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공학석사 학위논문

Development of Quasi-dimensional Combustion Model of SI Engines Equipped with Variable Valve Actuation

가변밸브가 장착된 가솔린 엔진의 성능 예측을 위한 유사차원 연소모델 개발

2014년 2월

서울대학교 대학원

기계항공공학부

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Development of Quasi-dimensional Combustion Model of SI Engines Equipped with Variable Valve Actuation

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이 논문을 공학석사 학위논문으로 제출함
2014년 2월

서울대학교 대학원
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Development of Quasi-dimensional Combustion Model of SI Engines Equipped with Variable Valve Actuation

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Abstract

Variable valve actuation technology has great potential of improving both fuel economy and performance of SI engines. However, due to numerous valve profiles and excessive calculation time, optimization of valve actuation under various engine operation conditions can hardly be achieved by experimental method or 3D engine simulation tools. Therefore, in this work, a quasi-dimensional combustion model for fast simulation of SI engines equipped with VVA was developed.

MIT quasi-dimensional combustion model was used as the base code. It was modified to reflect the effect of VVA on burning rate prediction which is mainly affected by turbulent intensity and residual gas fraction. STAR-CD and GT-

POWER were used to obtain base data of turbulent intensity and residual gas fraction under various valve lift, duration and timing. Using these base data, the turbulent flow energy model and boundary conditions of gas exchange process were modified to improve the accuracy of turbulent intensity and residual gas fraction prediction.

Model calibration based on experimental data was carried out. Simulation results showed that the modified model can predict burning rate with acceptable degree of accuracy under various valve lift, duration and timing. Finally, using the modified quasi-dimensional combustion model, the effect of VVA on fuel economy improvements was verified.

Keywords: Fuel economy, Quasi-dimensional combustion model, Variable valve actuation, Turbulent Intensity, Residual Gas Fraction, Burning rate

Student Number : 2012-20660

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Acronym

VVA	Variable Valve Actuation
IVO	Intake Valve Open
IVC	Intake Valve Close
EVO	Exhaust Valve Open
EVC	Exhaust Valve Close
ISFC	Indicated Specific Fuel Consumption
BSFC	Brake Specific Fuel Consumption
RGF	Residual Gas fraction
WOT	Wide Open Throttle
BMEP	Brake Mean Effective Pressure
IMEP	Indicated Mean Effective Pressure
CAD	Crank Angle Degree

Chapter 1. Introduction

1.1 Background and Literature Reviews

Due to soaring of crude oil price and global warming issues, major automobile producing and consuming countries have tightened regulation on fuel economy. In the USA, CAFÉ (corporate average fuel economy) regulation mandates average fuel mileage of 37.8mpg for passenger car and light duty truck by 2016[2]. Furthermore, this fuel economy requirement will be doubled by 2025. EU has regulated fuel economy not by direct standard on fuel economy but by CO2 emission standard. CO2 emission standard has been decreased over the time from 186g/km by 1995, to 140g/km by 2008, and to 95g/km by 2020.

To encounter these stringent fuel economy regulation, major vehicle manufacturers have developed highly energy efficient and zero emission vehicles such as hybrid vehicles, high efficiency CI engines, electric vehicles and fuel cell electric vehicles. Eventually, gasoline and diesel fuelled vehicles will be replaced by electric or fuel cell electric vehicles. However, production cost of electrically-powered vehicles is much higher than that of gasoline or diesel fuelled vehicles and infrastructure for recharging or refueling electrically-powered vehicles is not established enough to fully commercialize these kinds of vehicles[3]. Thus, traditional SI and CI engines

will be used as a main power source for at least another decade.

To increase fuel economy of SI engines, many new technologies such as down-sizing, gasoline direct injection, homogeneous charge compressed ignition, variable valve actuation, etc. have been developed. Among these, variable valve actuation (VVA) technology has regular cost effectiveness but has the potential of improving fuel economy, maximum torque and power by optimizing intake and exhaust valve lift, duration and timing[4].

In addition, VVA combined with other combustion technologies can create a synergy effect. Through VVA with spray-guided direct injection, charge stratified combustion can be realized[5]. Furthermore, VVA can be used as a control device of homogeneous charge compressed ignition[6].

Many preceding researches about the effects of VVA on engine performance have been carried out. SI engines with electro-hydraulic fully variable valve reduced CO2 emissions up to 10%, and achieved 10% more power and 15% more torque in the lower speed range [7]. 7% fuel economy improvement could be achieved by un-throttled engine operation with variable intake valve lift, duration and timing [8]. Using two-step variable valve actuation not fully variable valve actuation, fuel economy could be improved by 8% [9].

At the part load condition, fuel economy is mainly deteriorated by pumping loss[10] caused by pressure difference between intake and exhaust processes. For SI engines equipped with VVA, inducted air mass can be controlled not by throttle body but by intake valve lift, duration and timing. Therefore, pressure drop of inducted air across the throttle body can be eliminated by un-throttled engine operation. This leads to reduction of pumping loss. In addition, at high engine rpm and load, more torque and power can be achieved by optimizing valve lift, duration and timing to induct more air into the cylinder[11].

As mentioned above, not being confined to fuel economy improvement, VVA shows the great potential for improving overall engine performance. Therefore, many preceding researches have been conducted to optimize valve profile of VVA engines under different rpm and load conditions. However, optimization of valve profile has not been completed due to numerous combination of valve profile which is determined by lift, duration and timing of intake and exhaust valves. Furthermore, VVA has effects on not only inducted air mass but also turbulent intensity[12] and residual gas fraction[13] which changes burning rate drastically.

This means that optimization of VVA engines through experimental methods entails considerable expense and time. It is impossible to discover all the potential of VVA through experimental method only.

Due to vast amount of test case, commercial 3D engine simulation program needs excessive calculation time to optimize VVA. 1D engine simulation tools which have much shorter computing time than 3D simulation tools, use Wiebe function to model the combustion in SI engines. Since Wiebe function is a semi empirical law which does not consider physics related to burning rate, 1D simulation tools cannot predict burning rate variation precisely under various valve lift, duration and timing.

Therefore, it is necessary to develop a fast engine simulation tool which can predict the effect of VVA on SI engine performance.

1.2 Objectives

In this study, quasi-dimensional combustion model for SI engines equipped with VVA was developed. Model development process mainly focused on improvement of accuracy of burning rate prediction.

First, a laminar flame speed correlation which consider effect of diluent mixture on laminar flame speed and turbulent flame speed which contains exponential terms to describe early stage of flame development were introduced to quasi-dimensional combustion model.

And then, other important parameters for burning rate prediction which are influenced by VVA, such as turbulent intensity, residual gas fraction were investigated and modified. To estimate turbulent intensity and residual gas fraction under various valve lift, duration and timing, commercial 3D CFD program STAR-CD and 1D engine simulation program GT-POWER were used. Base on these results, turbulent intensity, residual gas fraction of quasi-dimensional combustion model were modified to reflect the effects of VVA on engine performance.

Adjustable constants were calibrated to make quasi-dimensional combustion model be applicable to VVA engine simulation. And potential of VVA on fuel economy improvement was validated.

Chapter 2. Investigation into base model

2.1 Overview of base model

Quasi-dimensional combustion model which had been developed by Poulos[14] was used as base model. Base model assumes flame propagates as a sphere centered at spark plug electrodes. So, it needs geometrical data about interaction of flame front with combustion chamber. Using this geometrical data, combustion chamber is divided into burned and unburned zone. To consider heat transfer effect, burned zone is composed of adiabatic core and thermal boundary layer. And this model integrates differential equations with time dimension which describe basic thermodynamics, combustion reaction, gas exchange process, turbulent energy cascade, NOx formation etc. Using these differential equations, air mass inducted, residual gas fraction, in-cylinder pressure, temperature, heat transfer rate, NOx, etc. can be calculated.

For simulation of SI engine with VVA, accuracy of burning rate prediction is the most significant part of simulation. Burning rate is mainly affected by laminar flame speed and turbulent intensity. As mentioned above, base model contains turbulent energy cascade and gas exchange model which determine turbulent intensity and

residual gas fraction. So, it can reflect the effect of VVA on burning rate under various rpms and loads.

2.2 Validation of base model

First, it needed to investigate the quantitative variation of burning rate under various valve lift, duration and timing, because base quasi-dimensional model had not been applied to VVA engine simulation. So, turbulent intensity and residual gas fraction of base quasi-dimensional model which determine burning rate were validated.

Turbulent intensity of base quasi-dimensional model was investigated at the condition listed below.

Table 1 — Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure [bar]	1	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	383	172
Valve close timing [CAD]	655	402

As shown in figure1, turbulent intensity of base model differs from STAR-CD simulation data acquired at the same condition. At the early stage of intake process,

turbulent intensity of base quasi-dimensional model is underestimated and base quasi-dimensional model is unable to reflect the effect of chocked flow which makes peak point of STAR-CD simulation result near 440 CAD. On the contrary to this, at the end of compression process, turbulent intensity of base quasi-dimensional model is overestimated.

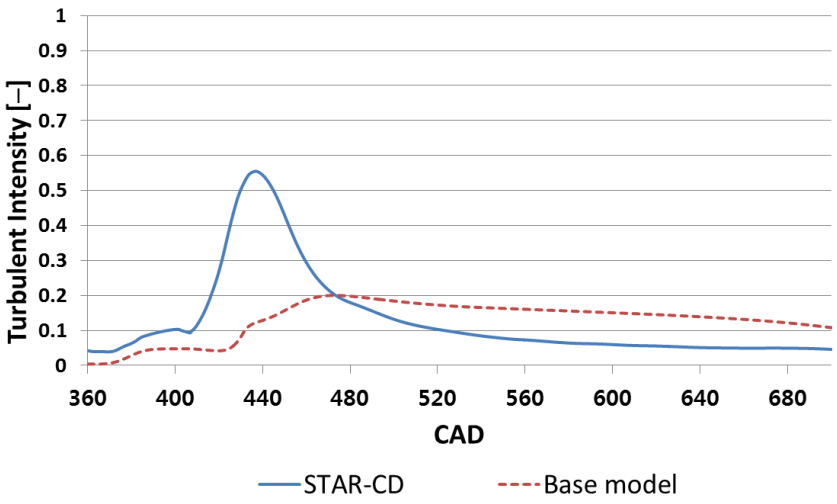


Fig.1 Turbulent intensity of base quasi-dimensional combustion model at condition listed in table 1

Residual gas fraction was validated. Test conditions are listed below

Table 2 — Test conditions of RGF validation

case #	rpm	Max Lift [mm]		IVO [CAD]	IVC [CAD]	EVO [CAD]	EVC [CAD]
		In cam	Ex cam				
1	2000	9	8	343	591	528	738
2	1000	9	8	354	602	530	740
3	1500	9	8	350	598	530	740
4	1500	2	8	372	578	470	730
5	1500	2	8	300	506	470	700
6	1500	2	8	310	516	470	700

As indicated in figure 2, base model calculation results show similar RGF trends of GT-POWER simulation results. However, RGF of base quasi-dimensional combustion model tends to be overestimated for 9 mm intake valve lift.

In summary, base model has a good potential for VVA engine simulation. But to improve the accuracy of burning rate prediction, parameters related to burning rate should be modified.

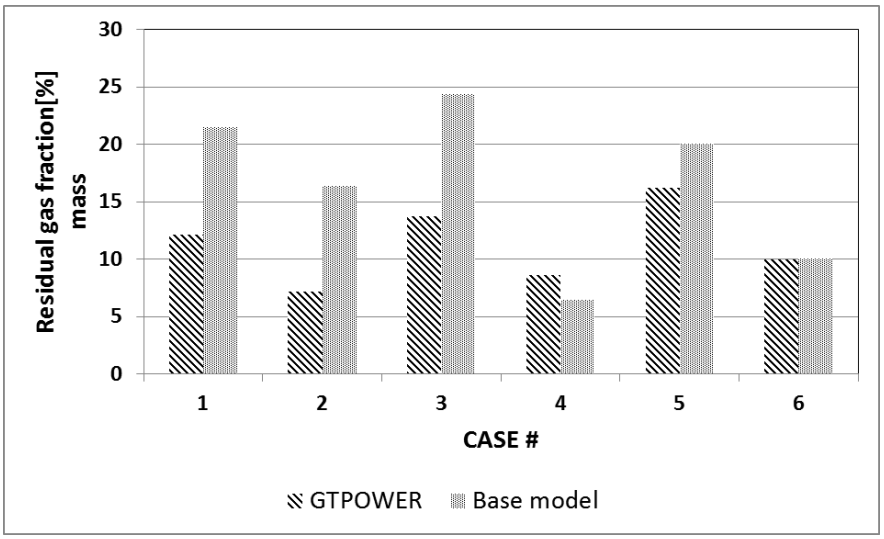


Fig.2 Residual gas fraction comparison of GTPOWER and base quasi-dimensional combustion model at conditions listed in table 2

Chapter 3. Model modification

3.1 Combustion model

Turbulent entrainment model is used to describe turbulent combustion process in quasi-dimensional combustion model. This combustion model had been formulated by Blizzard and Keck[15]. And then turbulent structure during combustion was modified by Tabazynski[16].

In this combustion model, it is assumed that combustion process consists of entrainment of unburned gas and burning of unburned gas already entrained within flame front area. As flame propagates, unburned gas is entrained into spherically shaped flame frontal area. Mass entrainment rate is expressed as below.

$$\frac{dm_e}{dt} = \rho_u A_f S_t$$

$$S_t = u' + S_l$$

$\mathbf{m}$	Mass entrained into the flame front
A_f	Flame frontal area
ρ_u	Density of unburned gas
S_t	Turbulent flame speed

$\boldsymbol{u'}$ Turbulent intensity

$\boldsymbol{S_l}$ Laminar flame speed

It is also assumed that within taylor micro scale, flame propagates at laminar flame speed. So, combustion characteristic time and burning rate are expressed as below.

$$\boldsymbol{\tau_b = \frac{\lambda}{S_l}}$$

$$\boldsymbol{\frac{dm_b}{dt} = \frac{m_e - m_b}{\tau_b}}$$

$\boldsymbol{\lambda}$ Taylor micro scale

In modified quasi-dimensional combustion model, it is assumed that there is no significant difference between turbulent entrainment rate and burning rate[17]. So, burning rate was set equal to turbulent entrainment rate.

$$\boldsymbol{\frac{dm_e}{dt} = \frac{dm_b}{dt}}$$

Turbulent flame speed of base model is represented simply as the sum of laminar flame speed and turbulent intensity. Because this correlation ignores the effect of ignition delay, in-cylinder pressure tends to be overestimated at the early stage of combustion[18]. So, developing flame speed proposed by H.Werweg[19] which

contains exponential terms to describe the early stage of flame development was introduced to the modified quasi-dimensional combustion model.

$$\frac{S_t}{S_l} = 1 + C_a \left[\frac{u'}{u'+s_l}\right]^{\frac{1}{2}} \cdot \left[1 - e^{-\frac{s_l}{u'+s_l}\cdot\frac{t}{\tau}}\right]^{\frac{1}{2}} \cdot \left[1 - e^{\frac{t}{\tau}}\right]^{\frac{1}{2}} \cdot \left(\frac{u'}{s_l}\right)^{\frac{5}{6}}$$

$\mathbf{t}$	Elapsed time from spark timing
C_a	Adjustable constant

Recommended value of $\mathbf{t}$ is 1.5 ms[20] and C_a should be calibrated to match experimental data.

Laminar flame speed correlation of Pae et al[21], was introduced to the modified quasi-dimensional combustion model also. This laminar flame speed correlation contains a parameter which reflects equivalence ratio variation on laminar flame speed.

3.2 Turbulent intensity

Zero-dimensional energy cascade model[22] is used to describe turbulent flow energy in base quasi-dimensional combustion model. During the intake process, fresh charge with fuel and residual gas is inducted into the cylinder and it also carries

mean kinetic energy. Mean kinetic energy is transformed to turbulent kinetic energy through a turbulent dissipation process. At the same time, turbulent kinetic energy is transformed to thermal energy through a viscous dissipation process. It is assumed that in-cylinder turbulent flow is homogeneous and isotropic.

Mean kinetic energy and turbulent kinetic energy can be calculated by the following equations.

$$\mathbf{K} = \frac{1}{2} m V_i^{\,2}$$

$$\mathbf{k} = \frac{1}{2} m u'^{2}$$

K	Mean kinetic energy
k	Turbulent kinetic energy
m	Trapped mass inside cylinder
$\boldsymbol{V_i}$	Mean flow velocity into the cylinder
$\boldsymbol{u'}$	Turbulent intensity

Rate of change of mean kinetic energy and turbulent kinetic energy can be expressed as follows,

$$\frac{\mathbf{dK}}{\mathbf{dt}} = \frac{\mathbf{1}}{\mathbf{2}}\dot{\boldsymbol{m}}_{\boldsymbol{i}}{V_{\boldsymbol{i}}}^{\mathbf{2}} - \boldsymbol{P} - \boldsymbol{K}\frac{\dot{\boldsymbol{m}}_{\boldsymbol{e}}}{\boldsymbol{m}}$$

$$\frac{\mathbf{dk}}{\mathbf{dt}} = \boldsymbol{P} - \boldsymbol{m\varepsilon} - \boldsymbol{k}\frac{\dot{\boldsymbol{m}}_{\boldsymbol{e}}}{\boldsymbol{m}}$$

$$\mathbf{P} = \mathbf{0.3307}\boldsymbol{C_\beta}\left(\frac{\boldsymbol{K}}{\boldsymbol{L}}\right)\left(\frac{\boldsymbol{k}}{\boldsymbol{m}}\right)^{\frac{\mathbf{1}}{\mathbf{2}}}$$

$$\boldsymbol{\varepsilon} = \frac{(\mathbf{2}\boldsymbol{k}/\mathbf{3}\boldsymbol{m})^{\frac{\mathbf{1}}{\mathbf{2}}}}{\boldsymbol{l}}$$

$\dot{\boldsymbol{m}}_{\boldsymbol{i}}$	Mass induction rate through the intake valve
$\dot{\boldsymbol{m}}_{\boldsymbol{e}}$	Mass outflow rate through the exhaust valve
$\boldsymbol{\epsilon}$	Turbulence dissipation rate per unit mass
$\mathbf{P}$	Turbulence energy production rate
$\boldsymbol{C_\beta}$	Adjustable constant
$\mathbf{L} = \mathbf{l}$	Integral length scale

3.2.1 Integral length scale

It is assumed that in-cylinder turbulence production process is similar to that of flat plate turbulent boundary layer. L is geometrical length scale and l is integral length scale. In this model, it is assumed that geometrical length scale is equal to integral

length scale. And its value is equal to stroke/2 during intake, compression and exhaust process.

For combustion process, by the rapid distortion theory, integral length scale and turbulent intensity are conserved. Integral length scale and turbulent intensity at spark timing can be determined by turbulent energy cascade model. Integral length scale and turbulent intensity can be described as (Subscript s means spark timing)

$$\frac{L}{L_s} = \left(\frac{\rho_{u,s}}{\rho_u}\right)^{\frac{1}{3}}$$

$$\frac{u'}{u'_s} = \left(\frac{\rho_u}{\rho_{u,s}}\right)^{\frac{1}{3}}$$

As mentioned above, integral length scale is determined by the chamber geometry. But because of this, integral length scale is overestimated[23] and the rate of change of the mean kinetic and turbulent kinetic energy is underestimated during intake and compression process.

Under various conditions of intake pressure, rpm, valve lift, duration and timing, more than 50 cases of integral length scale data were acquired using STAR-CD. These data shows that integral length scale is influenced by intake pressure, rpm, valve lift, duration and timing.

As indicated in figure 3, integral length scale does not vary significantly as intake pressure, rpm, valve lift, duration and timing change and it seems that integral length scale is mainly governed by combustion chamber geometry.

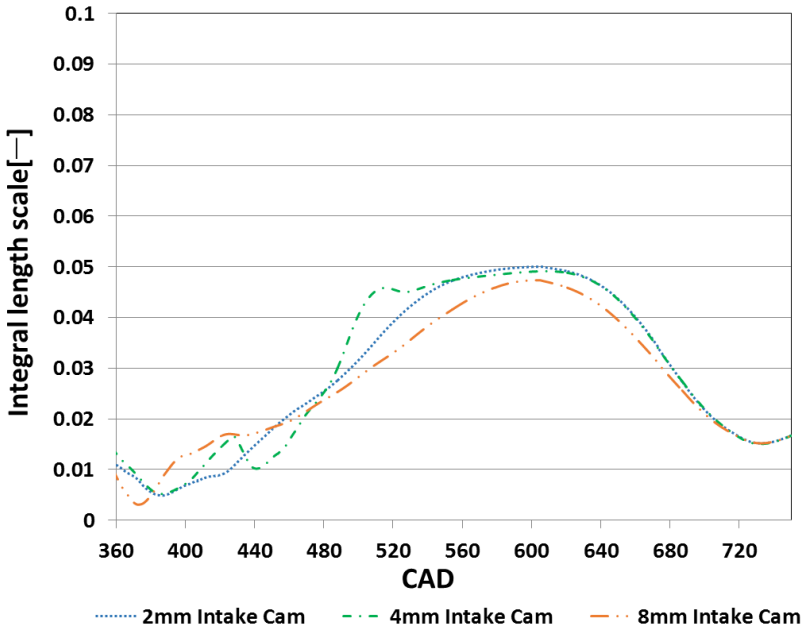


Fig.3 Integral length scale data obtained from STAR-CD under various rpm, valve actuation conditions

As shown in figure 4, integral length scale of base quasi-dimensional combustion model is approximately ten times larger than that of STAR CD results. So, integral length scale of the modified quasi-dimensional combustion model was changed to have same length order of STAR CD results.

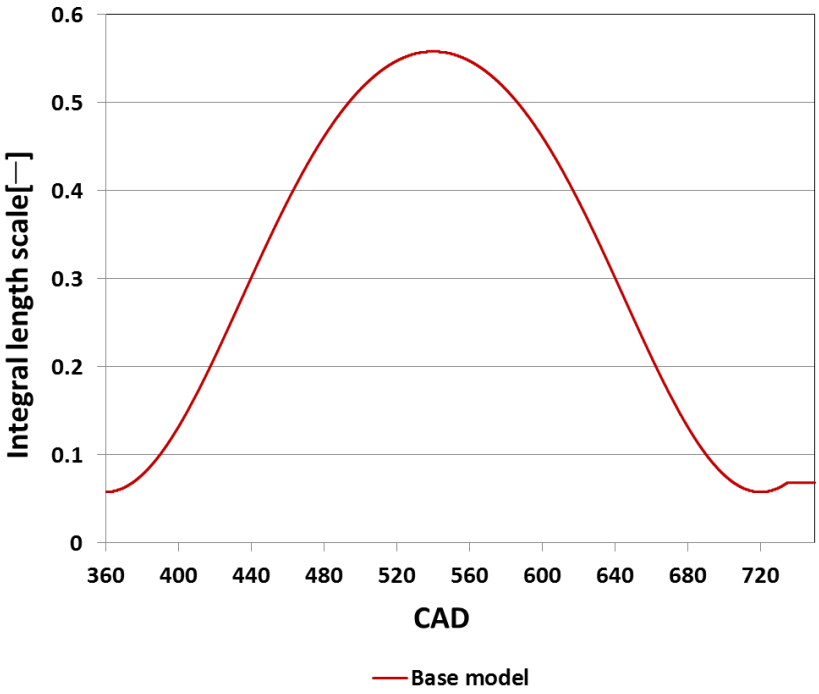


Fig.4 Integral length scale of base quasi-dimensional combustion model

In figure 5, modified integral length scale is represented. It is governed by combustion chamber geometry only and its maximum and minimum value is limited near TDC and BDC.

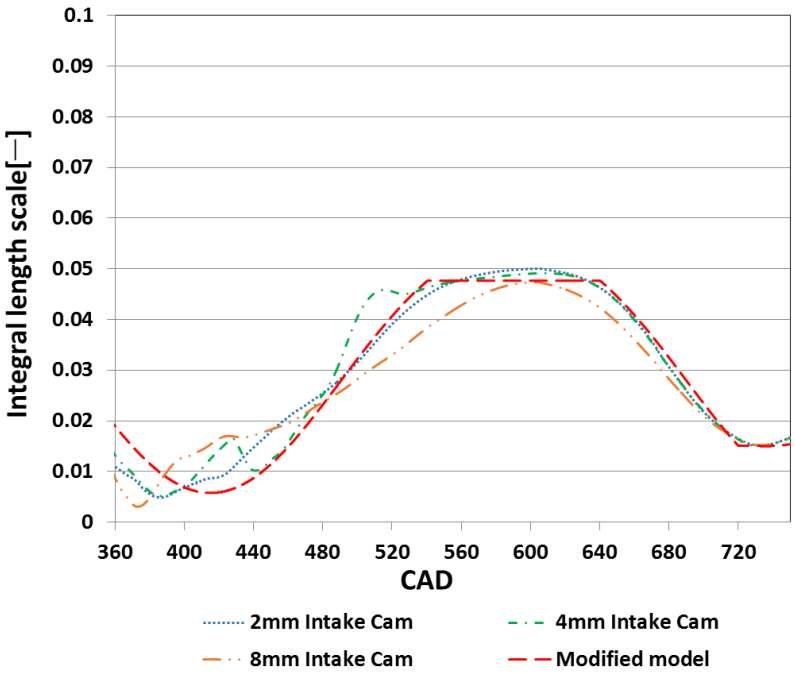


Fig.5 Integral length scale comparison of STAR-CD and the modified quasi-dimensional combustion model

3.2.2 Turbulent dissipation parameter

For simulation of SI engines with classic cam train, turbulent intensity is calibrated by adjustable constant C_β which is determined uniquely by combustion chamber geometry. But for VVA engine simulation, it is impossible to calibrate turbulent intensity through C_β only.

As shown in figure 1, for base quasi-dimensional combustion model, decreasing rate of turbulence intensity is too small at the end of intake process and its peak value during intake process is overestimated.

Therefore, to simulate VVA engine, another parameter which can control rate of change of turbulent intensity was introduced to the modified quasi-dimensional combustion model. This parameter is a function of rpm, valve lift, timing and duration.

$$\frac{\mathbf{dk}}{\mathbf{dt}} = \boldsymbol{P} - \boldsymbol{m}\boldsymbol{C_\gamma}\boldsymbol{\varepsilon} - \boldsymbol{k}\frac{\dot{\boldsymbol{m}}_{\boldsymbol{e}}}{\boldsymbol{m}}$$

$\boldsymbol{C_\gamma}$ Turbulent kinetic energy dissipation rate parameter

3.2.3 Turbulent intensity of the modified quasi-dimensional combustion model

After turbulence flow energy model modification, turbulent intensity of modified quasi-dimensional combustion model was validated. First, at base intake cam lift, timing and duration, rpm was changed from 1500 to 2500 by 500rpm. As shown in figures 6~8, simulation results of modified model show good agreement with 3D simulation results. And then, at base intake cam lift and duration, IVO was changed from 338 to 383 CAD. In figures 9,6,10, it is shown that modified model can reflect the effect of valve timing sweep on turbulent intensity.

Finally, Changing IVO from 338 to 403 CAD, turbulent intensity of VVA mode with 2mm cam lift and 124 CAD duration was validated. Figures 8~ 11 show that under VVA mode, the modified quasi-dimensional combustion model can predict turbulent intensity with acceptable accuracy near spark timing.

Table — 3 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure [bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	358	172
Valve close timing [CAD]	606	402

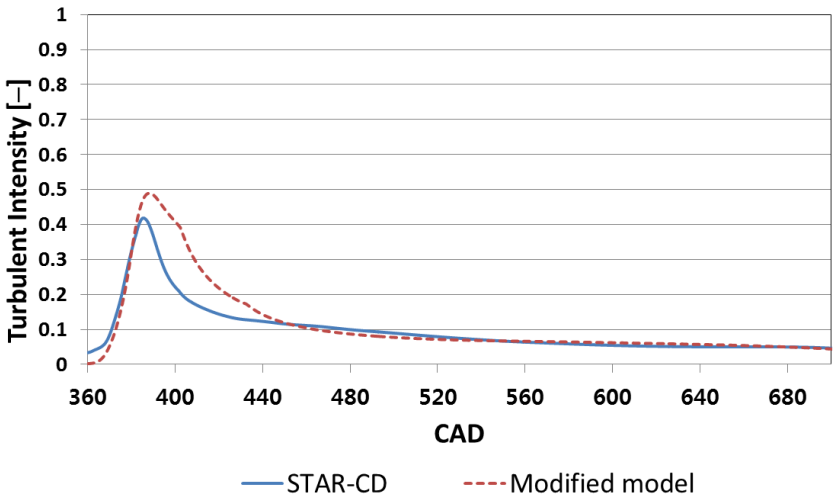


Fig.6 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 3

Table— 4 Test condition of turbulent intensity validation

rpm	2000	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	358	172
Valve close timing [CAD]	606	402

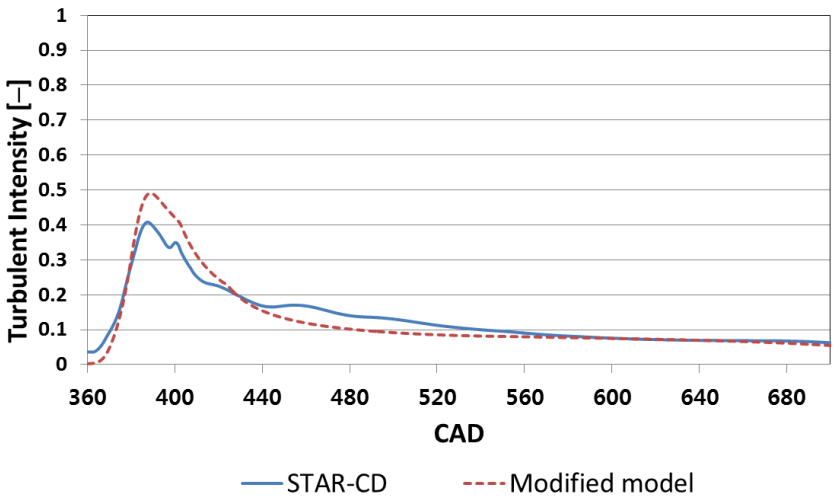


Fig.7 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 4

Table — 5 Test condition of turbulent intensity validation

rpm	2500	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	358	172
Valve close timing [CAD]	606	402

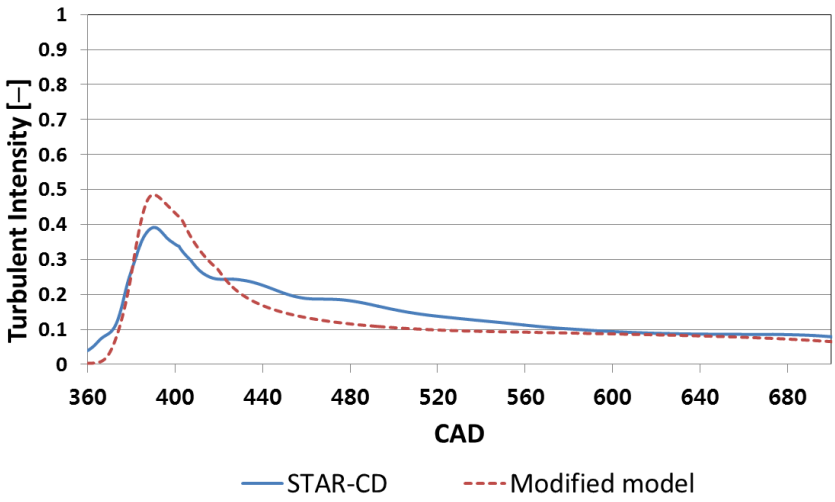


Fig.8 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 5

Table — 6 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	338	172
Valve close timing [CAD]	586	402

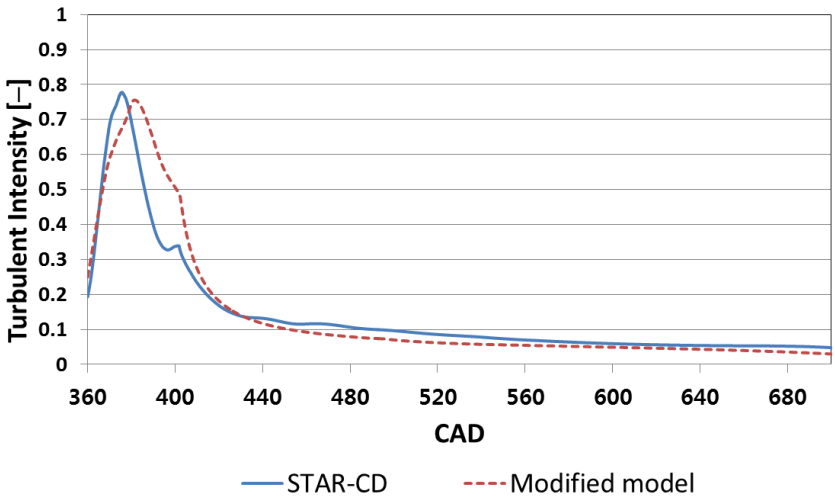


Fig.9 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 6

Table — 7 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure [bar]	1	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	383	172
Valve close timing [CAD]	655	402

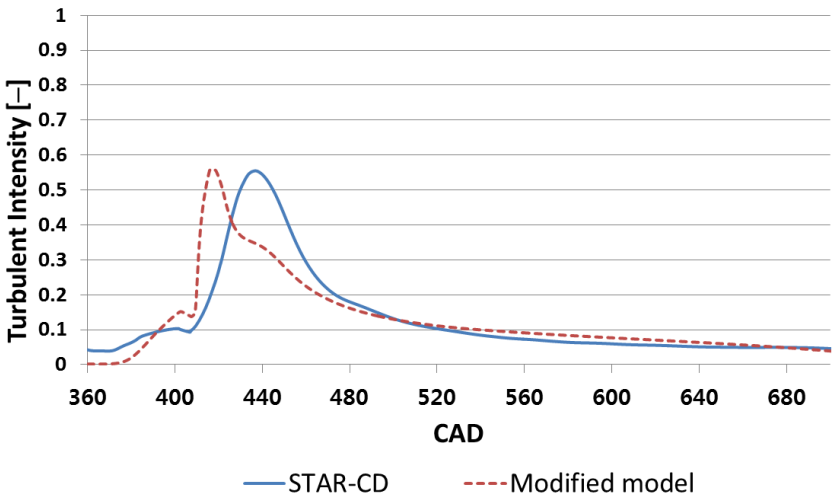


Fig.10 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 7

Table — 8 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	2	8.5
Valve open timing [CAD]	343	172
Valve close timing [CAD]	467	402

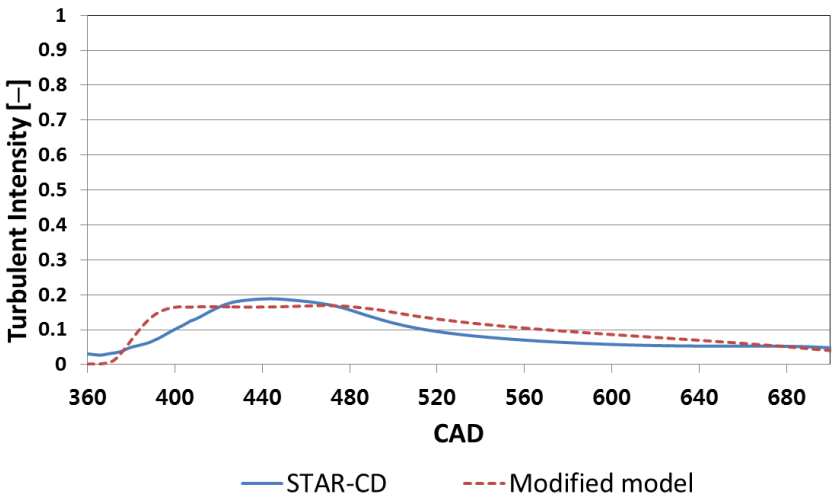


Fig.11 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 8

Table — 9 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	2	8.5
Valve open timing [CAD]	363	172
Valve close timing [CAD]	487	402

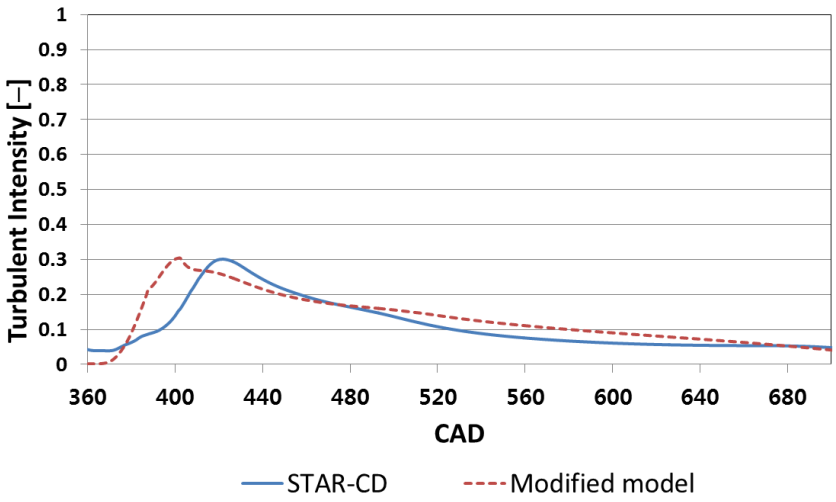


Fig.12 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 9

Table－10 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	2	8.5
Valve open timing [CAD]	383	172
Valve close timing [CAD]	507	402

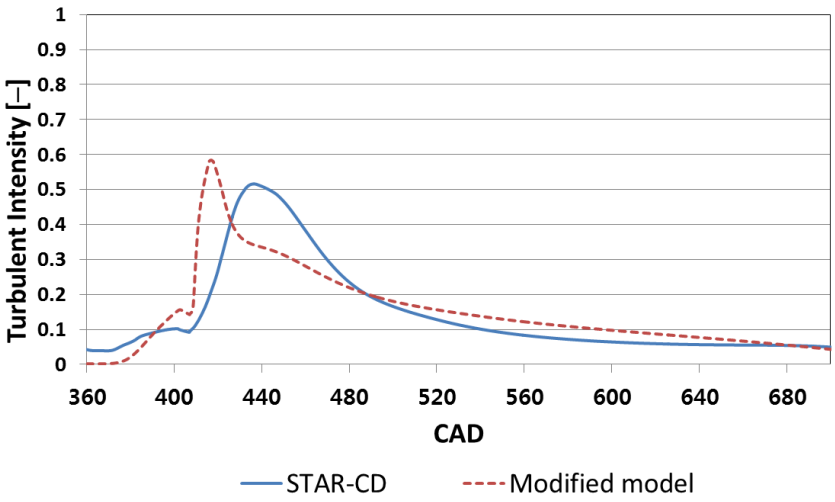


Fig.13 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 10

Table — 11 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure [bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	2	8.5
Valve open timing [CAD]	403	172
Valve close timing [CAD]	527	402

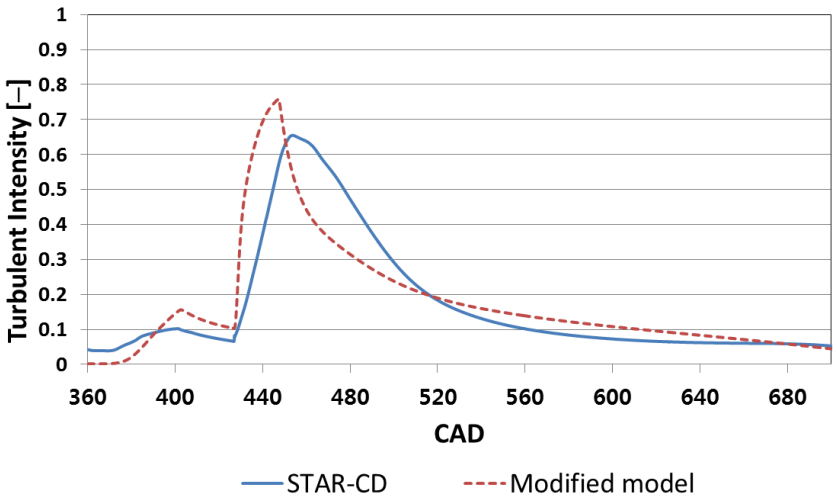


Fig.14 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 11

3.3 Residual gas fraction

Base model uses constant intake and exhaust manifold pressure as boundary conditions during gas exchange process. To predict RGF more precisely, intake and exhaust manifold pulsation was introduced to the modified quasi-dimensional combustion model. Intake and exhaust manifold pulsation data were acquired using GT-POWER under various conditions of rpm, load, valve lift, duration and timing.

As shown in figure 15, accuracy of RGF prediction is improved by introduction of pulsation effect. Test conditions are listed in table2.

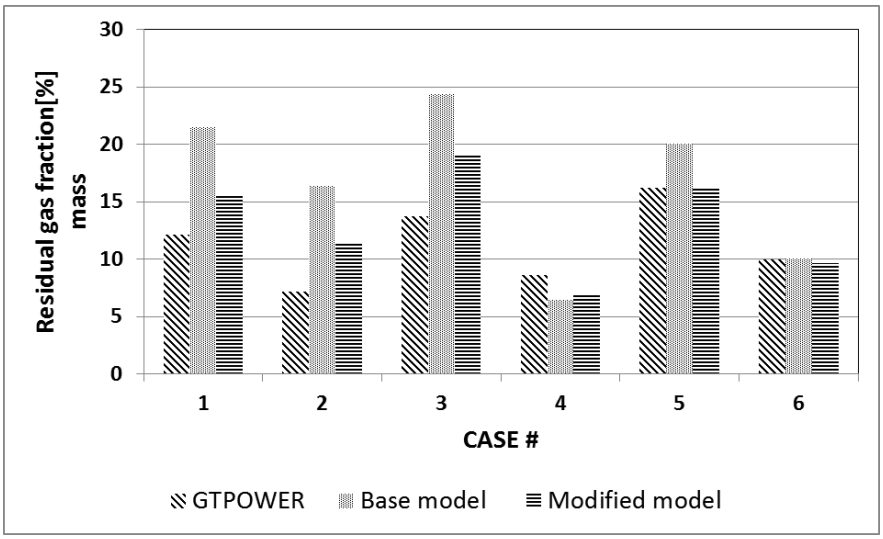


Fig.15 RGF comparison of base and modified model

Chapter 4. Model Calibration

4.1 Experimental setup

Adjustable constant C_a of developing flame speed was calibrated to match the experiments in IMEP and pressure. Since Single-cylinder engine used to get experimental data is not equipped with full VVA, two different type of intake cam shaft was used to simulate two different actuation mode of VVA. Engine specifications are listed in table 12.

Table — 12 Engine specifications

Item	Specification
Engine	Single-cylinder DOHC
Bore (mm)	86
Stroke (mm)	86
Displacement volume (cc)	499.5
Compression ratio	9.5
Injection type	Single DI(70bar)
Intake charge temperature	20℃
Coolant temperature	92℃

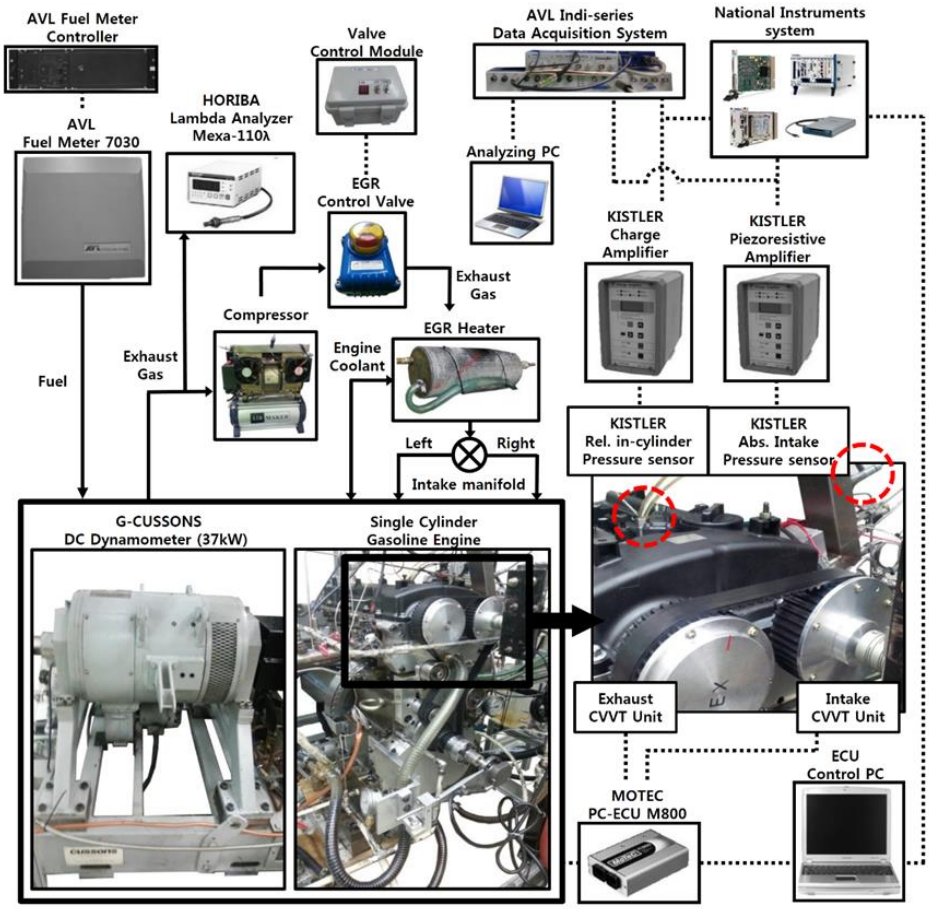


Fig.16 Schematic diagram of the experimental equipment

4.2 Model calibration

To simulate normal valve actuation mode, 9.5mm maximum lift intake cam shaft was used. And for VVA mode, 2mm maximum lift intake cam shaft was used. This VVA intake cam shaft has shorter duration than base mode intake cam shaft and its duration is 206CAD. Test conditions are listed in table 13.

Table—13 Experimental conditions of model calibration

	Case1(Base mode)		Case2(VVA mode)	
rpm	2000		1500	
IMEP [bar]	4.66		4.5	
Cam	IN	EX	IN	EX
Max Valve Lift [mm]	9.5	8.5	2	8.5
Valve open timing [CAD]	343	528	372	470
Valve close timing [CAD]	591	738	578	730

First, for base mode, adjustable constant of developing flame speed model was calibrated to match the experiment in IMEP and Pressure. Calibrated simulation result is presented in figure 17 as dashed green line and shows good agreement with experimental data.

For the same adjustable constant value, VVA mode of un-throttled operation at part load condition was validated. As shown in figure 18, pressure peak of the

modified quasi-dimensional combustion model is delayed by 3-4 CAD. But overall, simulation result showed that the modified quasi-dimensional combustion model can predict burning rate with acceptable degree of accuracy under various valve lift, duration and timing.

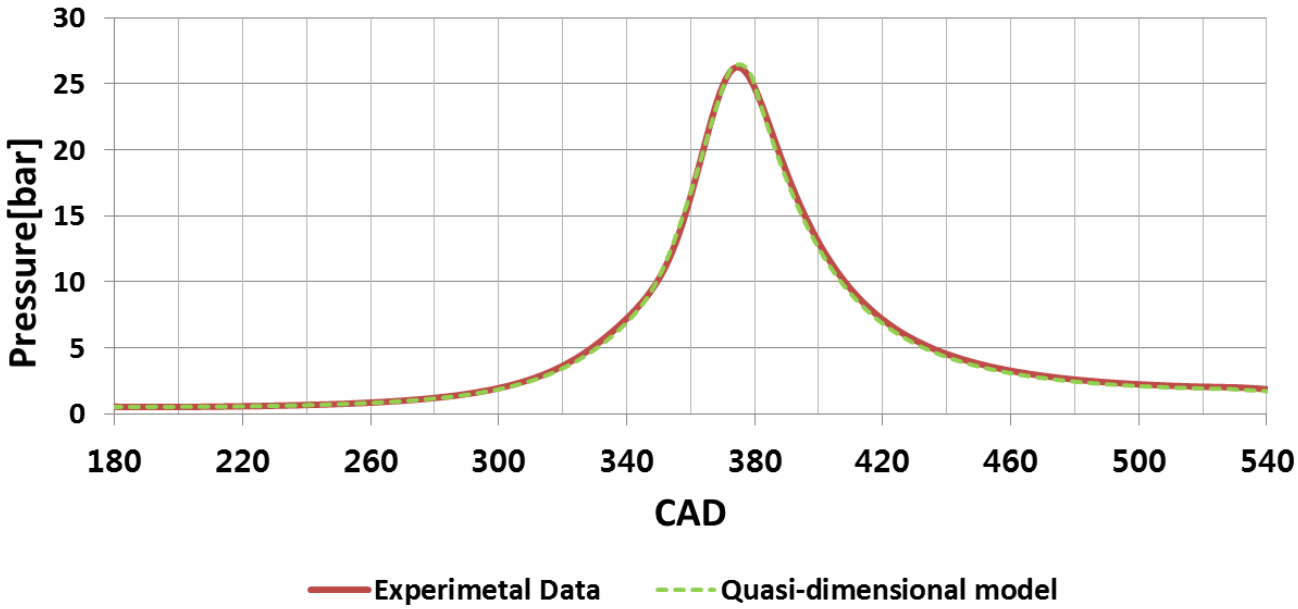


Fig.17 Calibration result under condition of case1

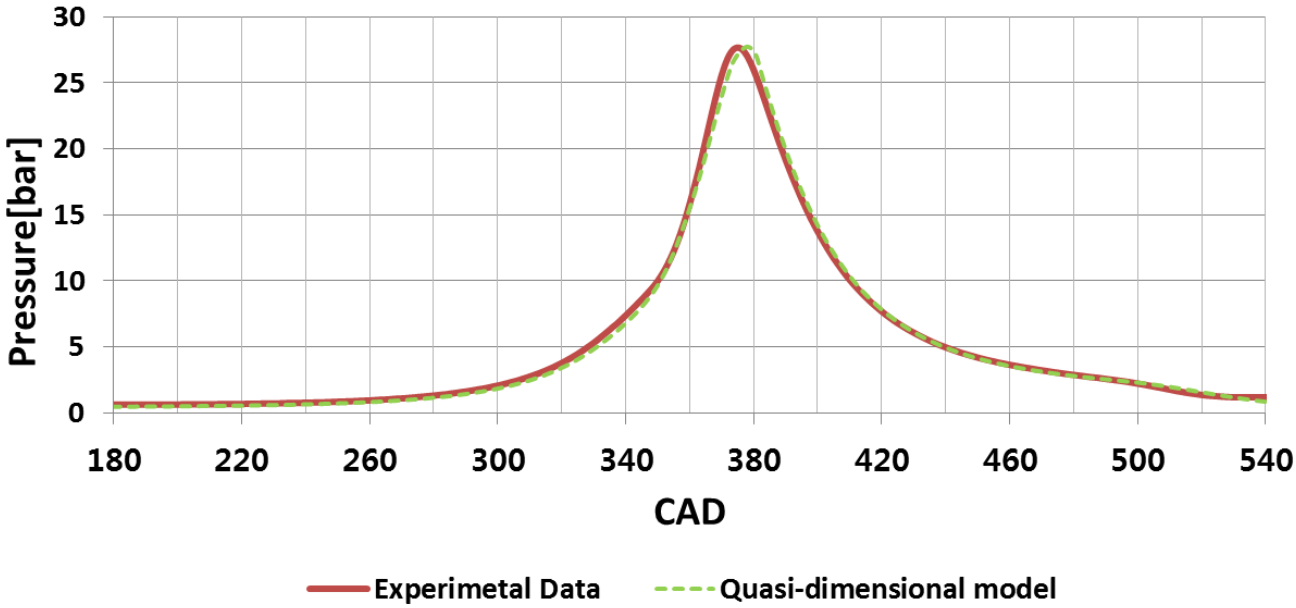


Fig.18 Validation result of calibrated model under condition of case2

4.3 Simulation results

Using the modified quasi-dimensional combustion model, the effect of VVA on fuel economy improvements was validated at part load condition. First, IMEP and ISFC of base cam actuation mode were calculated. And then, at the same rpm and IMEP, ISFC of VVA mode was calculated. As shown in table 14, under WOT condition, intake valve lift, duration and timing was changed to simulate VVA mode. Spark timing was changed to set MFB50 to aTDC7 under various intake valve lift, timing and duration.

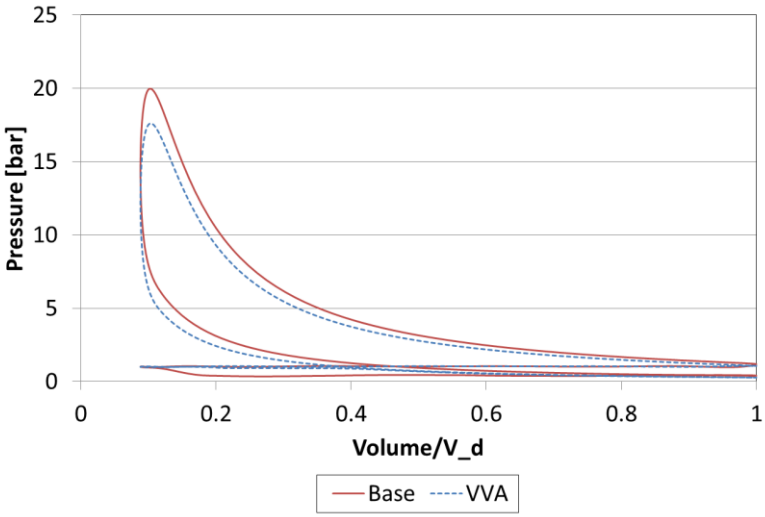
Table—14 Valve actuation conditions of base and VVA mode

	Base mode		VVA mode	
rpm	1500		1500	
IMEP[bar]	2.8		2.8	
Cam	IN	EX	IN	EX
Max Valve Lift [mm]	9.5	8.5	1.5	8.5
Valve open timing [CAD]	343	528	348	528
Valve close timing [CAD]	591	738	472	738

Table—15 ISFC comparison of base and VVA mode

RPM	IMEP [bar]	ISFC [g/kWh] (Base)	ISFC [g/kWh] (VVA)	Improvement
1500	2.8	278.5	253.6	8.94[%]

As shown in figure 19, decrease of peak pressure and reduction of pumping loss are identified under VVA condition Further fuel economy improvement will be possible, if novel optimization technique for VVA is added to quasi-dimensional combustion model.



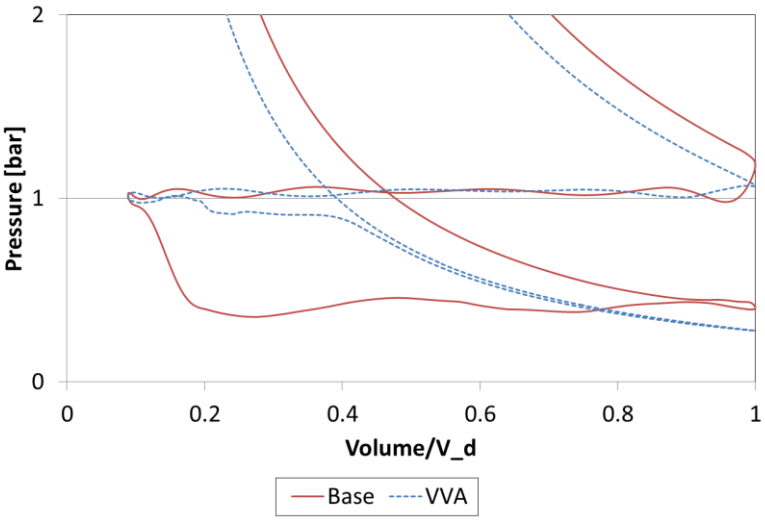


Fig.19 Pressure curve comparison of base and VVA mode at 1500rpm, 2.8bar IMEP

Chapter 5. Conclusions

In this study, to simulate SI engines equipped with variable valve actuation (VVA), the modified quasi-dimensional combustion model was suggested. Since VVA has effects on turbulent intensity and residual gas fraction which determine burning rate, it is important to predict burning rate precisely under various valve lift, duration and timing for VVA engine simulation.

STAR-CD and GT-POWER were used to obtain base data of turbulent intensity and residual gas fraction under various valve lift, duration and timing. Using these base data, the turbulent flow energy model and boundary conditions of gas exchange process were modified to improve the accuracy of turbulent intensity and residual gas fraction prediction.

A zero-dimensional energy cascade model was used to describe turbulent flow energy in a quasi-dimensional combustion model. First, to improve accuracy of turbulent intensity prediction, integral length scale was modified. According to the base data from STAR-CD, integral length scale did not vary significantly as intake pressure, rpm, valve lift, duration and timing change and it seems that integral length scale is mainly governed by combustion chamber geometry. Then, a parameter to control the turbulent kinetic energy dissipation rate was added to the zero-

dimensional energy cascade model. This parameter is a function of valve lift, duration and timing. Using this parameter and adjustable constant of turbulence energy production rate together, the modified quasi-dimensional combustion model can reflect the effect of VVA on turbulent intensity.

Residual gas fraction affects the burning rate indirectly by changing laminar flame speed. Using GT-POWER, pulsation data of intake and exhaust manifold were calculated and it was applied to the modified quasi-dimensional combustion model. By this method, the modified quasi-dimensional combustion model can predict residual gas fraction much precisely.

The newest laminar flame speed correlation was also introduced to the modified quasi-dimensional combustion model. This laminar flame speed correlation contains a parameter which considers the effect of equivalence ratio variation on laminar flame speed. To describe the early stage of flame propagation, H. Werweg's flame speed correlation was introduced.

Model calibration based on experimental data was carried out. Adjustable constant of developing flame model was calibrated under normal valve actuation conditions of intake valve lift, duration and timing. Then, with the same adjustable constant, the modified quasi-dimensional combustion model was applied to un-throttled operation at the part load condition. Simulation results showed that the modified

quasi-dimensional combustion model can predict burning rate with acceptable degree of accuracy under various valve lift, duration and timing.

Finally, using quasi-dimensional model developed in this study, the potential of fuel economy improvement at the part load condition was investigated. Decrease of peak pressure and reduction of pumping loss were identified under VVA engine operating conditions. Further fuel economy improvement will be possible, if novel optimization technique for VVA is added to the quasi-dimensional combustion model.

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가변 밸브 기구가 장착된 가솔린 엔진에 적용 가능한 유사차원 연소모델 개발

서울대학교 대학원
기계항공공학부
김 진 희

요 약

가변밸브기술(VVA)은 다양한 엔진 작동 영역에서 연비와 출력을 향상시킬 수 있는 차세대 엔진 기술로 각광받고 있다. 하지만 가변밸브는 밸브 프로파일 생성 자유도가 크고 이에 영향을 받는 변수가 매우 많기 때문에 실험적 방법으로 최적화 하는데 어려움이 있다. 동일한 이유로 과도한 계산시간이 소요되는 삼차원 엔진 시뮬레이션 기법으로도 밸브 프로파일을 최적화하기가 힘들다. 따라서 본 연구에서는 가변밸브가 장착된 가솔린 엔진의 성능을 빠르게 시뮬레이션 할 수 있는 유사차원 연소모델을 개발하였다.

가변 밸브는 난류강도, 잔류가스 등에 영향을 주어 연소율을 변화시키는데 연소율은 엔진 성능 예측에 있어 가장 중요한 인자이다. 본 연구에서 진행된 유사차원 연소모델 개발 과정은 이러한 가변밸브의 영향이 유사차원 시뮬레이션 결과에 정확히 반영되도록 하는데 주안점을 두었다.

3차원 엔진 해석프로그램인 STAR-CD와 1차원 엔진 해석프로그램 GT-POWER를 이용하여 다양한 밸브 작동모드에서 난류 강도와 잔류 가스데이터를 확보하였다. 이를 기반으로 유사차원 연소모델의 난류에너지 관계식과 가스 교환모델을 수정하여 난류강도와 잔류가스가 다양한 밸브 작동모드에서 정확히 예측될 수 있도록 하였다.

엔진 실험 데이터를 이용하여 화염 속도 모델의 상수를 보정하였고 저부하, un-throttled 작동 조건에 보정된 유사차원 연소모델을 적용하여 모델의 정확도를 평가하였다. 마지막으로, 본 연구에서 개발된 유사차원 연소모델을 저속, 저부하 엔진 작동 조건에 적용, 연비 개선 효과를 확인 하였다.

주요어: 연비, 가변밸브, 유사차원 연소모델, 난류강도, 잔류가스, 연소율

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Development of Quasi-dimensional Combustion Model of SI Engines Equipped with Variable Valve Actuation

가변밸브가 장착된 가솔린 엔진의 성능 예측을 위한 유사차원 연소모델 개발

2014년 2월

서울대학교 대학원

기계항공공학부

김 진 희

Development of Quasi-dimensional Combustion Model of SI Engines Equipped with Variable Valve Actuation

지도교수 민 경 덕

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Development of Quasi-dimensional Combustion Model of SI Engines Equipped with Variable Valve Actuation

Jinhee Kim
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Abstract

Variable valve actuation technology has great potential of improving both fuel economy and performance of SI engines. However, due to numerous valve profiles and excessive calculation time, optimization of valve actuation under various engine operation conditions can hardly be achieved by experimental method or 3D engine simulation tools. Therefore, in this work, a quasi-dimensional combustion model for fast simulation of SI engines equipped with VVA was developed.

MIT quasi-dimensional combustion model was used as the base code. It was modified to reflect the effect of VVA on burning rate prediction which is mainly affected by turbulent intensity and residual gas fraction. STAR-CD and GT-

POWER were used to obtain base data of turbulent intensity and residual gas fraction under various valve lift, duration and timing. Using these base data, the turbulent flow energy model and boundary conditions of gas exchange process were modified to improve the accuracy of turbulent intensity and residual gas fraction prediction.

Model calibration based on experimental data was carried out. Simulation results showed that the modified model can predict burning rate with acceptable degree of accuracy under various valve lift, duration and timing. Finally, using the modified quasi-dimensional combustion model, the effect of VVA on fuel economy improvements was verified.

Keywords: Fuel economy, Quasi-dimensional combustion model, Variable valve actuation, Turbulent Intensity, Residual Gas Fraction, Burning rate

Student Number : 2012-20660

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Acronym

VVA	Variable Valve Actuation
IVO	Intake Valve Open
IVC	Intake Valve Close
EVO	Exhaust Valve Open
EVC	Exhaust Valve Close
ISFC	Indicated Specific Fuel Consumption
BSFC	Brake Specific Fuel Consumption
RGF	Residual Gas fraction
WOT	Wide Open Throttle
BMEP	Brake Mean Effective Pressure
IMEP	Indicated Mean Effective Pressure
CAD	Crank Angle Degree

Chapter 1. Introduction

1.1 Background and Literature Reviews

Due to soaring of crude oil price and global warming issues, major automobile producing and consuming countries have tightened regulation on fuel economy. In the USA, CAFÉ (corporate average fuel economy) regulation mandates average fuel mileage of 37.8mpg for passenger car and light duty truck by 2016[2]. Furthermore, this fuel economy requirement will be doubled by 2025. EU has regulated fuel economy not by direct standard on fuel economy but by CO2 emission standard. CO2 emission standard has been decreased over the time from 186g/km by 1995, to 140g/km by 2008, and to 95g/km by 2020.

To encounter these stringent fuel economy regulation, major vehicle manufacturers have developed highly energy efficient and zero emission vehicles such as hybrid vehicles, high efficiency CI engines, electric vehicles and fuel cell electric vehicles. Eventually, gasoline and diesel fuelled vehicles will be replaced by electric or fuel cell electric vehicles. However, production cost of electrically-powered vehicles is much higher than that of gasoline or diesel fuelled vehicles and infrastructure for recharging or refueling electrically-powered vehicles is not established enough to fully commercialize these kinds of vehicles[3]. Thus, traditional SI and CI engines

will be used as a main power source for at least another decade.

To increase fuel economy of SI engines, many new technologies such as down-sizing, gasoline direct injection, homogeneous charge compressed ignition, variable valve actuation, etc. have been developed. Among these, variable valve actuation (VVA) technology has regular cost effectiveness but has the potential of improving fuel economy, maximum torque and power by optimizing intake and exhaust valve lift, duration and timing[4].

In addition, VVA combined with other combustion technologies can create a synergy effect. Through VVA with spray-guided direct injection, charge stratified combustion can be realized[5]. Furthermore, VVA can be used as a control device of homogeneous charge compressed ignition[6].

Many preceding researches about the effects of VVA on engine performance have been carried out. SI engines with electro-hydraulic fully variable valve reduced CO2 emissions up to 10%, and achieved 10% more power and 15% more torque in the lower speed range [7]. 7% fuel economy improvement could be achieved by un-throttled engine operation with variable intake valve lift, duration and timing [8]. Using two-step variable valve actuation not fully variable valve actuation, fuel economy could be improved by 8% [9].

At the part load condition, fuel economy is mainly deteriorated by pumping loss[10] caused by pressure difference between intake and exhaust processes. For SI engines equipped with VVA, inducted air mass can be controlled not by throttle body but by intake valve lift, duration and timing. Therefore, pressure drop of inducted air across the throttle body can be eliminated by un-throttled engine operation. This leads to reduction of pumping loss. In addition, at high engine rpm and load, more torque and power can be achieved by optimizing valve lift, duration and timing to induct more air into the cylinder[11].

As mentioned above, not being confined to fuel economy improvement, VVA shows the great potential for improving overall engine performance. Therefore, many preceding researches have been conducted to optimize valve profile of VVA engines under different rpm and load conditions. However, optimization of valve profile has not been completed due to numerous combination of valve profile which is determined by lift, duration and timing of intake and exhaust valves. Furthermore, VVA has effects on not only inducted air mass but also turbulent intensity[12] and residual gas fraction[13] which changes burning rate drastically.

This means that optimization of VVA engines through experimental methods entails considerable expense and time. It is impossible to discover all the potential of VVA through experimental method only.

Due to vast amount of test case, commercial 3D engine simulation program needs excessive calculation time to optimize VVA. 1D engine simulation tools which have much shorter computing time than 3D simulation tools, use Wiebe function to model the combustion in SI engines. Since Wiebe function is a semi empirical law which does not consider physics related to burning rate, 1D simulation tools cannot predict burning rate variation precisely under various valve lift, duration and timing.

Therefore, it is necessary to develop a fast engine simulation tool which can predict the effect of VVA on SI engine performance.

1.2 Objectives

In this study, quasi-dimensional combustion model for SI engines equipped with VVA was developed. Model development process mainly focused on improvement of accuracy of burning rate prediction.

First, a laminar flame speed correlation which consider effect of diluent mixture on laminar flame speed and turbulent flame speed which contains exponential terms to describe early stage of flame development were introduced to quasi-dimensional combustion model.

And then, other important parameters for burning rate prediction which are influenced by VVA, such as turbulent intensity, residual gas fraction were investigated and modified. To estimate turbulent intensity and residual gas fraction under various valve lift, duration and timing, commercial 3D CFD program STAR-CD and 1D engine simulation program GT-POWER were used. Base on these results, turbulent intensity, residual gas fraction of quasi-dimensional combustion model were modified to reflect the effects of VVA on engine performance.

Adjustable constants were calibrated to make quasi-dimensional combustion model be applicable to VVA engine simulation. And potential of VVA on fuel economy improvement was validated.

Chapter 2. Investigation into base model

2.1 Overview of base model

Quasi-dimensional combustion model which had been developed by Poulos[14] was used as base model. Base model assumes flame propagates as a sphere centered at spark plug electrodes. So, it needs geometrical data about interaction of flame front with combustion chamber. Using this geometrical data, combustion chamber is divided into burned and unburned zone. To consider heat transfer effect, burned zone is composed of adiabatic core and thermal boundary layer. And this model integrates differential equations with time dimension which describe basic thermodynamics, combustion reaction, gas exchange process, turbulent energy cascade, NOx formation etc. Using these differential equations, air mass inducted, residual gas fraction, in-cylinder pressure, temperature, heat transfer rate, NOx, etc. can be calculated.

For simulation of SI engine with VVA, accuracy of burning rate prediction is the most significant part of simulation. Burning rate is mainly affected by laminar flame speed and turbulent intensity. As mentioned above, base model contains turbulent energy cascade and gas exchange model which determine turbulent intensity and

residual gas fraction. So, it can reflect the effect of VVA on burning rate under various rpms and loads.

2.2 Validation of base model

First, it needed to investigate the quantitative variation of burning rate under various valve lift, duration and timing, because base quasi-dimensional model had not been applied to VVA engine simulation. So, turbulent intensity and residual gas fraction of base quasi-dimensional model which determine burning rate were validated.

Turbulent intensity of base quasi-dimensional model was investigated at the condition listed below.

Table 1 — Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure [bar]	1	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	383	172
Valve close timing [CAD]	655	402

As shown in figure1, turbulent intensity of base model differs from STAR-CD simulation data acquired at the same condition. At the early stage of intake process,

turbulent intensity of base quasi-dimensional model is underestimated and base quasi-dimensional model is unable to reflect the effect of chocked flow which makes peak point of STAR-CD simulation result near 440 CAD. On the contrary to this, at the end of compression process, turbulent intensity of base quasi-dimensional model is overestimated.

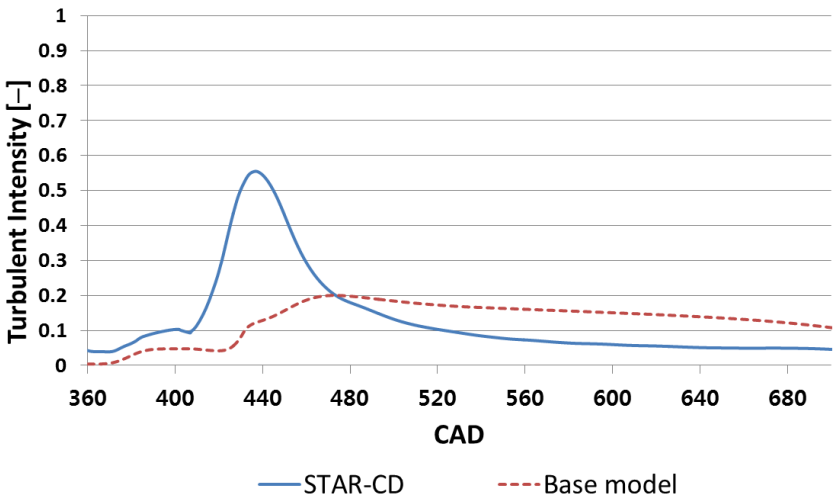


Fig.1 Turbulent intensity of base quasi-dimensional combustion model at condition listed in table 1

Residual gas fraction was validated. Test conditions are listed below

Table 2 — Test conditions of RGF validation

case #	rpm	Max Lift [mm]		IVO [CAD]	IVC [CAD]	EVO [CAD]	EVC [CAD]
		In cam	Ex cam				
1	2000	9	8	343	591	528	738
2	1000	9	8	354	602	530	740
3	1500	9	8	350	598	530	740
4	1500	2	8	372	578	470	730
5	1500	2	8	300	506	470	700
6	1500	2	8	310	516	470	700

As indicated in figure 2, base model calculation results show similar RGF trends of GT-POWER simulation results. However, RGF of base quasi-dimensional combustion model tends to be overestimated for 9 mm intake valve lift.

In summary, base model has a good potential for VVA engine simulation. But to improve the accuracy of burning rate prediction, parameters related to burning rate should be modified.

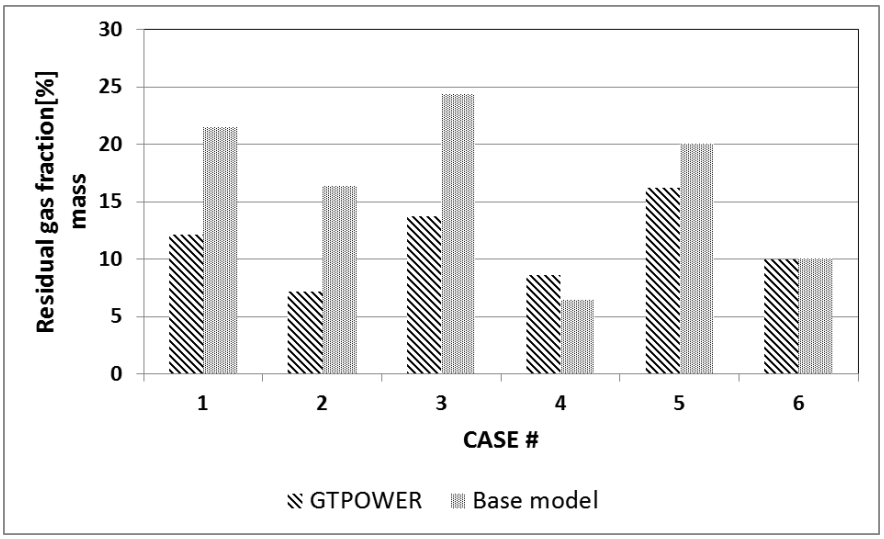


Fig.2 Residual gas fraction comparison of GTPOWER and base quasi-dimensional combustion model at conditions listed in table 2

Chapter 3. Model modification

3.1 Combustion model

Turbulent entrainment model is used to describe turbulent combustion process in quasi-dimensional combustion model. This combustion model had been formulated by Blizzard and Keck[15]. And then turbulent structure during combustion was modified by Tabazynski[16].

In this combustion model, it is assumed that combustion process consists of entrainment of unburned gas and burning of unburned gas already entrained within flame front area. As flame propagates, unburned gas is entrained into spherically shaped flame frontal area. Mass entrainment rate is expressed as below.

$$\frac{dm_e}{dt} = \rho_u A_f S_t$$

$$S_t = u' + S_l$$

$\mathbf{m}$	Mass entrained into the flame front
A_f	Flame frontal area
ρ_u	Density of unburned gas
S_t	Turbulent flame speed

$\boldsymbol{u'}$ Turbulent intensity

$\boldsymbol{S_l}$ Laminar flame speed

It is also assumed that within taylor micro scale, flame propagates at laminar flame speed. So, combustion characteristic time and burning rate are expressed as below.

$$\boldsymbol{\tau_b = \frac{\lambda}{S_l}}$$

$$\boldsymbol{\frac{dm_b}{dt} = \frac{m_e - m_b}{\tau_b}}$$

$\boldsymbol{\lambda}$ Taylor micro scale

In modified quasi-dimensional combustion model, it is assumed that there is no significant difference between turbulent entrainment rate and burning rate[17]. So, burning rate was set equal to turbulent entrainment rate.

$$\boldsymbol{\frac{dm_e}{dt} = \frac{dm_b}{dt}}$$

Turbulent flame speed of base model is represented simply as the sum of laminar flame speed and turbulent intensity. Because this correlation ignores the effect of ignition delay, in-cylinder pressure tends to be overestimated at the early stage of combustion[18]. So, developing flame speed proposed by H.Werweg[19] which

contains exponential terms to describe the early stage of flame development was introduced to the modified quasi-dimensional combustion model.

$$\frac{S_t}{S_l} = 1 + C_a \left[\frac{u'}{u'+s_l}\right]^{\frac{1}{2}} \cdot \left[1 - e^{-\frac{s_l}{u'+s_l}\cdot\frac{t}{\tau}}\right]^{\frac{1}{2}} \cdot \left[1 - e^{\frac{t}{\tau}}\right]^{\frac{1}{2}} \cdot \left(\frac{u'}{s_l}\right)^{\frac{5}{6}}$$

$\mathbf{t}$	Elapsed time from spark timing
C_a	Adjustable constant

Recommended value of $\mathbf{t}$ is 1.5 ms[20] and C_a should be calibrated to match experimental data.

Laminar flame speed correlation of Pae et al[21], was introduced to the modified quasi-dimensional combustion model also. This laminar flame speed correlation contains a parameter which reflects equivalence ratio variation on laminar flame speed.

3.2 Turbulent intensity

Zero-dimensional energy cascade model[22] is used to describe turbulent flow energy in base quasi-dimensional combustion model. During the intake process, fresh charge with fuel and residual gas is inducted into the cylinder and it also carries

mean kinetic energy. Mean kinetic energy is transformed to turbulent kinetic energy through a turbulent dissipation process. At the same time, turbulent kinetic energy is transformed to thermal energy through a viscous dissipation process. It is assumed that in-cylinder turbulent flow is homogeneous and isotropic.

Mean kinetic energy and turbulent kinetic energy can be calculated by the following equations.

$$\mathbf{K} = \frac{1}{2} m V_i^{\,2}$$

$$\mathbf{k} = \frac{1}{2} m u'^{2}$$

K	Mean kinetic energy
k	Turbulent kinetic energy
m	Trapped mass inside cylinder
$\boldsymbol{V_i}$	Mean flow velocity into the cylinder
$\boldsymbol{u'}$	Turbulent intensity

Rate of change of mean kinetic energy and turbulent kinetic energy can be expressed as follows,

$$\frac{\mathbf{dK}}{\mathbf{dt}} = \frac{\mathbf{1}}{\mathbf{2}}\dot{\boldsymbol{m}}_{\boldsymbol{i}}{V_{\boldsymbol{i}}}^{\mathbf{2}} - \boldsymbol{P} - \boldsymbol{K}\frac{\dot{\boldsymbol{m}}_{\boldsymbol{e}}}{\boldsymbol{m}}$$

$$\frac{\mathbf{dk}}{\mathbf{dt}} = \boldsymbol{P} - \boldsymbol{m\varepsilon} - \boldsymbol{k}\frac{\dot{\boldsymbol{m}}_{\boldsymbol{e}}}{\boldsymbol{m}}$$

$$\mathbf{P} = \mathbf{0.3307}\boldsymbol{C_\beta}\left(\frac{\boldsymbol{K}}{\boldsymbol{L}}\right)\left(\frac{\boldsymbol{k}}{\boldsymbol{m}}\right)^{\frac{\mathbf{1}}{\mathbf{2}}}$$

$$\boldsymbol{\varepsilon} = \frac{(\mathbf{2}\boldsymbol{k}/\mathbf{3}\boldsymbol{m})^{\frac{\mathbf{1}}{\mathbf{2}}}}{\boldsymbol{l}}$$

$\dot{\boldsymbol{m}}_{\boldsymbol{i}}$	Mass induction rate through the intake valve
$\dot{\boldsymbol{m}}_{\boldsymbol{e}}$	Mass outflow rate through the exhaust valve
$\boldsymbol{\epsilon}$	Turbulence dissipation rate per unit mass
$\mathbf{P}$	Turbulence energy production rate
$\boldsymbol{C_\beta}$	Adjustable constant
$\mathbf{L} = \mathbf{l}$	Integral length scale

3.2.1 Integral length scale

It is assumed that in-cylinder turbulence production process is similar to that of flat plate turbulent boundary layer. L is geometrical length scale and l is integral length scale. In this model, it is assumed that geometrical length scale is equal to integral

length scale. And its value is equal to stroke/2 during intake, compression and exhaust process.

For combustion process, by the rapid distortion theory, integral length scale and turbulent intensity are conserved. Integral length scale and turbulent intensity at spark timing can be determined by turbulent energy cascade model. Integral length scale and turbulent intensity can be described as (Subscript s means spark timing)

$$\frac{L}{L_s} = \left(\frac{\rho_{u,s}}{\rho_u}\right)^{\frac{1}{3}}$$

$$\frac{u'}{u'_s} = \left(\frac{\rho_u}{\rho_{u,s}}\right)^{\frac{1}{3}}$$

As mentioned above, integral length scale is determined by the chamber geometry. But because of this, integral length scale is overestimated[23] and the rate of change of the mean kinetic and turbulent kinetic energy is underestimated during intake and compression process.

Under various conditions of intake pressure, rpm, valve lift, duration and timing, more than 50 cases of integral length scale data were acquired using STAR-CD. These data shows that integral length scale is influenced by intake pressure, rpm, valve lift, duration and timing.

As indicated in figure 3, integral length scale does not vary significantly as intake pressure, rpm, valve lift, duration and timing change and it seems that integral length scale is mainly governed by combustion chamber geometry.

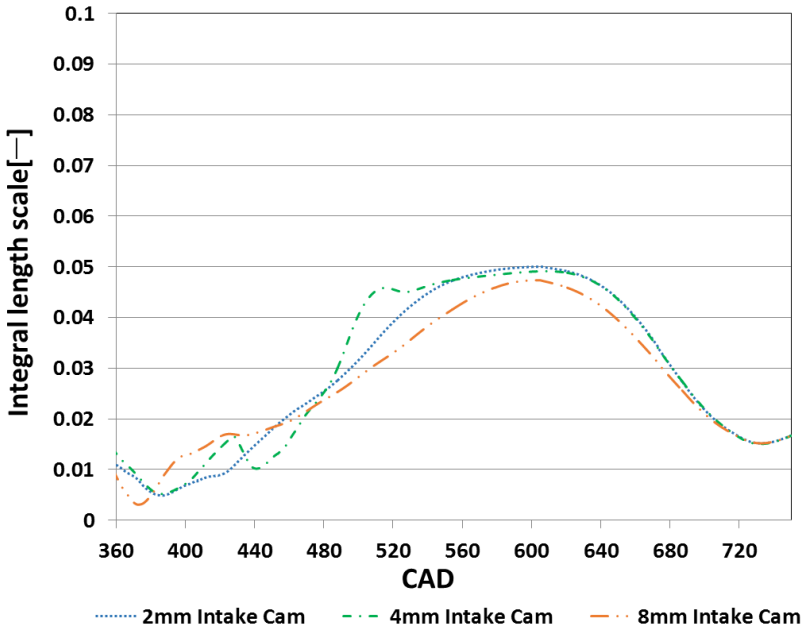


Fig.3 Integral length scale data obtained from STAR-CD under various rpm, valve actuation conditions

As shown in figure 4, integral length scale of base quasi-dimensional combustion model is approximately ten times larger than that of STAR CD results. So, integral length scale of the modified quasi-dimensional combustion model was changed to have same length order of STAR CD results.

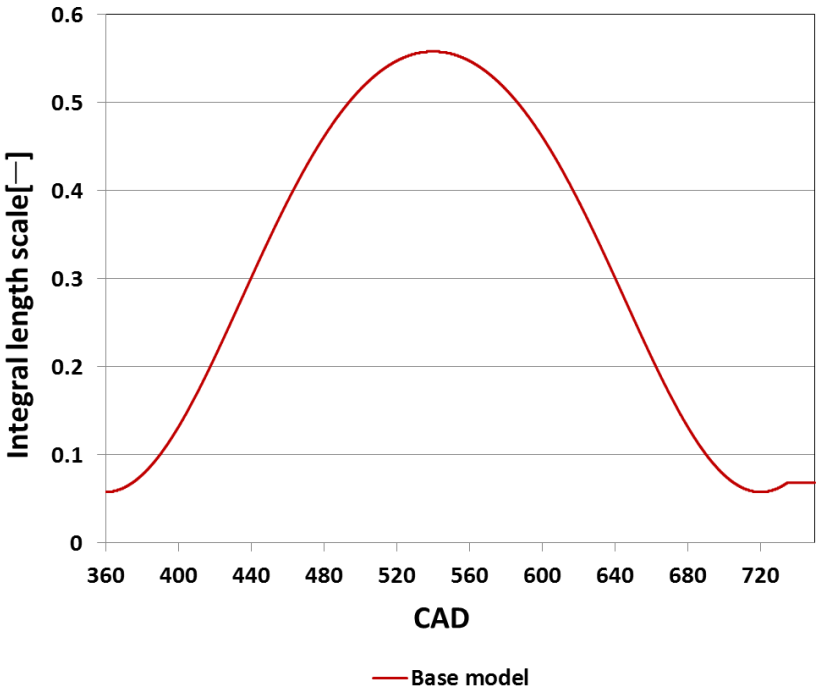


Fig.4 Integral length scale of base quasi-dimensional combustion model

In figure 5, modified integral length scale is represented. It is governed by combustion chamber geometry only and its maximum and minimum value is limited near TDC and BDC.

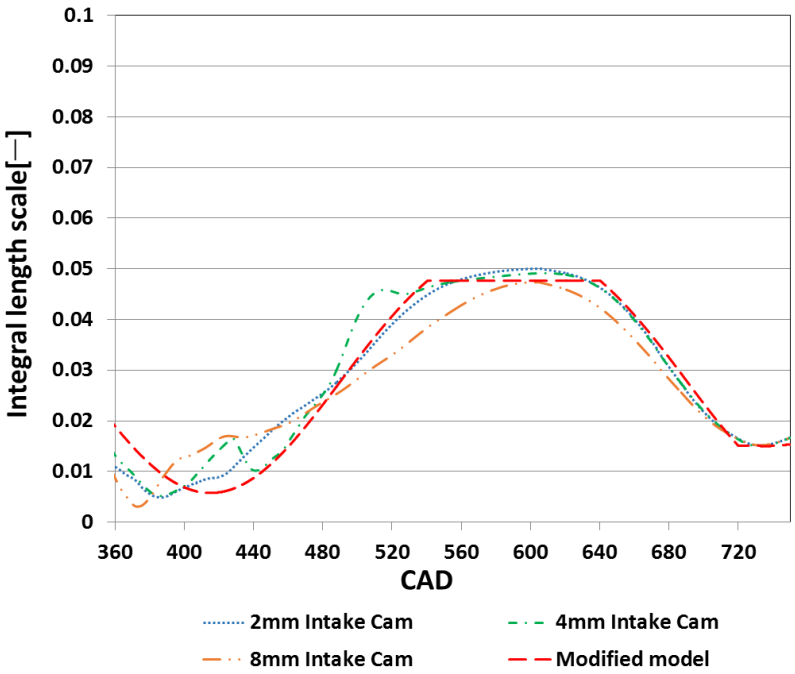


Fig.5 Integral length scale comparison of STAR-CD and the modified quasi-dimensional combustion model

3.2.2 Turbulent dissipation parameter

For simulation of SI engines with classic cam train, turbulent intensity is calibrated by adjustable constant C_β which is determined uniquely by combustion chamber geometry. But for VVA engine simulation, it is impossible to calibrate turbulent intensity through C_β only.

As shown in figure 1, for base quasi-dimensional combustion model, decreasing rate of turbulence intensity is too small at the end of intake process and its peak value during intake process is overestimated.

Therefore, to simulate VVA engine, another parameter which can control rate of change of turbulent intensity was introduced to the modified quasi-dimensional combustion model. This parameter is a function of rpm, valve lift, timing and duration.

$$\frac{\mathbf{dk}}{\mathbf{dt}} = \boldsymbol{P} - \boldsymbol{m}\boldsymbol{C_\gamma}\boldsymbol{\varepsilon} - \boldsymbol{k}\frac{\dot{\boldsymbol{m}}_{\boldsymbol{e}}}{\boldsymbol{m}}$$

$\boldsymbol{C_\gamma}$ Turbulent kinetic energy dissipation rate parameter

3.2.3 Turbulent intensity of the modified quasi-dimensional combustion model

After turbulence flow energy model modification, turbulent intensity of modified quasi-dimensional combustion model was validated. First, at base intake cam lift, timing and duration, rpm was changed from 1500 to 2500 by 500rpm. As shown in figures 6~8, simulation results of modified model show good agreement with 3D simulation results. And then, at base intake cam lift and duration, IVO was changed from 338 to 383 CAD. In figures 9,6,10, it is shown that modified model can reflect the effect of valve timing sweep on turbulent intensity.

Finally, Changing IVO from 338 to 403 CAD, turbulent intensity of VVA mode with 2mm cam lift and 124 CAD duration was validated. Figures 8~ 11 show that under VVA mode, the modified quasi-dimensional combustion model can predict turbulent intensity with acceptable accuracy near spark timing.

Table — 3 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure [bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	358	172
Valve close timing [CAD]	606	402

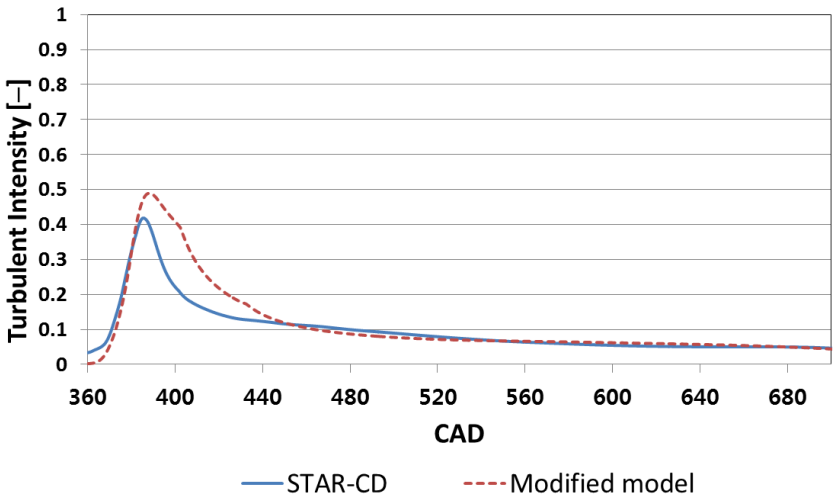


Fig.6 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 3

Table— 4 Test condition of turbulent intensity validation

rpm	2000	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	358	172
Valve close timing [CAD]	606	402

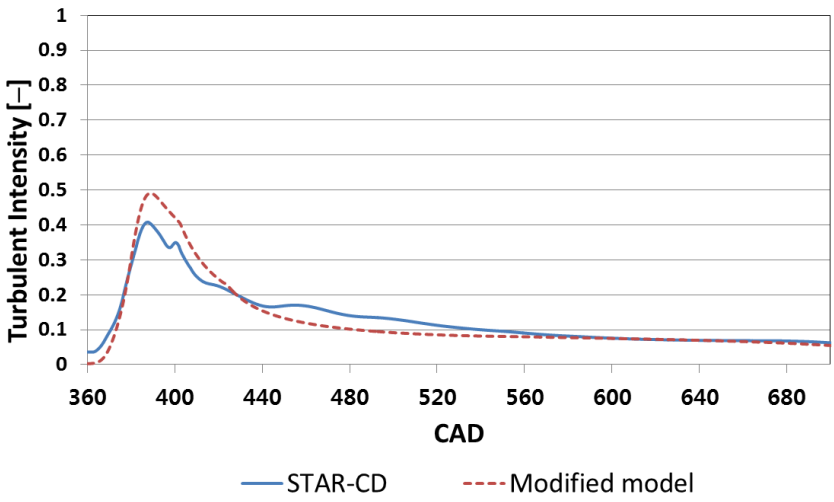


Fig.7 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 4

Table — 5 Test condition of turbulent intensity validation

rpm	2500	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	358	172
Valve close timing [CAD]	606	402

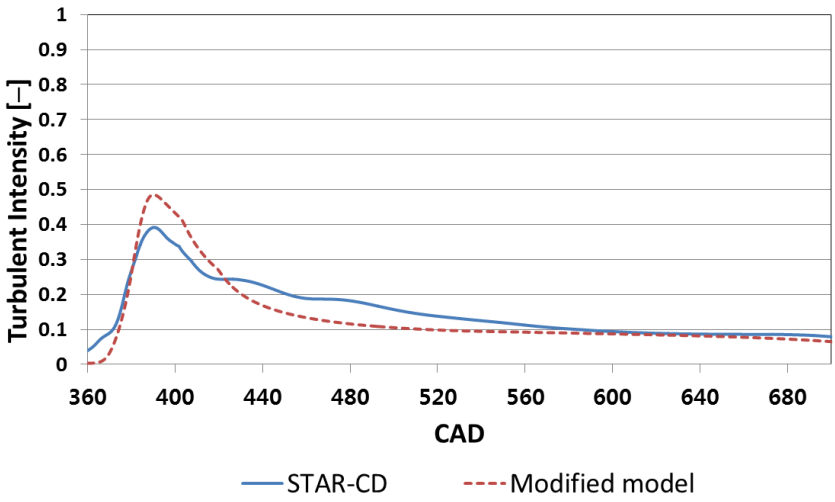


Fig.8 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 5

Table — 6 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	338	172
Valve close timing [CAD]	586	402

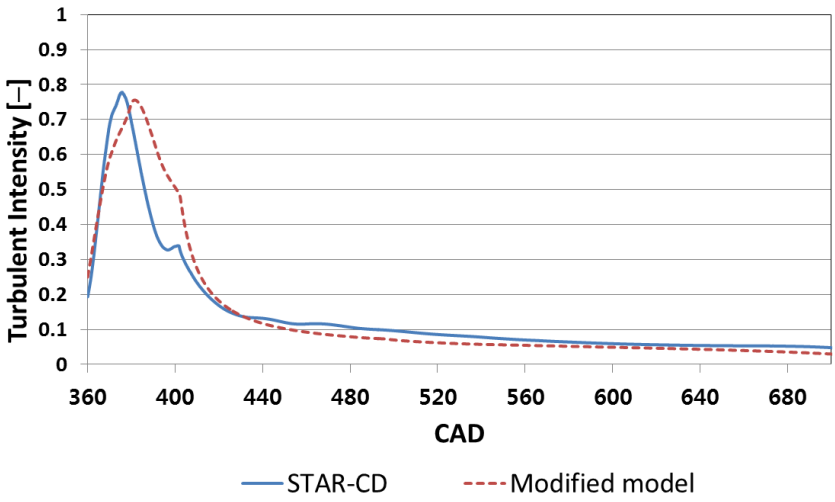


Fig.9 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 6

Table — 7 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure [bar]	1	
Cam	IN	EX
Max Valve Lift [mm]	9.5	8.5
Valve open timing [CAD]	383	172
Valve close timing [CAD]	655	402

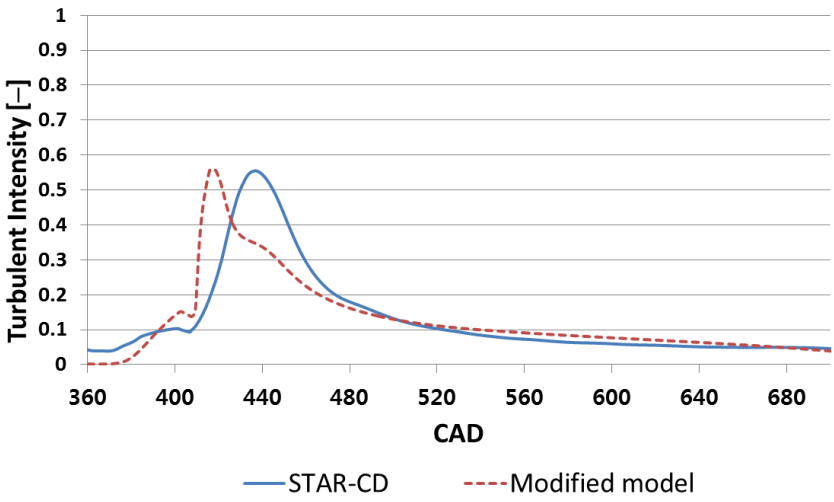


Fig.10 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 7

Table — 8 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	2	8.5
Valve open timing [CAD]	343	172
Valve close timing [CAD]	467	402

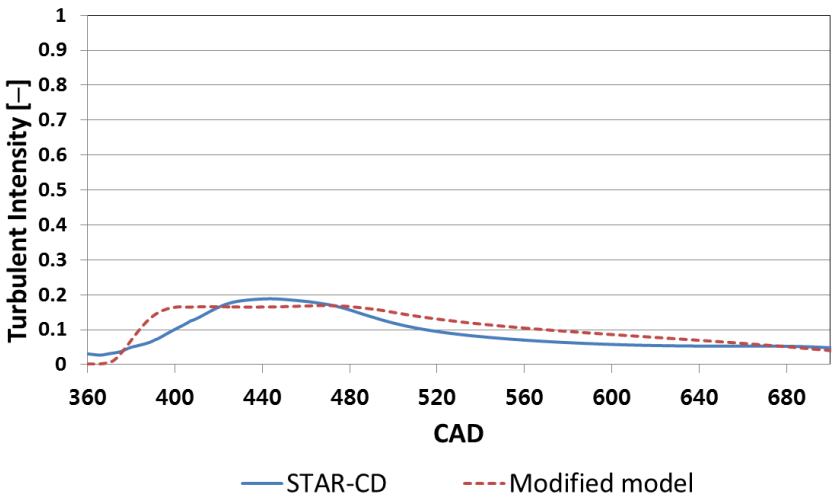


Fig.11 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 8

Table — 9 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	2	8.5
Valve open timing [CAD]	363	172
Valve close timing [CAD]	487	402

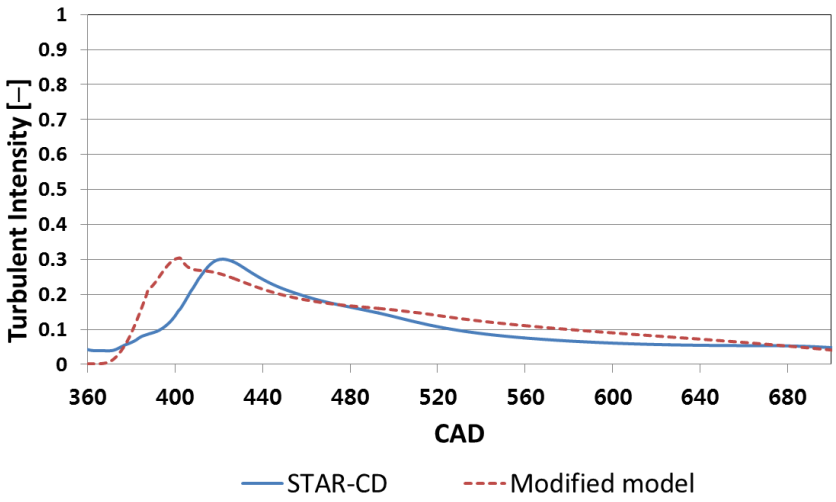


Fig.12 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 9

Table－10 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure[bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	2	8.5
Valve open timing [CAD]	383	172
Valve close timing [CAD]	507	402

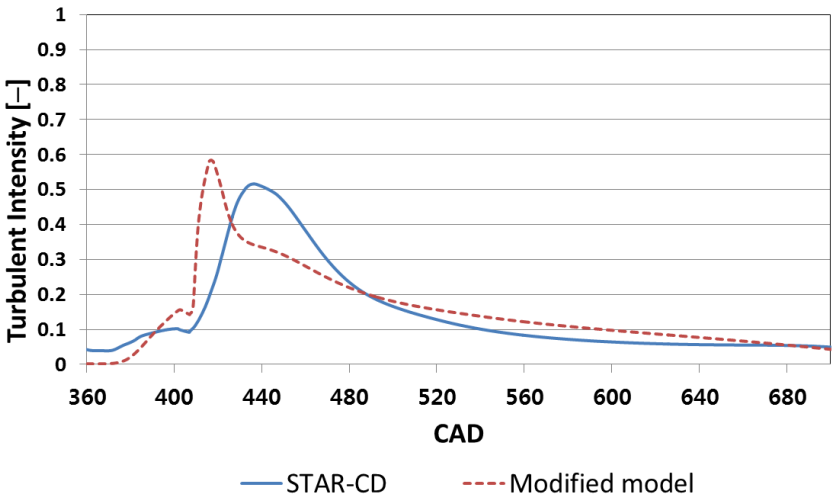


Fig.13 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 10

Table — 11 Test condition of turbulent intensity validation

rpm	1500	
Intake Pressure [bar]	0.6	
Cam	IN	EX
Max Valve Lift [mm]	2	8.5
Valve open timing [CAD]	403	172
Valve close timing [CAD]	527	402

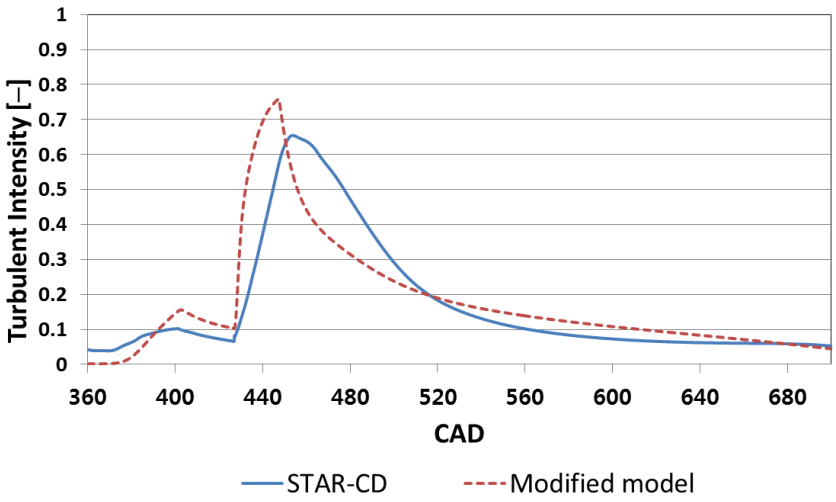


Fig.14 Turbulent intensity of the modified quasi-dimensional combustion model under the condition listed in table 11

3.3 Residual gas fraction

Base model uses constant intake and exhaust manifold pressure as boundary conditions during gas exchange process. To predict RGF more precisely, intake and exhaust manifold pulsation was introduced to the modified quasi-dimensional combustion model. Intake and exhaust manifold pulsation data were acquired using GT-POWER under various conditions of rpm, load, valve lift, duration and timing.

As shown in figure 15, accuracy of RGF prediction is improved by introduction of pulsation effect. Test conditions are listed in table2.

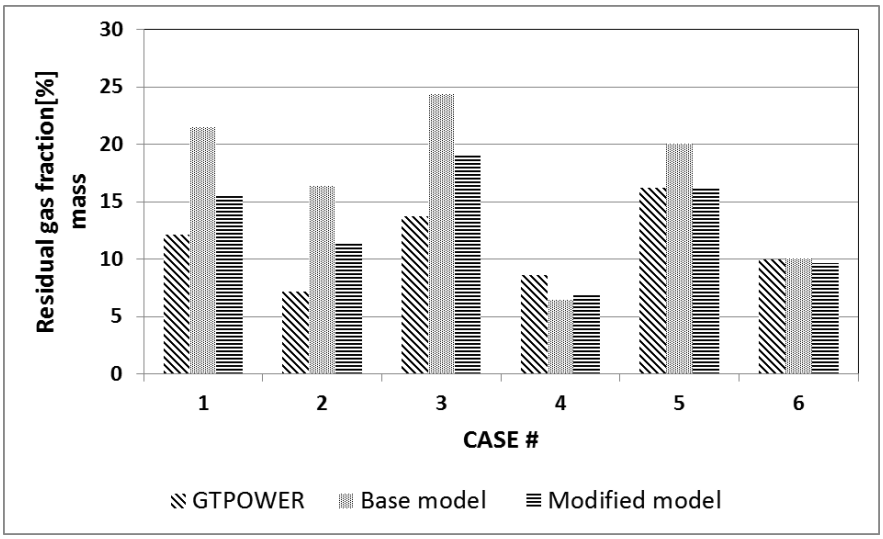


Fig.15 RGF comparison of base and modified model

Chapter 4. Model Calibration

4.1 Experimental setup

Adjustable constant C_a of developing flame speed was calibrated to match the experiments in IMEP and pressure. Since Single-cylinder engine used to get experimental data is not equipped with full VVA, two different type of intake cam shaft was used to simulate two different actuation mode of VVA. Engine specifications are listed in table 12.

Table — 12 Engine specifications

Item	Specification
Engine	Single-cylinder DOHC
Bore (mm)	86
Stroke (mm)	86
Displacement volume (cc)	499.5
Compression ratio	9.5
Injection type	Single DI(70bar)
Intake charge temperature	20℃
Coolant temperature	92℃

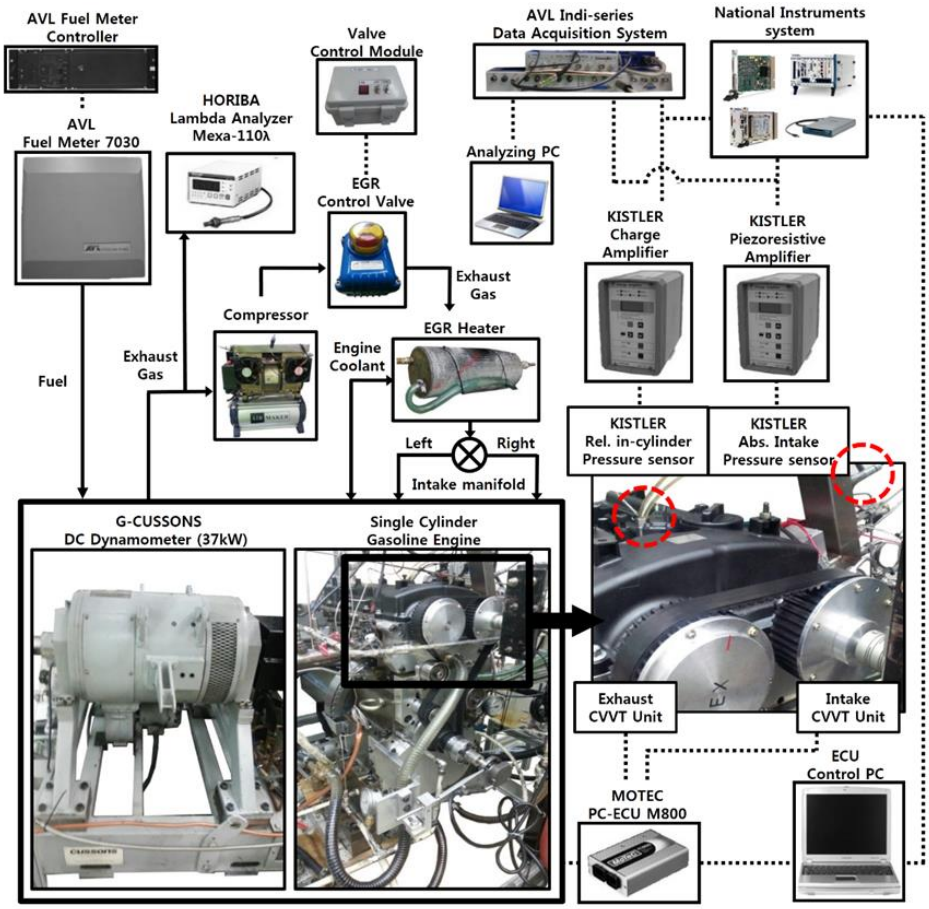


Fig.16 Schematic diagram of the experimental equipment

4.2 Model calibration

To simulate normal valve actuation mode, 9.5mm maximum lift intake cam shaft was used. And for VVA mode, 2mm maximum lift intake cam shaft was used. This VVA intake cam shaft has shorter duration than base mode intake cam shaft and its duration is 206CAD. Test conditions are listed in table 13.

Table—13 Experimental conditions of model calibration

	Case1(Base mode)		Case2(VVA mode)	
rpm	2000		1500	
IMEP [bar]	4.66		4.5	
Cam	IN	EX	IN	EX
Max Valve Lift [mm]	9.5	8.5	2	8.5
Valve open timing [CAD]	343	528	372	470
Valve close timing [CAD]	591	738	578	730

First, for base mode, adjustable constant of developing flame speed model was calibrated to match the experiment in IMEP and Pressure. Calibrated simulation result is presented in figure 17 as dashed green line and shows good agreement with experimental data.

For the same adjustable constant value, VVA mode of un-throttled operation at part load condition was validated. As shown in figure 18, pressure peak of the

modified quasi-dimensional combustion model is delayed by 3-4 CAD. But overall, simulation result showed that the modified quasi-dimensional combustion model can predict burning rate with acceptable degree of accuracy under various valve lift, duration and timing.

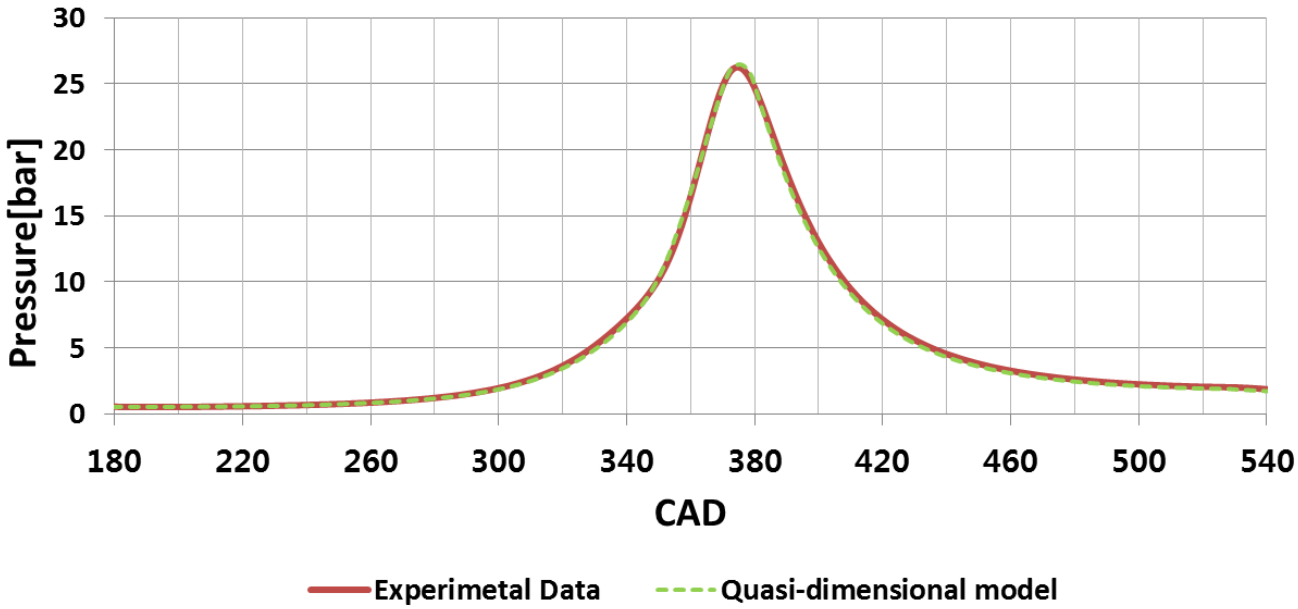


Fig.17 Calibration result under condition of case1

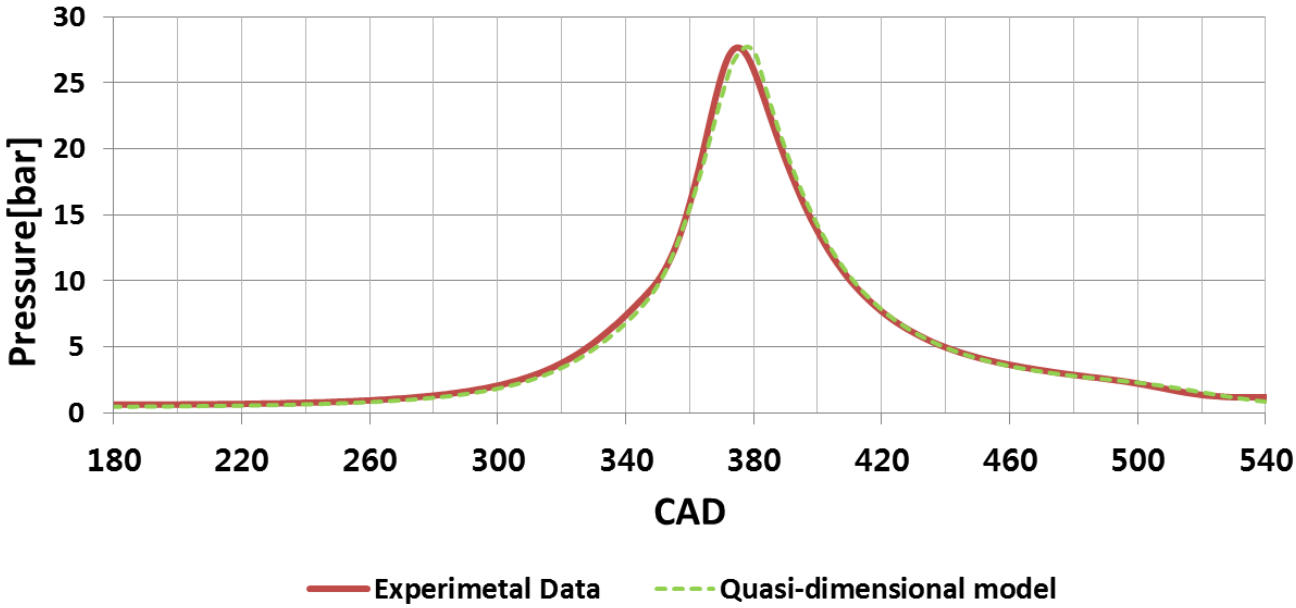


Fig.18 Validation result of calibrated model under condition of case2

4.3 Simulation results

Using the modified quasi-dimensional combustion model, the effect of VVA on fuel economy improvements was validated at part load condition. First, IMEP and ISFC of base cam actuation mode were calculated. And then, at the same rpm and IMEP, ISFC of VVA mode was calculated. As shown in table 14, under WOT condition, intake valve lift, duration and timing was changed to simulate VVA mode. Spark timing was changed to set MFB50 to aTDC7 under various intake valve lift, timing and duration.

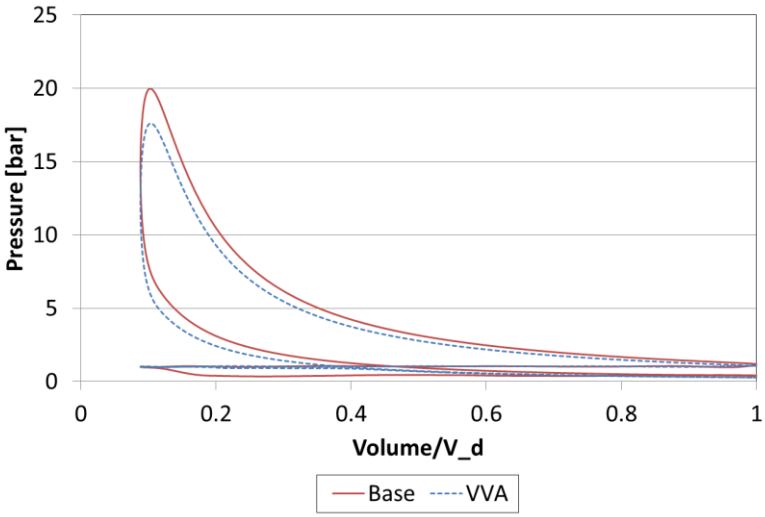
Table—14 Valve actuation conditions of base and VVA mode

	Base mode		VVA mode	
rpm	1500		1500	
IMEP[bar]	2.8		2.8	
Cam	IN	EX	IN	EX
Max Valve Lift [mm]	9.5	8.5	1.5	8.5
Valve open timing [CAD]	343	528	348	528
Valve close timing [CAD]	591	738	472	738

Table—15 ISFC comparison of base and VVA mode

RPM	IMEP [bar]	ISFC [g/kWh] (Base)	ISFC [g/kWh] (VVA)	Improvement
1500	2.8	278.5	253.6	8.94[%]

As shown in figure 19, decrease of peak pressure and reduction of pumping loss are identified under VVA condition Further fuel economy improvement will be possible, if novel optimization technique for VVA is added to quasi-dimensional combustion model.



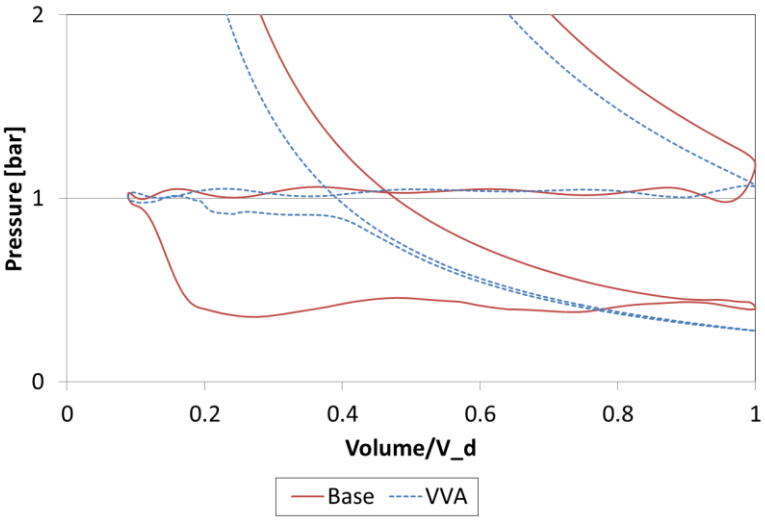


Fig.19 Pressure curve comparison of base and VVA mode at 1500rpm, 2.8bar IMEP

Chapter 5. Conclusions

In this study, to simulate SI engines equipped with variable valve actuation (VVA), the modified quasi-dimensional combustion model was suggested. Since VVA has effects on turbulent intensity and residual gas fraction which determine burning rate, it is important to predict burning rate precisely under various valve lift, duration and timing for VVA engine simulation.

STAR-CD and GT-POWER were used to obtain base data of turbulent intensity and residual gas fraction under various valve lift, duration and timing. Using these base data, the turbulent flow energy model and boundary conditions of gas exchange process were modified to improve the accuracy of turbulent intensity and residual gas fraction prediction.

A zero-dimensional energy cascade model was used to describe turbulent flow energy in a quasi-dimensional combustion model. First, to improve accuracy of turbulent intensity prediction, integral length scale was modified. According to the base data from STAR-CD, integral length scale did not vary significantly as intake pressure, rpm, valve lift, duration and timing change and it seems that integral length scale is mainly governed by combustion chamber geometry. Then, a parameter to control the turbulent kinetic energy dissipation rate was added to the zero-

dimensional energy cascade model. This parameter is a function of valve lift, duration and timing. Using this parameter and adjustable constant of turbulence energy production rate together, the modified quasi-dimensional combustion model can reflect the effect of VVA on turbulent intensity.

Residual gas fraction affects the burning rate indirectly by changing laminar flame speed. Using GT-POWER, pulsation data of intake and exhaust manifold were calculated and it was applied to the modified quasi-dimensional combustion model. By this method, the modified quasi-dimensional combustion model can predict residual gas fraction much precisely.

The newest laminar flame speed correlation was also introduced to the modified quasi-dimensional combustion model. This laminar flame speed correlation contains a parameter which considers the effect of equivalence ratio variation on laminar flame speed. To describe the early stage of flame propagation, H. Werweg's flame speed correlation was introduced.

Model calibration based on experimental data was carried out. Adjustable constant of developing flame model was calibrated under normal valve actuation conditions of intake valve lift, duration and timing. Then, with the same adjustable constant, the modified quasi-dimensional combustion model was applied to un-throttled operation at the part load condition. Simulation results showed that the modified

quasi-dimensional combustion model can predict burning rate with acceptable degree of accuracy under various valve lift, duration and timing.

Finally, using quasi-dimensional model developed in this study, the potential of fuel economy improvement at the part load condition was investigated. Decrease of peak pressure and reduction of pumping loss were identified under VVA engine operating conditions. Further fuel economy improvement will be possible, if novel optimization technique for VVA is added to the quasi-dimensional combustion model.

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가변 밸브 기구가 장착된 가솔린 엔진에 적용 가능한 유사차원 연소모델 개발

서울대학교 대학원
기계항공공학부
김 진 희

요 약

가변밸브기술(VVA)은 다양한 엔진 작동 영역에서 연비와 출력을 향상시킬 수 있는 차세대 엔진 기술로 각광받고 있다. 하지만 가변밸브는 밸브 프로파일 생성 자유도가 크고 이에 영향을 받는 변수가 매우 많기 때문에 실험적 방법으로 최적화 하는데 어려움이 있다. 동일한 이유로 과도한 계산시간이 소요되는 삼차원 엔진 시뮬레이션 기법으로도 밸브 프로파일을 최적화하기가 힘들다. 따라서 본 연구에서는 가변밸브가 장착된 가솔린 엔진의 성능을 빠르게 시뮬레이션 할 수 있는 유사차원 연소모델을 개발하였다.

가변 밸브는 난류강도, 잔류가스 등에 영향을 주어 연소율을 변화시키는데 연소율은 엔진 성능 예측에 있어 가장 중요한 인자이다. 본 연구에서 진행된 유사차원 연소모델 개발 과정은 이러한 가변밸브의 영향이 유사차원 시뮬레이션 결과에 정확히 반영되도록 하는데 주안점을 두었다.

3차원 엔진 해석프로그램인 STAR-CD와 1차원 엔진 해석프로그램 GT-POWER를 이용하여 다양한 밸브 작동모드에서 난류 강도와 잔류 가스데이터를 확보하였다. 이를 기반으로 유사차원 연소모델의 난류에너지 관계식과 가스 교환모델을 수정하여 난류강도와 잔류가스가 다양한 밸브 작동모드에서 정확히 예측될 수 있도록 하였다.

엔진 실험 데이터를 이용하여 화염 속도 모델의 상수를 보정하였고 저부하, un-throttled 작동 조건에 보정된 유사차원 연소모델을 적용하여 모델의 정확도를 평가하였다. 마지막으로, 본 연구에서 개발된 유사차원 연소모델을 저속, 저부하 엔진 작동 조건에 적용, 연비 개선 효과를 확인 하였다.

주요어: 연비, 가변밸브, 유사차원 연소모델, 난류강도, 잔류가스, 연소율

학번: 2012-20660