Single-Zone Thermodynamic Modelling of a Four Stroke Natural Gas Spark Ignition Engine

M. Mansha[1], E.M. Shahid[2], I. A. Chaudhry[3], A.H. Qureshi[4]

[1,2,3,4]Department of Mechanical Engineering, University of Engineering & Technology Lahore 54570, Pakistan
[1]mmanshaj@yahoo.com, [4]emshahid@uet.edu.pk

Abstract-This paper presents single zone thermodynamic simulation of closed part of spark ignition engine based on the Java Computer Applet. In the present work , finite Heat Release with Heat transfer library of the Applet has been used to find out indicated performance parameters at wide open throttle (WOT) condition for four stroke four cylinder 2.5 L dedicated natural gas engine, compression Ratio is 11, at various engine speeds (1000, 1500, 2000, 2500 & 3000 rpm). The same engine was also tested on dynamometer to determine the brake power at the same engine speeds at WOT condition. Brake power obtained through simulation was compared with the experimentally obtained brake power. The theoretical results obtained through simulation model are in close agreement with measured data. The results are almost matching up to 2000 rpm and at higher speeds i.e. from 2000 rpm to 3000 rpm the maximum deviation of around 7% has been found at 3000 rpm. The main reason of deviation at higher speeds may be due to the use of engine friction model which calculates engine friction based on motored engine.

Keywords-Single-zone model, Heat release analysis; Spark ignition engine

I. INTRODUCTION

Computer based thermodynamic & fluid dynamics models have long been used for predicting engine performance & emissions. The aim with modelling IC engines is: to predict engine performance without having to conduct tests, and to deduce the performance of parameters that can be difficult to measure. [1] remarked that the design of an engine and its controlling strategy often culminates as a trade-off between efficiency, emission and many practical constraints and modelling is a tool to envisage the performance of particular design. Mathematical models for spark ignition engines can be divided broadly in two groups- thermodynamics-based (single &multi zones) and fluid dynamics–based (CFD based). Other labels associated to thermodynamic-based models are: zero-dimensional, phenomenological and quasi-dimensional. In zero-dimensional (ZD) models Empirical Heat Release Models are used wherein time is the only variable (resulting in ordinary differential equation). As no flow modelling exists so geometric features of the fluid motion cannot be predicted in ZD models. In phenomenological models additional detail beyond the energy conservation equations is added for each phenomenon in turn and in quasi-dimensional models some specific geometric features, e.g., the spark-ignition engine flame or the diesel fuel spray shapes, are added to the basic thermodynamic approach. Separate sub models are also used for turbulent combustion to derive a Heat Release Model, [2].

Fluid dynamic-based models are often called multidimensional models owing to their intrinsic ability to provide in depth geometric information on the flow field based on solution of the governing flow equations. These models solve numerically the equations for mass, momentum, energy and species conservation in three dimension (Navier-Stokes equations), in order to predict the flame propagation. As these equations are dependent on spatial co-ordinates, so these equations take the form of partial differential equations. The progression of engine models starts from the ideal cycle calculations in the 1950's to simple component matching models in the 1960's, full thermodynamic models during the late 1970's, and multi-zone & multidimensional combustion models in the 1980's and 1990's as described by [3].

II. SINGLE ZONE THERMODYNAMIC MODELLING

Single-zone (Zero Dimensional) model is based on the First law of thermodynamics and can be used as either predictive or diagnostic tool. The model does not account for combustion chamber geometry except in global manner through volume (V). Combustion in single-zone model can be approximated as heat addition process and the heat addition is specified as a function of crank angle CA (as predictive tool). The outcome of the model is temperature and pressure which is the function of crank angle. The heat release-time curve employed (combustion rate model) is a Semi Empirical Weibe function or Cosine curve, it describes the fraction of mass of fuel consumes which depends upon crank angle. The model does not involve any flow field detail, so geometric features of fluid motion cannot be predicted. In single zone model analysis, it is assumed that the specific internal energy of the charge is a function of temperature only. The cylinder-gas composition is modelled assuming as uniform and gas mixture is perfect. Average gases properties are assumed with no distinguish between burned & unburned gases. The heat transfer coefficient is calculated by Woschni correlation. The model, it is supposed that the unburned and burned gasses mix within no time. [4] are of the view that these models are fast and cheap in terms of computation.

A. Mathematical Formulations

When applied to closed part of the engine and ignoring the crevice mass flow, the first law of thermodynamic may be written as

$$\delta Q_{add} - \delta Q_{loss} - \delta W = dU \tag{1}$$

Where

$$\delta Q = \delta Q_{add} - \delta Q_{loss}$$

δQin = Total amount of heat input to cylinder by combustion of fuel per cycle = mf LHV

δQadd = Total amount of heat added from the fuel to the system.

Upon differentiating, (1) becomes

$$\delta Q_{add} - \delta Q_{loss} - PdV = \frac{C_v}{R}\left(PdV + VdP\right) \tag{2}$$

With respect to crank angle θ, (2) can be written as

$$\frac{dP}{d\theta} = \frac{\gamma - 1}{V}\frac{dQ_{add}}{d\theta} - \frac{dQ_{loss}}{d\theta} - \frac{\gamma PdV}{Vd\theta} \tag{3}$$

Where the rate of heat release as a function of crank angle is given as

$$\frac{dQ_{add}}{d\theta} = \frac{Q_{in}dx_b}{d\theta} = \frac{Q_{in}}{\theta_d}\left(1 - x_b\right)\left[\frac{\theta - \theta_s}{\theta_d}\right]^{n-1} \tag{4}$$

Where xb is the Weibe function which is used to determine the mass fraction burned Vs crank angle using finite heat release model. The Weibe function is described as

$$x_b\left(\theta\right) = 1 - \exp\left[-a\left\{\frac{\theta - \theta_s}{\theta_d}\right\}^n\right] \tag{5}$$

Where θ is crank angle, θs is start of heat release, θd is duration of heat release, n is Weibe form factor and a is Weibe efficiency factor. Values of n = 3 & a = 5 are best fit to the experimental data.

$$\frac{dP}{d\theta} = \frac{\gamma - 1}{V}\left[\frac{dQ_{in}dx_b}{d\theta} - \frac{dQ_{loss}}{d\theta}\right] - \frac{\gamma PdV}{Vd\theta} \tag{6}$$

Where dQw/dθ is heat transfer rate at any crank angle θ to the exposed cylinder wall

$$\frac{dQ_{loss}}{d\theta} = \frac{h_g(\theta)A_w(\theta)(T_g(\theta) - T_w)}{N} \tag{7}$$

Equation (8) is known as Newtonian convection equation and is used to determine the heat transfer rate at any crank angle (θ) to the exposed cylinder wall at any engine speed N. Alternatively the (7) may also be written as

$$\frac{dQ_{loss}}{d\theta} = \frac{hgAw}{\omega}(T_g - T_{w)}\frac{\pi}{180} \tag{8}$$

Where dQloss is differential heat transfer to cylinder wall, hg is instantaneous area averaged heat transfer coefficient; Aw is exposed cylinder area which is the sum of cylinder bore area, the cylinder head area and the piston crown area, Tg is cylinder gas temperature, Tw is cylinder wall temperature, ω

is engine Speed (rad/sec) and N is engine Speed (rpm)

$$\frac{dP}{d\theta} = \frac{\gamma - 1}{V}\left[\frac{dQ_{in}dx_b}{d\theta} - \frac{hA}{\omega}(T_g - T_w)\frac{\omega}{180}\right] - \frac{\gamma PdV}{Vd\theta} \tag{9}$$

The area of the cylinder can be calculated using the crank slider model (Fig. 1.) as

$$A = \frac{\pi}{2}b^2 + \pi b\frac{s}{2}\left[R + 1 - \cos\theta + (R^2 - \sin^2\theta)^{\frac{1}{2}}\right] \tag{10}$$

Where, R = L/a,

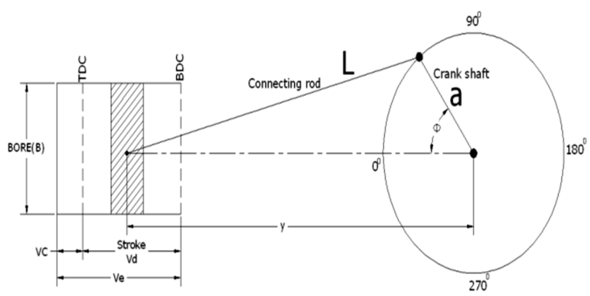

Fig. 1. Crank slider Model

The temperature of the gas, Tg, will be an average temperature. The heat transfer coefficient given by Woschni is:

$$h = 3.26b^{-0.2}P^{0.8}T^{-0.55}v^{0.8} \tag{11}$$

By using the above equations indicated quantities can be determined. To convert the indicated performance to brake performance, it is necessary to predict the frictional losses. The total frictional work of engine comprises of three components, pumping work, rubbing friction work and accessories work.

To evaluate frictional mean effective pressure of automotive four stroke engines following correlation has been employed by Alla (2002).

$$\text{FMEP (bar)} = 0.97 + 0.15\,[N/100] + 0.05\,[N/1000]2 \quad (12)$$

Equation (12) is used by Heywood (1998) and Richard (1999), to determine frictional mean effective pressure (FMEF) of motoring four-stroke four cylinders engine. The brake mean effective pressure (BMEP) can be found by using the following equation.

$$\text{BMEP (bar)} = \text{IMEP} - \text{FMEP} \quad (13)$$

Where IMEP is the indicated mean effective pressure, which is the work delivered to the piston over the compression & expansion stroke, per cycle per unit displacement volume. The brake power is then determined by the following equation.

$$\text{Brake Power (KW)} = \text{BMEP} \times \text{Vd}\,[N/2] \times n\,\{\text{for } 4 - \text{stroke}\} \quad (14)$$

Where, Vd is the engine displacement, N is engine revolution per minute and n is the number of cylinders.

In order to determine the heat input Qin in (J), the mass of the fuel entering the cylinder has to be calculated. The equivalence ratio can be defined as

Φ = (mf/ma) act/ (mf/ma) s

$$\Phi = (\text{mf/ma})\ \text{act}/\ (\text{mf/ma})\ \text{s} \quad (15)$$

$$\text{ma, act} = \Phi\ (\text{mf/ma})\text{s} * \text{ma,act} \quad (16)$$

$$\text{Qin} = \text{HHVfuel} * \text{mf, act} \quad (17)$$

However in the simulation lower heating value has been used as input condition

qin = Qin/mass of gas mixture inducted, which may be written as [5]

$$= \Phi\text{Fs}/1 + \Phi\text{Fs} * [\text{qc} - 3890\ (\Phi - 1)] \quad \text{for } \Phi > 1 \quad (18)$$

$$\text{qin (kJ/kg mix)} = \Phi\text{Fs}/1 + \Phi\text{Fs} * \text{qc} \quad \text{for } \Phi \le 1 \quad (19)$$

Where qc is the heat of combustion of fuel (here LHV)

For hydrocarbon fuel

$$\gamma = 1.4 - 0.16\,\Phi \quad (20)$$

The mass of air can be expressed from displacement volume

$$\text{ma, act} = \rho\text{Vd} \quad (21)$$

TABLE I

CONDITIONS FOR SIMULATION

Parameters	Values
Pressure inside manifold (near intake valve)	0.98 bar
Inlet temperature T_i	323 K (experimental value)
Mean cylinder wall temperature	423 K
Specific heat ratio γ	Using Eq-20
Air density ρ_{air}	1.2 kg/m^3
Equivalence ratio Φ	1
Heat of combustion of methane (q_c) LHV	50.01 MJ /kg
q_{in} (kJ/kg $_{mix}$)	Using Eq-18
Stoichiometric Fuel-air ratio of methane (F_s)	0.05814
Spark timing (θ_s)	35deg bTDC
Duration of combustion (θ_d)	60 degaTDC
Weibe efficiency factor (a)	5
Weibe form factor (n)	3
Engine Geometric Parameters	
Stroke (S)	95 mm
Bore (b)	91.1mm
Connecting rod length mm	158mm
Compression ratio (r)	11
Engine speed	1000, 1500, 2000, 2500, 3000 rpm

III. SIMULATION RESULTS & DISCUSSION

Figs. 2-5 show the simulation results i.e. indicated work, indicated mean effective pressure, indicated power and maximum pressure obtained though single zone thermodynamic heat release model using Woschni heat transfer correlation. These figures illustrate the increasing trends of all the quantities with increase in engine speed. Fig. 6 shows the calculated brake power based on (12)~(14) at various engine speeds. The brake power obtained through single-zone computational model was compared and verified against the experimentally measured brake power in Fig. 7. The computational results are in close agreement with measured data. The results are almost matching up to 2000 rpm; however at higher speeds i.e from 2000 rpm to 3000 rpm the maximum deviation of around 7% has been found between simulation & experimental results, which is well within the published data [6], [7], [8]. The simulation results start departing from the experimental results at 2000 rpm to 3000 rpm with maximum 7% over prediction at 3000 rpm. The main reason of deviation at higher speeds may be due to the use of engine friction model which calculate engine friction based on motoring the engine. Further in actual engine, natural gas with 87% methane has been used while in the model simulation 100% methane has been employed

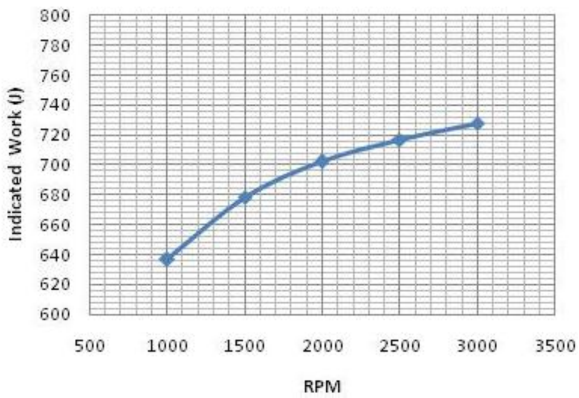

Fig. 2. Indicated work Vs Engine speed

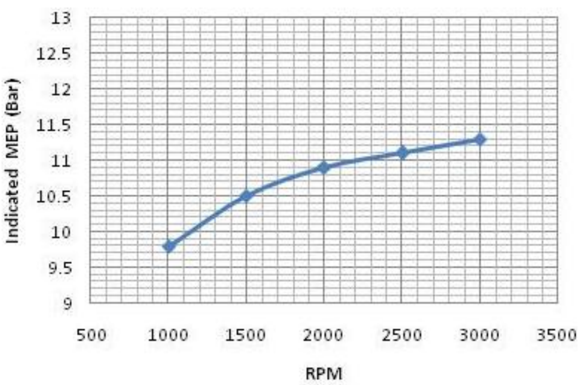

Fig. 3. Indicated `mean effective pressure (IMEP) Vs Engine speed

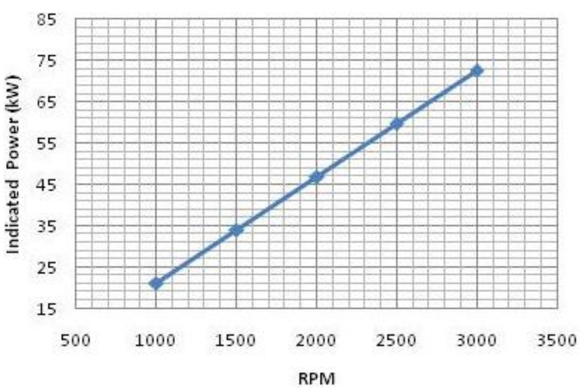

Fig. 4. Indicated power Vs Engine speed

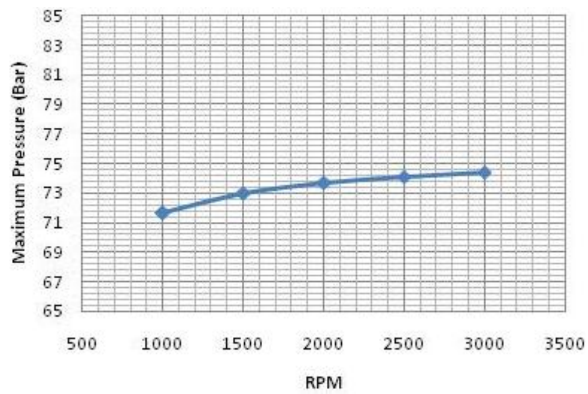

Fig. 5. Maximum pressure Vs Engine speed

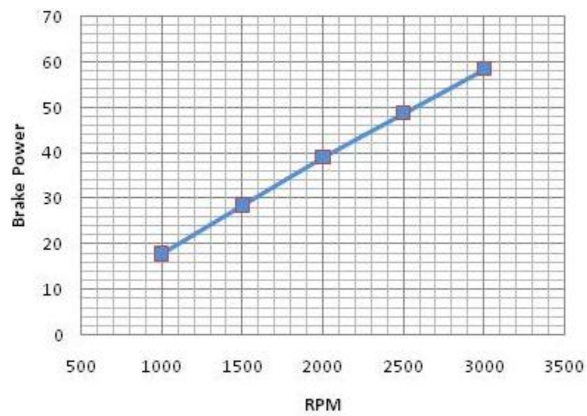

Fig. 6. Calculated brake power based on (12)-(14) at various engine speeds

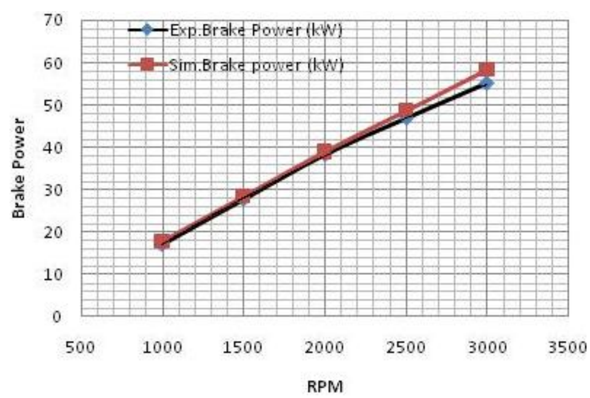

Fig. 7. Comparison of experimental & Simulation results

IV. Conclusions

Single-zone thermodynamic model is simple model based on the first law analysis which can safely be used during the early stages of engine development to determine the performance of the engine. As no flow modeling exists so geometric features of the fluid motion cannot be predicted in the model. Comparisons have been made between computational results obtained though the model and experimental results to confirm the reliability and accuracy of the model for predicting brake power of spark ignition engine running on natural gas. The computational results are in close agreement with measured data. The results are almost matching up to 2000 rpm; however at higher speeds i.e from 2000 rpm to 3000 rpm the maximum deviation of around 7% has been found between simulation & experimental results, which is well within the range.

Acknowledgment

Finally I would like to acknowledge Muhammad Faisal for his cooperation and support.

References

[1] C. Borganakke, P. Puzinauskas, P. Xiao (1986), Technical Report-1 on Spark Ignition Engine Simulation Models, Report No. UM-MEAM-86-35 University of Michigan.

[2] J. B. Heywood (1988), Internal Combustion Engine Fundamentals. New York: McGraw-Hill.

[3] A. Chow, M. L. Wyszynoki (1999), Proceeding Institute of Mechanical Engineer, Thermodynamic modeling of complete engine systems- a review, Vol. 213 Part D, Journal of Automobile Engineering April 1, 1999 vol. 213 no. 4 403-415.

[4] S. Verhelst, C. G. W. Sheppard (2009), Energy Conversion and Management, Multi-Zone Thermodynamic Modeling of Spark-Ignition Engine Combustion- An Overview, Vol. 50, NO. 5. 1326-1335.

[5] S. Richard (1999), Introduction to Internal Combustion Engines 3rd ed. Macmillan Press ltd London,

[6] C. R. Ferguson, K. T. Allan (2004), Internal Combustion Engines Applied Thermo sciences, 2nd ed, Wiley Indea.

[7] M. H. W. Barnes (1975), conference proceeding, Institute of Mechanical Engineering London, A designer's viewpoint, in passenger car engines.

[8] G. H. AbdAlla (2002), Computer Simulation of a four stroke spark ignition engine, Energy Conservation and Management, Vol. 43, NO.8. 1043-1061.