THESIS

VOLUMETRIC EFFICIENCY MODELING OF A FOUR STROKE IC ENGINE

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ABSTRACT

VOLUMETRIC EFFICIENCY MODELING OF A FOUR STROKE IC ENGINE

Volumetric efficiency, one of the most important engine performance parameters, is influenced by several engine parameters such as valve timing, valve lift, intake and exhaust runner length, and intake and exhaust pressure. To explore how these parameters impact volumetric efficiency, a 1D unsteady thermal fluid analysis is performed to determine the instantaneous cylinder pressure and the mass flow rate through the intake and exhaust valves during the intake and exhaust processes. In addition to a MATLAB implementation, a GT-Power model is also developed for validation.

A synthetic intake pressure pulse is then added to the model to explore the engine intake tuning effect. The results predict a volumetric efficiency increase by about 9% when this pulse enters the cylinder near bottom dead center (BDC). The prediction is consistent with an acoustic model predicting that the maximum volumetric efficiency is reached when the natural frequency of the intake system equals the frequency of the intake process.

The GT-Power model is also used to validate the relationship between the engine speed and intake runner length for optimal volumetric efficiency. The results from GT-Power do not agree with the Helmholtz model very well. Finally, the effect of the intake valve timing, valve lift and intake and exhaust pressure on volumetric efficiency are also determined.

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CHAPTER 1

Introduction

1.1. Background

Internal combustion (IC) engines are widely used in vehicles for our daily transportation including cars, buses and motorcycles. They are also one of the critical power sources for shipment of goods and agricultural products. The convenient and efficient lifestyle made possible by IC engines has also resulted in a deteriorating environment due to IC engine emissions. Even though the use of clean energy for power is increasing, IC engine emissions are still a major source of air pollution and waste of energy. Therefore there is a need to improve IC engines' efficiency.

IC engines have been studied since their invention in 1876. Presently, computer aided techniques can predict engine performance to help in the design of more efficient engines. However, for researchers, there is still an important need to use basic theory to determine the interaction between the thermal fluid parameters in IC engines in order to make further progress in optimizing IC engine performance.

1.1.1. Valve Events. Valve events such as opening and closing and valve timing play a critical role in the intake and exhaust processes of IC engines. Therefore they have a major influence on the volumetric efficiency. To predict overall engine performance, valve events and timing should be modeled for different engine operating conditions. However, for conventional IC engines, the cam profile is fixed as a function of the crank angle, which results in non-flexible valve events.

As suggested by previous research on valve timing, a variable valve timing (VVT) system has great potential to enhance engine overall performance by controlling valve lift, phase and timing at any point on the engine map. One of the most important restrictions preventing adoption of VVT systems is the complexity and commercial cost. More research needs to be done to determine the effect of valve timing so to optimize the performance of a VVT mechanism.

1.1.2. Intake and Exhaust Manifold Gasdynamics.

In order to predict the effect of the engine intake and exhaust pressures, we must employ intake and exhaust manifold gasdynamics. The pressure waves created by periodic valve opening and closing can produce a positive or negative impact on engine breathing processes. We can take advantage of the pressure waves to 'tune' the intake by adjusting the engine intake runner length for a specific engine speed range. To achieve this, we need to analyze the flow behavior in the intake and exhaust runner and the engine cylinder.

The Helmholtz resonator is used for engine intake tuning, as it can predict the resonant frequency of the intake manifold and the cylinder while the intake valve is open. An acoustic model based on wave traveling and reflection in a tube is also helpful while the intake valve is closed.

1.2. Literature Review

1.2.1. Valve Operation.

A detailed assessment of intake and exhaust valve operation was presented by G. B. Parvate-Patil et al. [12]. As reported by Asmus [3], late intake valve closing helped to increase volumetric efficiency at low engine speed but penalized it at high engine speed.

From the study by Ham et al. [8] on the effect of intake valve lift on in-cylinder turbulence intensity and burn rate, the intake valve closing timing was found to be the most dominant parameter that affect engine breathing, and thus the volumetric efficiency.

Tsu [16] presented an analysis of the intake and exhaust processes with valve overlap included. He derived a general open system energy equation including inlet and exhaust processes, which connected cylinder pressure, volume and mass flow rate through intake and exhaust valves. A computer model based on his mathematical analysis was built to predict engine performance. Experimental data from 3 different engines were collected to compare with the theoretical results. Most of the predicted results agreed well with the experimental data.

A similar open system energy equation was derived by P. Ram Reddy et al. [13] by applying a control volume method to the cylinder during the intake and exhaust processes. The effects of the compression ratio, valve area and engine speed on volumetric efficiency were investigated in his mathematical model. Several conclusions were drawn that, 1) the compression ratio had a strong influence on the duration of choked flow; 2) the derived equation could predict a subsonic flow condition very well; 3) both intake and exhaust valve area affect cylinder pressure during the aspiration process.

1.2.2. Engine Intake Tuning. In 1999, Blair [4] reviewed the progress on engine intake tuning. Based on his review, engine intake tuning was not on engine laboratories research agendas until the late 1940's. Breakthroughs were facilitated by computer aided numerical analysis in the 1950's. The prediction of engine gas dynamics by computer modeling enabled people in the 1960's and 1970's to optimize engine performance. Excellent commercial software such as Wave [2] and GT-Power [1] became available in the 1980's and were used for engine design and research.

Malkhede et al. [10] employed the Helmholtz resonance theory and acoustic theory to maximize the volumetric efficiency of an IC engine through intake manifold tuning using a GT-Power model. A FFT analysis of pressure waves in the intake runner was presented. In his paper, intake system pressure waves were separated into two distinct phases, one was during the intake valve opening phase, the other was during the intake valve closing phase. In the two phases, Helmholtz resonance theory and acoustic theory were applied respectively. In the GT-Power model, it was found that by reducing intake runner length linearly from 13.7 to 3.1 times the stroke length, the volumetric efficiency would increase as the engine speed increased from 1200 rpm to 2600 rpm.

A Helmholtz theory and reflective wave theory were applied in research by Sammut et al. [14] to explore the relative contributions of intake and exhaust tuning to engine aspiration process. He showed that volumetric efficiency would be increased most by intake tuning, which provided an increase in the cylinder pressure at intake valve closing (IVC).

Ohata [11] performed several experiments on a fuel injected engine to study the effect of the inlet pressure on the volumetric efficiency. The results indicated that the volumetric efficiency was primarily determined by the inlet pressure profile during the short period before the intake valve was closed. A rectangular inlet pressure pulse was created, and the pulse effect was determined by varying the crank angle at which the pulse entered the cylinder. The volumetric efficiency had a steep increase when this pulse entered the cylinder near BDC, while there was only a small increase at other pulse crank angles.

Engelman [6] calculated the resonant frequency of the Helmholtz resonator consisting of the intake pipe and the cylinder at its mid-stroke piston position. He did the tuning experiment on a Lauson Model TLC-349 engine and concluded that, for a 4-stroke engine, the tuning effect could be achieved if the resonant frequency of the Helmholtz resonator was approximately twice the speed of the engine. This was then confirmed by experiments done by Thompson [15] on a Cummins V-6 diesel engine.

1.3. Motivation and Outline of the Thesis

From the paper review, it is suggested that both the intake and exhaust system geometry will affect the engine performance. The intake system parameters play a dominant role in influencing the engine volumetric efficiency. The review indicates that it is possible to predict the volumetric efficiency from the general open system energy equation. The Helmholtz resonance theory and a rectangular inlet pressure pulse can be used to explore the intake tuning effect.

This thesis consists of 4 parts, introduction, model theory, results, and conclusion. The governing equations for the valve mass flow rate, cylinder pressure during the breathing period, etc. are elucidated in Chapter 2. Modeling details and engine specifications are also clarified in Chapter 2. Chapter 3 shows the results of the volumetric efficiency modeling using MATLAB as well as GT-Power. Within Chapter 4, conclusions drawn from the research of this thesis and suggestions for the future work are presented.

CHAPTER 2

Engine Modeling

The MATLAB code includes 4 strokes, which are combustion, expansion, exhaust, and intake process. The heat release and heat transfer model are applied for the combustion process. The expansion process is assumed to be isentropic. An open system control volume energy equation is adopted to analyze the intake and exhaust processes.

To be consistent with the MATLAB model, intake and pipes are only included when investigating the effect of intake runner length on volumetric efficiency.

2.1. Engine Performance Parameters

There are several engine parameters that are used to evaluate engine performance, including output torque, brake efficiency, fuel consumption, indicated work, and volumetric efficiency. The most important parameter related to engine aspiration process is volumetric efficiency, as the overall engine thermal efficiency is influenced by the volumetric efficiency.

For a 4-stroke spark ignition (SI) IC engine, the volumetric efficiency is defined as the air and fuel mass inducted into the cylinder relative to the mass of a reference intake manifold condition that would occupy the displacement volume (V_d) of the cylinder [7]. To express it in Equation form:

$$E_v = \frac{M_{in}}{\rho_i V_d} \tag{1}$$

in which M_{in} is the total mass inducted into the cylinder and ρ_i is the density in the intake manifold.

In the MATLAB code, the total mass inducted into the cylinder is calculated by the integral of mass flow rate through the intake valve (dm_i) from IVO to IVC, denoted by Equation (2):

$$M_{in} = \int_{IVO}^{IVC} dm_i \tag{2}$$

Another equation proposed by Livengood et al. [9] relates intake and exhaust pressure ratio with volumetric efficiency as presented in Equation (3). P_e and P_i are intake and exhaust pressures respectively. γ is the ratio of specific heat for gas of interest. cr is the compression ratio of the cylinder. This Equation relies on the ideal cycle assumption: valve events occur at cylinder dead center; cylinder pressure is equal to exhaust pressure and intake pressure during the entire exhaust and intake processes, respectively. The effect of valve overlap is not accounted in this Equation.

$$E_v = 1 + \frac{1 - \frac{P_e}{P_i}}{\gamma(cr - 1)} \tag{3}$$

We included Equation (3) to explore the effect of the intake and exhaust pressures on the volumetric efficiency and compare that with the result from Equation (1).

From Equation (1) and (2), we can see the volumetric efficiency is a function of valve events and timing. So it is important to study the valve events and timing in order to investigate the volumetric efficiency.

2.2. Valve Specification

Intake and exhaust valve lift, valve timing and valve diameter all impact the volumetric efficiency. Within this thesis, we'll focus on the intake system. In our model, the effects of the intake valve timing and valve lift on the volumetric efficiency are explored. The goal of this section is to explain how these two parameters are set up in our model. The theory of the valve flow model will also be demonstrated.

2.2.1. Valve Timing.

The timing for valve motion is denoted using crank angle degree (CAD). There are four valve timings consisting of intake valve opening (IVO), exhaust valve opening (EVO), intake valve closing (IVC) and exhaust valve closing (EVC). The intake valve typically opens before top dead center (TDC), and close after bottom dead center(BDC). The intake valve events put forth the intake stroke. For exhaust valve, it'll open typically before BDC, and close after TDC. The exhaust stroke occurs between the exhaust valve timings. Typical valve timings of an 4-stroke engine are illustrated in Figure 2.1

The four strokes are denoted by the solid spiral line in Figure 2.1. The simulation starts from the compression period and is followed by expansion, exhaust and intake processes sequentially. The crank angle for starting the simulation is a negative value (CAD of IVC - 720 CAD). The CAD of the end of the compression period is zero. Note that the CAD of IVC is larger than 540 CAD.

Eight different intake and exhaust arrangements were introduced with plenty of details in reference [12]. In this paper, the potential of adjusting valve timing to increase volumetric efficiency were presented. In the literature review, it shows that intake valve timing is the most significant parameter affecting low speed and high speed volumetric efficiency.

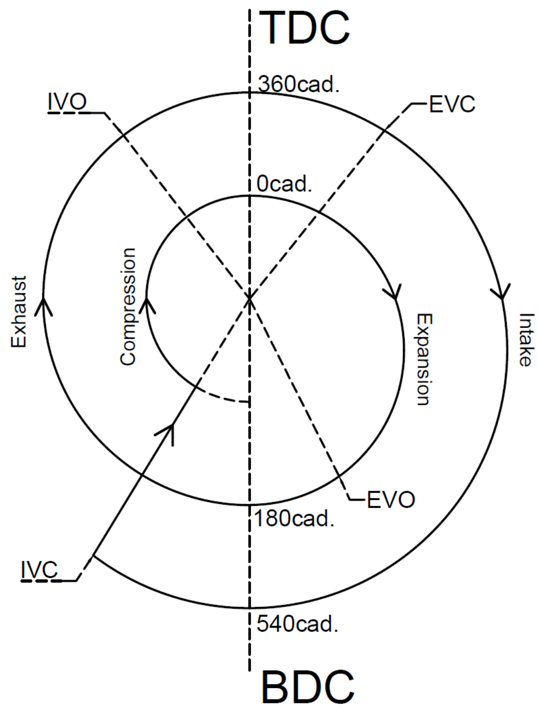

FIGURE 2.1. Typical intake and exhaust valve timing

Here are eight valve timing events illustrated in reference [12]:

a). Early intake valve opening (EIVO)

b). Late intake valve opening (LIVO)

c). Early intake valve closing (EIVC)

d). Late intake valve closing (LIVC)

e). Early exhaust valve opening (EEVO)

f). Late exhaust valve opening (LEVO)

g). Early exhaust valve closing (EEVC)

h). Late exhaust valve closing (LEVC)

For this thesis, three groups of intake valve timing are simulated in MATLAB as well as in GT-Power. The detailed parameter values are presented with results in Chapter 4.

2.2.2. Valve Lift. The valve lift is simplified using a sine function in Equation (4), where θ_{VO} is the CAD when valve opens and θ_d is the crank angle degree duration from valve opening to valve closing. The plot of the default valve lift and timing of the model is shown in Figure 2.2

$$L(\theta) = L\sin(180\frac{\theta - \theta_{VO}}{\theta_d}) \tag{4}$$

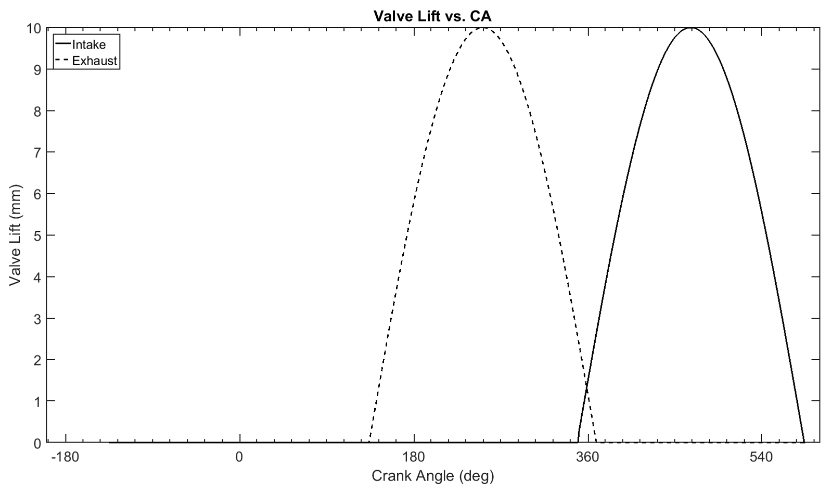

Figure 2.2. Default intake and exhaust valve timing and lift in MATLAB

Three pairs of valve lift are tested, as shown in Table 2.1.

Table 2.1. Intake and exhaust valve lift for three cases

Case Number	1	2	3
Intake Valve Lift (mm)	8	10	12
Exhaust Valve Lift (mm)	10	10	10

2.2.3. Valve Flow. The inlet and exhaust processes are modeled as a 1D unsteady mass flow through a nozzle [16]. The mass flow rate through the valve is given by Equation (5), where the subscript u and d represent upstream and downstream conditions respectively. P

refers to pressure, ρ is density. c stands for the speed of sound and A_f is the effective valve area.

$$\frac{dM}{dt} = \rho_u A_f c_u \sqrt{\frac{2}{\gamma - 1}[(\frac{P_d}{P_u})^{\frac{2}{\gamma}} - (\frac{P_d}{P_u})^{\frac{\gamma+1}{\gamma}}]} \tag{5}$$

For inward flow into the cylinder, the upstream condition refers to the condition in the intake or exhaust port, and the downstream condition refers to the condition in the cylinder. These two conditions switch roles when it comes to outward flow out of the cylinder. In the model, we use inlet pressure as the pressure in the intake port and exhaust pressure as the pressure in the exhaust port. The cylinder condition is assumed to be homogeneous.

2.2.3.1. *Choked Flow.* When the difference between upstream and downstream pressure is too large, the flow will get choked at the sonic velocity. The condition for choked flow is

$$\frac{P_u}{P_d} = (\frac{\gamma + 1}{2})^{\frac{\gamma}{\gamma - 1}}$$

The mass flow rate for choked flow is given by [7]

$$\frac{dM}{dt} = \rho_u A_f c_u (\frac{2}{\gamma + 1})^{\frac{\gamma+1}{2(\gamma-1)}}$$

2.2.3.2. *Discharge Coefficient.* Mass flow rate through valves highly depends on the effective valve area. The effective valve area is calculated by the product of discharge coefficient or flow coefficient with reference valve area [1]. If discharge coefficient C_d is adopted, the reference valve area will be the curtain area. If flow coefficient C_f is employed, the reference valve area will be the seat area. A detailed experimental methodology and numerical simulation of the intake discharge coefficient are presented in reference [5].

With the seat diameter denoted by d and valve lift by l, the curtain area is

$$A_c = \pi d l$$

in which the speed of sound is calculated by

$$c = \sqrt{\gamma R T}$$

The effective valve area is indicated by

$$A_f = C_d A_v \tag{6}$$

The seat area is calculated by Equation (7).

$$A_s = \frac{\pi}{4} d^2 \tag{7}$$

The corresponding effective valve area will be

$$A_f = C_f \frac{\pi}{4} d^2 \tag{8}$$

$$C_f = -0.001 + 3.5477 \frac{l}{d} - 6.566 (\frac{l}{d})^2 \tag{9}$$

In our simulation, we employed the default flow coefficient in GT-Power and applied that to our MATLAB model. The default flow arrays from GT-Power are plotted in Figure 2.3.

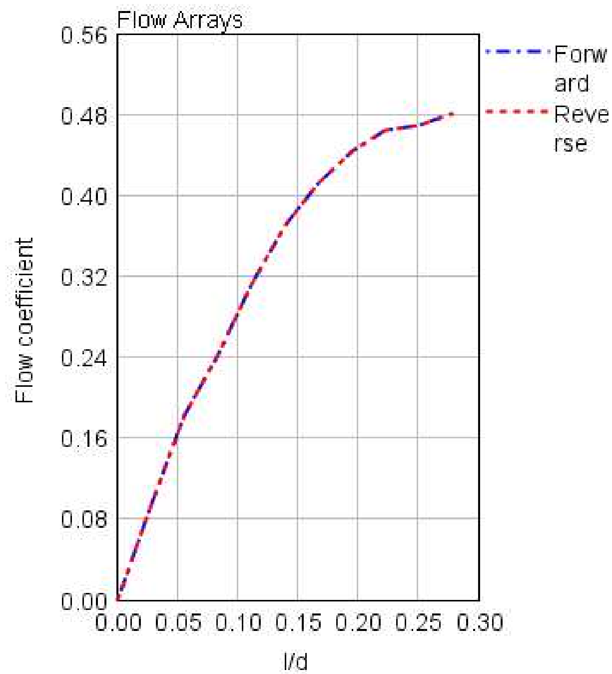

FIGURE 2.3. Discharge coefficient vs. l/d

A fitted polynomial in Equation (9) is calculated from the default flow array data from GT-Power [1].

2.3. Open System Energy Equation

During intake and exhaust processes, with mass flowing into and out of the cylinder, the properties in the cylinder were determined using an open system energy equation [13]. Figure 2.4 is the diagram of an open system for the engine cylinder in two states from time t to $t + dt$.

For inward flow, from t to $t + dt$, the energy Equation is:

$$dE = dH - PdV + dQ \tag{10}$$

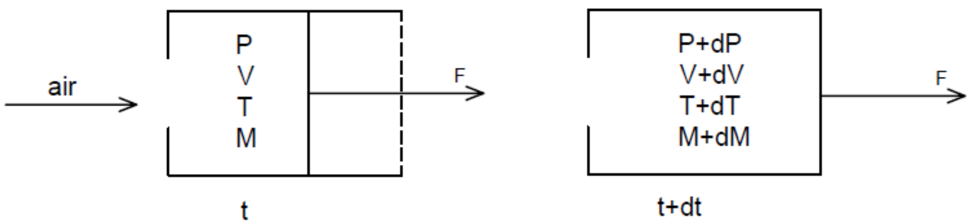

FIGURE 2.4. Open system analysis for engine cylinder

The term dE is the internal energy difference of the cylinder contents between two states, with

$$dE = C_v d(MT) \tag{11}$$

dH is the energy of the mass which flows into the cylinder

$$dH = C_p T_a dM \tag{12}$$

The subscript a represents air. T is the temperature. C_p and C_v are the specific heats for gas in constant pressure and constant volume processes respectively. dQ refers to heat transferred into or out of the cylinder during intake process, which was assumed small in our analysis.

We use the ideal gas law, which relates the temperature, mass, volume with the pressure, to replace the cylinder mass and temperature variation with the pressure and volume variation. We have Equation (13), where R_s is the mass specific gas constant of the cylinder contents.

$$d(MT) = d(\frac{PV}{R_s}) \tag{13}$$

$$C_v d(\frac{PV}{R_s}) = C_p T_a dM - PdV \tag{14}$$

Combining Equation (11), (12) and (13) with Equation (10), results in Equation!(14).

Knowing that $C_p - C_v = R$ and $C_p/C_v = \gamma$, we finally get

$$VdP + \gamma PdV = R_s \gamma T_a dM \tag{15}$$

If we do the same analysis for outward flow, the result is

$$VdP + \gamma PdV = R_s \gamma T_c dM \tag{16}$$

Note that dM is a negative value for outward flow. For the outward flow Equation the upstream and downstream condition will switch roles. Thus the properties of the mass flowing out of the cylinder are the cylinder content's properties. We assume the same specific gas constant and specific heat ratio for the gas in the cylinder, intake manifold, and exhaust manifold. During the intake and exhaust processes, there can be inward or outward flow depending on the pressure differences.

From cylinder geometry, the volume change is

$$\frac{dV}{d\theta} = \frac{V_d}{2} \frac{\sin\theta(1+\sin\theta)}{\sqrt{(\frac{L_r}{a})^2 - \sin^2\theta}} \frac{\pi}{180} \tag{17}$$

The term θ is the crank angle degree, L_r is the connecting rod length and a is the crank throw radius of the engine. Multiplying the left hand side of the mass flow Equation (5) by $dt/d\theta$ and the right hand side by $1/\omega$ (engine angular velocity), we get

$$\frac{dM}{d\theta} = \frac{\rho_u A_f c_u}{\omega} \sqrt{\frac{2}{\gamma - 1}[(\frac{P_d}{P_u})^{\frac{2}{\gamma}} - (\frac{P_d}{P_u})^{\frac{\gamma+1}{\gamma}}]} \tag{18}$$

Doing the same manipulation for choked flow, results in

$$\frac{dM}{d\theta} = \frac{\rho_u A_f c_u}{\omega} (\frac{2}{\gamma + 1})^{\frac{\gamma+1}{2(\gamma-1)}} \tag{19}$$

There are four cases for the engine aspiration processes: normal inward flow, choked inward flow, normal outward flow and choked outward flow. They all can occur in the three periods of the aspiration processes, which are the intake, exhaust and overlap periods. By including both inward and outward flow, the general energy Equation (20) turns up:

$$\frac{dP}{d\theta} = -\frac{R_s \gamma P}{V}\frac{dV}{d\theta} + \frac{R_s \gamma C_1}{V}\frac{dM_i}{d\theta} + \frac{R_s \gamma C_2}{V}\frac{dM_e}{d\theta} \tag{20}$$

In which, the subscript i refers the intake valve and e refers to the exhaust valve. In different periods, there are different constants C_1 and C_2 as in Table 2.2. The subscript c refers to the cylinder condition.

TABLE 2.2. Constants for general equation in different periods

Periods	Exhaust Only	Overlap	Intake Only
Inward Flow	$C_1 = 0$ $C_2 = T_e$	$C_1 = T_i$ $C_2 = T_e$	$C_1 = T_i$ $C_2 = 0$
Outward Flow	$C_1 = 0$ $C_2 = T_c$	$C_2 = T_c$ $C_2 = T_c$	$C_1 = T_c$ $C_2 = 0$

For clarification, during the overlap period, if the intake and exhaust pressure are not the same, we may have inward flow and outward flow at the same time. The corresponding C_1 and C_2 will be decided based on the relative pressure differences.

By replacing $dV/d\theta$ and $dM/d\theta$ with Equation (17) and (18) or (19) in our general Equation, we can solve for $dP/d\theta$ during the aspiration processes, and get the curve of cylinder pressure versus crank angle.

2.4. Helmholtz Resonator

The Helmholtz Resonator (HR) was introduced in 1850s by Hermann Von Helmholtz in his book 'On the Sensation of Tone'. A typical HR is presented in Figure 2.5. The air in the neck acts as an air plug and the air in the cavity acts like a spring. When the "air plug" is triggered to push or pull the "spring", an oscillation of this system will occur. A resonant frequency can be calculated from this analogy.

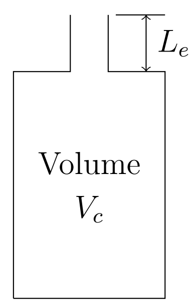

Figure 2.5. A typical Helmholtz Resonator

The resonance frequency [17] of a typical Helmholtz resonator is given by

$$f = \frac{c}{2\pi}\sqrt{\frac{A_{pipe}}{L_e V_c}} \tag{21}$$

where c is the speed of sound, A_{pipe} is the effective cross sectional area of the neck and L_e is the effective length of the neck.

The above frequency is derived for a typical HR with constant cavity volume V_c. For an engine cylinder, with piston motion, the cavity volume is varying. Thus the Equation must be modified for the variable volume condition.

From research by Engleman [6], "The amount of the change of resonant frequency depends more upon the amount of modulation than upon the modulating frequency". He proposed the natural frequency at mid-stroke piston position as mean static natural frequency to represent the natural frequency of the intake system.

$$f = \frac{c}{2\pi}\sqrt{\frac{A_{pipe}}{L_e V_{midstroke}}} \tag{22}$$

With the cylinder volume at mid-stroke as

$$V_{midstroke} = \frac{V_d}{2} + V_{cl} = \frac{V_d}{2}\frac{cr+1}{cr-1}$$

We get

$$f = \frac{\sqrt{2}c}{2\pi}\sqrt{\frac{A_{pipe}}{L_e V_d}}\sqrt{\frac{cr-1}{cr+1}} \tag{23}$$

The assumptions used in this analysis are the same as in HR: 1) friction and heat transfer are negligible; 2) the gas in the cylinder acts as a linear spring; 3) the gas in the intake runner is incompressible. One resonant oscillation during the intake period is expected [6] [15], as concluded by Engleman and Thompson. More oscillations than that will impair the tuning effect. This conclusion leads us to the Equation that the natural frequency for the intake system should be twice the frequency of the engine piston motion. (The period of the HR equals the period of the intake process). Frequency of engine piston motion is $f_p = \frac{N}{60}$.Thus we have $f = 2f_p$.

Then we obtain an Equation relating engine speed and intake runner length.

$$N = \frac{15\sqrt{2}c}{\pi}\sqrt{\frac{A_{pipe}}{L_e V_d}}\sqrt{\frac{cr-1}{cr+1}} \tag{24}$$

From Equation (24), with fixed pipe diameter and engine specifications, we can determine the optimum engine speed in terms of the pipe length or vice versa. However, HR is only applicable when cylinder and intake pipe size are less than the sound wave length.

2.5. Effect of Inlet Pressure Pulse

To investigate the effect of an inlet pressure pulse on volumetric efficiency, a synthetic square pressure pulse with 20 CAD width and 15 kPa intensity pressure was created. Figure 3.27 is the graphical representation for an inlet pressure pulse.

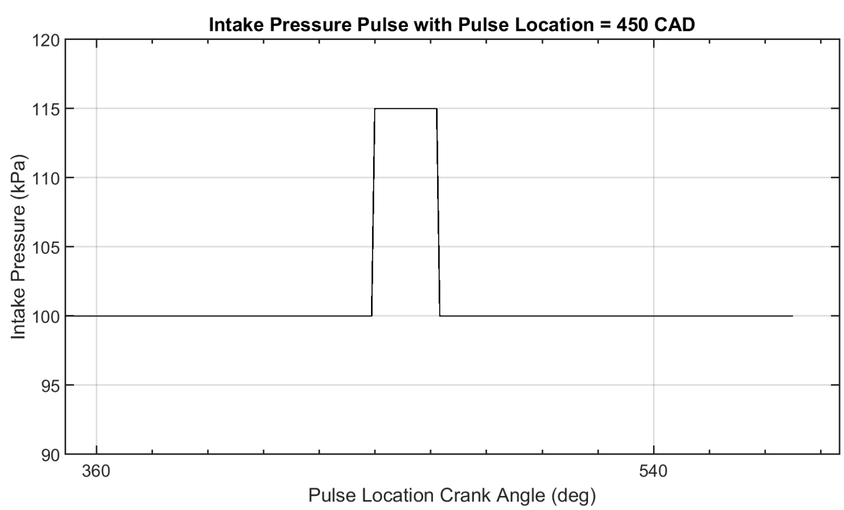

FIGURE 2.6. A pressure pulse example, pulse location=450 CAD

The inlet pressure pulse was used to simulate the pressure pulse caused by inlet valve opening. A rarefraction wave will be produced when the intake valve opens, with the rarefraction wave traveling away from the cylinder and as it encounters the intake manifold inlet, it changes its sign and returns as a compression wave. Depending on the engine and intake system geometries, it can reach the intake valve at different crank angles, which will result in different gains in the volumetric efficiency.

In addition to the pressure pulse created by intake valve opening, intake valve closing will also produce a pressure pulse (IVC pulse). IVC pulse is a compression pressure wave. Similar to IVO pulse, it'll change its sign and return as a rarefraction pressure wave. However the effect on volumetric efficiency is relatively small. Therefore, in the thesis, we'll investigate the effect of the IVO pressure wave on volumetric efficiency and compare to the Helmholtz resonator analysis.

2.6. Heat Release and Expansion

The combustion and expansion model are also included in the MATLAB code. The Weibe function is included for heat release [7]

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

where θ_s is the crank angle for start of heat release, θ_d is the duration of heat release. n and a are Weibe form factor and effeciency factor respectively.

A Woschnif function is applied to simulate the heat transfer during combustion process. The expansion process is assumed to be an insentropic process.

2.7. Engine Specification

Table 2.3. Engine Geometries

Engine Geometries	Value
Bore (mm)	100
Stroke (mm)	100
Connection Rod Length (mm)	250
Compression Ratio	10

Table 2.4. Default Engine Parameters

Engine Parameters	Value
Intake Valve Open (crank angle degree)	10 btdc
Intake Valve Close (crank angle degree)	45 abdc
Exhaust Valve Open (crank angle degree)	45 bbdc
Exhaust Valve Close (crank angle degree)	10 atbc
Intake Valve Maximum Lift (mm)	10
Exhaust Valve Maximum Lift (mm)	10
Intake Valve Diameter (mm)	50
Exhaust Valve Diameter (mm)	40
Start of heat release (crank angle degree)	35 btdc
Heat release duration (crank angle degree)	60

Table 2.3 lists the geometric specifications of the general engine for the simulations. Table 2.4 are the default parameter values used in the MATLAB and GT-Power models. The properties of the mass flowing into and out of the cylinder during the intake and exhaust processes are assumed to be the properties of air.

2.8. MATLAB and GT-Power Valve Calibration

Figure 2.7 is a schematic of the single cylinder GT-Power model. In GT-Power, intake and exhaust ports must be included. However there were no intake and exhaust ports model in the MATLAB simulation. In order to minimize the discrepancies between the two models,

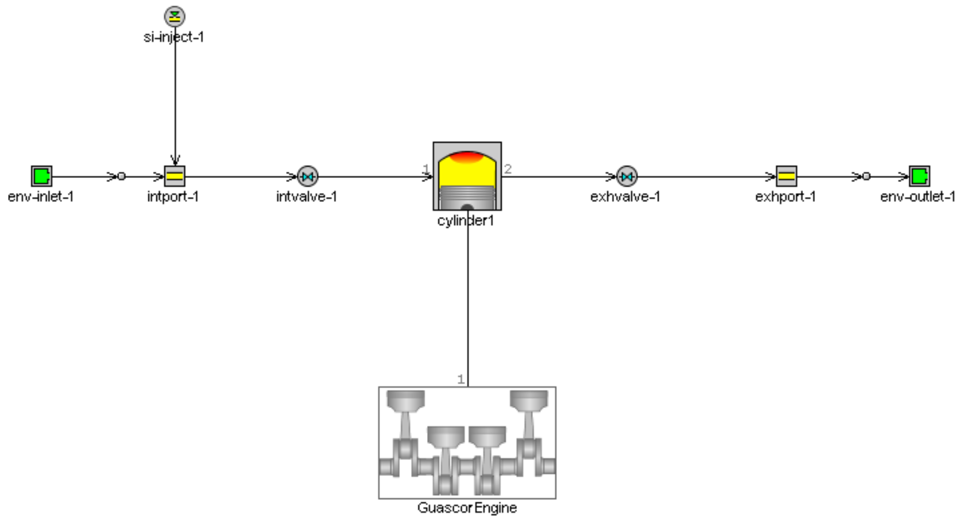

FIGURE 2.7. GT-Power model without intake and exhaust runner

the intake and exhaust ports in the GT-Power model were set to be small, just 2 mm of length.

As the valve lift in the MATLAB simulation is a simple sine function, a calibration between the MATLAB and GT-Power valve timing and lift setup is performed by changing the angle multiplier in the GT-Power valve module. The valve lifts after calibration are plotted in Figure 2.8. In the simplified MATLAB model, valve lift is abrupt as there are no ramp periods at the beginning of intake stroke and at the end before valve closing as in the GT-Power valve module.

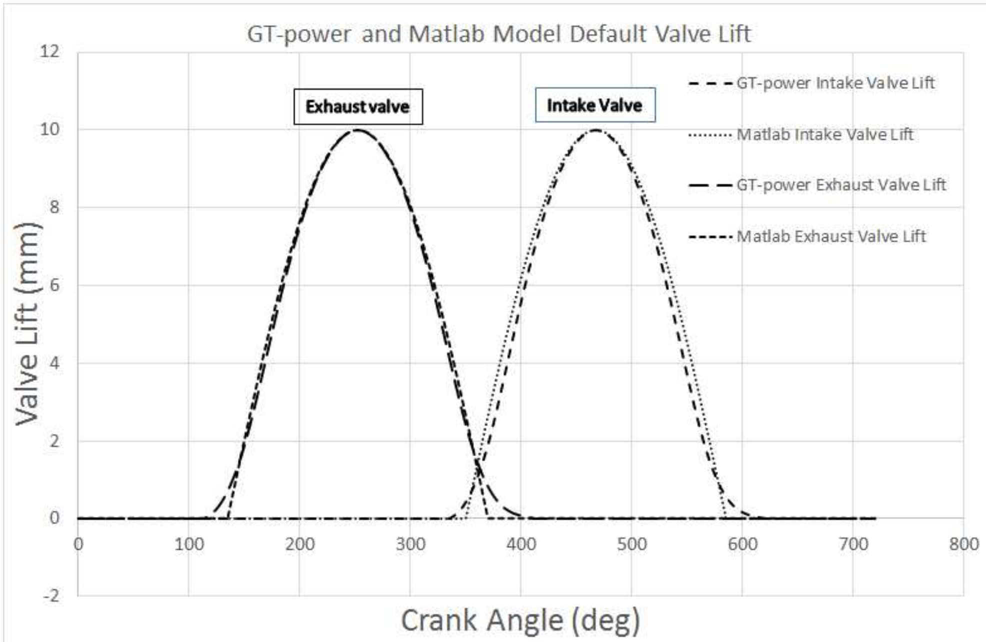

FIGURE 2.8. GT-Power and MATLAB default valve lift

CHAPTER 3

Results

The purpose of this chapter is to show how the intake system parameters affect the intake pressure drop, mass flow rate, and volumetric efficiency. Results from the MATLAB and GT-Power calculations are illustrated in this chapter. Most of the results are given as plots for the volumetric efficiency as a function of the engine speed or other intake system parameters.

3.1. Effect of Engine Speed on Volumetric Efficiency

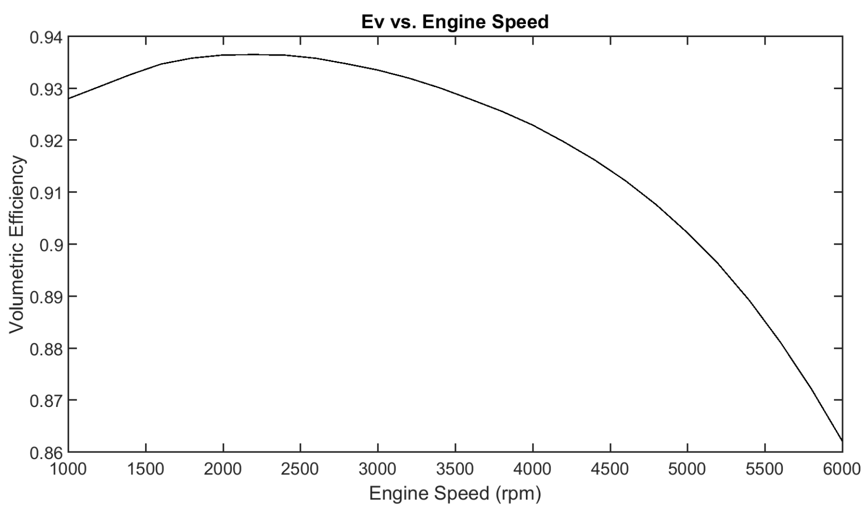

Figure 3.1. Volumetric efficiency vs. engine speed (MATLAB)

Figure 3.1 and 3.2 are the plots of the volumetric efficiency by varying engine speed from 1000 rpm to 6000 rpm. For the baseline engine, the volumetric efficiency increases and then decreases monotonically over the 1000 rpm to 6000 rpm engine speed range. Note that a maximum volumetric efficiency for the simple model is around 2500 rpm, and about 4000 rpm for the more complex GT-Power model.

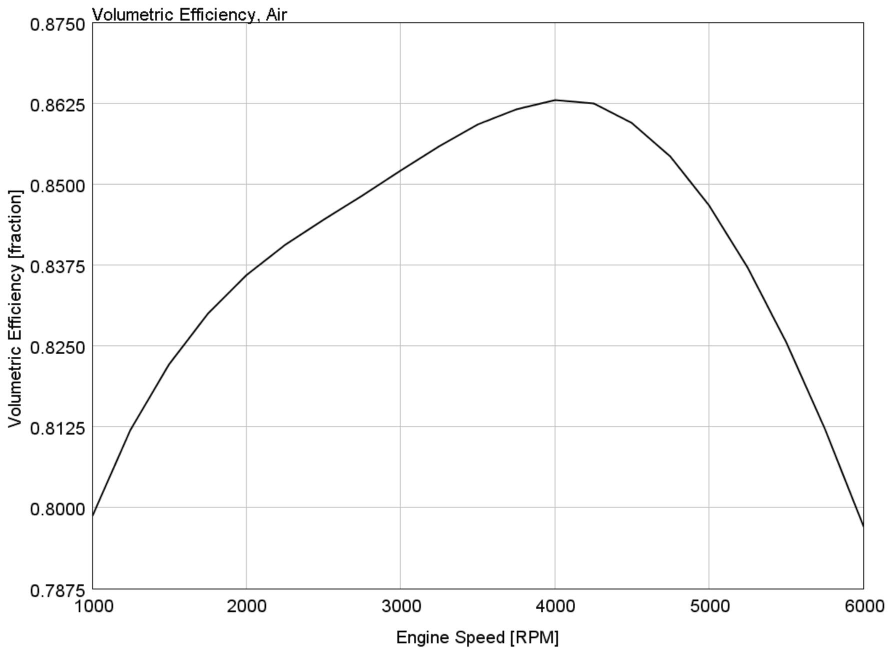

FIGURE 3.2. Volumetric efficiency vs. engine speed (GT-Power)

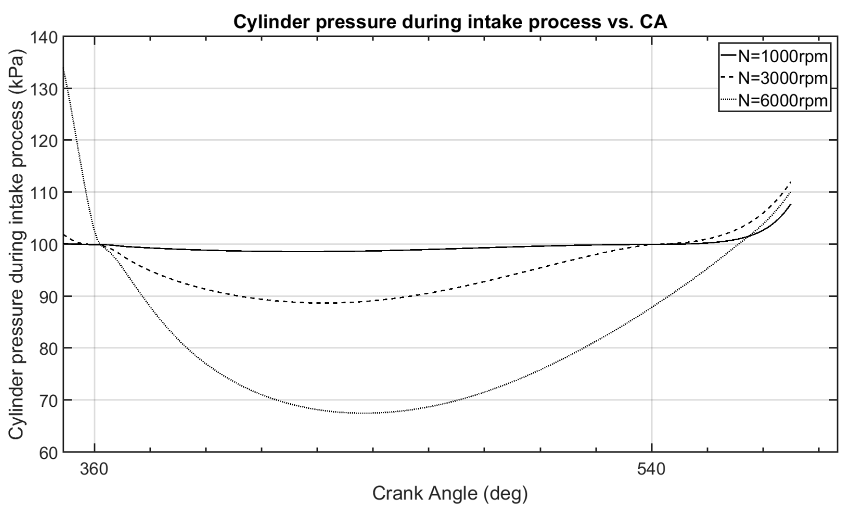

FIGURE 3.3. Cylinder pressure during intake process vs. CAD (MATLAB)

Figure 3.3 presents the cylinder pressure versus crank angle during the intake process from the TDC (360^o) to BDC (540^o) for different engine speeds. A higher engine speed can result in a larger pressure drop across the intake valve during the intake process. This can

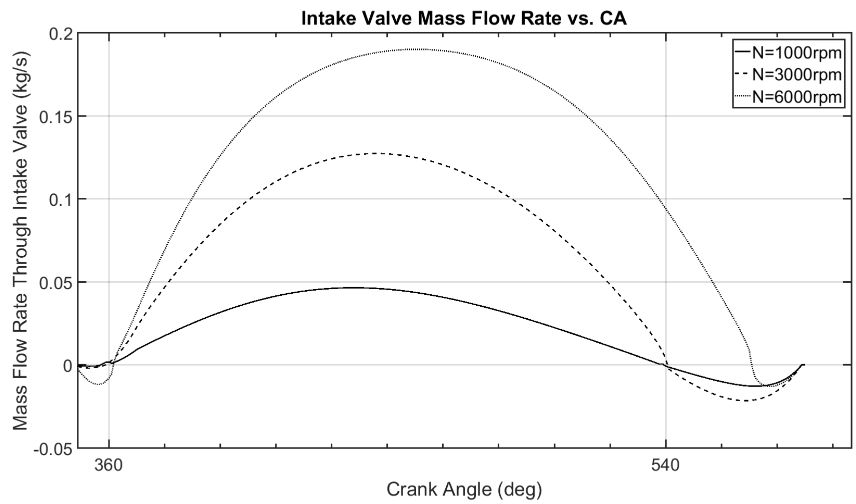

FIGURE 3.4. Mass flow rate through intake valve (kg/s) vs. CAD (MATLAB)

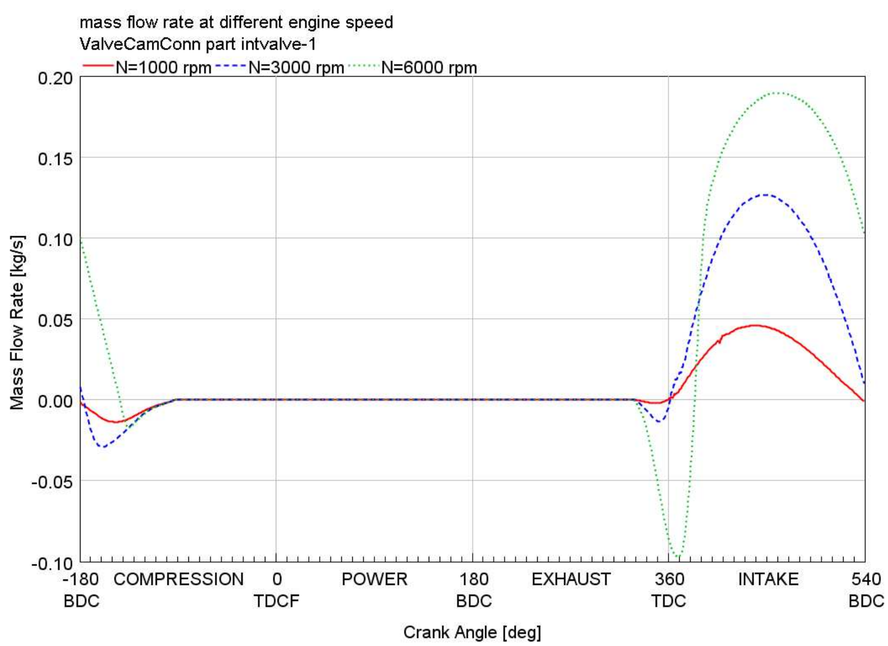

FIGURE 3.5. Mass flow rate through intake valve (kg/s) vs. CAD (GT-Power)

result in a higher inlet mass flow rate. Also note that at BDC (540^o) at 6000 rpm, there still is a pressure drop across the inlet valve.

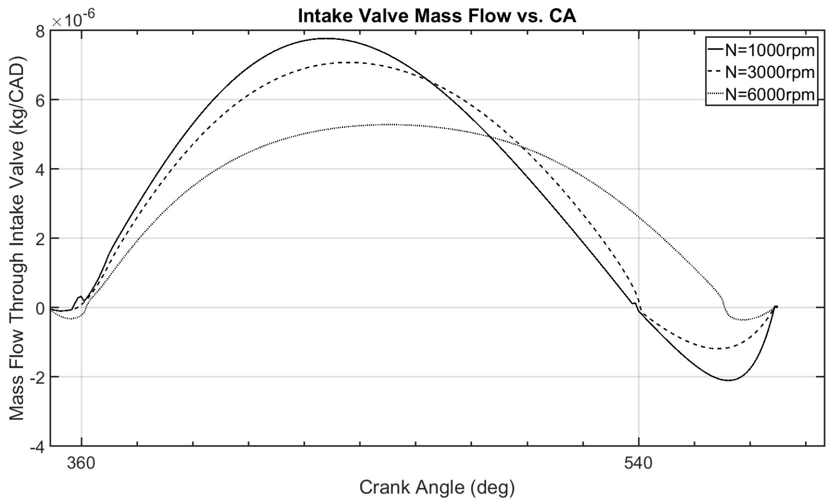

FIGURE 3.6. Mass flow rate through intake valve (kg/CAD) vs. CAD (MATLAB)

Figure 3.4 and 3.5 are plots of the mass flow rate through the intake valve for different engine speeds. We can see from the mass flow rate plots that a higher engine speed will result in a higher intake valve mass flow rate. Note the back flow before TDC due to the greater cylinder pressure. However, if we convert to mass flow per degree instead of per second by multiplying by $1/\omega$, we'll get Figure 3.6 which shows a lower mass flow per crank angle degree resulting from the higher engine speed.

As we are using the integral of mass flow rate over the entire intake valve opening duration to calculate the volumetric efficiency, Figure 3.6 clearly shows why the volumetric efficiency decreases at higher engine speed.

3.2. Effect of Intake Valve Lift on Volumetric Efficiency

Effect of intake valve lift on volumetric efficiency are explored in this section. Volumetric efficiency vs. engine speed with different valve lifts are plotted in Figure 3.7 and Figure 3.8. The three different maximum intake valve lift cases are 8 mm, 10 mm and 12 mm.

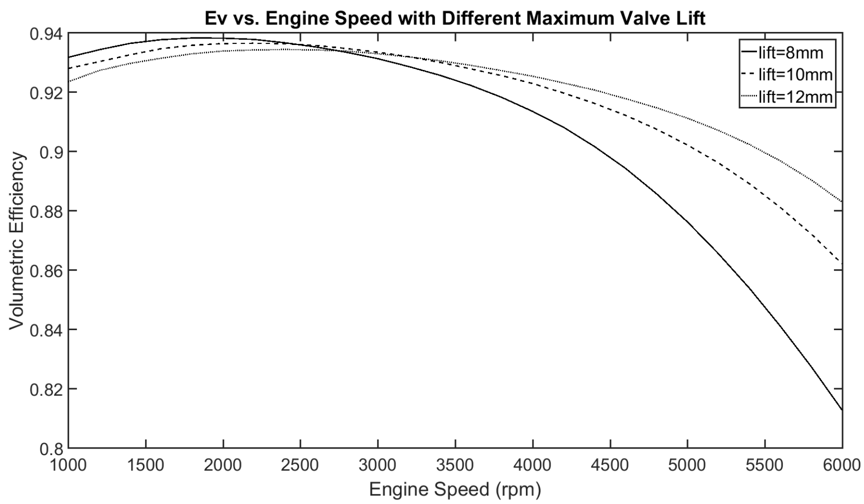

FIGURE 3.7. Volumetric efficiency vs. engine speed with different valve lift (MATLAB)

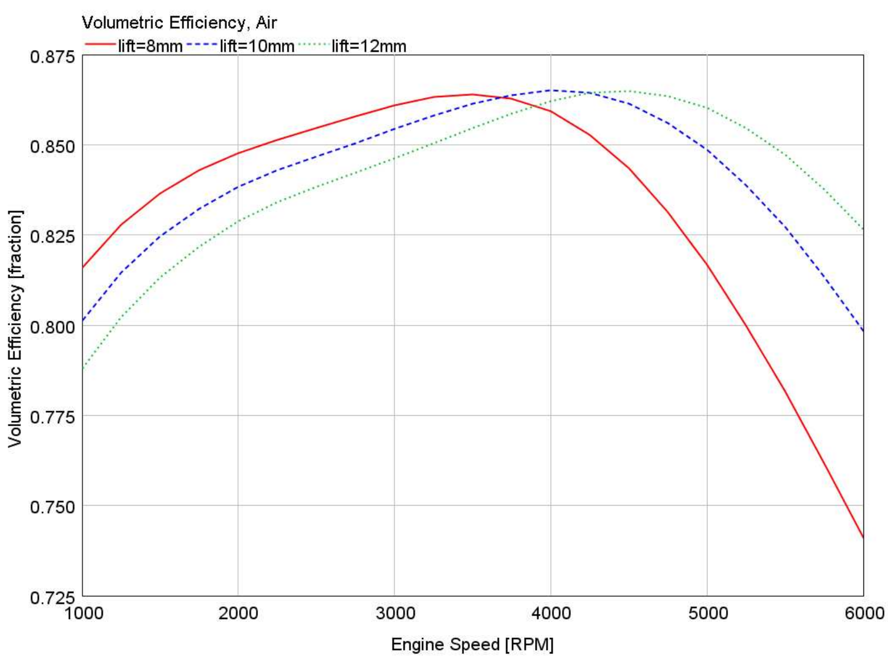

FIGURE 3.8. Volumetric efficiency vs. engine speed with different valve lift (GT-Power)

All three curves have a peak volumetric efficiency at a different engine speed. From the MATLAB model, with the maximum valve lift increasing from 8 mm to 12 mm, the engine

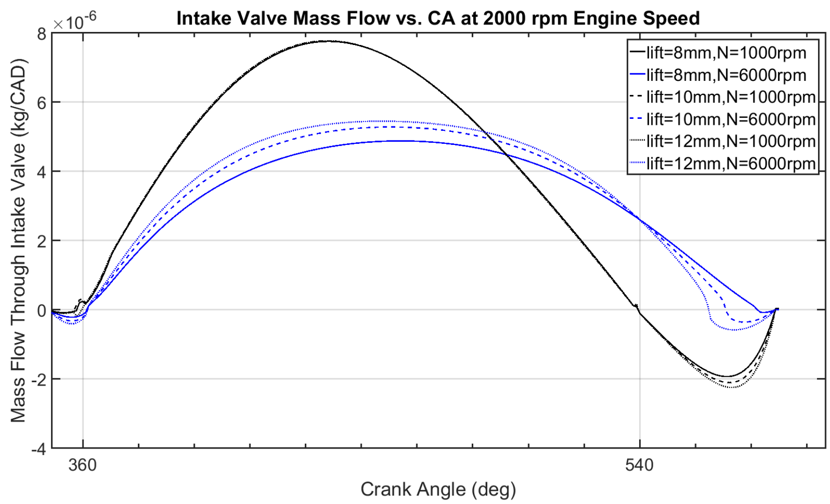

FIGURE 3.9. Mass flow rate through intake valve with different valve lift (MATLAB)

speed for peak volumetric efficiency increases while the peak volumetric efficiency itself is slightly decreased. Similarly with the GT-Power model, the peak volumetric efficiency occurs at high engine speed as the valve lift increases while the volumetric efficiency itself increases, which is opposite to that of the MATLAB model. The volumetric efficiency curves are shifted from left to right with higher maximum valve lift. A lower valve lift will result in a better volumetric efficiency at low engine speed, and higher valve lift has a greater volumetric efficiency at high engine speed.

To further examine the reason for the decrease in the volumetric efficiency at high engine speeds, the mass flow per degree through the intake valves for three different valve lifts at two different engine speeds are plotted in Figure3.9. At the high engine speed, the mass flow per degree greatly decreases relative to the low engine speed.

At the high engine speed, a low valve lift result in a smaller mass flow per crank angle during most of the open duration of the intake stroke. Since volumetric efficiency is calculated based on the integral of mass flow into the cylinder through the intake valve, a smaller mass flow per crank angle will result in a lower volumetric efficiency.

3.3. Effect of Intake Valve Timing on Volumetric Efficiency

The effect of the intake valve timing on the volumetric efficiency are discussed in this section. Three groups of valve timing are tested both in MATLAB and GT-Power.

The first group of valve timing retards the intake valve timing to late IVO as in Figure 3.10, keeping duration and exhaust valve timing fixed. Table 3.1 shows the five intake valve timing cases.

Table 3.1. Five intake valve timing cases

Case Number	Intake Valve Opening (IVO) CAD BTDC	Intake Valve Closing (IVC) CAD ABDC
1	40	30
2	30	40
3	20	50
4	10	60
5	0	70

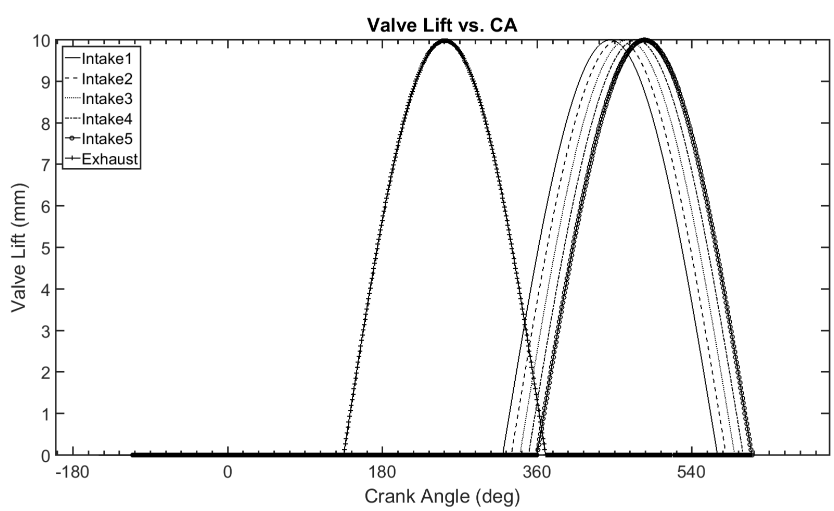

Figure 3.10. Five different intake valve timing cases in Table 3.1

Figure 3.11 and 3.12 are the plots of the volumetric efficiency as a function of the engine speed of these five cases. The legend shows the case number. We can see from both the MATLAB and GT-Power plots that all five curves have a peak volumetric efficiency at a

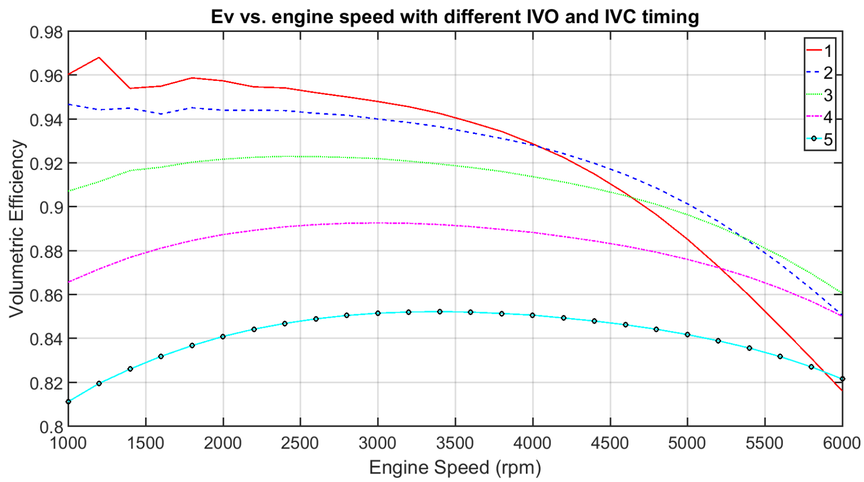

FIGURE 3.11. Volumetric efficiency vs. engine speed with different intake valve timing cases in Table 3.1(MATLAB)

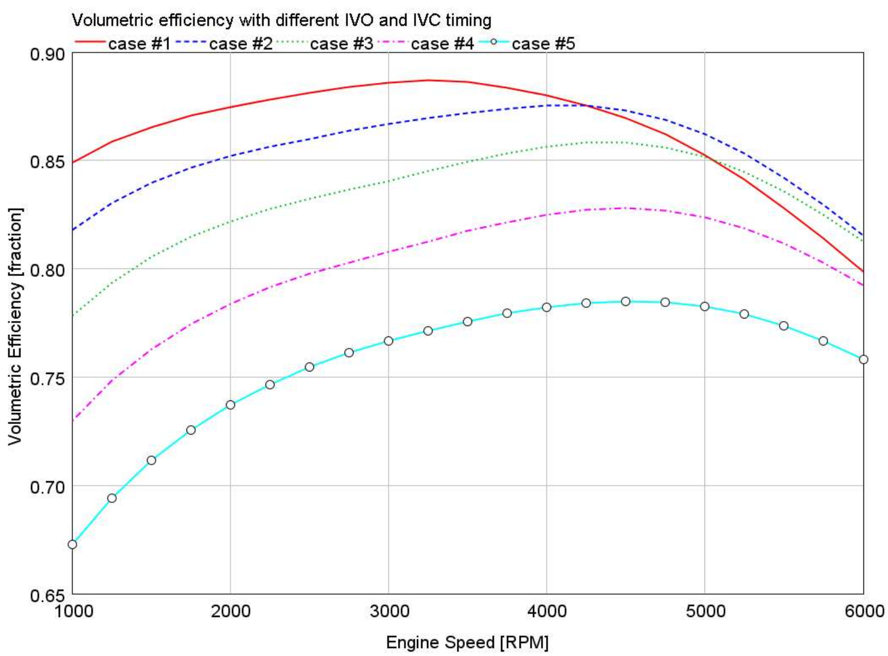

FIGURE 3.12. Volumetric efficiency vs. engine speed with different intake valve timing cases in Table 3.1 (GT-Power)

different engine speed. By retarding the intake valve opening and closing timing, the engine speed for corresponding peak volumetric efficiency increases. However, the overall volumetric efficiency decreases. At a low engine speed, the volumetric efficiency decreases with the intake

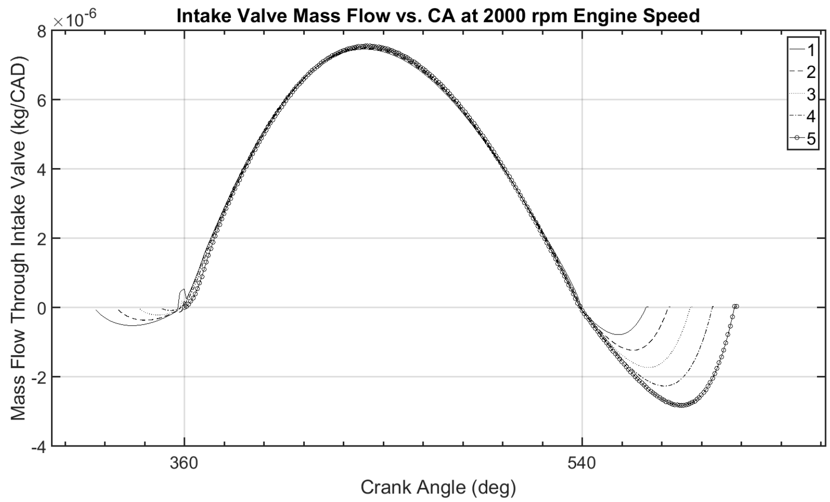

FIGURE 3.13. Intake valve mass flow vs. CAD with different intake valve timing cases of Table 3.1 at 2000 rpm engine speed (MATLAB)

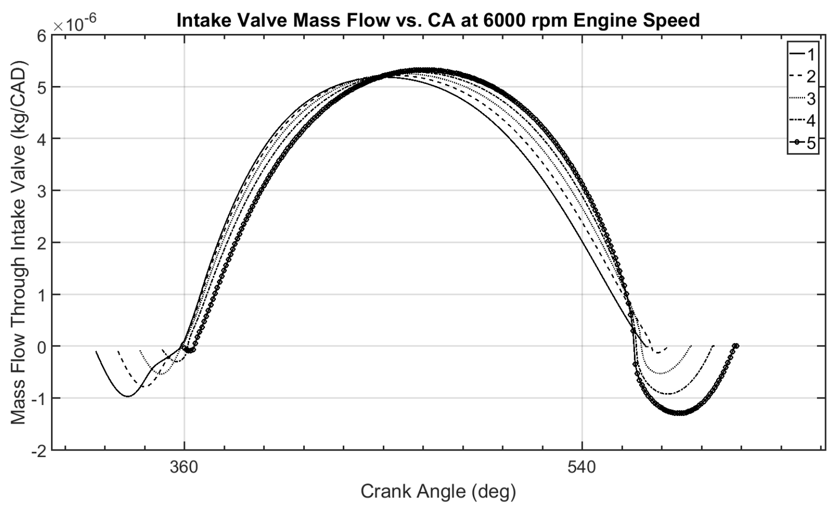

FIGURE 3.14. Intake valve mass flow vs. CAD with different intake valve timing cases of Table 3.1 at 6000 rpm engine speed (MATLAB)

valve timing shifting to late IVO. At high engine speed, the difference between the volumetric efficiency of the different cases is not so considerable as at low engine speed.

Figure 3.13 is the intake valve mass flow per crank angle for five cases at 2000 rpm engine speed and Figure 3.14 is that at 6000 rpm engine speed. The major difference for five cases is the backward flow from the cylinder to the intake before TDC and after BDC. At high engine speed, there is more forward flow at BDC and more backward flow before TDC than that at low engine speed.

Table 3.2 is the second group of 5 intake valve timing cases. Each case has the same intake valve opening timing with intake valve closing crank angle increasing from 30 degree to 70 degree ABDC.

TABLE 3.2. Five intake valve timing cases

Case Number	Intake Valve Opening (IVO) CAD BTDC	Intake Valve Closing (IVC) CAD ABDC
1	20	30
2	20	40
3	20	50
4	20	60
5	20	70

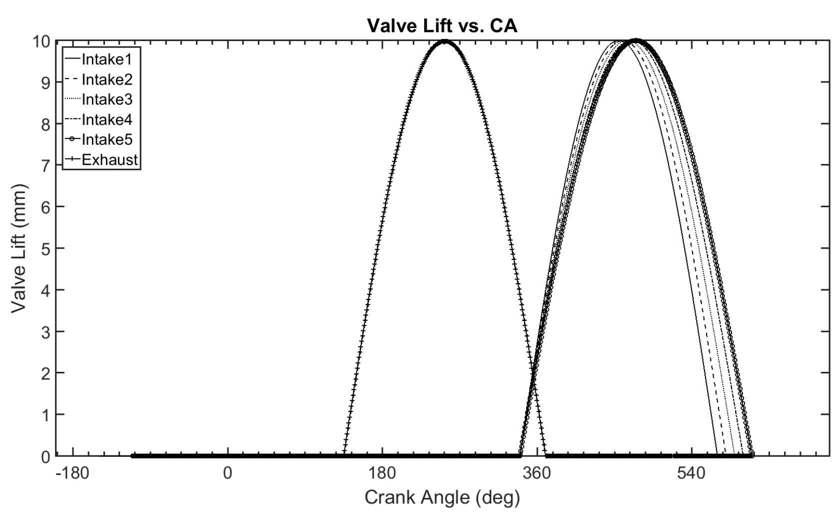

FIGURE 3.15. Five different intake valve timing cases in Table 3.2

Figure 3.16 and 3.17 are plots of the volumetric efficiency of the 5 cases in Table 3.2. The five curves of the volumetric efficiency in this group look very similar with that from the five cases in Table 3.1, except for a little bit difference at high engine speed.

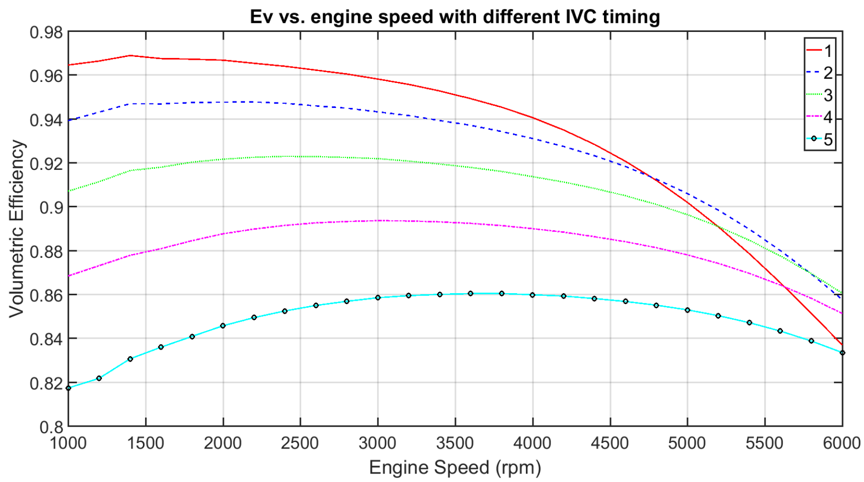

FIGURE 3.16. Volumetric efficiency vs. engine speed with different intake valve timing cases of Table 3.2 (MATLAB)

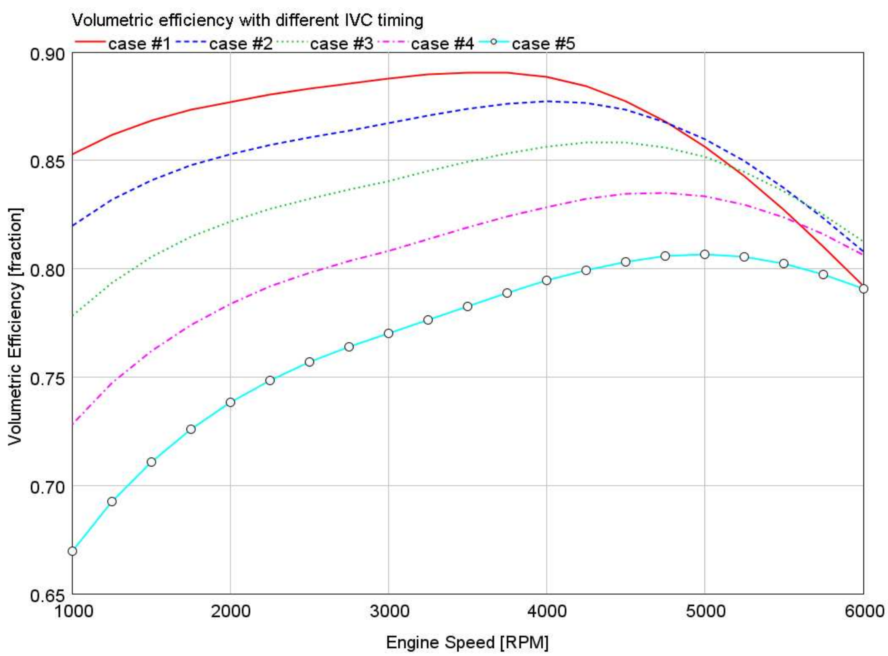

FIGURE 3.17. Volumetric efficiency vs. engine speed with different intake valve timing cases of Table 3.2 (GT-Power)

Figure 3.18 and Figure 3.19 show the intake mass flow per crank angle degree of the second five cases at 2000 and 6000 rpm engine speed respectively. It is the backward flow after BDC that result in the difference in the volumetric efficiency.

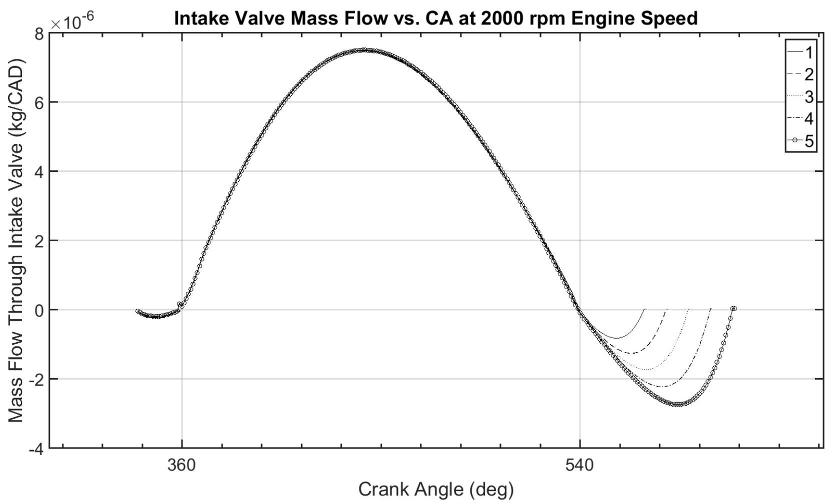

FIGURE 3.18. Intake valve mass flow vs. CAD with different intake valve timing cases of Table 3.2 at 2000 rpm engine speed (MATLAB)

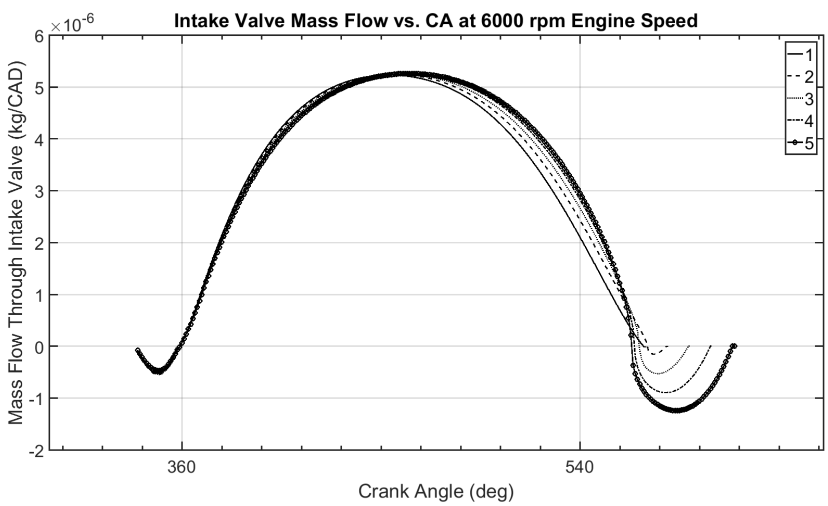

FIGURE 3.19. Intake valve mass flow vs. CAD with different intake valve timing cases of Table 3.2 at 6000 rpm engine speed (MATLAB)

The above two groups of intake valve timing have the same intake valve closing timing and different intake valve opening timing. This brought about the comparable results from two groups.

To see the effect of intake valve opening on the volumetric efficiency, the following 5 cases are investigated. In Table 3.3, these are five cases which adjust the IVO timing from 40 to 0 degree BTDC while keeping the intake valve closing timing fixed.

Figures 3.21 and 3.22 are the plots of the volumetric efficiency of the five cases in Table 3.3. The five curves from this group do not differentiate from each other that much. The results from the MATLAB model of different cases are very close at both low and high engine speeds. The results from the GT-Power model differentiate at high engine speed.

TABLE 3.3. Five intake valve timing cases

Case Number	Intake Valve Opening (IVO) CAD BTDC	Intake Valve Closing (IVC) CAD ABDC
1	40	50
2	30	50
3	20	50
4	10	50
5	0	50

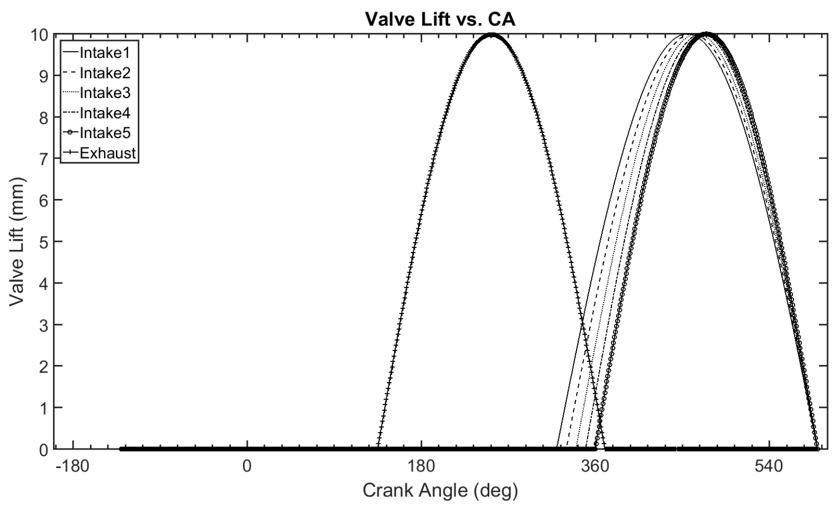

FIGURE 3.20. Five different intake valve timing cases in Table 3.1

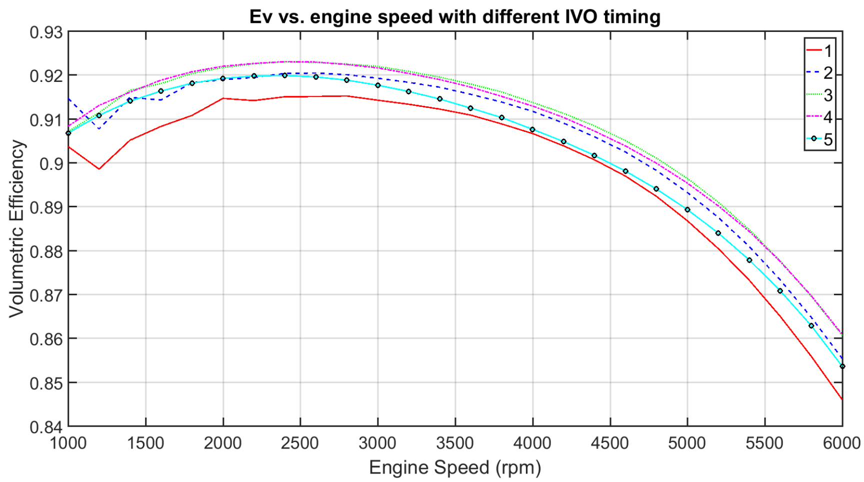

FIGURE 3.21. Volumetric efficiency vs. engine speed with different intake valve timing cases in Table 3.3 (MATLAB)

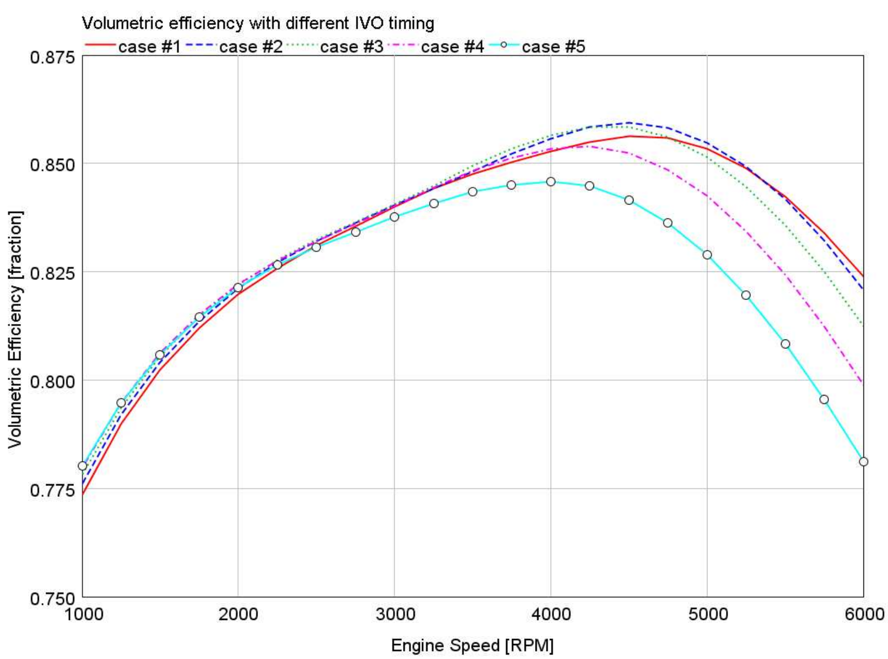

FIGURE 3.22. Volumetric efficiency vs. engine speed with different intake valve timing cases in Table 3.3 (GT-Power)

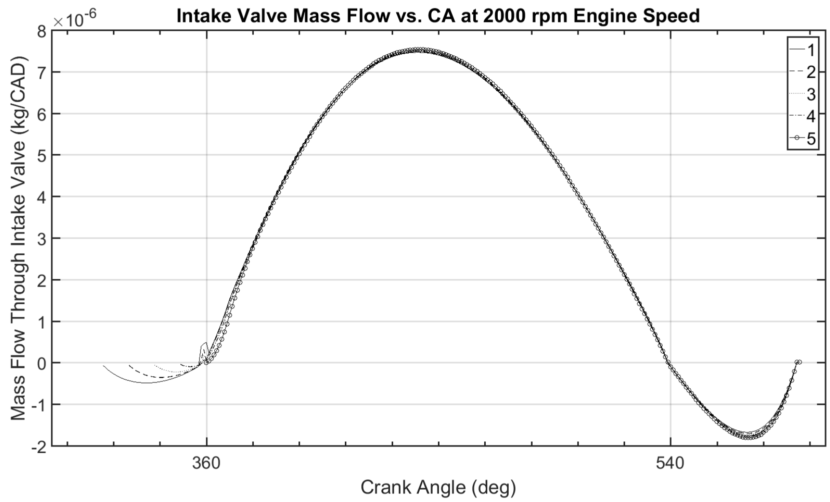

FIGURE 3.23. Intake valve mass flow vs. CAD with different intake valve timing cases of Table 3.3 at 2000 rpm engine speed (MATLAB)

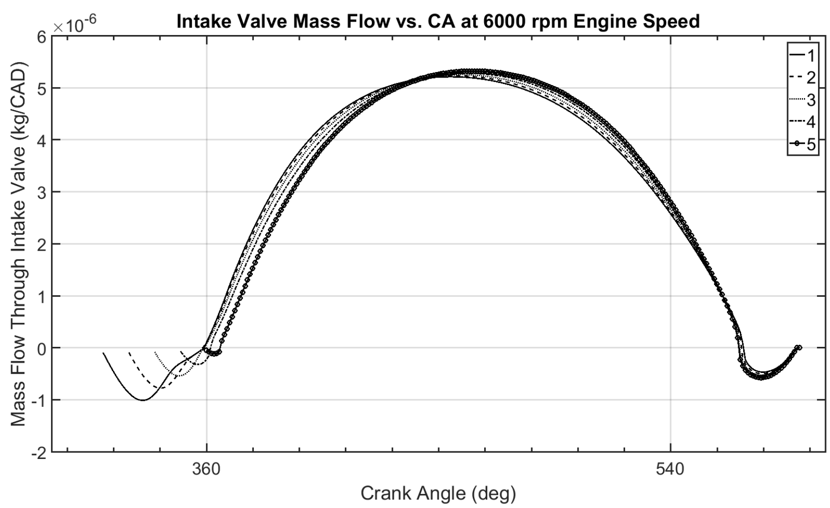

FIGURE 3.24. Intake valve mass flow vs. CAD with different intake valve timing cases of Table 3.3 at 6000 rpm engine speed (MATLAB)

3.4. Effect of Inlet and Exhaust Pressure on Volumetric Efficiency

In this section, the effect of the inlet and exhaust pressure ratio on the volumetric efficiency is investigated. By varying the intake pressure P_i from 50 kPa to 200 kPa and fixing the exhaust pressure P_e as 100 kPa, the volumetric efficiency versus the intake and exhaust pressure ratio are plotted in Figures 3.25 and 3.26.

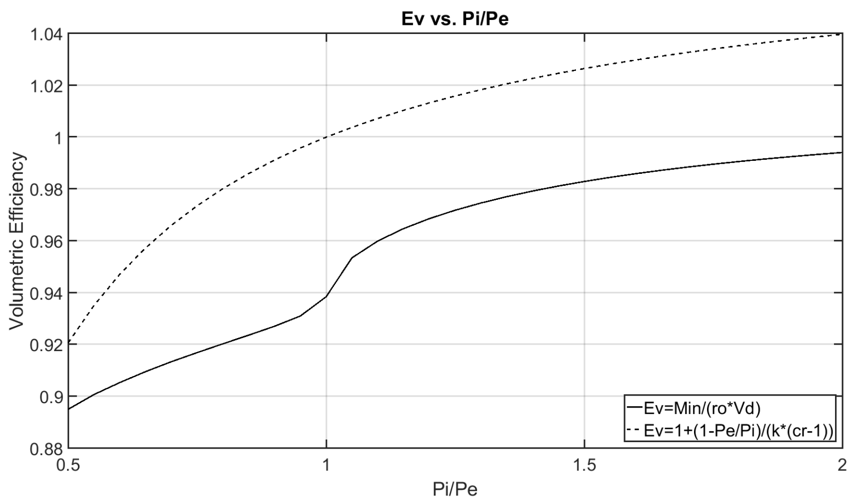

Figure 3.25. Volumetric efficiency vs. intake and exhaust pressure ratio (MATLAB)

In Figure 3.25, the solid line is the volumetric efficiency calculated from Equation (1) and the dashed line is from Equation (3). The two curves both increase with P_i/P_e increases. We can see from Figure 3.25, it is possible to use Equation (3) to predict the volumetric efficiency with some calibration against the idealized cycle assumption. The volumetric efficiency curve from GT-Power and the solid line from MATLAB both have a 'jump' effect at $P_i/P_e = 1$. When the intake and exhaust pressure ratio is larger than 1, there will be a steep increase in volumetric efficiency. However, when the pressure ratio is larger than 1.2, the curves will be smoother.

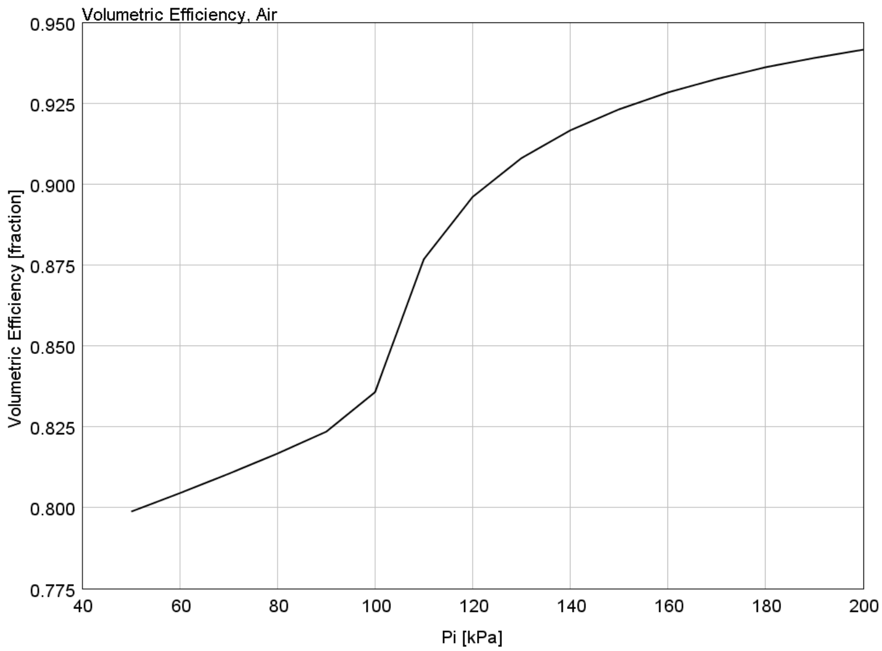

FIGURE 3.26. Volumetric efficiency vs. intake and exhaust pressure ratio (GT-Power)

3.5. Effect of Intake Pressure Pulse on Volumetric Efficiency

To understand intake tuning behavior, the effect of an intake pressure pulse on the volumetric efficiency is studied using the MATLAB code. Figure 3.27 is an example of an intake pressure pulse, of which the pulse location is 450 CAD. Pulse location is defined as the degree of crank angle when the intake pressure pulse starts to enter the cylinder.

Figure 3.28 is the plot of the volumetric efficiency as a function of the location of a positive pressure pulse ranging from BTDC to ABDC at 2000 rpm engine speed. A positive pressure pulse means that the pulse pressure is higher than the constant intake pressure by 15 kPa as in Figure 3.27, while a negative pressure pulse means a lower pressure than the constant pressure by 15 kPa.

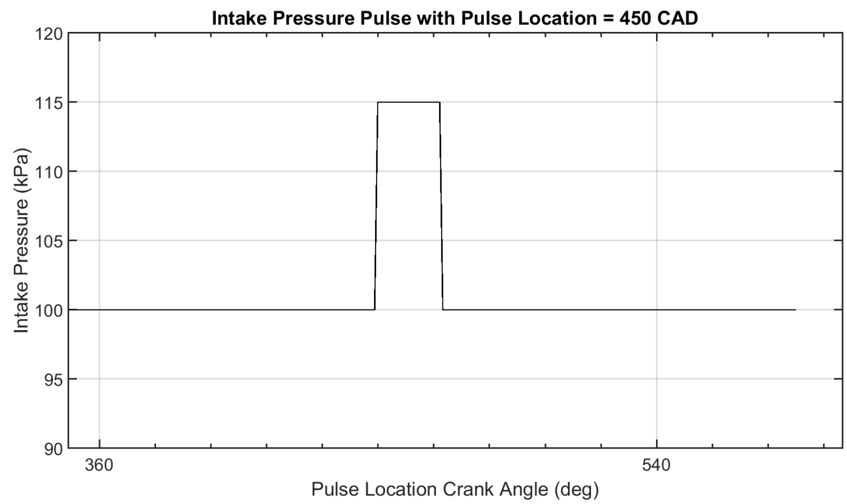

FIGURE 3.27. A pressure pulse example, pulse location = 450 CAD

The horizontal line in Figure 3.28 is the volumetric efficiency baseline without any intake pressure pulse. It indicates that with a positive intake pressure pulse, the volumetric efficiency will increase no matter when the pulse enters the cylinder. It is also clear that when the pulse enters the cylinder near 540 CAD (BDC), it will have a strong supercharge effect that increases the volumetric efficiency by about 9%. While a negative pressure pulse will have a strong adverse impact if it enters the cylinder near BDC, as shown in Figure 3.29. Note that the negative pulse can also result in a volumetric efficiency increase for certain pulse location. This may result from the lower cylinder pressure caused by the negative pressure pulse.

To understand why the supercharge or the adverse effect only happens near BDC, the intake valve mass flow per crank angle degree for four different positive pressure pulse locations are plotted in Figure 3.30. Figure 3.31 plots the mass flow rate, cylinder pressure, and intake pressure for 450 CAD pulse location. The positive pressure pulse will create a positive mass flow into the cylinder. This increase the cylinder pressure to above the intake pressure, so there is a short following period of reverse flow back into the intake.

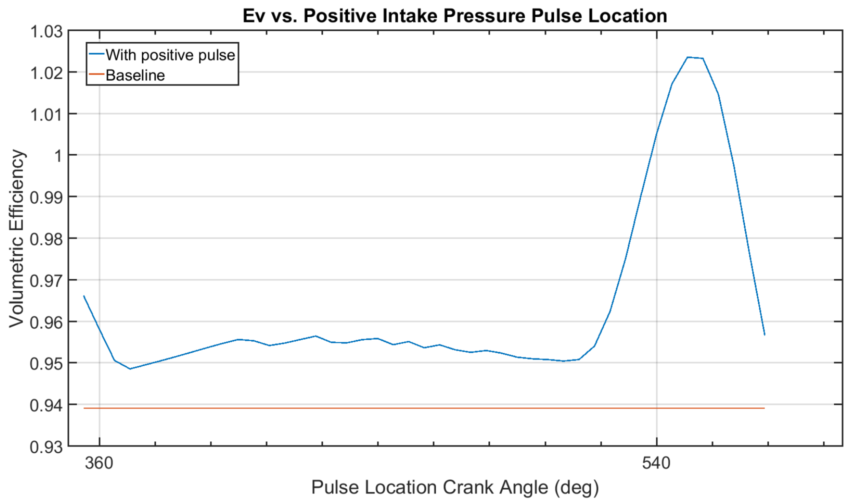

FIGURE 3.28. Volumetric efficiency with different positive pressure pulse location

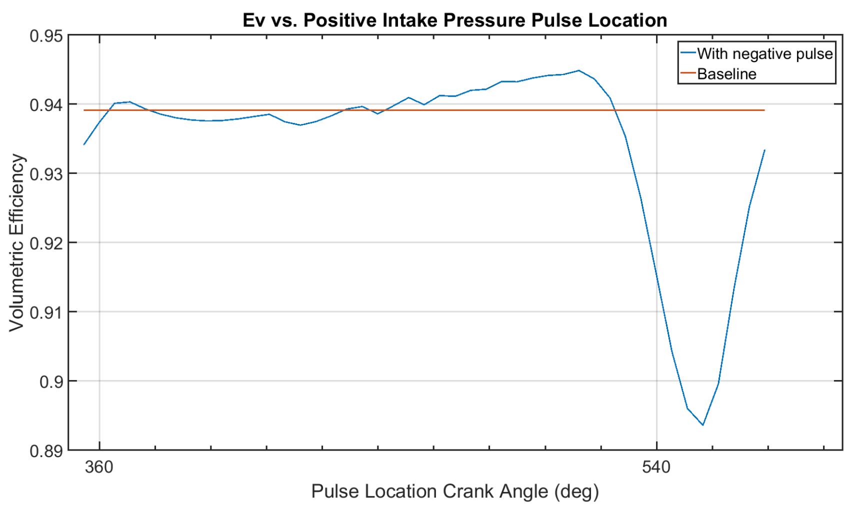

FIGURE 3.29. Volumetric efficiency with different negative pressure pulse location

From Equation (1) and (2), we can see the increase of the volumetric efficiency resulting from the pressure pulse is actually the area of the positive mass flow above the baseline minus the area of the reverse mass flow below the baseline. The net area of the mass flow near BDC is larger than that at other crank angle away from BDC.

To further demonstrate why the net area will be maximized near BDC, the cylinder pressures with different pulse locations are plotted in Figure 3.32. The intake pressure pulse will also create a pressure rise in the cylinder. The difference between the near-BDC pulse

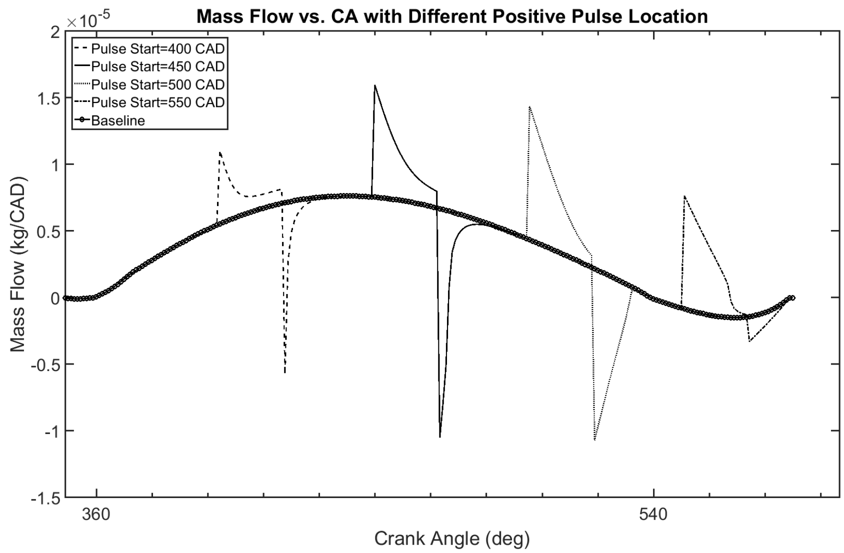

FIGURE 3.30. Intake mass flow rate with 4 different positive pressure pulse location

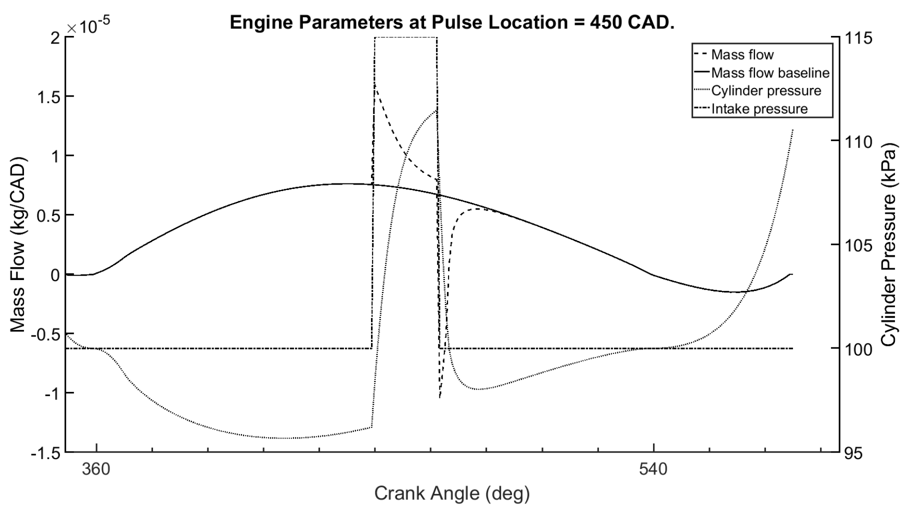

FIGURE 3.31. Engine parameters for pulse location = 450 CAD

and the faraway-BDC pulse is that the cylinder pressure rise caused by the former one has no time to go back to the baseline. It will stay at a higher pressure when the intake valve closes.

The pressure pulse which starts at 550 degrees has the maximum volumetric efficiency as shown in Figure 3.32. We can tell from Figure 3.32 the pulse which starts at 550 degrees results in the largest cylinder pressure when intake valve closes. If we use the ideal gas law

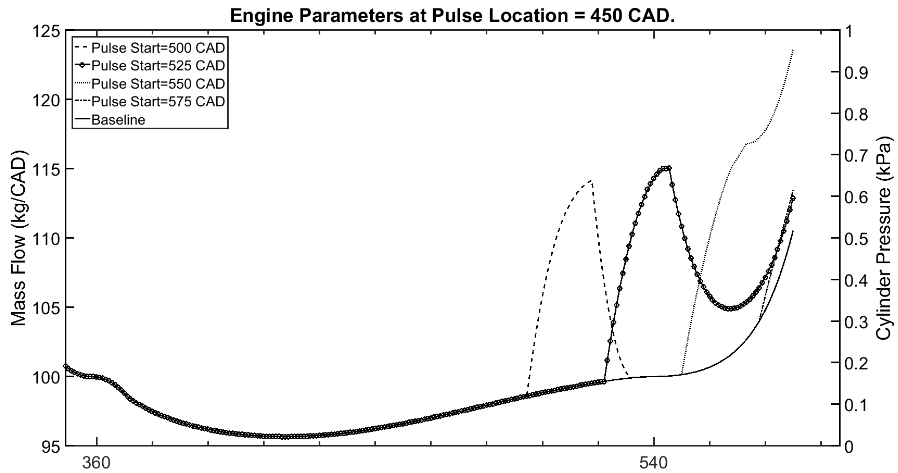

FIGURE 3.32. Cylinder pressure during intake process with different positive pressure pulse location

to calculate the mass trapped in the cylinder at intake valve closing, a larger pressure will give a higher total mass if the temperature and gas constant are fixed.

3.6. Effect of Intake Runner Length on Volumetric Efficiency

The effect of the intake runner length on the volumetric efficiency is explored in GT-Power and MATLAB. The GT-Power model can give a more realistic prediction of the intake tuning effect.

Figure 3.33 is the plot of the volumetric efficiency in terms of engine speed with different intake runner length. A pressure pulse with 15 kPa magnitude and 20 crank angle degrees width is created to enter cylinder with different pulse locations. The pulse location is a function of the intake runner length given by Equation (25).

$$Pulse\ location = \frac{12\pi N}{\sqrt{2}c\sqrt{\frac{A_{pipe}}{L_e V_d}}\sqrt{\frac{cr-1}{cr+1}}} \tag{25}$$

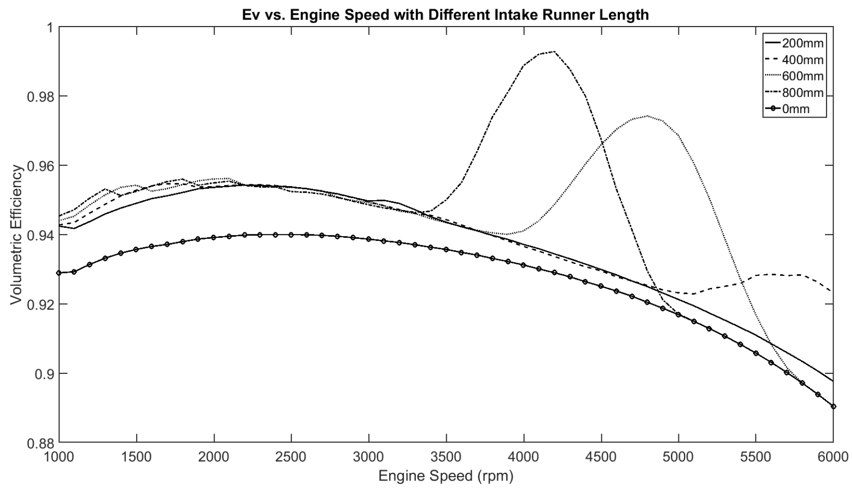

FIGURE 3.33. Volumetric efficiency vs. engine speed with different intake runner length(MATLAB)

The plot of optimum intake runner length in terms of engine speed is presented in Figure 3.34. With lower engine speed, a longer intake runner should be adopted.

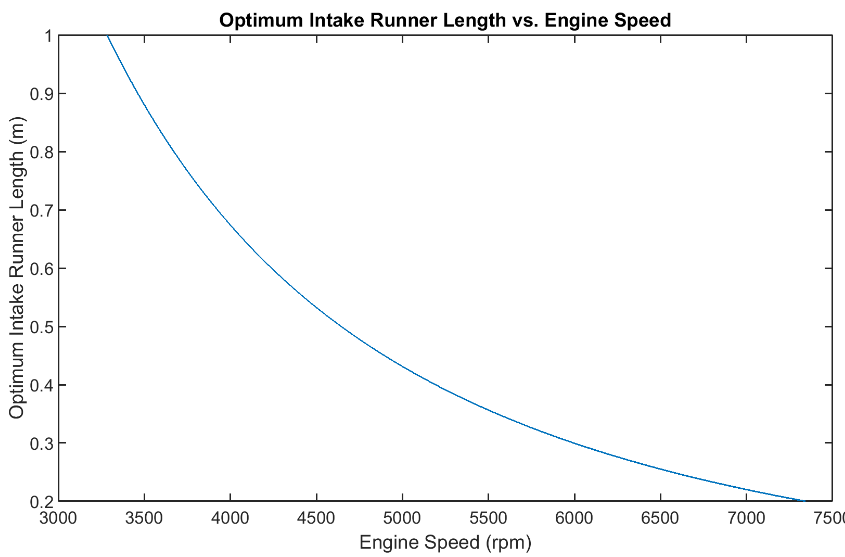

FIGURE 3.34. Optimum intake runner length vs. engine speed

The pulse model is just an idealized model to study the basic behavior of the intake tuning. In actuality, both intake valve opening and intake valve closing will create pressure pulses traveling along the intake runner and they will superimpose with each other to impact the volumetric efficiency.

Figure 3.35 is the configuration of the GT-Power model we use to explore the intake tuning effect. An intake runner is added to the previous GT-Power model. The intake runner length is varied from 200 mm to 800 mm.

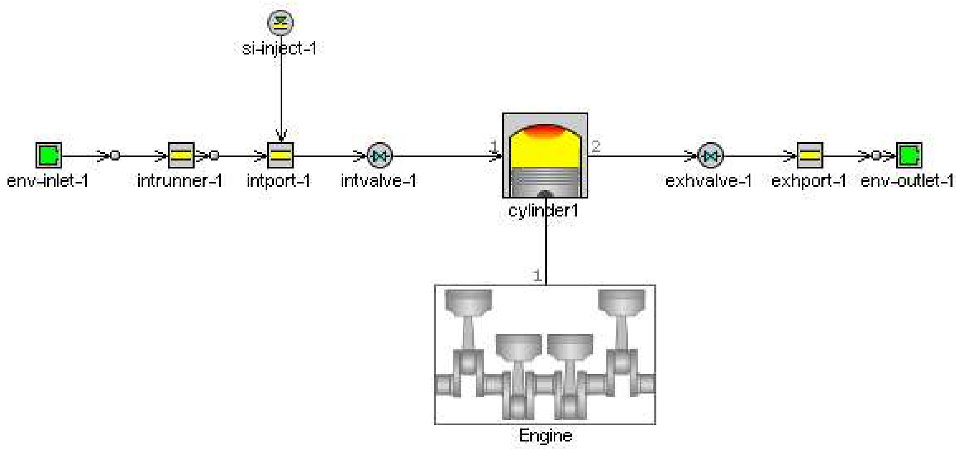

Figure 3.35. GT-Power model with intake runner

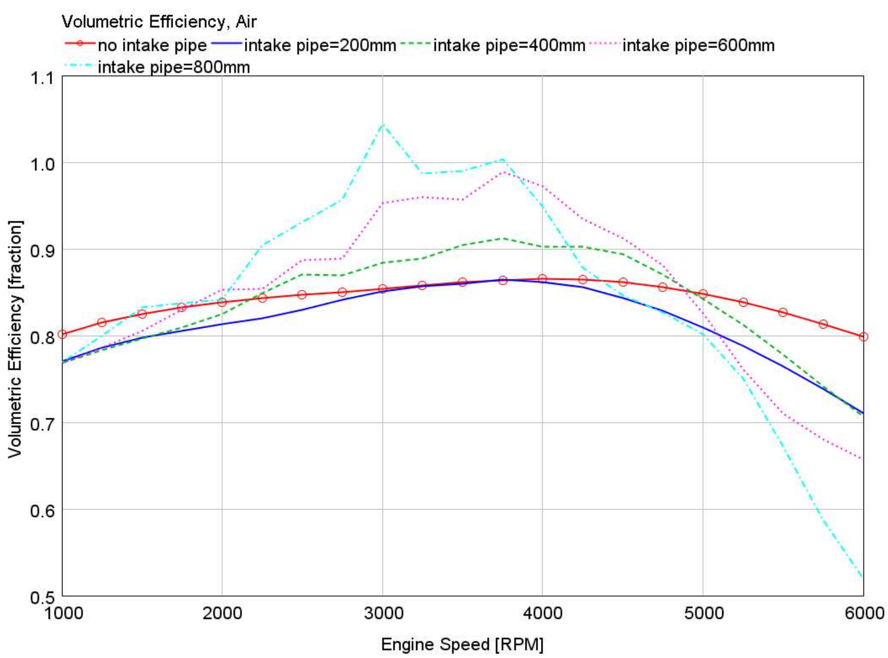

Figure 3.36. Volumetric efficiency vs. engine speed with different intake runner length(GT-Power)

We can see from Figure 3.36, with intake runner length increasing from 200 mm to 800 mm, the peak volumetric efficiency increases and the optimum engine speed decreases. This prediction agrees well with that from Figure 3.33.

Table 3.4 shows the optimal engine speed with different intake runner length from the GT-Power simulation and Helmholtz model prediction. They do not agree so well. The major factors which can explain the differences are: 1) the Helmholtz resonator is a simple idealized model and it can not accurately predict the gas dynamics in the IC engine; 2) the Helmholtz model does not include the IVC pulse effect.

TABLE 3.4. Optimal engine speed and the intake runner length

Intake runner length (mm)	Optimal engine speed (rpm) from GT-Power simulation	Optimal engine speed (rpm) from Helmholtz resonator model prediction
200	3750	6767
400	3750	4976
600	3750	4779
800	3000	3592

CHAPTER 4

Conclusions

A MATLAB and a GT-Power model based on a general square engine are built to study the effect of the intake system parameters on the volumetric efficiency. The mass flow model through the valves including choked flow are included in the MATLAB model. It is able to predict the cylinder pressure during the aspiration processes. Heat transfer and pressure loss in the intake and exhaust runner are neglected. Except for intake tuning effect section, intake and exhaust runner are excluded from both MATLAB and GT-Power model.

The influence of the engine speed, intake valve timing and lift, intake and exhaust pressure ratio, intake pressure pulse and intake runner length on the volumetric efficiency are tested in two models. Although the exact value from the two models doesn't agree very well with each other because of the simplicity of the MATLAB model, the tendency of the curves from two models are similar and can help to explain the parameter effects on the volumetric efficiency.

The following conclusions can be drawn from results from two models:

(1) By increasing intake valve lift, the engine speed for peak volumetric efficiency will increase.

(2) IVC timing has much more dominant influence on volumetric efficiency than IVO timing.

(3) Volumetric efficiency will decrease with a retarded IVC time at low to normal engine speed, due to reverse flow. It will increase at higher engine speeds.

(4) With a positive pressure pulse entering the cylinder, the volumetric efficiency will increase. The supercharge effect can be achieved if the positive pressure pulse enters

cylinder near BDC, or right before intake valve close, increasing the cylinder pressure and mass.

(5) A Helmholtz model is adopted to predict the relationship between optimum intake runner length and engine speed. It does not agree very well with the GT-Power model.

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