside combustion chamber such as auto-ignition and etc. The model developed in this work concern to the combustion of a 4-stroke, single cylinder, Ricardo E6 research engine and is the first step in developing of a comprehensive program aimed at accurately simulating spark ignition engine combustion over a wide range of operating conditions. The accuracy of the present model is confirmed by comparison with experimental result that have established by the effect of engine-operating conditions on the model outputs. The influences of engine-operating variables on the model outputs are components of a parametric study.

Keywords: *Engine-Model-Combustion*

I. Introduction

The simulation of the physical process in engine combustion chambers has found increasing interest during recent years. To validate new design concepts through experimental work takes a long-time and high cost especially during prototype developing stage. The computer simulation techniques are useful alternative way, provided that the simulation model is accurate and fast enough to execute. A quasi-dimensional model for engine combustion was first developed in the 1970 [1] and still used more frequently that multi used more frequently than multi-dimensional simulations. However there is an important distinction between the state of the multi-dimensional simulation of engine combustion and direct numerical solution of the instantaneous governing equations. For combustion in engines, direct numerical simulation is not yet applicable due to the extensive and continuous progressive and spatial scales that must be considered, practical limitations on computation time and cost, and very limited possibility for experimental validation. These models are dependent on coefficients that have not always been validated in relevant experiments due to the difficulty of finding data allowing the validation of an isolated-modeled process.

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II. Simulation Model

The calculation in the current work is achieved by step by step evaluation of the physical and thermodynamic states of the charge in the combustion chamber consecutive points in the engine cycle. Analytical techniques based on iterative procedures. This procedure is used at the end of each step to determine from known initial values. The mean values of thermodynamic properties during compression stroke.

Consequently, three distinct periods are present:

1. Compression of the charge from is state at inlet valve closing to the crank angle at which mixture is ignited following the spark plug firing.
2. Duration of the combustion process while all the charge is being completely burnt.
3. Experimental of the products of combustion to the crank angle at which the exhaust valve opens.

During compression the charge is assumed to be a homogenous mixture of air and fuel vapor. For numerical calculation procedures the step length during this stage of the engine cycle was considered 0.1 deg of crankshaft rotation and each step corresponds with conversion of the chemical energy of the unburned charge (end-gas). Two separate zones (quasi-dimensional model) having equal pressure are involved: the unburned mixture and the burned product.

Again each step during the combustion process, which is divided into two sub-stages: the combustion stage and the piston movement and heat transfer stage, was consider for each 0.1 degree step of crankshaft rotation. The expansion process is essentially similar to the compression process except that burned charge properties replace with those of the unburned charge. Each step again occupies 0.1 deg of crankshaft rotation. All gas mixture are assumed to behave as ideal gases. The thermodynamic properties data for the species of mixture are taken from Joint-Army-Navy-Air Force (JANEF) [3].

III. Calculation of Mass Burning Rate

Mass fraction burn can be calculated either by the Wiebe function or by assuming of flame front travels (flame-speed) through the combustion chamber at the rate given by the following equation [3]:

$$\frac{dm_b}{dt} = \rho_u A_f u_t \qquad (1)$$

Where ρ_u is density of unburned mixture, A_f is the area of flame front, U_t is the velocity of turbulent flame and t is the time. A good approximation for the flame front area is to assume that it is semi-spherical ($2\pi r^2$).

IV. Flame Speed

Laminar flame speed is defined as the velocity of flame front through the end-gas. There are many experimental relations for calculating laminar flame speed. Kuehl [4] suggested an empirical equation for laminar flame speed as a function of pressure, temperature of burned and unburned zones as:

$$U_l = \frac{2.4256*10^4}{[(10^4/T_b)+(900/T_u)^{4.938}} P^{-0.09876} \quad (2)$$

Where U_l is the laminar flame speed (m/s), T_b and T_u are temperatures of burned and unburned mixture ($°K$) and ρ is the pressure of mixture (Pa). Keck and Metghalchi [5] suggestion is another experimental relation for laminar flame speed:

$$U_l = \dot{U}_l \left(\left(\frac{T_u}{T_o}\right)^a \left(\frac{P}{P_0}\right)^b (1-2.1r)\right) \quad (3)$$

Where T_0=298 $°K$, P_0=1 atm, r is the residual gas fraction in the cylinder from the previous cycle and $\dot{U}_l$ is the reference laminar flame speed (m/s), a and b are coefficients that for propane and isooctane are as follow:

$$\alpha = 2.18 - 0.8(\phi - 1) \quad (4)$$

$$b = -0.16 - 0.22(\phi - 1) \quad (5)$$

C_3H_8:

$$\dot{U}_l = 0.342 - 1.387(\phi - 1.08)^2 \quad (6)$$

C_8H_{18}:

$$\dot{U}_l = 0.263 - 1.387(\phi - 1.13)^2 \quad (7)$$

Where ϕ is the equivalence ratio.

V. Turbulent Flame Speed

The velocity of flame propagation increasing in turbulent conditions compared to laminar flame speed. A simple model for turbulent flame speed is to multiply the laminar flame speed by a coefficient [7]:

$$U_t = f U_l \quad (8)$$

Where U_t turbulent flame speed (m/s) and f is a coefficient. Hiroyasu and Kadota [6] have been suggested an empirical equation for calculation of that is a function of engine speed (N):

$$f = 1 + 0.002N$$

Where N is the engine speed (rpm)

VI. Governing Equations of Each Control Volume

The first law of thermodynamic for two-zone combustion model can be written as:

$$U_0 = m x_b [(C_{Vb} T_b + h_b) + (1 - x_b)(C_{v,u} T_u + h_u) + W + Q \quad (9)$$

(9)

Where U_0 is the internal energy, x_b is the mass fraction burned, C is the specific heat, and h is the enthalpy. For each zone, burned and unburned the first law of thermodynamic can be written in following forms:

$$\frac{d(m_u e_u)}{d\theta} = \frac{dQ_u}{d\theta} - P\frac{dV_u}{d\theta} + h_u \frac{dm_u}{d\theta} \quad (10)$$

$$\frac{d(m_b e_{b)}}{d\theta} = \frac{dQ_b}{d\theta} - P\frac{dV_b}{d\theta} + h_b \frac{dm_b}{d\theta} \quad (11)$$

$$e_u = e^0 u + \int_{T_0}^{T} C_{vu} dT \quad :$$

$$e_u = e^0 b + \int_{T_0}^{T} C_{vb} \quad (12)$$

The pressure is assured uniform and is the same in the burned and unburned zones. The total charge is treated as an ideal gas so by using the ideal gas law, the pressure can be written as:

$$P = \frac{m_u T_u R_u + m_b T_b R_b}{V_{total}} \quad (13)$$

Where T_u is the temperature of unburned mixture and T_b is the temperature in the burned mixture. The mass of charge is the sum of the masses of burned and unburned mixture:

$$m = m_b + m_u \quad (14)$$

So:

$$\dot{m}_u + \dot{m}_b = 0 \quad (15)$$

The first law of thermodynamic for the whole control volume (two zones) can be written as:

$$(e_b - e_u)\frac{dm_b}{d\theta} + m_u C_{vu}\frac{dT_u}{d\theta} + m_b C_{vb}\frac{dT_b}{d\theta} + P\frac{dV}{d\theta} + \frac{dQ}{d\theta} = 0 \quad (16)$$

The derivative of the state can be written as:

$$\frac{dV}{d\theta} = \left(\frac{V_b}{m_b} - \frac{V_u}{m_u}\right)\frac{dm_b}{d\theta} + \frac{m_u R_u}{P}\frac{dT_u}{d\theta} + \frac{m_b R_b}{P}\frac{dT_b}{d\theta} - \frac{V}{P}\frac{dP}{d\theta} \quad (17)$$

From the equations (18 to 20) ideal gas relations for each zone, $\frac{dP}{d\theta}, \frac{dT_u}{d\theta}, \frac{dT_b}{d\theta}$, can be calculated from:

$$\frac{dT_u}{d\theta} = \frac{V_u}{m_u C_{pu}} \frac{dP}{d\theta} - \frac{1}{m_u C_{pu}} \frac{dQ_u}{d\theta} \quad (18)$$

$$\frac{dT_b}{d\theta} = \frac{P}{m_b R_b} \left(\frac{dV}{d\theta} - \left(\frac{R_b T_b}{P} - \frac{R_u T_u}{P}\right) \frac{dm_b}{d\theta} - \right.$$

$$\left. \frac{R_u}{C_{pu}} \frac{V_u}{P} \frac{dP}{d\theta} + \frac{1}{P} \frac{R_u}{C_{pu}} \frac{dQ_u}{d\theta} + \frac{V}{P} \frac{dP}{d\theta} \right) \quad (19)$$

$$\frac{dP}{d\theta}\left(\frac{C_{vu}}{C_{pu}} V_u - \frac{C_{vb}}{C_{pu}} V_b + \frac{C_{vb}}{R_b} V\right) - \left(\frac{C_{vu}}{C_{pu}} - \frac{C_{vb}}{C_{pu}} \frac{R_u}{R_b}\right)$$

$$\frac{dQ_u}{d\theta} - \left(1 + \frac{C_{vb}}{R_b}\right) P \frac{dV}{d\theta} + \frac{dm_b}{d\theta} [(e_b - e_u) - C_{vb}$$

$$(T_b - T_u \frac{R_u}{R_b})] - \frac{dQ}{d\theta} \quad (20)$$

In these equations $\frac{dQ_u}{d\theta}, \frac{dQ_b}{d\theta}$ are calculated from the heat transfer relations. When the temperature is known the values of R_u, R_b, C_{vu}, C_{vb} and C_{pu} are calculated from the thermodynamic properties of burned and unburned mixtures.

VII. Taking Data

The combustion model has been developed for a 4 stroke, single-cylinder spark-ignition, Ricardo E6 research engine with variable compression ratio. This section describes the experimental and theoretical investigations of engine output and combustion simulated model results affected by variable parameters. This model has been developed at full operating conditions. Combustion chamber is considered a two zone control volume. The combustion process is modeled thermodynamically with a mass burned rate through an appropriate flame propagation model. The experiments were performed in a Ricardo E6 engine over a wide range of operating conditions. The experimental results and engine operating condition were used to generate the necessary data for validation of model. The combustion model was run at the same condition in order to compare and math the experimental results with the model outputs.

The accuracy of the present model is confirmed by comparison with experimental results that have established the effects of equivalence ratio, compression ratio, speed, and spark timing on cylinder pressure diagram. The influences of engine-operating variable on these diagrams are components of a parametric study. The mathematical model of the compression, combustion and expansion process reported in this work is an attempt to combine the basic characteristics of engine combustion.

VIII. Parametric Studies

To optimize the theoretical designs and the experiments, parametric studies can be conducted in order to investigate the influence and the interaction of each parameter. There are a large number of combinations of the model parameters that can be investigated. To analyses the effect of different parameters on the combustion process within the overall model a series of simulations were carried out by changing the importance of combustion parameters such as compression ratio, spark timing, equivalence ratio, combustion duration, and speed of the engine.

Figure 1 and 2 show the effect of varying each parameter on the simulation results (model output signals). A set of simulation tests was carried out for each replaced parameter, and the output signals were recorded. As seen from these selected typical figures although the predicted cylinder pressure is a response to the important operating parameters such as compression ratio, spark timing, fuel air ratio, speed, and combustion duration. But the pressure is more sensitive to the spark timing and compression ratio. As seen from the figures 1and 2. Also we can see that the rate of pressure rise differs when the parameters vary and may be able to use this specification for prediction of some phenomena such as knock

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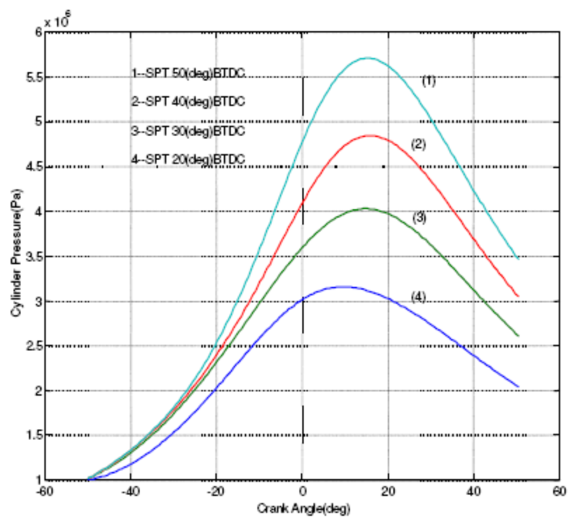

Figure 1: Variation of cylinder pressure with compression ratio (Combustion duration)

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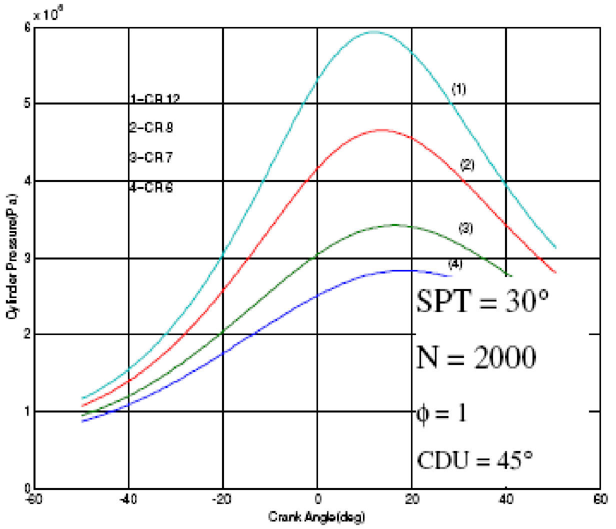

Figure 2: Variation of cylinder pressure with spark timing (SPT) in combustion duration.

IX. Typical Results of Model

In this section typical output results of the model are discussed. The variables that are going to be varied are spark timing (SPT), compression ratio (CR), Equivalence ratio (ϕ) and combustion duration (CDU). In each run one variable is changed and its effect on the pressure diagram is shown. These results include the effects of compression due to flame propagation and piston movement, heat transfer between mixture and cylinder wall in the form of convection and radiation, and heat release due to the chemical reactions in the gas.

X. Variation of Pressure with Compression Ratio

Figure 1 shows the variation if cylinder pressure diagram due to the varying the compression ratio from 6:1 to 12:1. Increasing compression ratio is associated with increasing pressure as shown in the figure 1.

Decrease combustion chamber volume with increasing compression ratio is the reason of increase pressure in the case.

XI. Variation of Pressure with Spark Timing

Figure 2 shows the variation of pressure diagram due to varying the spark timing from 20, 30, 40, and 50° BTDC. As shown in the figure advancing of spark timing strongly affects the spark pressure and its angular position. The research is well known: when spark in advanced causes the burning gas more compressed by rising piston consequently, increases the cylinder pressure. Also it is possible to shown the variation of cylinder pressure with equivalence ratio, combustion duration and engine speed. It was found that the equivalence ratio affects the maximum pressure of cycle the reason is as the mixture is made rich the temperature of the combustion increases due to excess of energy and so following this, the maximum pressure will increase. However at richer mixtures the temperature and also pressure will decrease because of insufficient air for burning. Increasing combustion duration is associated with the decreasing in cylinder pressure. Because of increasing combustion duration causes longer time for burning of mixture that causes decrease in pressure. Increasing speed was found to causes the cylinder pressure to decrease because increasing engine speed causes increased heat transfer from the gas to the cylinder wall and consequently causes the pressure to decrease.

XII. Model Adjustment and Validation

This is a practical approach, which utilizes the real data obtained from the engine to assess model validity. The model adjustment and comparison between the simulated response by the model and the measured response from the engine is a fundamental feature of model validation. In broad terms, "model validation" here is an adjustment process, which involves the comparison between the model and the actual monitored signal from the engine. Adjustment of model is carried out in conjunction with "Cross-Validation", which is to adequate for an independent set of measured input/output data from this process. To adjust the model and compare the simulated data of the overall model with real data a series of tests were carried out on the engine under normal conditions and the pressure signals from the pressure transducer were recorded. For matching simulated data with real data, model was run at the same operating conditions by varying some adjusting parameters.

Complete validation of the model in term of pressure diagrams has been carried out. A typical comparison between the experimental data and simulated cylinder pressure for the range of different operating conditions is presented. The predicted and measured cylinder pressure of tests 1 and 2 are given in figure 3 and 4.

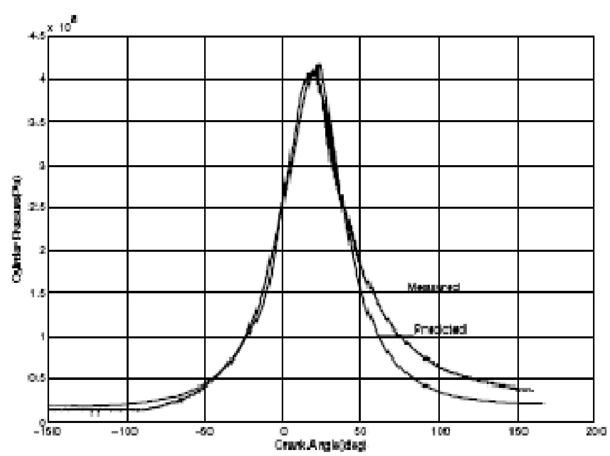

Figure 3: Measured and predicted cylinder pressure (test 1).

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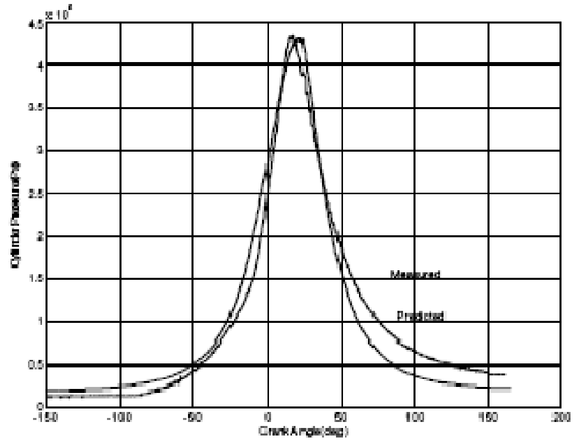

Figure 4: Measured and predicted cylinder pressure (test 2).

Table 1: Specification of Ricardo E6

Engine's Part	Specification
Bore	76.2 (mm)
Stroke	111 (mm)
Crank Radius	55.5 (mm)
Connecting Rod Length	241 (mm)
Swept Volume	507 (cm3)
Normal Speed	1000-3000 (rpm)
Compression Ratio	4.5-20
Inlet Valve Opens	10° Deg BTDC
Inlet Valve Closes	36° Deg ABDC
Inlet Valve Tappet Clearance	0.152 (mm)
Maximum Lift of Inlet Valve	10.6 (mm)
Exhaust Valve Opens	43 °Deg BBDC
Exhaust Valve Closes	8° Deg ATDC
Exhaust Valve Tappet Clearance	0.203 (mm)
Maximum Lift of Exhaust Valve	10.48 (mm)

XIII. Summary and Conclusion

A quasi-dimensional model for the compression, combustion and expansion processes in a spark ignition engine has been developed. The model is validated against data from a Ricardo E6 single cylinder research engine. The accuracy of the model is first tested with comparison of simulated and experimental data. The accuracy determinations are considered with regard to speed, spark timing, compression ratio, and equivalence ratio in cylinder pressure versus crank angle diagrams. The model accurately simulated the trends noted experimentally when these parameters are varied. After model adjustment for matching the simulated pressure data with experimental data the following results are drawn

1. Heat transfer model of Annand [8] was found has better influence on matching experimental data than models of Woshni [9] and Eichelberg [10].
2. Cylinder wall pressure (T_w) was found around 400 °K.
3. Inlet gas pressure (P0) was found around % 95 of atmospheric pressure.
4. It was found the inlet gas temperature (T0) around 300 °K. Combustion duration (CDU) 45 to 50 degrees.

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