An Analytical Model of Spark Ignition Engine for Performance Prediction

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Abstract

The objective of this research is to develop mathematical model of spark ignition engine using cylinder-by-cylinder model approach in order to predict the performances; torque and power. The method is based on ideal Otto cycle and modified by equations which affect the performances. The model consists of set of tuning parameters such as engine physical geometries, ignition advanced, air/fuel ratio, etc. It is also developed under Matlab/Simulink. The results from simulation are verified with the data from commercial engines.

Keyword: Spark ignition engine, Gasoline engine, Engine modeling, Cylinder-by-cylinder engine model, Engine simulation

1. Introduction

Four-stroke spark ignition engine was developed by Otto in 1876. This engine provided power output 3 HP. Since that point, the engine developing has been done continuously over 100 years. Some engines can provide power output more than 1,000 HP.

However, developments of spark ignition engine for last 100 years were very slow due to lots of uncontrollable parameter, such as physical geometries, ignition advanced, valve timing, combustion characteristic, influencing the performances. Many studies on effects of each parameter were done by experiments. But that approach needs lot of expenses and time such as building test engine, setting up laboratory, etc.

Another approach is simulation method that allows engine designer to change and test many different parameters without building real parts or even real engines. The engine model can be used in various ways including designing engine control system or ECU, designing transmission, etc. However, computational model is unable to obtain the exact characteristics because there are no perfect models. There are many complicated processes that take place in the engine. It simply estimates the trend of those characteristics and effects with very low cost and less time consumption that is very helpful for speeding up the engine development process before making real one.

The basic operation of a four stroke engine involves intake, compression, expansion (or power), and exhaust strokes as shown in Fig.1.

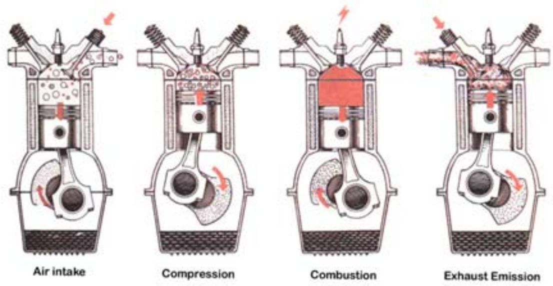
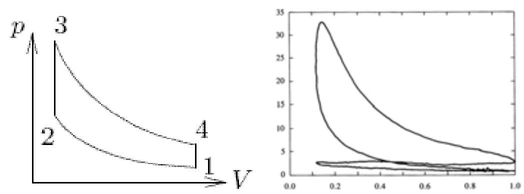


Figure 1. Basic four stroke cycle

With models for each of these processes, a simulation of complete engine cycle can be built up and be analyzed to provide information on engine performances. These ideal models that describe characteristic of each process are proposed [1]. However the calculation needs information from each state as shown in Fig. 2a which sometime could not obtain from real condition as shown in Fig. 2b.



(a) (b)

Figure 2. Pressure-volume diagram of Otto cycle: (a) ideal; (b) real

Overall engine work can be determined by integrating the area under the pressure-volume diagram or P-V diagram. So many previous works concerned mainly prediction the pressure inside the combustion chamber [2, 3 and 4]. But the pressure and volume are influenced by engine geometries during variation of crank angle. So the pressure and displacement volume are needed to convert as functions of crank angle. Kuo [3] and Kirkpatrick [5] proposed the method that can calculate the pressure and volume at any crank angle. The combustion process can be described by Wiebe function [1].

The results from Kuo [3] and Zeng et al [4] indicated that heat transfer from inside the cylinder to engine cooling water had much influences on the pressure inside the cylinder. So the heat transfer function is needed to take into account in the model.

Many researches reported that the mass of mixture that flows into the cylinder during intake stroke is very importance parameter [1, 2, 3 and 6] because it affects amount of fuel which mixes with the air. This mass can be determined by combining the ideal gas law and volumetric efficiency. However it is very difficult to evaluate because they are affected by many factors, such as manifold geometries and valve timing [1]. So Eriksson et al [2] and Kuo [3] assumed that the pressure inside manifold and inside the cylinder are the same, and neglect effect of volumetric efficiency. But Kuo [3] used corrective equation from real experiment to compensate the errors. While Zeng et al [6] took the effect of volumetric efficiency into account. However the data were obtained from the real experiment and stored in a 3-D table.

Combining of those methods that are mentioned above can predict the engine performances precisely only if some testing data are known, mainly the volumetric efficiency. So this research tries to use the zero-dimension model to predict the volumetric efficiency in order to reduce the testing data dependence.

2. The model

The engine to be modeled is a straight four cylinder gasoline engine which has no Exhaust Gas Recirculation (EGR) and turbo system as shown in Fig. 3. The chosen model bases on pressure inside the cylinder prediction and also the cylinder-by-cylinder model is used since the objective of this research intends to develop the model improving the engine efficiency, the mean value model which is meaningless in physical terms cannot achieve this goal.

Another consideration for model selection is empirical equations and physical based equations. Since there are no perfect equations which can describe phenomena in the engine, both formulae are used in the model.

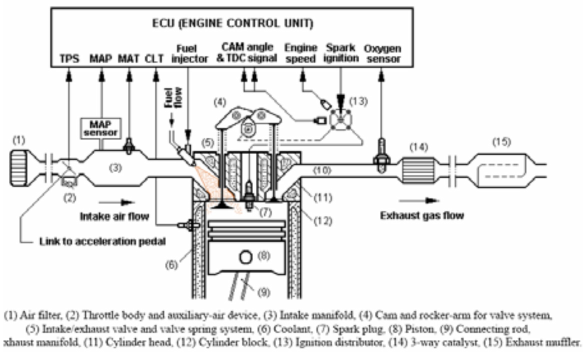


Figure 3. Schematic of gasoline engine [6]

2.1 Model overview

Fig. 4 shows the overview of the model. Engine geometries, such as bore, stroke, compression ratio, etc., are calculated to obtain physical information such as displacement volume, area and volume variation as function of crank angle, etc. That information will be used for cylinder pressure prediction with another line of information about heat input. Heat energy needs data from amount of flow in mass and burn characteristic which is described by Wiebe function. Predicted pressure will be used to determine temperature inside cylinder and also heat transfer from cylinder to wall chamber. Rate of heat loss will be fed back to the pressure prediction function. Resulted pressure will be converted to indicated mean effective pressure subtracted by mean friction, then work and power will be known finally. The details of each module are described in following section.

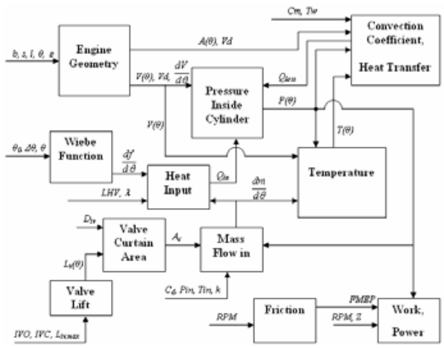


Figure 4. Model overview

2.2 Crank slider model

The volume of the piston cylinder can be determined as a function of crank angle from the compression ratio, the stroke, bore and connecting rod length. The geometric parameters of the piston cylinder can be described by the crank slider model which is represented in Fig. 5.

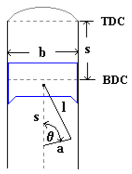


Figure 5. Piston cylinder

Where b = bore θ = crank angle
s = stroke a = crank radius (= ½ s)
l = connecting rod length
TDC = top dead center
BDC = bottom dead center

The equations of volume ($V(\theta)$) and area ($A(\theta)$) that relate to crank angle are described as following equation:

$$V(\theta)=\frac{V_d}{\varepsilon\text{-}1}+\frac{V_d}{2}[\frac{l}{a}+1-\cos\theta-(\left(\frac{l}{a}\right)^2-\sin^2\theta)^{\frac{1}{2}}] \quad (1)$$

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

Where V_d = displacement volume
ε = compression ratio

2.3 Cylinder pressure model

This model is derived from the first law of thermodynamics. The pressure is derived as function of crank angle also [5].

$$\frac{dP}{d\theta} = \frac{k-1}{V}\frac{\partial Q}{d\theta} - k\frac{P}{V}\frac{dV}{d\theta} \qquad (3)$$

Where P = pressure inside cylinder
k = specific heat ratio c_p/c_v
Q = heat addition

$\frac{dV}{d\theta}$ can be determined from Eq. (1) by taking derivative with respect to the crank angle, θ. For heat release term, $\frac{\partial Q}{d\theta}$, the Wiebe function for the burn fraction is used.

2.4 Wiebe function

The mass fraction burned profiles as a function of crank angle in each individual cycle. It has a characteristic S-Shape. The rate at which fuel-air mixture burns increases from a low value intermediately following the spark discharge to a maximum about halfway through the burning process and then decreases to a close to zero as the combustion process ends.

A functional form often used to represent the mass fraction burned versus crank angle curve is the Wiebe function [1]:

$$f(\theta) = 1 - \exp\left[-5\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^3\right] \qquad (4)$$

Where $f(\theta)$ = the fraction of heat added
θ = the crank angle
θ_0 = angle of the start of the heat addition
$\Delta\theta$ = the duration of the heat addition (length of burn)

The heat release, ∂Q, over the crank angle change, $\Delta\theta$, is:

$$\frac{\partial Q}{d\theta} = Q_{in}\frac{df}{d\theta} \qquad (5)$$

Where Q_{in} = overall heat input

Then take the derivative of the heat release function, $f(\theta)$, with respect to crank angle, being $\frac{df}{d\theta}$ from Eq. (4).

2.5 Burning duration

Mixture burning rate is strongly influenced by engine speed [1]. It is well established that the duration of combustion in crank angle degrees only increases slowly with increasing engine speed. Fig. 6 shows how intervals of burning characteristics are. The burning rate throughout the combustion process does not increase linearly as engine speed. Additionally, increasing in-cylinder gas velocities (e.g. with intake generated swirl) increases the burning rate. Increasing engine speed and introducing swirl both increase the levels of turbulence in the engine cylinder.

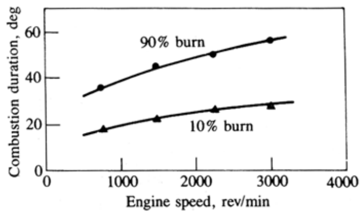


Figure 6. Effect of engine speed on flame-development angle 10% and 90% burn [1]

However, the swirl characteristic is not able to determine easily depending on manifold geometries and piston head design. Therefore, the results in Fig. 6, which obtained from experiment, are used to estimate the burning duration, $\Delta\theta$.

$$\Delta\theta = -1.6189\left(\frac{N}{1000}\right)^2 + 19.886\left(\frac{N}{1000}\right) + 39.951 \qquad (6)$$

Where N = engine speed (rpm)

2.6 Heat transfer

In spark ignition engines, the primary heat transfer mechanism from the cylinder gases to the wall is convection, with only 5% from radiation. Using a Newtonian model, the heat loss to the wall is given by:

$$Q_{loss} = hA(T_g - T_w) \qquad (7)$$

Where h = convection heat transfer coefficient
T_g = temperature of the cylinder gas
T_w = cylinder wall temperature
A = exposed combustion chamber surface area

When determining the heat release term, $\frac{\partial Q}{d\theta}$, the heat loss to the walls has to be taken into account. From Eq. (3) and (5), the pressure over crank angle changes now becomes [5]:

$$\frac{dP}{d\theta} = \frac{k-1}{V}\left[Q_{in}\frac{df}{d\theta} - \frac{hA}{6N}(T_g - T_w)\right] - k\frac{P}{V}\frac{dV}{d\theta} \qquad (8)$$

2.7 Heat transfer coefficient correlations

The heat transfer coefficient (h) is needed in order to calculate heat loss from the cylinder in Eq. (7). For design purposes, simplified analyzes are often performed using empirical heat transfer correlations such as Annand, Woschni and Hohenberg [1]. These give at most estimates of the surface-averaged heat transfer coefficient, which are defined in terms of the bulk gas temperature and used with this to calculate surface-averaged or total heat flux. Annand assumed a constant characteristic gas velocity equal to the mean piston speed, while Woschni assumed that the average gas velocity should be proportional to the mean piston speed. Hohenberg examined Woschni's formula and made changes to give better predictions. Kleeman et al [10] compared these empirical correlations with Computational Fluid Dynamic (CFD) prediction by using Standard Wall Function (SWF) and Modified Wall Function (MWF) methods. They founded that the MWF method gave the most accurate results. So using these simple correlations will introduce some errors. However MWF can only be obtained from CFD because the CFD program can calculate the instantaneous local heat fluxes which are not uniform throughout the combustion chamber, while this research assumes the charge are burned completely throughout the cylinder at the same time. According to the results from Kleeman, the Hohenberg correlation is selected because it is closest to MWF and also there is no empirical form of MWF. The Hohenberg correlation is following equation [4]:

$$h = 130 \cdot V^{-0.06} P^{0.8} T^{-0.4} (C_m + 1.4)^{0.8} \tag{9}$$

Where V = cylinder volume
P = pressure inside cylinder
T = temperature inside cylinder
C_m = mean piston speed

2.8 Volumetric efficiency

The intake system – the air filter, carburetor, and throttle plate (in a spark ignition engine), intake manifold, intake port, intake valve – restricts the amount of air which an engine of given displacement can induct. The parameter used to measure the effectiveness of an engine's induction process is the volumetric efficiency (η_v). Volumetric efficiency is only used with four-stroke engines which have distinct induction process. It is defined as the volume of air which is drawn into the intake system divided by the volume which is displaced by the piston. Typical maximum values of volumetric efficiency for naturally aspirated engines are in the range 80 to 90% [1]. The volumetric efficiency for diesels is somewhat higher than the spark ignition engines as shown in Fig. 7.

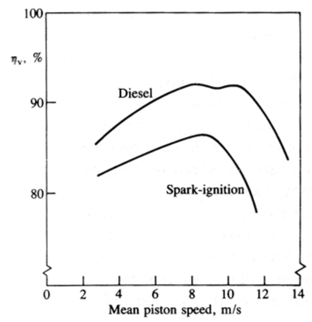


Figure 7. Volumetric efficiency versus mean piston speed for a four-cylinder indirect-injection diesel and a six-cylinder spark-ignition engine [1]

According to the Fig. 7, there are no such a model can predict the trend of the volumetric efficiency exactly because it is affected by the following fuel, engine design and engine operating variables [1]:

- Fuel type, air/fuel ratio, fraction of fuel vaporized in the intake system, and fuel heat of vaporization
- Mixture temperature as influenced by heat transfer
- Ratio of exhaust to inlet manifold pressures
- Compression ratio
- Engine speed
- Intake and exhaust manifold and port design
- Intake and exhaust valve geometry, size, lift and timings

The manifold and valve geometry design has great effects on the volumetric efficiency since the designs of each engine model have never been the same. So this research tries to develop the model to predict η_v from those factors as explain in following section.

2.9 Flow through valves

The valve, or valve and port together, is usually the most important flow restriction in the intake and the exhaust system of four-stroke cycle engines. In this research considers only the intake valve in order to determine η_v. The mass flow rate ($\dot{m}$) through a poppet valve is usually described by the equation for compressible flow through a flow restriction, Eq. (10). This equation is derived from a one-dimensional isentropic flow analysis, and real gas flow effects are included by means of an experimentally determined discharge coefficient (C_D).

$$\dot{m} = \frac{C_D A_R p_0}{(RT_0)^{0.5}} \left(\frac{p_T}{p_0}\right)^{1/k} \left\{ \frac{2k}{k-1} \left[1 - \left(\frac{p_T}{p_0}\right)^{(k-1)/k} \right] \right\}^{0.5} \tag{10}$$

Where C_D = discharge coefficient
A_R = reference area
p_0 = upstream stagnation pressure
T_0 = stagnation temperature
p_T = pressure at restriction
R = gas constant 287 J/kg/K

When the flow is choked, the pressure ratio ($\frac{p_T}{p_0}$) will not lower than the following value so called critical pressure ratio.

$$\left(\frac{p_T}{p_0}\right)_{Critic} \geq \left[\frac{2}{k+1}\right]^{k/(k-1)} \tag{11}$$

For the mass flow of the mixture into the cylinder through the intake valve, p_0 is the intake manifold pressure, assumed to be less than atmospheric due to loss from the air filter, and p_T is the cylinder pressure [1, 11]. T_0 is also temperature in the intake manifold. For A_R, the most convenient reference area in practice is the so called valve curtain area since it varies linearly with valve lift and is simple to determine [1, 5].

$$A_C = \pi D_v L_v \tag{12}$$

According to Eq. (8), the Eq. (10) should be converted into function of crank angle also by dividing with *6N*.

$$\frac{dm}{d\theta} = \frac{C_D A_R p_0}{6N(RT_0)^{0.5}} \left(\frac{p_T}{p_0}\right)^{1/k} \left\{ \frac{2k}{k-1} \left[1 - \left(\frac{p_T}{p_0}\right)^{(k-1)/k} \right] \right\}^{0.5} \tag{13}$$

The Eq. (13) is integrated to obtain amount of mass over a crank angle degree and then converted becoming pressure of mixture inside cylinder by using ideal gas equation of state, $PV = mRT$. The pressure is added with integrated pressure from Eq. (8) to obtain p_T in Eq. (11).

2.10 Valve lift

The fundamental the valve lift design is to satisfy an engine breathing requirement at the design speeds. However, it is philosophies and secrecies of each carmaker. One of the valve lift design uses polynomial function, for example, Hermann, McCartan and Blair (HMB) technique [12] uses up to 11th order polynomial functions. The alternative approach is the G. P. Blair (GPB) method which considers jerk characteristic of valve motion [12].

Which method is employed depends on how smooth of the lift and/or acceleration diagrams are otherwise the forces and impacts on the cam follower mechanism will be considered. In other words, a good mathematical smoothing technique within the valve lift design process is absolutely essential, may be degree by degree level. So this research assumes to use cosine function instead in order to reduce complexity as following equation:

$$L_v(\theta) = \frac{L_{iv,\max}(1+\cos\varphi)}{2} \tag{14}$$

$$\varphi = \frac{\pi(IVO - IVC + 2\theta + 540)}{IVO + IVC + 180} \tag{15}$$

Where $L_v(\theta)$ = valve lift function
$L_{iv,max}$ = maximum inlet valve lift
θ = crank angle
IVO = inlet valve open angle before TDC
IVC = inlet valve close angle after BDC

2.11 Discharge coefficient

Although the flow through valve is dynamic behavior, steady flow discharge coefficient results can be used to predict dynamic performance with reasonable precision [1]. The data are correlated into following equation.

$$C_D = 107.78\left(\frac{L_v}{D_{iv}}\right)^4 - 77.204\left(\frac{L_v}{D_{iv}}\right)^3 + 14.1\left(\frac{L_v}{D_{iv}}\right)^2 - 1.01\left(\frac{L_v}{D_{iv}}\right) + 0.6687 \tag{16}$$

Where L_v = valve lift function
D_{iv} = inlet valve diameter

2.12 Charge Heating

According to the conduction heat transfer, the heat inside the cylinder transfers to inlet manifold via connecting ports and intake valves. This heat is absorbed by air/fuel mixture directly and makes the mixture expand its volume due to increasing of temperature. Finally, the volumetric efficiency decreases automatically. Generally, the residence times, length of inlet manifold and manifold geometries are the main factors which influence how much heat can be transfer to the mixture. However, the length and geometries of manifold cannot be determined easily and the mixture velocity is dynamic behavior. So assumption in this research is the inlet temperature of the mixture must be lower than the mean cylinder wall temperature at lowest engine speed and must be greater than ambient temperature at highest engine speed. The basis is assumed to be function of engine speed linearly in order to reduce complexity as following.

$$T_{in} = -0.0177N + 413.96 \tag{17}$$

Where T_{in} = mixture temperature inside manifold

2.13 Heat input

Overall heat input (Q_{in}) can be determined by using following equation:

$$Q_{in} = \frac{HV \int_{IVO}^{IVC} m(\theta)\,d\theta}{1 + m_{air,stoich}} \tag{18}$$

Where HV = heating value of fuel
$m_{air,stoich}$ = theoretical amount of air requirement

2.14 Friction

Friction losses influence the indicated power and useful output, the brake power. Barnes-Moss [13] tested several four-stroke cycle four cylinder SI engines between 845 and 2000 cm^3 displacement at wide-open throttle. The total friction mean effective pressure (p_f) per cycle for a given engine geometry will vary with the engine speed. The data are correlated into following equation [1]:

$$p_f = 0.05\left(\frac{N}{1000}\right)^2 + 0.15\left(\frac{N}{1000}\right) + 0.97 \tag{19}$$

2.15 Assumptions and simulation conditions

The assumptions in this research are described as following. The model is derived from four-stroke, spark ignition engine without turbo or super charger and exhaust gas recirculation (EGR) system. The engine cycle is a closed system fuel-air cycle and only the compression and power strokes affect the output of the engine. The engine uses multi-port injection system for adding fuel. All the fuel in the system reacts completely with air simultaneously. Fuel-air mixture is assumed to be an ideal gas all the time. There is no any effect of combustion chamber design. The conditions for simulation are summarized in Table 1.

Table 1. Simulation conditions

Parameters	Value
Pressure inside intake manifold	90 kPa
Ambient temperature	35°C
Mean cylinder wall temperature	400 K [11, 15]
specific heat ratio (k)	1.3 [1, 3]
Air molecular weight	28.97 g/mol
Air density	1.2 kg/m^3
Gasoline molecular weight	114 g/mol
Gasoline heating value	44,000 kJ/kg
Air/fuel ratio	14.6:1
Equivalent air/fuel ratio	1

All engine geometries are obtained from Mercedes Benz model year 1969 series which all data are summarized in Appendix A. The model is developed under Matlab/Simulink and simulated between 1,000 to 6,000 rpm of engine speed range. The maximum brake torque (MBT) method is used in order to obtain maximum output.

3. Results and discussions

The results from simulations are shown in Fig. 8-15.

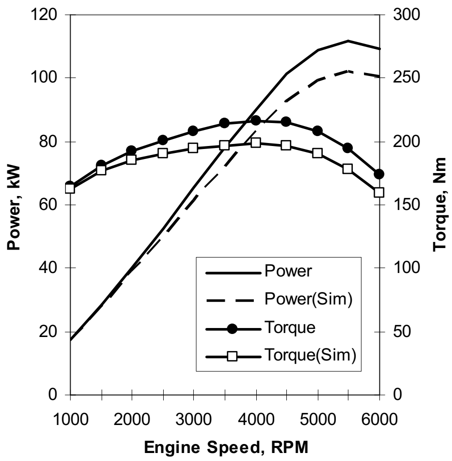


Figure 8. Mercedes Benz 250SE

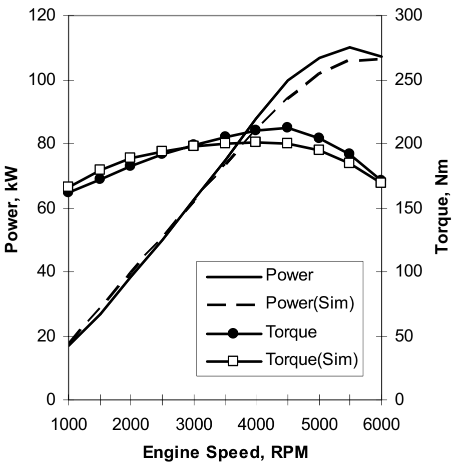


Figure 9. Mercedes Benz 250E/8

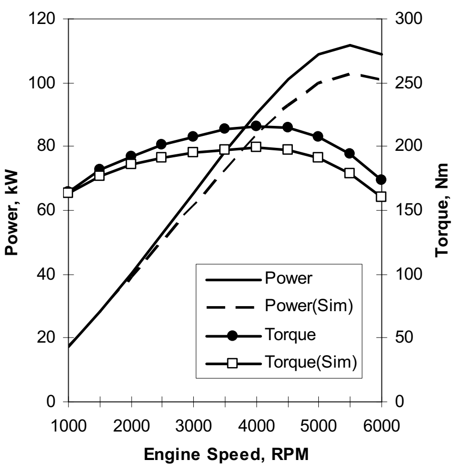


Figure 10. Mercedes Benz 250SL

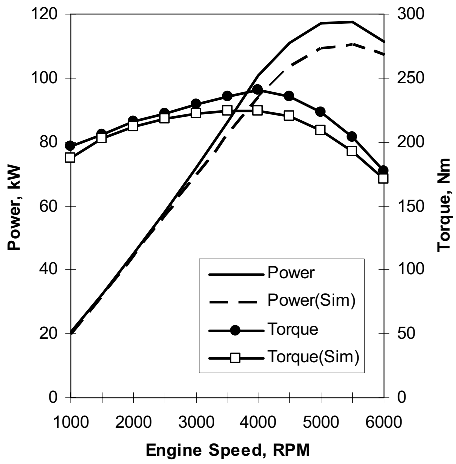


Figure 11. Mercedes Benz 280SE/8

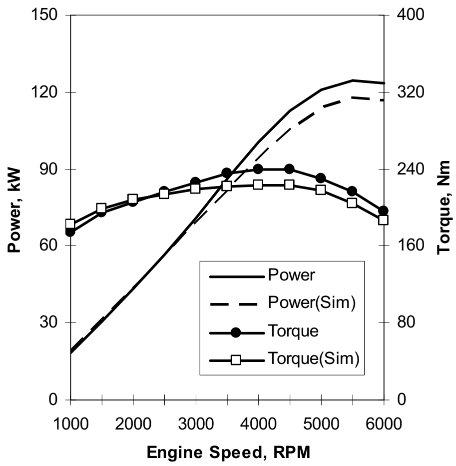


Figure 12. Mercedes Benz 280SL/8

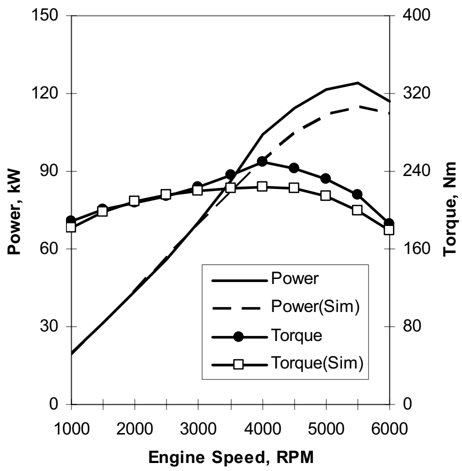


Figure 13. Mercedes Benz 300SEL/8

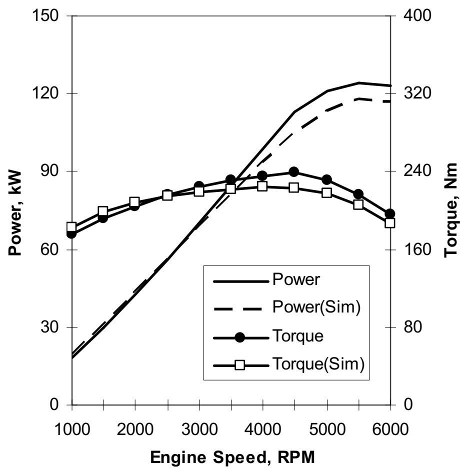


Figure 14. Mercedes Benz 300SEL

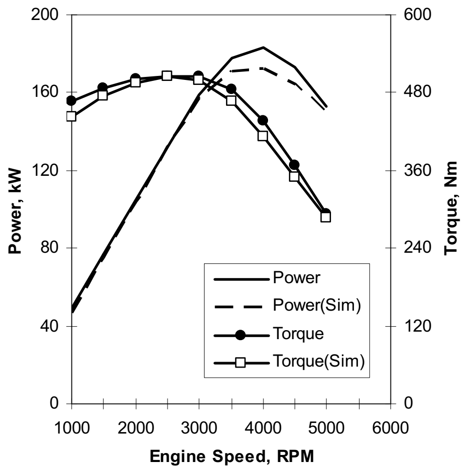


Figure 15. Mercedes Benz 600

According to the results, torque and power characteristics have same relations when comparing to the references in each graph. At low engine speed, the simulated values are almost equal or greater, while the simulated values at high speed are lower than the references in all cases. The errors come from volumetric efficiency determination of mixture charging process. The η_v indicates how much mixture is drawn into cylinder. However the mixture charging process has different characteristics depending on various factors such as residual gas, charge heating, ram effect, etc. If those factors are known, the simulated results will come closer to real characteristics as mention in Sec. 2.8. This research can determine effects of charge heating, backflow, flow friction, choking and valve timing layout. But this model still lacks tuning and ram effects which have great influences at mid and high speed. The amounts of error in all cases are summarized in Fig. 16 by using equation below.

$$\%error = \frac{Sim - Ref}{Ref} \tag{20}$$

Where Sim = simulated values
Ref = reference value from data manual

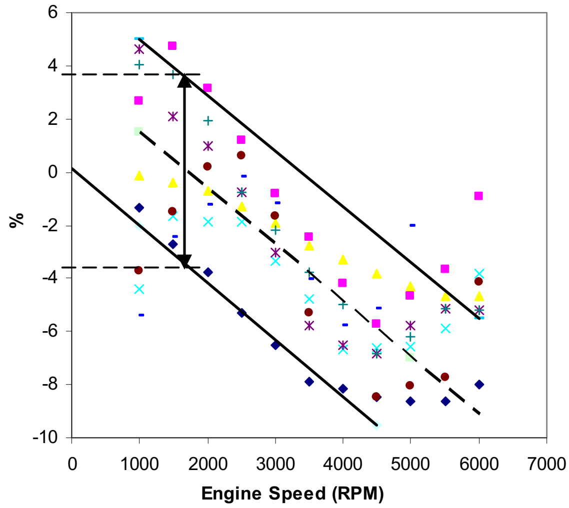


Figure 16. Relative errors

Trends of all errors are almost the same. At low speed (1,000-2,500 rpm), it may cause from inappropriate assumptions of charge heating. It should be non-linear function which is a function of residence time, length and geometries of the inlet manifold. At mid and high speed (2,500-6,000 rpm), the errors can be compensated by adding tuning and ram effects which can increase the volumetric efficiency. The overall errors in all 8 cases are from -9% to +5%. When considering at each engine speed, the errors are only about less than ±4% in all cases.

This model can be classified so called Zero-dimensional (0D) model which neglects effects of hardware dimensions. The errors of 0D model always more than Three-dimensional (3D) model, such as 3D CFD, even 0D model can predict and explain the trends of system real behaviors. However, rough data need and less time consumption are strength of 0D model. This model needs only 1-2 seconds to calculate the results for each engine speed. Guizzetti et al [16] used 3D CFD to simulate V6 3.2l engine of the Alfa Romeo 156. They spent around 5.8-7.1 days to compute 1 task. Although 0D model can roughly predict the engine performances and interpret the effects of each parameter, engine optimization still needs more details and more accuracy results from 3D model.

4. Conclusions

This model can predict the performance curves of various engine models. The volumetric efficiency has great influences on the performance curve. It is results of combining of many effects. However, this model does not include tuning and ram effects which introduce the errors at mid and high engine speed. The overall errors are in between -9% to +5%. However, the errors are less than ±4% when consider at each engine speed.

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Appendix A: Engine geometric data

<table>
<tr><th rowspan="2">Parameters</th><th colspan="8">Mercedes Benz Model Year 1969 Series</th></tr>
<tr><th>250SE</th><th>250SL</th><th>250E/8</th><th>280SE/8</th><th>280SL/8</th><th>300SEL/8</th><th>300SEL</th><th>600</th></tr>
<tr><td>No. of Cylinders</td><td colspan="7">6</td><td>8</td></tr>
<tr><td>No. of Valves per Cylinder</td><td colspan="8">2</td></tr>
<tr><td>Displacement (cc.)</td><td colspan="3">2,500</td><td colspan="3">2,800</td><td>3,000</td><td>6,300</td></tr>
<tr><td>Bore × Stroke (mm.)</td><td colspan="3">82 × 78.8</td><td colspan="3">86.5 × 78.8</td><td>85 × 88</td><td>103 × 95</td></tr>
<tr><td>Compression Ratio</td><td>9.3 : 1</td><td colspan="5">9.5 : 1</td><td>8.8 : 1</td><td>9 : 1</td></tr>
<tr><td>Intake Valve Open before TDC (degree)</td><td colspan="2">11°</td><td>16°</td><td>11°</td><td colspan="2">12°</td><td>18°</td><td>2.5°</td></tr>
<tr><td>Intake Valve Close after BDC (degree)</td><td colspan="2">53°</td><td>46°</td><td>47°</td><td colspan="2">56°</td><td>58°</td><td>52.5°</td></tr>
<tr><td>Maximum Valve Lift (mm.)</td><td colspan="2">8.5</td><td colspan="2">9.5</td><td colspan="2">10</td><td colspan="2">8</td></tr>
<tr><td>Inlet Valve Diameter (mm.)</td><td colspan="5">41.2</td><td colspan="2">49</td><td>48.95</td></tr>
</table>