

A Quasi-Dimensional Combustion Model and Experimental Investigation For Small size Single Cylinder Four Stroke Spark Ignition Engine Using Ethanol and Gasoline Mixture

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Abstract

A quasi-dimensional, a small size single cylinder four stroke spark ignition engine simulation is developed for studies of engine performance and the ethanol and gasoline mixture of fuel (E20 and E85), air, unburned mixture, and burned products zones (A two zone heat release). The first law of thermodynamics is applied to each zone as following a four stroke engine. Phenomenological models are used to simulate fuel injection, break-up, evaporation, air entrainment, and ignition delay. Premixed burning is modeled based on the new chemical kinetics model, while diffusion burning depends on the rate of fuel-air mixing. This paper describes the model, and illustrates the behavior of critical sub-models of heat transfer between the unburned-burned zones. It is shown that predicted cylinder pressure and heat release profiles are in good agreement with experimental investigation data of as power, torque, specific fuel consumption and specific fuel economy under rates engine speed and load conditions. Furthermore, the applicability of the simulation for performance studies under a range of engine speed and loads demonstrated. The results of the study as the power and torque were calculated by the methods differ by around 10% with experiment investigation. And the results of specific fuel consumption and fuel economy are less than 10% with experiment investigation.

Keywords: Quasi-Dimensional, Combustion Model, Chemical Kinetics, unburned-burned zones

1. Introduction

Spark ignition engine combustion is a complex, unsteady, heterogeneous three dimensional processes. The complexity arises from the strong interaction of many phenomena such as fuel spray, chemical kinetics, turbulence which is not completely understood and

constitutes difficult modeling problems in their own right. A number of modeling approaches have been proposed and tested with various degrees of success. In zero-dimensional single-zone models, the control volume is defined to coincide with the cylinder volume [1]. The cylinder charge is assumed to be uniform in both

composition and temperature, at all times during the cycle. The fuel injected into the cylinder is assumed to mix instantaneously with the cylinder charge, which is treated as an ideal gas. Combustion is modeled as an empirical heat addition process. However, this type of model cannot be used to account for fuel evaporation (and its effect on pressure), air entrainment into the fuel spray, and spatial variation in mixture composition and temperature.

Comprehensive, multi-dimensional models of spark ignition combustion have evolved over the past fifteen years [2, 3, 12 and 13]. However, phenomenological sub-models describing fuel spray processes are still included in these models, and some of the input variables are based on estimates and speculation. Furthermore, computer time and storage constraints place severe restrictions on the routine use of these codes for design purposes. In addition, for practical spark ignition fuels which are blends of a large number of hydrocarbons, detailed reaction mechanisms have yet to be identified (Heywood [12]). As a consequence, these schemes need to be calibrated with experimental data to give acceptable results for different engine designs and operating conditions.

As an intermediate step between zero-dimensional and multidimensional models, (*Non liner equation from the engine cycle*), quasi-dimensional, multi-zone models have been effectively used to model spark ignition engine combustion systems [2 and 3]. The Quasi-dimensional, multi-zone models combine some of the advantages of zero-dimensional, single-zone and multi-dimensional models. These models are based upon a blend of fundamental data. Quasi-

dimensional models offer the fastest and least expensive means of generating the information required to support design and development.

Time dependent (*Non liner equation from kinetic combustion*), quasi-dimensional modeling has been an active area of research since the early heat release studies [5, 7 and 10]. They predicted heat release rate in a quiescent chamber spark ignition engine allowing for fuel/air mixing. Krieger [1] predicted proposed an air induction model based on the rate of change of fuel mass (see fig. 1). The combustion rate given by this model was only related to the total amount of air entrainment during premixed combustion and was independent of the fuel-air mixing process. Depcik [11] assumed that the fuel injected into the cylinder vaporizes instantaneously. However, this approach is valid only if the combustion chamber conditions are near the critical point of the fuel.

The objective of this paper is to develop a quasi-dimensional, zonal combustion model and implement it into a comprehensive simulation of spark ignition engine performance [10 and 11]. In the present effort, the combustion chamber will be partitioned into fuel, air, mixing and burned zones. Phenomenological models will be used to simulate fuel injection, break-up, evaporation, air entrainment, and ignition delay. Pre-mixed burning will be modeled based on the novel chemical kinetics [8], while diffusion burning will be controlled by the rate of fuel-air mixing. The applicability of the simulation for performance studies (variation in power (kW), torque (N.m), specific energy consumption (g/kWh) and fuel consumption (L/hr)) under a range of speeds and loads will be demonstrated.

2. Model Description

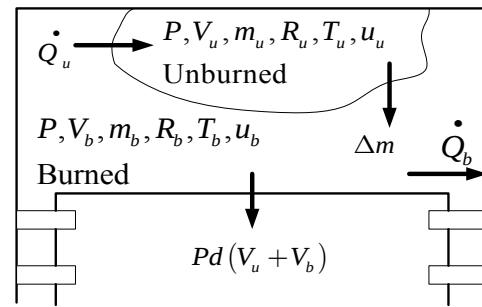
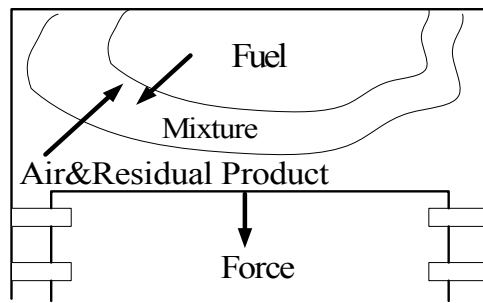


Fig. 1a. the mixing zone is created as air is entrained Fig. 1b. Combustion starts.
Fig.1 Definition of fresh air, fuel injection, mixture unburned and burned with combustion products zones.

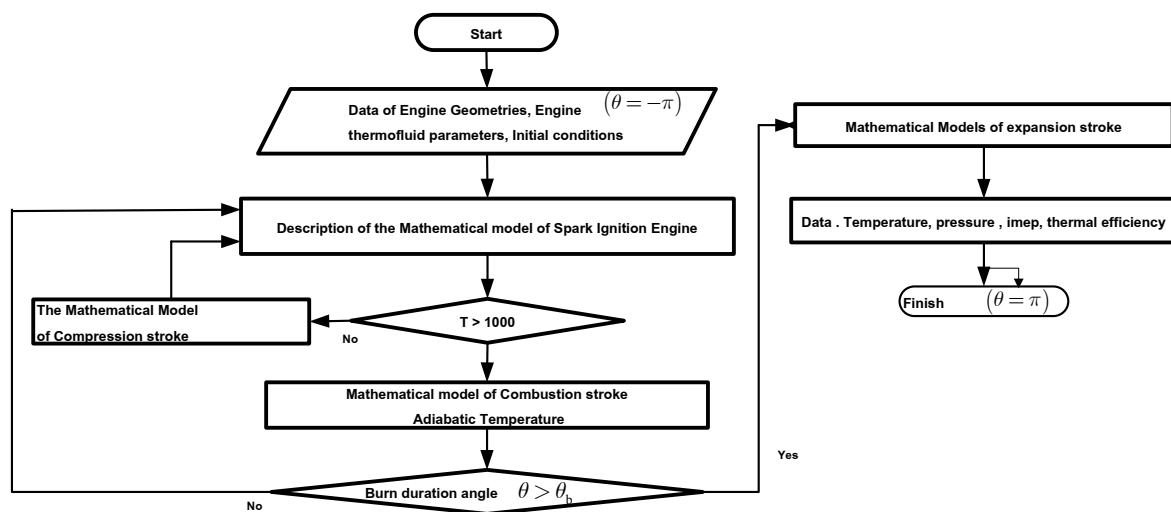


Fig.2 Computational sequence of Compression Stroke (Injection, Evaporation, Entrainment) Combustion stroke (ignition delay, The novel chemical kinetics [8]) and Expansion stroke.

During the course of the combustion process, this model partitions the cylinder contents into fuel, air, mixing, and combustion products zones, as illustrated in Fig.1. prior to fuel injection, the combustion chamber contains a mixture of air and residual combustion products (stage Fig 1a). Next, the fuel which is injected into the combustion chamber is divided into many droplet parcels according to time increment. Each of these parcels according to the time increment. Each of these parcels retains its identity during the subsequent processes. These liquid fuel droplet parcels form a fuel zone (stage Fig 1a). The fuel droplet

starts evaporation immediately following injection. Vaporized fuel and entrained air form the mixing zone (stage Fig 1b). The entrainment of air from the air zone into the mixing zone is calculated based on fuel spray tip penetration and spray angle. Before ignition, the mixing zone is divided into an unburned zone, which contains only fuel vapor and air, and a burned zone, which contains residual combustion products. After a short time period of ignition delay, combustion starts (stage Fig 1b). Combustion products only transfer from the burned zone.

Combustion proceeds in two stages. First, the burning rate is controlled by premixed

combustion. Second, the burning rate is controlled by diffusion combustion. The ignition delay is calculated for each fuel-air parcel. At the end of the ignition delay of the first parcel, combustion of the parcel starts. Premixed combustion is controlled by chemical kinetics. When all fuel parcels which have evaporated until the lapse of the ignition delay of the first parcel are burned, premixed combustion is completed. After the ignition delay period, more fuel is injected into the chamber, evaporated and mixed with air to form combustible mixture. Diffusion combustion is controlled by fuel-air mixing rate, and continues until combustion is completed. The computational sequence of injection, evaporation, entrainment, ignition delay, and combustion is shown in fig.2.

The model assumptions are: [6, 13 and 14]

- 1) Each zone is in thermodynamic equilibrium.
- 2) The gas in all zones behaves as ideal gases.
- 3) Pressure is spatially uniform throughout the chamber, but changes with time.
- 4) Each zone is assumed to have its own temperature and composition. These properties are assumed to be uniform within each zone, and vary with time.
- 5) No mass transfer occurs from the mixing zone to the air and fuel zones. No fuel is transferred directly to the air zone.
- 6) Only vaporized fuel is transferred into the mixing zone; the fuel droplets are all contained in the fuel zone.
- 7) The temperatures of the burned and unburned zones are initialized to the bulk gas temperature at the time of injection.
- 8) There is no heat transfer between the unburned and burned zones.

2.1 The thermodynamic model

The mathematical engine model was applicable in detail mass and energy balance for four stroke spark ignition engine and is based on the thermodynamic analysis of the ideal fuel-air cycle. The method is assumed that the fuel-air is supplied from a perfect system and mixed completely.

Mass balance, the equation of state for an ideal gas is

$$PV = mRT \quad (1)$$

The rate of change of mass within any open system is the net flux of mass across the system boundaries. Hence for a control volume enclosing the air-fuel mixture, we have

$$\dot{m} = \sum_i \dot{m}_i \quad (2)$$

From the first law of thermodynamics, the open system can be modified as following equation from Ferguson [13].

$$\dot{E} = \dot{Q} - \dot{W} + \sum_i \dot{m}_i (H_i) \quad (3)$$

Taking the derivative of Eq. 2 and 3 as the function of the crank angle θ yields as

$$\frac{dm}{d\theta} = \sum_i \frac{dm_i}{d\theta} \quad (4)$$

$$\frac{d(mu)}{d\theta} = \frac{dQ}{d\theta} - \frac{dW}{d\theta} + \sum_i \frac{dm_i}{d\theta} (h_i) \quad (5)$$

2.2 Thermodynamic properties

The specific heat change as the function of the crank angle in eq (2) and (5), those indicate the enthalpy change with respect to temperature and pressure, which was obtained from curve fitted polynomial equation. Table 1 shows the thermodynamics properties expressed as a function of length, pressure and temperature. Where, Lewis [15] has represented the thermodynamic properties of fuels and air

Ferguson [13] proposed the thermodynamic properties of air and combustion products.

Table 1 Thermodynamic properties [12, 14]

Internal energy, entropy, enthalpy [12] $\frac{du}{dL} = \left(c_P - \frac{Pv}{T} \frac{\partial \ln v}{\partial \ln T} \right) \frac{dT}{dL} - \left(v \left(\frac{\partial \ln v}{\partial \ln T} + \frac{\partial \ln v}{\partial \ln P} \right) \right) \frac{dP}{dL}$ $\frac{dv}{dL} = \frac{v}{T} \frac{\partial \ln v}{\partial \ln T} \frac{dT}{dL} - \frac{v}{P} \frac{\partial \ln v}{\partial \ln P} \frac{dP}{dL}$ $\frac{ds}{dL} = \left(\frac{C_P}{T} \right) \frac{dT}{dL} - \frac{v}{T} \frac{\partial \ln v}{\partial \ln T} \frac{dP}{dL} \left(\frac{\partial h}{\partial T} \right)_P = C_P$
Fuels [14] $\frac{c_P}{R} = a_0 + b_0 T + c_0 T^2$ $\frac{h}{RT} = a_0 + \frac{b_0}{2} T + \frac{c_0}{3} T^2 + \frac{d_0}{T}$ $\frac{s^0}{R} = a_0 \ln T + b_0 T + \frac{c_0}{2} T^2 + e_0$
Air and Combustion product [12, 14] $\frac{c_P}{R} = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4$ $\frac{h}{RT} = a_1 \ln T + \frac{a_2}{2} T + \frac{a_3}{3} T^2 + \frac{a_4}{4} T^3 + \frac{a_5}{5} T^4 + \frac{a_6}{T}$ $\frac{s^0}{R} = a_1 \ln T + a_1 T + \frac{a_3}{3} T^2 + \frac{a_4}{4} T^3 + \frac{a_5}{4} T^4 + a_7$

2.3 The model of Combustion Engine

In the combustion stroke, we consider the temperature in the term of burned mixture (T_b) and unburned (T_u) as separate open systems. Recalling Eq.(5) and combining all the derivatives of thermodynamic properties will enable the pressure and temperature to be expressed as a function of length, pressure, unburned gas temperature and burned gas temperature.

$$\frac{dP}{dL}, \frac{dT_b}{dL}, \frac{dT_u}{dL} = f_1(L, P, T_b, T_u) \quad (6)$$

Modified algorithm from Ferguson [13] as following the arbitrary heat release conditions and solving the above equations with appropriate input data enable determination of the indicated work, enthalpy and heat loss throughout the system. Those can be expressed as a function of pressure and temperature.

$$\frac{dP}{dL} = \frac{A + B + C}{D + E} \quad (7)$$

$$\frac{dT_b}{dL} = \frac{-h \left(\frac{\pi b^2}{2} + \frac{4V}{b} \right) x^{1/2} (T_b - T_W)}{vmc_{pb}x} + \frac{v_b}{c_{pb}} \left(\frac{\partial \ln v_b}{\partial \ln T_b} \right) \left(\frac{A + B + C}{D + E} \right) \quad (8)$$

$$\begin{aligned} & + \frac{h_u - h_b}{xc_{pb}} \left[\frac{dx}{dL} - (x - x^2) \frac{C}{v} \right] \\ & - h \left(\frac{\pi b^2}{2} + \frac{4V}{b} \right) (1 - x^{1/2}) (T_u - T_W) \\ \frac{dT_u}{dL} = & \frac{vmc_{pb}(1 - x)}{vmc_{pb}(1 - x)} \\ & + \frac{v_u}{c_{pu}} \left(\frac{\partial \ln v_u}{\partial \ln T_u} \right) \left(\frac{A + B + C}{D + E} \right) \end{aligned} \quad (9)$$

From the derivatives of Eq. (7–9) let us defined the constant term with respect to the combination of the thermodynamic properties equations are:

$$A = \frac{1}{m} \left(\frac{dV}{dL} + \frac{VC}{v} \right) \quad (10)$$

$$B = h \frac{\left(\frac{dV}{dL} + \frac{VC}{v} \right)}{vm} \left[\frac{v_b}{c_{pb}} \frac{\partial \ln v_b}{\partial \ln T_b} x^{1/2} \frac{T_b - T_W}{T_b} + \frac{v_u}{c_{pu}} \frac{\partial \ln v_u}{\partial \ln T_u} (1 - x^{1/2}) \frac{T_b - T_W}{T_b} \right] \quad (11)$$

$$C = -(v_b - v_u) \frac{dx}{dL} - v_b \frac{\partial \ln v_b}{\partial \ln T_b} \frac{h_u - h_b}{c_{pb} T_b} \left[\frac{dx}{dL} - \frac{(x - x^2)C}{v} \right] \quad (12)$$

$$D = x \left[\frac{v_b^2}{c_{pb} T_b} \left(\frac{\partial \ln v_b}{\partial \ln T_b} \right)^2 + \frac{v_b}{P} \frac{\partial \ln v_b}{\partial \ln P} \right] \quad (13)$$

$$E = (1 - x) \left[\frac{v_u^2}{c_{pu} T_u} \left(\frac{\partial \ln v_u}{\partial \ln T_u} \right)^2 + \frac{v_u}{P} \frac{\partial \ln v_u}{\partial \ln P} \right] \quad (14)$$

2.4 Chemical equilibrium with the fuel mixture

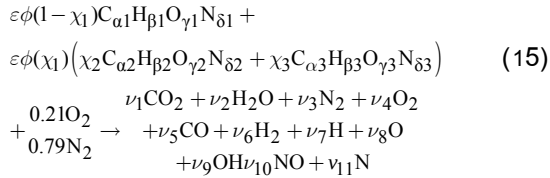
The basis of the equilibrium combustion products with fuel mixture model $C_a H_\beta O_\gamma N_\delta$ is a solution to the atom balance equations from the chemical reaction equation of fuel and air forming and The subscript 1,2&3 represent the one of the fuel (Gasoline), the fuel in the second (Ethanol) and third (Base Benzene) of E20, E85 respectively as shown in Table 2.

Table 2 The E20 and E85 Thailand Fuel [9]

Name	Ratio E20	Ratio E85
Ethanol (99.5%)	20	85
Base Benzene	80	15

This mixture equation is given in Eq.15 for the condition of equivalence ratio ϕ , where ν_1

through ν_{11} are mole fractions of the product species, ε is the molar fuel-air ratio required to react with one mole of air. This process based on the work of modified Ferguson [13] as given below



Where χ_1 is the molar ratio of multi-fuel having the condition ($0 < \chi_1 < 1$). χ_2 and χ_3 are the fuel in the second and third which having the condition ($\chi_2 + \chi_3 = 1$). There is the conservation of 4 atoms, C, H, O and N from mixture equation, so atom balancing can be written,

$$C \quad \varepsilon\phi(1-\chi_1)\alpha1 + \varepsilon\phi\chi_1\chi_2\alpha2 + \varepsilon\phi\chi_1\chi_3\alpha3 = (y_1 + y_5)N \quad (16)$$

$$H \quad \varepsilon\phi(1-\chi_1)\beta1 + \varepsilon\phi\chi_1\chi_2\beta2 + \varepsilon\phi\chi_1\chi_3\beta3 = (2y_2 + 2y_6 + y_7 + y_9)N \quad (17)$$

$$O \quad \varepsilon\phi(1-\chi_1)\gamma1 + \varepsilon\phi\chi_1\chi_2\gamma2 + \varepsilon\phi\chi_1\chi_3\gamma3 + 0.42 = (2y_1 + y_2 + 2y_4 + y_5 + y_8 + y_9 + y_{10})N \quad (18)$$

$$N \quad \varepsilon\phi(1-\chi_1)\delta1 + \varepsilon\phi\chi_1\chi_2\delta2 + \varepsilon\phi\chi_1\chi_3\delta3 + 1.58 = (2y_3 + y_{10} + y_{11})N \quad (19)$$

Where, N is the total number of moles and y is the mole fraction. Thus, the total number of mole fraction must be equal to one and this gives:

$$\sum_{i=1}^{11} y_i - 1 = 0 \quad (20)$$

The expression for atom balance of each equation can be eliminated by dividing eq. (17-19) by eq. 16. The equation can be written as next equation.

$$2y_2 + 2y_6 + y_7 + y_9 - [d_1(y_1 + y_5)] = 0 \quad (21)$$

$$2y_1 + y_2 + 2y_4 + y_5 + y_8 + y_9 + y_{10} - [d_2(y_1 + y_5)] = 0 \quad (22)$$

$$2y_2 + y_{10} + y_{11} - [d_3(y_1 + y_5)] = 0 \quad (23)$$

where

$$\begin{aligned} d_1 &= \frac{(1-\chi_1)\beta1 + \chi_1\chi_2\beta2 + \chi_1\chi_3\beta3}{(1-\chi_1)\alpha1 + \chi_1\chi_2\alpha2 + \chi_1\chi_3\alpha3} \\ d_2 &= \frac{(1-\chi_1)\gamma1 + \chi_1\chi_2\gamma2 + \chi_1\chi_3\gamma3 + \frac{0.42}{\varepsilon\phi}}{(1-\chi_1)\alpha1 + \chi_1\chi_2\alpha2 + \chi_1\chi_3\alpha3} \\ d_3 &= \frac{(1-\chi_1)\delta1 + \chi_1\chi_2\delta2 + \chi_1\chi_3\delta3 + \frac{1.58}{\varepsilon\phi}}{(1-\chi_1)\alpha1 + \chi_1\chi_2\alpha2 + \chi_1\chi_3\alpha3} \end{aligned}$$

Eq. (20-23) have 11 unknowns ($y_1, y_2, y_3, \dots, y_{11}$), therefore in order to solve for these 11 unknowns other 7 more equations are needed which may be derived from the consideration of equilibrium among products. The equilibrium constant can be related to the partial pressure of the reactants and products. And the partial pressure of a component is defined relative to the total pressure and the mole fraction, thus the equilibrium constant can be rewritten as Table 3.

Table 3 The dissociation effect

$$\begin{aligned} CO_2 &\leftrightarrow CO + \frac{1}{2}O_2 & K_1 &= \frac{y_5 y_4^{1/2} P^{1/2}}{y_1} \\ H_2 + \frac{1}{2}O_2 &\leftrightarrow H_2O & K_2 &= \frac{y_2}{y_4^{1/2} y_6 P^{1/2}} \\ \frac{1}{2}H_2 + \frac{1}{2}O_2 &\leftrightarrow OH & K_3 &= \frac{y_9}{y_4^{1/2} y_6^{1/2}} \\ \frac{1}{2}H_2 &\leftrightarrow H & K_4 &= \frac{y_7 P^{1/2}}{y_6^{1/2}} \\ \frac{1}{2}O_2 &\leftrightarrow O & K_5 &= \frac{y_8 P^{1/2}}{y_4^{1/2}} \\ \frac{1}{2}N_2 &\leftrightarrow N & K_6 &= \frac{y_{11} P^{1/2}}{y_3^{1/2}} \\ \frac{1}{2}O_2 + \frac{1}{2}N_2 &\leftrightarrow NO & K_7 &= \frac{y_{10}}{y_4^{1/2} y_3^{1/2}} \end{aligned}$$

Equilibrium constant in Table 3, K_1 through K_7 are curve fitted form Gills [8] and their expressions are of the form,

$$K_p = \exp\left(\frac{\Delta a_1}{T}(\ln T - 1) + \frac{\Delta a_2}{2} + \frac{\Delta a_3 T^2}{6} + \frac{\Delta a_4 T^3}{12} + \frac{\Delta a_5 T^4}{20} - \frac{\Delta a_6}{T} + \Delta a_7\right) \quad (24)$$

Through algebraic manipulations, the ten equations can be reduced into four equations with four unknowns. The equations are nonlinear and solved by using the Newton-Raphson method [14]. Each of these may be expanded in Taylor's series $f_j(y_3, y_4, y_5, y_6) = 0$ where $j = 1, 2, 3, 4$ (neglecting the second order and higher order) as,

$$f_j + \frac{\partial f_j}{\partial y_3} \Delta y_3 + \frac{\partial f_j}{\partial y_4} \Delta y_4 + \frac{\partial f_j}{\partial y_5} \Delta y_5 + \frac{\partial f_j}{\partial y_6} \Delta y_6 = 0 \quad (25)$$

Functions f_j are evaluated from the solution of interested functions (Eq.20-23) the independent set of derivatives is obtained by

solution of matrix equation that results from differentiating with respect to mole fraction. The above can be arranged as set of linear equations in the matrix form,

$$\left[\frac{\partial f_i}{\partial y_i} \right] [\Delta y] - [-f] = 0 \quad (26)$$

This set of linear equations can then be solved for y_3, y_4, y_5, y_6 and iterative procedures undertaken until the corrections are less than a specified tolerance (δ). For convenience, defining following partial derivatives and defining the constant values for a simple studied and the Jacobian by Turn [14] of solution are given as Table 4.

Table 4 The Jacobian of solution

$D_{ij} = \frac{\partial f_i}{\partial y_j} \quad i=1,2,7,8,9,10,11$ $j=3,4,5,6$	
$D_{14} = \frac{\partial f_1}{\partial y_4} = \frac{1}{2} \frac{c_1 y_5}{y_4^{1/2}}$	$D_{24} = \frac{\partial f_2}{\partial y_4} = \frac{1}{2} \frac{c_2 y_6}{y_4^{1/2}}$
$D_{15} = \frac{\partial f_1}{\partial y_5} = c_1 y_4^{1/2}$	$D_{26} = \frac{\partial f_2}{\partial y_6} = c_2 y_4^{1/2}$
$D_{103} = \frac{\partial f_{10}}{\partial y_3} = \frac{1}{2} \frac{c_1 y_4^{1/2}}{y_3^{1/2}}$	$D_{104} = \frac{\partial f_{10}}{\partial y_4} = \frac{1}{2} \frac{c_1 y_3^{1/2}}{y_4^{1/2}}$
$\frac{\partial f_1}{\partial y_3} = 1 + D_{103} + D_{113}$	$\frac{\partial f_2}{\partial y_3} = 0$
$\frac{\partial f_1}{\partial y_4} = D_{14} + D_{24} + 1 + D_{84} + D_{104} + D_{94}$	$\frac{\partial f_2}{\partial y_4} = 2D_{24} + D_{94} - d_1 D_{14}$
$\frac{\partial f_1}{\partial y_5} = D_{15} + 1$	$\frac{\partial f_2}{\partial y_5} = -d_1 D_{15} - d_1$
$\frac{\partial f_1}{\partial y_6} = D_{26} + 1 + D_{76} + D_{96}$	$\frac{\partial f_2}{\partial y_6} = 2D_{26} + 2 + D_{76} + D_{96}$
$\frac{\partial f_3}{\partial y_3} = D_{103}$	$\frac{\partial f_4}{\partial y_3} = 2 + D_{103} + D_{113}$
$\frac{\partial f_3}{\partial y_4} = 2D_{14} + D_{24} + 2 + D_{84} + D_{94} + D_{104} - d_2 D_{14}$	$\frac{\partial f_4}{\partial y_4} = D_{104} - d_3 D_{14}$
$\frac{\partial f_3}{\partial y_5} = 2D_{15} + 1 - d_2 D_{15} - d_2$	$\frac{\partial f_4}{\partial y_5} = d_3 D_{15} - d_3$
$\frac{\partial f_3}{\partial y_6} = D_{26} - D_{96}$	$\frac{\partial f_4}{\partial y_6} = 0$

Eq.26 may be solved using by Gauss elimination. The second approximation is then $\{y\}_{k+1} = \{y\}_k + \{\delta\}_k \quad y=3,4,5,6$. The process of forming the jacobian, solving Eq.24 and calculating new values for $\{y\}$ is repeated until a stop criterion is met, results in the molar concentrations of the 11 product species.

3. Results and Discussion

To illustrate the behavior of the combustion model, and at the same time validate the overall engine simulation, a study was performed for a representative, direct-injection, four-stroke, spark ignition engine. The basic parameters of this engine are listed in Table 1. Rated speed and load conditions were selected for the validation run since complete test data, including cylinder pressure trace, were available for this point. The rated operating conditions are speed of 2000-4500 rpm, fueling rate of 15-25 g/sec, and injection timing of 346.0 crank angle degrees.

Table 5 Baseline Spark Ignition Engine, [9].

Bore, b (m)	0.068
Stroke (m)	0.045
Stroke/2 (rod Length)	0.25
Compression ratio	8.5
Equivalence ratio	1.1
Intake valve opening	11 deg BTC
Intake valve opening	40 deg. ABC
Exhaust valve opening	44 deg. BBC
Exhaust valve opening	26 deg. ATC
Fuel	Model, Gasoline Fig. 3, 4 and 5 Model 1, Test 1 is fuel E20 Model 2, Test 2 is fuel E85 Output, Fig.6, varied speed (rpm)) Output Fig.7, varied load (%)

Fig. 3 shows that the assumption (Model, gasoline 100%) predicted cylinder pressure trace falls in very good agreement with the experimental data. Heat release analysis of the cylinder pressure trace reveals the details of the combustion process. The predicted heat release profile shown in Fig. 4 exhibits the double-peak shape which characterizes direct injection spark

ignition combustion. The first peak due to premixed combustion strongly depends on the amount of fuel that mixes with air to form a combustible mixture during the ignition delay period. Premixed combustion is primarily controlled by the chemical reaction rate. The constants used for the chemical reaction rate are listed by Gill [8]. Premixed combustion is completed when all fuel injected during the ignition delay period is burned completely. When premixed combustion is over, heat release rates momentarily drop. The second peak due to diffusion combustion is controlled by air-fuel mixing rate. Diffusion combustion continues until combustion is completed.

Figure 5 assumptions to use compare the temperatures of the unburned mixture zone and burned product zone with the bulk gas temperature (obtained from a single zone analysis). The temperature in the burned product zone is close to the adiabatic flame temperature throughout the duration of the combustion process. The temperature in the unburned zone is close to that of the air zone. It reaches its peak close to TDC, and drop with piston expansion. The bulk mean gas temperature clearly falls between the burned and unburned temperature. It rapidly increases when combustion starts, and reaches its peak value when the diffusion controlled burning rate is at its peak value.

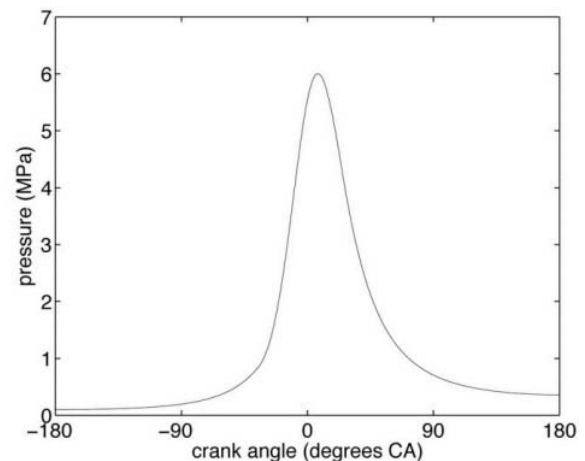


Fig.3 Pressure VS Crank Angle

A quasi-dimensional, a small size single cylinder spark ignition engine

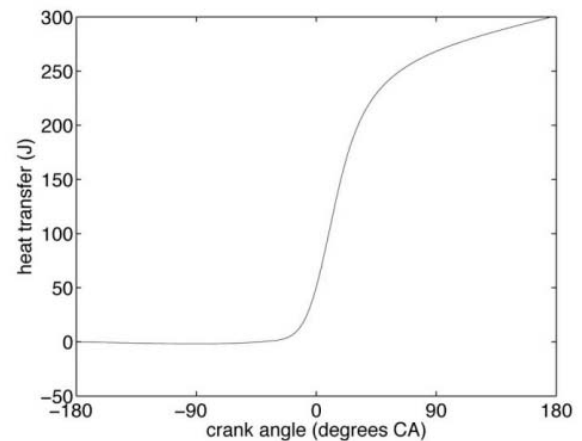


Fig. 4 Heat release rate VS crank angle

A quasi-dimensional, a small size single cylinder spark ignition engine

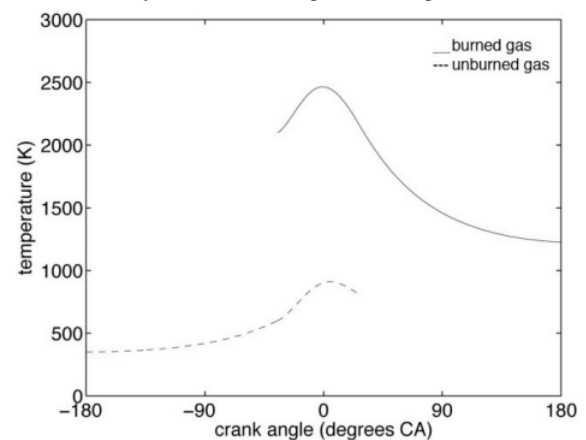


Fig. 5 Temperature VS Crank Angle

A quasi-dimensional, a small size single cylinder spark ignition engine

4. Parametric Studies

4.1 Effect of Engine Speed

For this study, the engine speed was varied between 2000-4000 rpm, while maintaining free load at constant assumption injection timing (Combustion duration angle 40 deg.). Operation condition was defined at an overall fuel-air equivalence ratio of 1.1. Thus, the fueling rate was adjusted as speed (and thus airflow) was varied in order to maintain 1.1.

Figure 6 shows the variation in power (kW), torque (N.m), specific energy consumption (g/kWh) and fuel consumption (L/h) with engine speed. At low engine speeds, charge heating in the cylinder and manifold is more predominant due to longer gas residence times, Furthermore; the increased chance for backflows tends to decrease power and torque. On the other hand, frictional flow losses increase with the square of engine speed. In addition, when choking occurs, further increase flow rate. These factors tend to decrease power and torque at high engine speeds. As a result, the optimum volumetric efficiency occurs between 3500-4000 (rpm) as the single cylinder engine.

The variation in power (kW) with speed is certainly reflected on the corresponding variation of FC (L/hr). However, as shown in Fig. 6, SFC (g/kWh) larger relative changes between its minimum and maximum values than the breathing characteristic. Hence, combustion and heat transfer related factors should be explaining the observed trend.

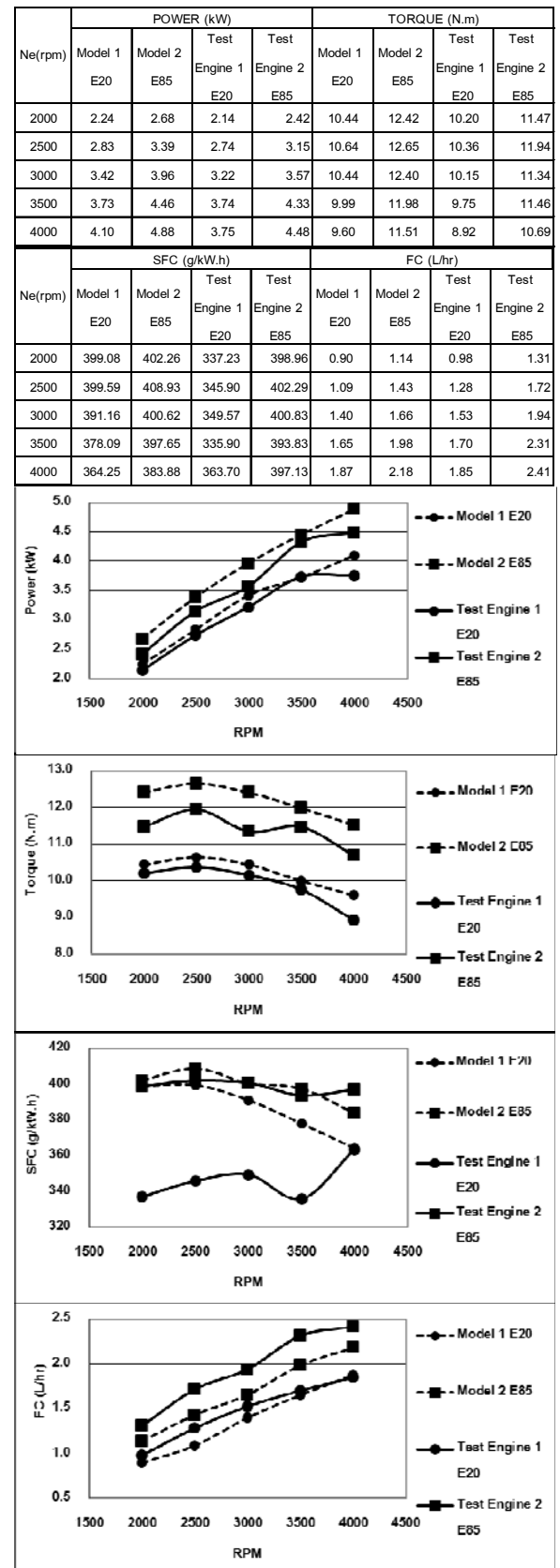


Fig. 6 Engine performance parameters as a function of speed.

4.2 Effect of Load

Figure 7 shows the variation in power (kW), torque (N.m), Specific energy consumption (g/kWh) and fuel consumption (L/hr) with a constant speed of 2500 rpm is used to study the effect of load on engine performance. In order to vary load, the fueling rate is changed. At full load, the fuel mass injected per cylinder results in an overall fuel-air equivalence ratio of 1.1.

The lower the load, the lower the amount of fuel injected, and thus the leaner the overall fuel-air equivalence ratio 1.1.

Engine friction is a strong function of engine speed. As load is decreased at a constant engine speed, the importance of friction (and mechanical losses) as a fraction of fuel energy input increases. Therefore, power and torque decreases with decreasing load. SFC (g/kWh) and FC (L/hr) increases with increasing load. This trend is depicted in Fig. 7.

5. Conclusions

A quasi-dimensional multi-zone combustion model has been developed for studies of engine performance. The zonal combustion model partitions the cylinder into fuel, air, unburned mixture, and combustion products zones. Phenomenological sub-models are used to simulate assumption fuel injection, assumption break-up, assumption evaporation, assumption air entrainment, and assumption ignition delay. Premixed burning is modeled based on chemical kinetics, while diffusion burning is controlled by the rate of fuel-air mixing. Due to its expanded physical basis and the increased resolution of the combustion chamber, the model can provide

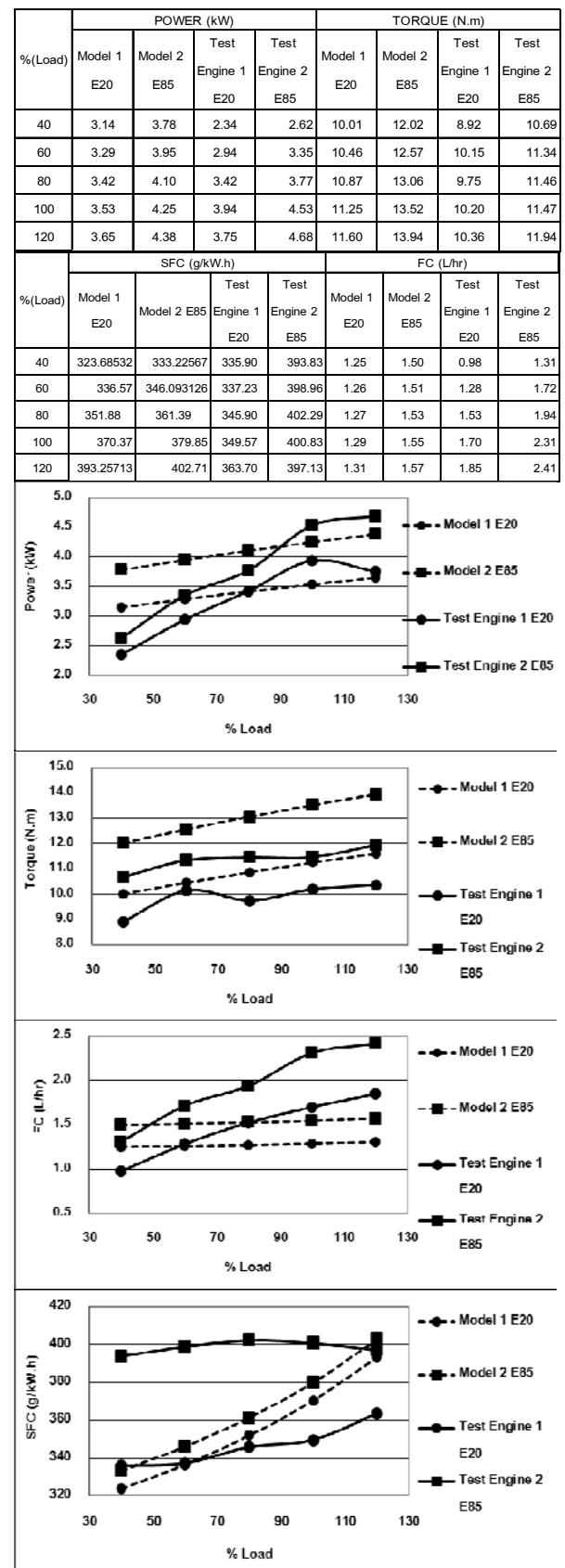


Fig. 7 Engine performance parameters
as a function of load.

a level of accuracy which surpasses that of single-zone, zero-dimensional models

The model has been shown to predict cylinder pressure and heat release shapes in good agreement with experimental data, under rated speed and load conditions. Further, its behavior over a range of speeds and load is reasonable. While less fundamental than multi-dimensional combustion models, the quasi-dimensional model can still generate cost and time-effective solutions, directly applicable to engineering problems.

The results of the study as the power and torque were calculated by the methods differ by around 10%. Typical deviations results of specific fuel consumption and fuel economy are less than 5% with experiment investigation.

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8. Nomenclature

A	Heat area
H	Enthalpy
K	Equilibrium constant
Q	Heat
T	Temperature
U	Internal energy
V	Volume
x	Burned faction
y	Mole faction
ε	Ratio of mole fuel per mole air
ϕ	Equilibrium ratio
θ	Crank Angle
ω	Engine speed

Subscript

b	Burned
s	Start
u	Unburned
w	Wall