

Single Cylinder Four Stroke Spark Ignition Engine - Thermodynamic Simulation Model

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Abstract: - The performance of an engine whose basic design parameters are known can be predicted with the assistance of simulation programs. A thermodynamic model for the simulation of a single cylinder spark ignition engine running on Iso-octane fuel is presented. The simulation consists of compression, combustion and expansion processes. The model is based on the classical two-zone approach, wherein parameters like heat transfer from the cylinder, blow-by energy loss and heat release rate are also considered. The general fuel is specified by way of its C-H-O-N values. Curve-fit coefficients are then employed to simulate air and fuel data along with fuel-air composition and practical chemical equilibrium routines. The calculated data is then used to plot the various thermodynamic parameters with respect to crank angle. Simulation program is usable to set for varies compression ratios, engine speed, and engine sizes.

Key words: Spark Ignition (SI) engine, Simulation model, Thermodynamic Modeling of Engines.

I. INTRODUCTION

The thermodynamic simulation model of spark ignition (SI) engines are one of the most effective tools for the analysis of engine performance, parametric examinations and assistance to new developments. Researchers have been spending essential effort to improve the performance of SI engines and about internal combustion engines (ICE). Generally, engine simulations are practice in two ways, which are fluid dynamic based and thermodynamic based models. Thermodynamic cycle models are based on the thermodynamic analysis of the content of cylinder during the engine cycle. In these models, the first law of thermodynamics is applied to open system of air-fuel and residual gas mixture into the manifolds and cylinders and it is zero dimensioned. Therefore, the equations leading the model are made of basic differential equalities [4, 6, and 8].

The present trend is towards the development of comprehensive 3-D models, which describes the functioning of engines at a very high level of detail and accuracy; however, these require substantial computational power. There are several instances where theoretical methods, which are based on a limited set of experimental data, are preferred. From these considerations, the need for a simple, fast and accurate engine simulation model is quite evident.

Nomenclature

A	area exposed to heat transfer (m^2)
b	bore of cylinder (m)
c_p	specific heat at constant pressure
C	blow-by coefficient (s^{-1})
E	total energy (kJ)
ΔG	Standard-state Gibbs free energy
h	specific enthalpy ($kJ \cdot kg^{-1}$)
K	Equilibrium constant
m	mass (kg)
N_u	Nusselt number
p	pressure (Pa)
Q	heat transfer (kJ)
r	compression ratio

In this paper a two-zone, Zero-dimensional model was used to simulate the engine operations. The most important assumptions were, a) The working medium was considered, in general, to be a mixture of 10 species (CO₂, H₂O, N₂, O₂, CO, H₂, H, O, OH, NO) and fuel vapor. b) All 10 species were considered as ideal gases. And c) The fuel is limited to C-H-O-N species.

II. THERMODYNAMIC MODEL

For the present study, a Zero-dimensional combustion model is employed. The combustion chamber is divided into two zones consisting of unburned gas (mixture of fuel, air and residuals) and burned gas (mixture of 10 product species), each under uniform composition. This model assumes that at any instant of time during the combustion, the cylinder volume is divided into burned and unburned zones by an infinitesimally thin flame-front with a spherical shape. The burned gases are assumed to be in chemical equilibrium during combustion. A Wiebe function specifies the fuel burn rate and controls the rate at which mixtures from the unburned zone is converted to the burned zone. Mass and energy conservation relations and equations of state form the principle governing equations. Also considering crank angle as the independent variable, thus form the base of our thermodynamic model.

A. Mass and Energy 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

$$\dot{m} = \sum_k \dot{m}_k \quad (2)$$

The first law of thermodynamics to an open system yields the energy equation as

$$\dot{E} = \dot{Q} - \dot{W} + \sum_k \dot{m}_k h_k \quad (3)$$

Equations (2) and (3) can be written as

$$\frac{dm}{d\theta} = \sum_k \frac{dm_k}{d\theta} \quad (4)$$

$$\frac{d(mu)}{d\theta} = \frac{dQ}{d\theta} - p \frac{dV}{d\theta} + \sum_k h_k \frac{d m_k}{d\theta} \quad (5)$$

B. Air and Combustion Products Data

Colin R. Ferguson and Kirkpatrick [4] proposed the following expressions that were curve-fitted to polynomials by minimizing the least squares error. The function will employ for any given species is

$$\frac{c_p}{R} = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4 \quad (6)$$

$$\frac{h}{RT} = a_1 + \frac{a_2}{2} T + \frac{a_3}{3} T^2 + \frac{a_4}{4} T^3 + \frac{a_5}{5} T^4 + \frac{a_6}{T} \quad (7)$$

$$\frac{s}{R} = a_1 \ln T + a_2 T + \frac{a_3}{2} T^2 + \frac{a_4}{3} T^3 + \frac{a_5}{4} T^4 + a_7 \quad (8)$$

where c_p is the specific heat at constant pressure, h is the specific enthalpy and s is the specific entropy.

The coefficients a_1 to a_7 are calculated over two different temperature ranges: 1) $300 < T < 1000$ K; and 2) $1000 < T < 5000$ K and can be sourced from [6, 8].

The most combustion models are based on the assumption that the burned mixture is in equilibrium [6]. The following are species of interest during combustion: CO_2 , H_2O , N_2 , O_2 , CO , H_2 , H , O , OH and NO [4].

C. Fuel Data

Heywood [6] has represented the thermodynamic properties of fuels (in vapor phase).

$$\frac{c_p}{R} = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 \frac{1}{T^2} \quad (9)$$

$$\frac{h}{RT} = a_1 + \frac{a_2}{2} T + \frac{a_3}{3} T^2 + \frac{a_4}{4} T^3 - a_5 \frac{1}{T^2} + \frac{a_6}{T} \quad (10)$$

$$\frac{s}{R} = a_1 \ln T + a_2 T + \frac{a_3}{2} T^2 + \frac{a_4}{3} T^3 - \frac{a_5}{2} \frac{1}{T^2} + a_7 \quad (11)$$

The following relation is proposed for deriving the properties like specific heats and enthalpies for various species, with ξ referring to the property and x is the burnt mass fraction.

$$\xi_{mixture} = \sum_{k=1}^N x_k \xi_k \quad (12)$$

D. Equivalence Ratio

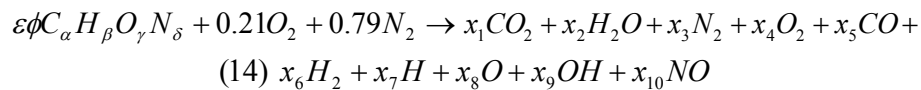
When modeling with a single fuel, the equivalence ratio is given by [4]

$$\phi = \left(\frac{F}{Air} \right)_{Act} / \left(\frac{F}{Air} \right)_{st} \quad (13)$$

where subscript Act. refers to actual and St. refers to stoichiometric.

E. Practical Chemical Equilibrium

Under the assumption of atmospheric air composition (21%v Oxygen and 79%v Nitrogen), and provided $\phi < 3$, the only species that are important because of dissociation are O, H, OH, and NO [4]. The combustion reaction thus becomes



where x_1 to x_{10} represents mole fractions of the products.

The atom balance for the various elements gives

$$C : \varepsilon\phi\alpha = (x_1 + x_5)N \quad (15)$$

$$H : \varepsilon\phi\beta = (2y_2 + 2y_6 + y_7 + y_9)N \quad (16)$$

$$O : \varepsilon\phi\gamma + 0.42 = \left(\begin{array}{l} 2y_1 + y_2 + 2y_4 + y_5 \\ + y_8 + y_9 + y_{10} \end{array} \right) N \quad (17)$$

$$N : \varepsilon\phi\delta + 1.58 = (2y_3 + y_{10})N \quad (18)$$

where $N = \sum_{i=1}^{10} x_i$ is the total number of moles. The constraint that the mole fraction of all the products adds up to unity requires that

$$\sum_{i=1}^{10} x_i = 1 \quad (19)$$

To solve for the unknowns, we need equations that are provided by the criteria of equilibrium among the products, which are expressed by the following hypothetical relations

$$\frac{1}{2}H_2 \leftrightarrow H \quad K_1 = \frac{x_7 P^{0.5}}{x_6^{0.5}} \quad (20)$$

$$\frac{1}{2}O_2 \leftrightarrow O \quad K_2 = \frac{x_8 P^{0.5}}{x_4^{0.5}} \quad (21)$$

$$\frac{1}{2}H_2 + \frac{1}{2}O_2 \leftrightarrow OH \quad K_3 = \frac{x_9}{x_4^{0.5} x_6^{0.5}} \quad (22)$$

$$\frac{1}{2}O_2 + \frac{1}{2}N_2 \leftrightarrow NO \quad K_4 = \frac{x_{10}}{x_4^{0.5} x_3^{0.5}} \quad (23)$$

$$H_2 + \frac{1}{2}O_2 \leftrightarrow H_2O \quad K_5 = \frac{x_2}{x_4^{0.5} x_6^{0.5}} \quad (24)$$

$$CO + \frac{1}{2}O_2 \leftrightarrow CO_2 \quad K_6 = \frac{x_1}{x_5 x_4^{0.5} P^{0.5}} \quad (25)$$

The equilibrium constant K, for the above are determined from the Gibbs free energy as

$$-\frac{\Delta G}{RT} = \ln K_p \quad (26)$$

where ΔG is the standard-state Gibbs free energy. The values of the equilibrium constants can be obtained from [4, 18]

F. Thermal Properties

We consider the unburned and burnt mixture zones as separate open systems. Therefore, the specific internal energy, u and specific volume, v is expressed as

$$u = \frac{U}{m} = xv_b + (1-x)u_u \quad (27)$$

$$v = \frac{V}{m} = xv_b + (1-x)v_u \quad (28)$$

subscripts b and u refers to burnt gas and unburned gas respectively.

Assuming that the pressures of burnt and unburned gases are equal, v_b and v_u are functions of T_b , T_u and p . Hence

$$\frac{dv_b}{d\theta} = \frac{\partial v_b}{\partial T_b} \frac{dT_b}{d\theta} + \frac{\partial v_b}{\partial p} \frac{dp}{d\theta} \quad (29)$$

$$\frac{dv_u}{d\theta} = \frac{\partial v_u}{\partial T_u} \frac{dT_u}{d\theta} + \frac{\partial v_u}{\partial p} \frac{dp}{d\theta} \quad (30)$$

Substituting the logarithmic derivatives into the Eq. (29) and Eq. (30) yields

$$\frac{dv_b}{d\theta} = \frac{v_b}{T_b} \frac{\partial \ln v_b}{\partial \ln T_b} \frac{dT_b}{d\theta} + \frac{v_b}{p} \frac{\partial \ln v_b}{\partial \ln p} \frac{dp}{d\theta} \quad (31)$$

$$\frac{dv_u}{d\theta} = \frac{v_u}{T_u} \frac{\partial \ln v_u}{\partial \ln T_u} \frac{dT_u}{d\theta} + \frac{v_u}{p} \frac{\partial \ln v_u}{\partial \ln p} \frac{dp}{d\theta} \quad (32)$$

Similarly, the internal energies of both the burnt and unburned gases, under the same pressure condition and including the logarithmic derivatives can be written as

$$\frac{du_b}{d\theta} = \left(c_{pb} - \frac{pv_b}{T_b} \frac{\partial \ln v_b}{\partial \ln T_b} \right) \frac{dT_b}{d\theta} - v_b \left(\frac{\partial \ln v_b}{\partial \ln T_b} + \frac{\partial \ln v_b}{\partial \ln p} \right) \frac{dp}{d\theta} \quad (33)$$

$$\frac{du_u}{d\theta} = \left(c_{pu} - \frac{pv_u}{T_u} \frac{\partial \ln v_u}{\partial \ln T_u} \right) \frac{dT_u}{d\theta} - v_u \left(\frac{\partial \ln v_u}{\partial \ln T_u} + \frac{\partial \ln v_u}{\partial \ln p} \right) \frac{dp}{d\theta} \quad (34)$$

G. Mass in Control Volume

The mass at any angle is

$$m = m_1 \exp \left[-C(\theta - \theta_1) / \omega \right] \quad (35)$$

where C is the blow-by constant. The initial mass m_1 at $\theta = \theta_1$ (start of compression) is specified from knowledge of the volumetric efficiency and residual fraction.

The volume of the cylinder at any crank angle instant is given by [4]

$$V = V_c \left\{ 1 + \frac{r-1}{2} \left[\frac{1 - \cos \theta + \frac{1}{\varepsilon} \left[1 - (1 - \varepsilon^2 \sin^2 \theta)^{0.5} \right]}{\varepsilon} \right] \right\} \quad (36)$$

V_c is clearance volume, r is the compression ratio and

$$\varepsilon = \frac{\text{stroke}}{2 * \text{length of connecting rod}}$$

H. Fuel Burning Rate Model

The Wiebe function represents the mass fraction burned, x_b versus crank angle and defined as [6]

$$x_b(\theta) = 1 - \exp \left[-a \left(\frac{\theta - \theta_s}{\theta_d} \right)^n \right] \quad (37)$$

where θ = crank angle, θ_s = start of heat release, θ_d = duration of heat release, n = Wiebe form factor and a = Wiebe efficiency factor. The parameters a and n are adjustable parameters used to fit experimental data. The present simulation uses $a=5$ and $n=3$, these values have been reported in [6].

I. Heat Transfer Model

Heat transfer into the system is expressed in terms of heat loss from the burned and unburned gas respectively.

$$\frac{dQ}{d\theta} = \frac{-Q_l}{\omega} = \frac{-Q_b - Q_u}{\omega} \quad (38)$$

To express the heat loss in terms of temperature requires the introduction of a heat transfer coefficient h ,

$$Q_b = hA_b(T_b - T_w) \quad (39)$$

$$Q_u = hA_u(T_u - T_w) \quad (40)$$

where A_b and A_u are the areas of burned and unburned gas in contact with the cylinder walls at temperature T_w . We have the following relations to calculate the areas of A_b and A_u

$$A_b = \left(\frac{\pi b^2}{2} + \frac{4V}{b} \right) x^{0.5} \quad (41)$$

$$A_u = \left(\frac{\pi b^2}{2} + \frac{4V}{b} \right) (1 - x)^{0.5} \quad (42)$$

The fraction of cylinder area contacted by burned gas is assumed to be proportional to the square root of the mass fraction burned to reflect the fact, because of the density difference between burned and unburned gas, the

burned gas occupies a larger volume fraction of the cylinder than the unburned gas. We assumed that $h_u = h_b = h$ = constant in this paper.

This simulation model has the convenience to adapt the heat transfer correlation proposed by Woschni [7].

J. Blow-by Energy Loss

Enthalpy loss due to blow-by is expressed as [4]

$$h_l = (1 - x^2)h_u + x^2 h_b \quad (43)$$

Early in the combustion process, unburned gas leaks past the rings. Late in the combustion process, burned gas leaks past in the rings. The above indicates that more leaking is due to the unburned gas compared with the burnt gas in the early stage of combustion.

K. Principle Governing Equations

Differentiating (28) with respect to crank angle and incorporating (31) and (32), we have

$$\frac{1}{m} \frac{dV}{d\theta} + \frac{VC}{m\omega} = x \frac{v_b}{T_b} \frac{\partial \ln v_b}{\partial \ln T_b} \frac{dT_b}{d\theta} + (1-x) \frac{v_u}{T_u} \frac{\partial \ln v_u}{\partial \ln T_u} \frac{dT_u}{d\theta} + \left[x \frac{v_b}{p} \frac{\partial \ln v_b}{\partial \ln p} + (1-x) \frac{v_u}{p} \frac{\partial \ln v_u}{\partial \ln p} \right] \frac{dp}{d\theta} + (v_b - v_u) \frac{dx}{d\theta} \quad (44)$$

Expressing the heat loss of burnt and unburned gases as a function of the rate of change of specific entropy gives

$$-Q_b = m\omega x T_b \frac{ds_b}{d\theta} \quad (45)$$

$$-Q_u = m\omega(1-x) T_u \frac{ds_u}{d\theta} \quad (46)$$

Where

$$\frac{ds_b}{d\theta} = \left(\frac{c_{pb}}{T_b} \right) \frac{dT_b}{d\theta} - \frac{v_b}{T_b} \frac{\partial \ln v_b}{\partial \ln T_b} \frac{dp}{d\theta} \quad (47)$$

$$\frac{ds_u}{d\theta} = \left(\frac{c_{pu}}{T_u} \right) \frac{dT_u}{d\theta} - \frac{v_u}{T_u} \frac{\partial \ln v_u}{\partial \ln T_u} \frac{dp}{d\theta} \quad (48)$$

Expressing the heat loss of burnt and unburned gases as a function of the rate of change of specific entropy by combining (39)-(42) and (45)-(48)

$$c_{pb} = \frac{dT_b}{d\theta} - v_b \frac{\partial \ln v_b}{\partial \ln T_b} \frac{dp}{d\theta} = \frac{-hA_b(T_b - T_w)}{m\omega x} \quad (49)$$

$$c_{pu} = \frac{dT_u}{d\theta} - v_u \frac{\partial \ln v_u}{\partial \ln T_u} \frac{dp}{d\theta} = \frac{-hA_u(T_u - T_w)}{m\omega(1-x)} \quad (50)$$

Differentiating Equations (36) and (37) and incorporating with Equations (4), (27), (28), (31)-(34) and (38)-(43) into Equation (5), we have the following relations [4]

$$\frac{dp}{d\theta} = \frac{A + B + C}{D + E} \quad (51)$$

$$\frac{dT_b}{d\theta} = \frac{-hA_b(T_b - T_w)}{m\omega c_{pb}x} + \frac{v_b}{c_{pb}} \frac{\partial \ln v_b}{\partial \ln T_b} \frac{dp}{d\theta} + \frac{h_u - h_b}{xc_{pb}} \left[\frac{dx}{d\theta} - (x - x^2) \frac{C}{\omega} \right] \quad (52)$$

$$\frac{dT_u}{d\theta} = \frac{-hA_u(T_u - T_w)}{m\omega c_{pu}(1-x)} + \frac{v_u}{c_{pu}} \frac{\partial \ln v_u}{\partial \ln T_u} \frac{dp}{d\theta} \quad (53)$$

Where

$$A = \frac{1}{m} \left(\frac{dV}{d\theta} + \frac{VC}{\omega} \right) \quad (54)$$

$$B = \frac{h}{m\omega} \left[\frac{v_b}{c_{pb}} \frac{\partial \ln v_b}{\partial \ln T_b} \frac{A_b(T_b - T_w)}{T_b} + \frac{v_u}{c_{pu}} \frac{\partial \ln v_u}{\partial \ln T_u} \frac{A_u(T_u - T_w)}{T_u} \right] \quad (55)$$

$$C = -(v_b - v_u) \frac{dx}{d\theta} - v_b \frac{\partial \ln v_b}{\partial \ln T_b} \frac{h_u - h_b}{c_{pb}T_b} \left[\frac{dx}{d\theta} - \frac{(x - x^2)C}{\omega} \right] \quad (56)$$

$$D = x \left[\frac{v_b^2}{c_{pb}T_b} \left(\frac{\partial \ln v_b}{\partial \ln T_b} \right)^2 + \frac{v_b}{p} \frac{\partial \ln v_b}{\partial \ln p} \right] \quad (57)$$

$$E = (1-x) \left[\frac{v_u^2}{c_{pu}T_u} \left(\frac{\partial \ln v_u}{\partial \ln T_u} \right)^2 + \frac{v_u}{p} \frac{\partial \ln v_u}{\partial \ln p} \right] \quad (58)$$

Equations (54)-(58) are functions of θ , p , T_b and T_u respectively.

III. RESULTS

The results obtained from the thermodynamic simulation model are presented. We simulate Pressure, Temperature, Work, Heat Transfer and Heat flux for a single cylinder; four stroke SI engine running on Isooctane (C_8H_{18}). We have assumed a bore of 0.1m, stroke of 0.15m. The engine is assumed to operate at 2000 rpm, with an equivalence ratio of 0.8.

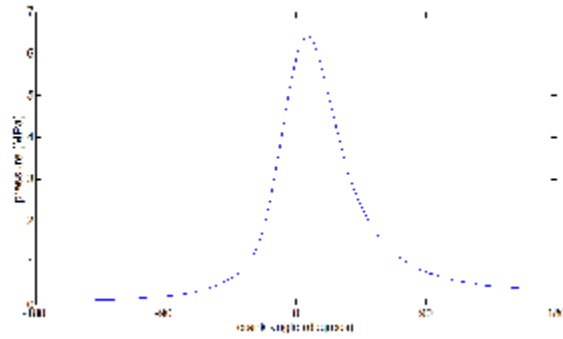


Figure 1: Pressure v/s Crank Angle

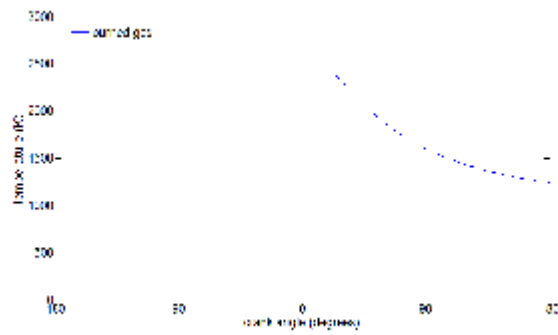


Figure 2: Temperature v/s Crank Angle

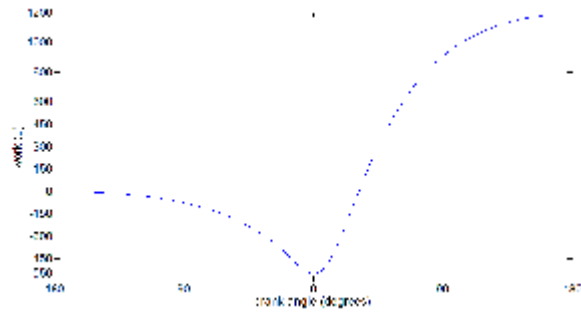


Figure 3: Work v/s Crank Angle

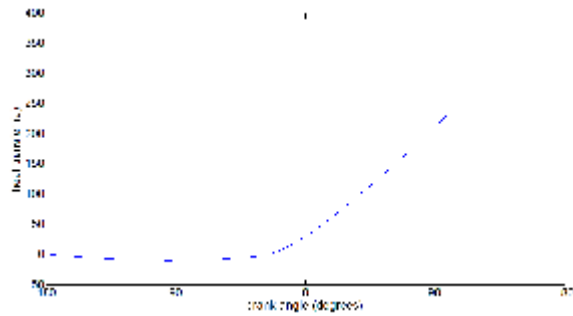


Figure 4: Heat Transfer v/s Crank Angle

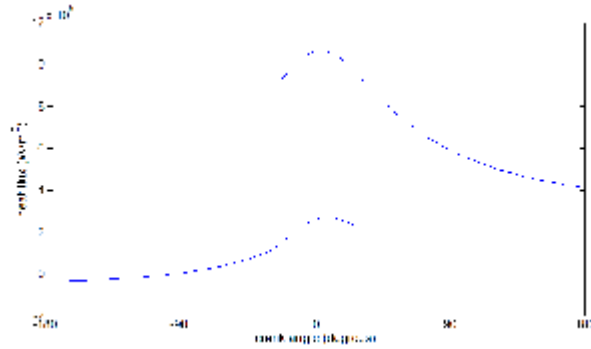


Figure 5: Heat Flux v/s Crank Angle

IV. CONCLUSION

A thermodynamic simulation model has been developed to simulate a single cylinder 4-stroke cycle of a spark ignition engine fuelled with Isooctane. The program written from this simulation model can be used to assist in the design of a spark ignition engine for alternative fuels as well. The results obtained can be used as a first-degree approximation and is useful in numerous engineering applications including general design predictions.

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