CHAPTER 4

Thermodynamics Modeling of Gas Engine Performance

The objective of this chapter is to explicate a mathematical model in performance of spark ignition engine with producer gas as fuel. This chapter is divided into two main topics, zero – dimensional model and producer gas engine modeling. The first topic describes principle of thermodynamics model that is single zone cylinder model. The second topic provides modeling of producer gas engine.

4.1 Zero – Dimensional Model

For operation of SI engine with producer gas, the producer gas must be mixed with air by adjusting air gas mixer before entering the engine cylinder on intake stroke. On the compression stroke, the piston to compress gas in cylinder is moved upward. Until closing at the top dead center position, the spark plug will ignite and combustion occurs throughout the combustion chamber. The combustion in cylinder causes the piston to move downward. The crankshaft is in rotating motion, and develops power output. Using a mathematical model is another approach that can solve the problem. This reduces need for the experiment and simulation that is invisible to the naked eye. It may be used to simulate engine operation in conditions that are risk of damage. Therefore, the benefit of mathematical model is to reduce the cost and duration time in the experiment (Sekmen et al, 2007).

Generally, there are three mathematical models related to engine performance, including thermodynamics model, phenomenological model, and multidimensional model. In this research, thermodynamics model of single zone cylinder model is chosen. The zero dimensional thermodynamic model calculates using the first law of thermodynamics and mass balance. The advantage includes quick calculation, optimum conditions of performance. Examination of various engine performance parameters may be achieved (Hatte et al, 2012). For analysis of single zone cylinder model, the substance in the cylinder is assumed to be homogeneous and unsteady state. The combustion chamber is control volume. The mass and energy balance are based on the first law of thermodynamics. Figure 4.1 shows analysis of heat and work on engine cylinder. The variation of volume, heat transfer and mass loss on cylinder pressure is called heat release analysis. The basis energy conservation equation of the open system for cylinder pressure of engine (Klein, 2007) it expressed as

$$dU = dQ - dW + \sum_i h_i \, dm_i \tag{4.1}$$

where dU is the change of internal energy on mass in the system based on temperature, dQ is the change of heat release inside the engine cylinder, dW is the work generated by the system and $\sum_i h_i \, dm_i$ is the enthalpy flux passes through the boundary. The dm_i is mass flow which mean; 1) the flow in and out on both valve; 2) the flow in direct injection of fuel on cylinder; 3) the flow of piston ring blow-by; 4) the flow in and out of crevice regions, h_i is the mass specific enthalpy flow. The application of the single zone cylinder model is necessary to assume for the calculations. The common assumptions (Jagadish et al., 2011) are as follow

1. The specific heat is constant
2. The motion of the gas in the cylinder is in equilibrium state
3. The mass in cylinder is constant
4. The crank speed is uniform
5. The gas in cylinder is air, following is ideal gas law
6. The pressure and temperature in cylinder are uniform and vary with crank angle

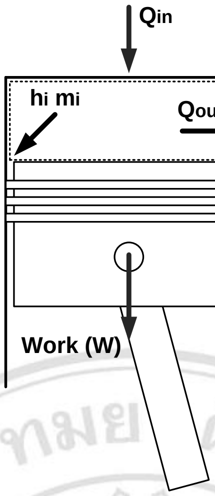

Figure 4.1 Analysis of heat and work on cylinder of internal combustion engine

4.2 Modeling Overview

Mathematical modeling of the producer gas engine used is a single zone cylinder thermodynamics model. The model combines physical based and empirical equations that can give desirable outcome (Sitthiracha et al, 2006). Figure 4.2 shows the overview model of producer gas engine. First step of calculation is input of the gas component, and engine geometric and initial conditions of the experiment. Next, heating value, gas flow rate, Wiebe function, pressure and temperature in cylinder are calculated with specific heat, specific heat ratio based on ideal gas law and first law of thermodynamic. Calculation of specific heat based on gas temperature to find new specific heat ratio is performed. The difference between initial and new specific heat ratio should be less than 0.05. The work based on crank angle up to the end of combustion is derived, then calculation of the indicated power and friction loss of the engine are carried out. Finally, brake power, torque, thermal efficiency and brake specific fuel consumption are worked out.

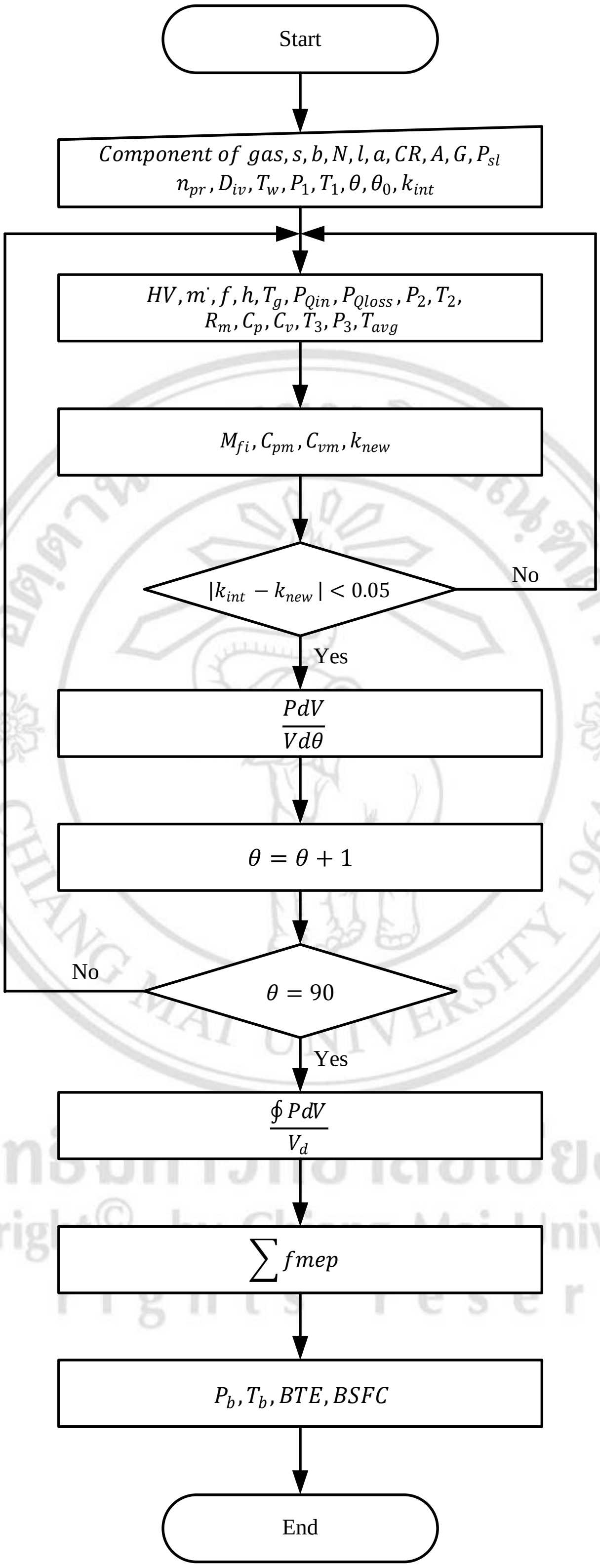

Figure 4.2 Flow chart of overview of producer gas engine modeling

4.3 Model of Producer Gas Engine

4.3.1 Cylinder pressure

Pressure in the cylinder has effect on performance of the internal combustion engine. For the producer gas engine, the pressure in cylinder can be derived from the first law of thermodynamics based on the closed-system. The cylinder pressure versus small crank angle (Ferguson et al, 1998) is shown in Eq. (4.2).

$$\frac{dP}{d\theta} = \frac{k-1}{V}\frac{dQ}{d\theta} - k\frac{PdV}{Vd\theta} \tag{4.2}$$

where, P is the pressure inside cylinder, θ is crank angle, k is specific heat ratio, Q is heat releases, V is the cylinder volume and as a function of crank angle considering basic geometry of reciprocating in Figure 4.3, the cylinder volume is given as:

$$V(\theta) = \frac{V_d}{r_c - 1} + \frac{V_d}{2}\left[\frac{l}{a} + 1 - cos\theta - \left(\left(\frac{l}{a}\right)^2 - \sin^2\theta\right)^{0.5}\right] \tag{4.3}$$

where, V_d is displacement volume, r_c is compression ratio, l is connecting rod length, a is crank radius equal of $(0.5s)$ and s is stroke. Meanwhile, surface area in cylinder at any crank angle express is

$$A(\theta) = \frac{\pi}{2}b^2 + \pi b\frac{s}{2}\left[\frac{l}{a} + 1 - cos\theta + \left(\left(\frac{l}{a}\right)^2 - \sin^2\theta\right)^{0.5}\right] \tag{4.4}$$

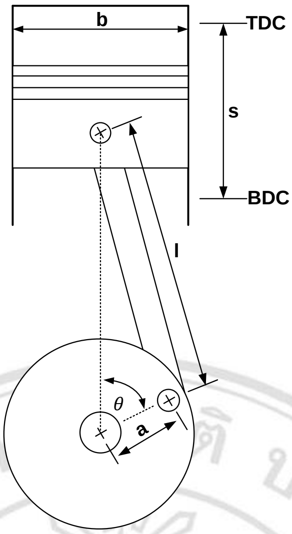

Figure 4.3 Basic geometry of reciprocating

4.3.2 Heat input

The total amount of heat input to cylinder versus changes in the crank angle (Hatte et al, 2012) is shown in Eq. (4.5).

$$\frac{\partial Q}{d\theta} = HV \int_{IVO}^{IVC} \dot{m}\, d\theta \frac{df}{d\theta} \tag{4.5}$$

where, HV is heating value, $\dot{m}$ is producer gas flow rate, IVO and IVC is inlet valve open and close before and after TDC, $f(\theta)$ is the Wiebe function. Producer gas flow rate through an intake valve was derived from basic fuel consumption with empirical equation from examination. That equation is related to engine speed and volume efficiency. The flow rate of producer gas in SI engine is given as:

$$\dot{m} = 0.00378V_d(0.105N^2 - 0.792N - 0.0015N^3) \tag{4.6}$$

where, N is engine speeds and the Wiebe function is used to determine the combustion rate of the fuel (Heywood, 1988), expressed as:

$$f(\theta) = 1 - exp\left[-5\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^3\right] \tag{4.7}$$

where, θ is crank angle, θ_0 is start of heat release angle, $\Delta\theta$ is duration of heat release and can be determined from this equation.

$$\Delta\theta = -1.618\left(\frac{N}{1000}\right)^2 + 19.866\left(\frac{N}{1000}\right) + 39.395 \tag{4.8}$$

4.3.3 Heat transfer

The heat transfer is necessary for the internal combustion engine to maintain cylinder walls, pistons and piston ring. The majority of heat transfer in the cylinder is convection heat transfer. Important parameters are heat transfer coefficient, surface area, and gas temperature in cylinder. The heat transfer in cylinder (Lata et al, 2010) can be determined as follows

$$Q_{loss} = hA\left(T_g - T_w\right) \tag{4.9}$$

where, h is heat transfer coefficient, A is surface area of combustion chamber T_g is gas temperature in cylinder, T_w is cylinder wall temperature. The heat transfer coefficient is instantaneous area average heat transfer coefficient derived from Woschni (1967), shown in Eq. (4.10).

$$h = 0.82b^{-0.2}(P10^{-3}c)\ 0.8T_g^{-0.53} \tag{4.10}$$

where, b is bore cylinder, c is equal of 6.18. The gas temperature is calculated using following equation from Sitthiracha (2006) while, engine speed in a range of 1000-6000 rpm.

$$T_g = 3.395\left(\frac{N}{1000}\right)^3 - 51.9\left(\frac{N}{1000}\right)^2 + 279.49\left(\frac{N}{1000}\right) + 676.21 \tag{4.11}$$

4.3.4 Ideal gas equation of state

The molecular mass is the mass of one mole of compound in grams per mole. For the gas mixture, molecular mass is dependent on the molar mass of gas species. The sum of molar mass gas fraction is equal to average molecular mass of gas and is given as

$$M_m = \sum_{i=1}^{n} y_i\, M_i \tag{4.12}$$

when M_i is molar mass of gas species, y_i is mole fraction of gas and n is amount of gas component. Meanwhile, the average molecular mass is related average gas constant that shown in Eq. (4.13) and (4.14).

$$R_m = \frac{R_u}{M_m} \tag{4.13}$$

$$R_m = \sum_{i=1}^{n} mf_i\, R_i \tag{4.14}$$

when R_u is universal gas constant is equal of 8.314 J/mole-K, R_i is average gas constant of gas species, mf_i is mole fraction that is defined as

$$mf_i = \frac{y_i M_i}{\sum_{i=1}^{n} y_i M_i} \tag{4.15}$$

The familiar relationships between average gas constant, specific heat for a constant volume and specific heat ratio is defined as

$$C_{vm} = \frac{R_m}{k-1} \tag{4.16}$$

The specific heat for a constant pressure obtained from the relationship of average gas constant with specific heat for a constant volume is determined in Eq. (4.17). It can be obtained from table of gas temperature with specific heat for a constant pressure in gas species shows in Appendix B that defined in Eq. (4.18).

$$C_{pm} = C_{vm} + R_m \tag{4.17}$$

$$C_{pm} = \sum_{i=1}^{n} mf_i + C_{pi} \tag{4.18}$$

In this work, the gaseous fuel is producer gas, and gas species include CO, H_2, CH_4, N_2, CO_2, O_2. Therefore, the specific heat for a constant pressure of gas species on temperature (Cengel et al, 2007) can be shown in Eqs. (4.19 - 4.24).

$$C_{p\_CO} = 0.1775\ \ln(T_{avg}) - 0.0441 \tag{4.19}$$

$$C_{p\_H_2} = 0.0000006(T_{avg})^2 + 0.003(T_{avg}) \quad (4.20)$$

$$C_{p\_CH_4} = 2.341\ \ln(T_{avg}) - 11.713 \quad (4.21)$$

$$C_{p\_O_2} = 0.1484\ \ln(T_{avg}) + 0.059 \quad (4.22)$$

$$C_{p\_CO_2} = 0.2591\ \ln(T_{avg}) - 0.5696 \quad (4.23)$$

$$C_{p\_N_2} = 0.1782\ \ln(T_{avg}) - 0.0633 \quad (4.24)$$

when T_{avg} is average gas temperature of crank angle. Finally, specific heat ratio of producer gas in cylinder determine as follow

$$k_m = \frac{C_{pm}}{C_{vm}} \quad (4.25)$$

4.3.5 The relation of temperature, pressure and volume on the Otto cycle

The Otto cycle or constant volume cycle is a four basic process, including isentropic compression, constant volume heat addition, isentropic expansion, constant volume heat rejection. The relation of temperature, pressure, volume and heat involved with mathematic model of engine can be as following Eq. (4.26).

$$Q_{in} = mC_v(T_3 - T_2) \quad (4.26)$$

when Q_{in} is heat addition on engine combustion that base on constant volume, T_3 is higher temperature of end process, T_2 is temperature before starting to burn. Determination of T_2 can be analyzed from relation of isentropic compression process as shown in Eq. (4.27). Meanwhile, the pressure is defined in Eq. (4.28) (Dogahe, 2012).

$$T_2 = T_1(r_c)^{k-1} \tag{4.27}$$

$$P_2 = P_1(r_c)^{k} \tag{4.28}$$

when T_1 and P_1 are initial temperature and pressure of ambient respectively. The average temperature between T_2 and T_3 is defined as

$$T_{avg} = \frac{T_3 + T_2}{2} \tag{4.29}$$

4.3.6 Indicated and brake mean effective pressure

The sums of pressure in cylinder are indicated mean effective pressure ($imep$) (Hatte et al, 2012). The equation is given as.

$$imep = \frac{\oint PdV}{V_d} \tag{4.30}$$

Therefore, brake mean effective pressure ($bmep$) can be calculated from Eq. (4.31) when $\sum fmep$ is the sum of friction on engine work.

$$bmep = imep - \sum_{i=1}^{n} fmep \tag{4.31}$$

4.3.7 Friction

The friction loss in an internal combustion engine can be analyzed by three components, including the mechanic friction, the pumping work and accessory work. Calculation of engine friction is based on empirical equations (Raut, 2013). The detail frictions caused by engine test are include bearing friction ($fmep_1$), piston and ring friction ($fmep_2$), wall tension ring friction ($fmep_3$), valve gear friction ($fmep_4$), pumping loss ($fmep_5$), combustion chamber and wall pumping loss ($fmep_6$). The equations of all friction loss are shown in Eqs. (4.32-4.37).

$$fmep_1 = 0.0564\left(\frac{b}{s}\right)\left(\frac{N}{1000}\right) \tag{4.32}$$

$$fmep_2 = 12.85\left(\frac{P_s}{bs}\right)\left(\frac{100S_i}{1000}\right) \tag{4.33}$$

$$fmep_3 = 10\left(\frac{0.377sn_p}{b^2}\right) \tag{4.34}$$

$$fmep_4 = 0.226\left(30 - \frac{4N}{1000}\right)\left(\frac{GD_{iv}}{b^2s}\right) \tag{4.35}$$

$$fmep_5 = 0.0275\left(\frac{N}{1000}\right)^{1.5} \tag{4.36}$$

$$fmep_6 = 0.0915\sqrt{\frac{imep}{11.45}}\left(\frac{N}{1000}\right)^{1.7} \tag{4.37}$$

where, P_s is piston skirt length, S_i is mean piston speed, n_p is number of piston ring, G is number of intake valve per cylinder, D_{iv} is Intake valve diameter.

4.3.8 Torque and brake power

The brake power and torque can be determined by following equations

$$P_b = 0.5bmepNV_d \tag{4.38}$$

$$T_b = \frac{P_b}{2\pi N} \tag{4.39}$$

4.3.9 Brake thermal efficiency and brake specific fuel consumption

The brake thermal efficiency (BTE) and brake specific fuel consumption ($BSFC$) can be determined by following equations (Semin et al 2008).

$$BTE = \frac{P_b}{\dot{m_b} HV} \tag{4.40}$$

$$BSFC = \frac{\dot{m_b}}{P_b} \tag{4.41}$$

where, $\dot{m_b}$ is biomass (charcoal) consumption

4.3.10 Simulation conditions

Model validation was carried out to verify the experimental results. A small SI engine converted from CI engine was used to operate on producer gas. The engine was a single cylinder, four strokes, 598cc with bath tub combustion chamber. The detailed specifications of small producer gas engine are shown in Table 4.1. The power output was measured by a dynamometer set and monitored by a display panel. The best experimental conditions used to develop mathematical models were on full load and 14: 1 of CR, the engine speed between 1000-2000 rpm. Producer gas was derived from longan charcoal. The composition of the gas was of CO 30.5±2%, H_2 8.5 ± 2%, CH_4, 0.35%, CO_2 4.8±1%, O_2, 6.3±0:5%, and the balance Nitrogen. The mean calorific value of the producer gas was 4.64 MJ/Nm^3. Parametric study was based on numerical solution to find the engine performance.

Table 4.1 Engine and operational specifications in simulation

Engine type	SI engine, 4 stroke, single cylinder
Fuel	Producer gas
Compression ratio	14:1
Bore × Stroke (mm)	92×90
Connecting rod length (m)	0.143
Crank radius (m)	0.0413
Clearance volume (m^3)	4.60×10^{-5}
Swept volume (m^3)	5.98×10^{-3}
Rated output (kW)	3.2 @ 1700 rpm
Ambient pressure (kPa)	0.92 kPa
Ambien temperature (K)	308
Mean wall temperature (K)	400
Air density (kg/m^3)	1.2
Air/fuel ratio	1.2:1
Equivalent air/fuel ratio	1
Duration of combustion	90°