공학석사학위논문

NOx 배출량 예측을 위한 물리 기반의 EGR 시스템 포함 디젤 엔진 모델 개발

Developing Physics-based Diesel Engine Model with EGR system for Predicting NOx Emissions

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Abstract

Developing Physics-based Diesel Engine Model with EGR system for Predicting NOx emissions

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Emissions from tailpipe of passenger cars are getting attention as a primary source of air pollution; thereafter regulations for the vehicle emissions are getting more stringent. The limit for NOx emissions is decreasing and the vehicle test environment is being more aggressive that covers a wide range of engine operation. In addition, real-driving NOx emissions are included in the Euro 6d regulation recently, and therefore efforts to reduce the NOx emissions from the vehicle have been made by several approaches. Especially, many researches are conducted to understand and predict the NOx formation in various engine operation

points. However, most of the studies lack physical understanding of engine operations and NOx formations or require huge computational costs to cover a broad range of engine operations. In this study, the diesel engine model that includes exhaust gas recirculation(EGR) system was developed to predict the NOx emissions with 0D/1D physics based model. Each engine part was implemented based on the measured data from the test result of the target vehicle. First, EGR system with EGR cooler and EGR control system was modeled with the estimated target EGR fraction map. Second, the specifications of injector such as fuel mass injected and the timing of injection were collected from the measured data and entered in the model. Third, the effect by swirl control valve that enhances the fuel-air mixing was reflected in the combustion model. Finally, the diesel combustion was calibrated based on the data from the combustion analyzer such as in-cylinder pressure profile. The engine IMEP and NOx emissions from the engine showed good agreement with the measured data in a wide range of engine speed and load.

Keywords: Diesel engine, NOx, Combustion model, EGR system,

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1. Introduction

1.1. Research background

Regulations for the tailpipe emissions from on-road vehicles are getting more stringent recently. European emission standards announced by the European Union(EU) restrict carbon monoxide(CO), nitrogen oxides(NOx), unburned hydrocarbon(HC) and PM emissions from tailpipe of passenger cars. The limits of these emissions are decreasing as new regulations are updated. For example, Euro 6 NOx emissions limit for light-duty diesel vehicles is 0.08g/km which is almost half of Euro 5 limit, 0.18g/km[1].

Also, the test conditions for vehicle emissions are being more aggressive so that the driving conditions can reproduce the real driving environment. For instance, EU suggested a new test cycle named Worldwide Harmonized Light Vehicle Test Cycle(WLTC) that can replace the New European Driving Cycle(NEDC), an old lab test cycle designed in the 1980s. WLTC is a more dynamic cycle compared with NEDC; it covers higher speed and acceleration while driving vehicles as shown in Figure 1.1[2,3]. Moreover, new regulations for Real Driving Emissions(RDE) came into effect in 2017 for new type passenger car models approved in Europe[4]. RDE measurements are conducted on the road with Portable Emissions Measurement System(PEMS) mounted on the vehicles, while WLTC runs the vehicle on the dynamometer.

In such trends, efforts to reduce the NOx emissions from diesel engine cars by auto-maker have reinforced recently. For example, many auto companies are developing and adopting aftertreatment systems such as Lean NOx Trap(LNT) and Selective Catalytic Reduction(SCR) on their diesel cars. In addition, study for predicting NOx emissions from diesel engines is widely conducted by many researchers. Considering huge costs and effort are spent for testing and tuning the engines for NOx emissions, these works are also meaningful for an industrial purpose not only for academic aspect. Besides, since recent regulations for vehicle emissions include RDE emissions as mentioned above, the impact of real driving conditions such as weather conditions or altitude change of the test route cannot be ignored[5]. However, these effects are hard to be evaluated only by running the real vehicle one-by-one, and therefore, a predictive model for diesel engine NOx emissions can be a substitute for assessing the characteristics of NOx emissions in real driving conditions.

1.2. Previous research

Since NOx formation during combustion in diesel engines is a complex phenomenon, diverse approaches to predict the emissions were proposed. Many studies suggested computational techniques or mathematical models employing artificial neural network[6,7] or fuzzy model[8] that can predict the NOx emissions based on the engine test data. These models require

less computational costs and most of them aim to predict the engine out NOx emissions in real-time that can be utilized for the aftertreatment control system via vehicle electronic control unit(ECU). However, physical situations of the engine operation is hard to be understood from these techniques. Some studies utilized computational fluid dynamics(CFD) to understand the combustion and NOx formation in diesel engines[9], but this approach takes a huge amount of time and computational cost to study a wide range of the engine operation and test conditions. Other research suggested an equation with some factors relevant to the NOx formation such as burned zone temperature, and then optimize the coefficients of the equation with experimental data[10]. However, it is hard to say that this single equation is still valid in various driving conditions and environments.

1.3. Research objective

A physics-based engine model to predict NOx emissions at each engine operation points was developed in this study. The model includes 0D simulation for in-cylinder diesel combustion and 1D model for the paths of intake and exhaust gas flow. The combustion model used in this study is a physics-based model that treats fuel injection, air entrainment into the fuel spray zone, fuel evaporation and combustion requiring relatively less computational power than CFD models. Exhaust gas

recirculation(EGR) system which can affect the NOx formation by controlling the composition of intake gas is also included in the model.

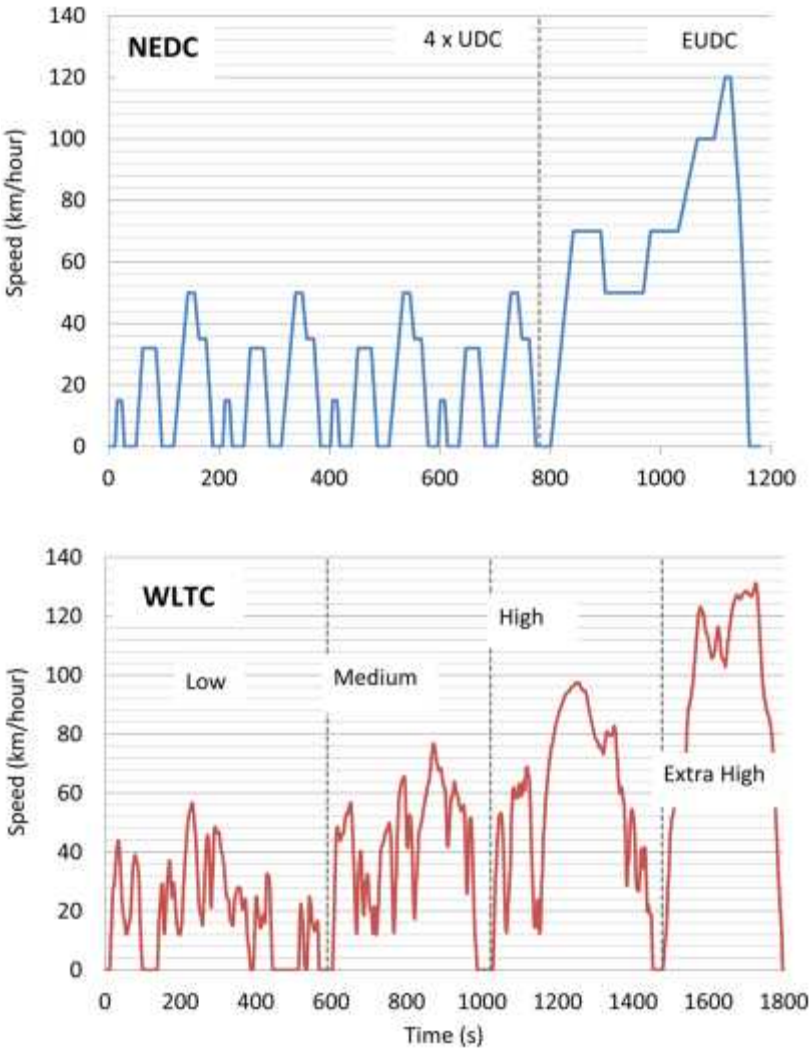


Figure 1.1 NEDC and WLTC with vehicle speeds[2]

2. Methodology

2.1. Measured data from target vehicle

Hyundai Santa Fe(2.0L, '19) was selected as a target vehicle for the engine model. This model contains Hyundai R engine which is an inline-four diesel engine with variable geometry turbocharger(VGT) and intercooler as boosting system, and EGR cooler and EGR valve as EGR system. LNT and SCR are mounted as NOx aftertreatment and diesel particulate filter(DPF) is utilized for soot emissions reduction.

Basic specifications of the engine were accessible from the website of Hyundai and enumerated on Table 2.1. Some dimensions of engine parts such as coolers and manifolds were required for the engine model, but not provided online and therefore measured directly. Various types of data related to the engine performance and parts operation were measured while running the vehicle on the dynamometer at each engine operation points(engine RPM and torque). Basic operational data such as engine load, RPM, valve position and fuel injection rate were displayed on on-board diagnostics(OBD). Auxiliary sensors were additionally equipped on each engine part to collect the pressures, temperatures, and gas compositions. The engine test on the dynamometer was conducted for 80 operation points covering a wide range of engine RPM and load.

Additional dynamometer test was implemented with combustion analyzer(AVL IndiCom) for 43 operation points which are all included in 80 operation points measured in the previous test. In-cylinder pressure profile, heat-release rate profile and some combustion related factors such as CA50 were calculated by the combustion analyzer.

2.2. Modeling tool and scope

In this study, GT-Power(Gamma Technologies) software (Figure 2.1) was utilized to build the diesel engine model. GT-Power is a 0D/1D-based multi-physics CAE system simulation software, which provides the basic design templates for the major engine parts such as cylinders, valves, manifolds/ports, and injectors. Especially, the templates for engine cylinder comprises wall temperature calculation model, heat transfer model, diesel combustion model, and emissions(NOx, soot, CO and HC) formation model.

The engine model we developed includes intercooler, EGR cooler, EGR control system, manifolds, ports, and four cylinders with injectors and intake/exhaust valves as displayed in Figure 2.2. Though turbocharger and aftertreatment system(LNT, DPF, SCR) are mounted on the target vehicle, they were excluded from the scope of the model in this study since they are too complex to be included in the entire model that demands high computation times and costs. Therefore, the inlet boundary

condition of the model is the temperature and the pressure of the compressor outlet of the turbocharger, and the outlet boundary condition corresponds to the temperature and the pressure of the turbine inlet of turbocharger.

Table 2.1 Specifications of Hyundai R engine

Displacement volume [cm^3]	1995
Compression ratio	16:1
Bore[mm]	84
Stroke[mm]	90
IVO timing [aTDC]	8~16
IVC timing [aBDC]	3~11
EVO timing [bBDC]	28~36
EVC timing [bTDC]	13~21

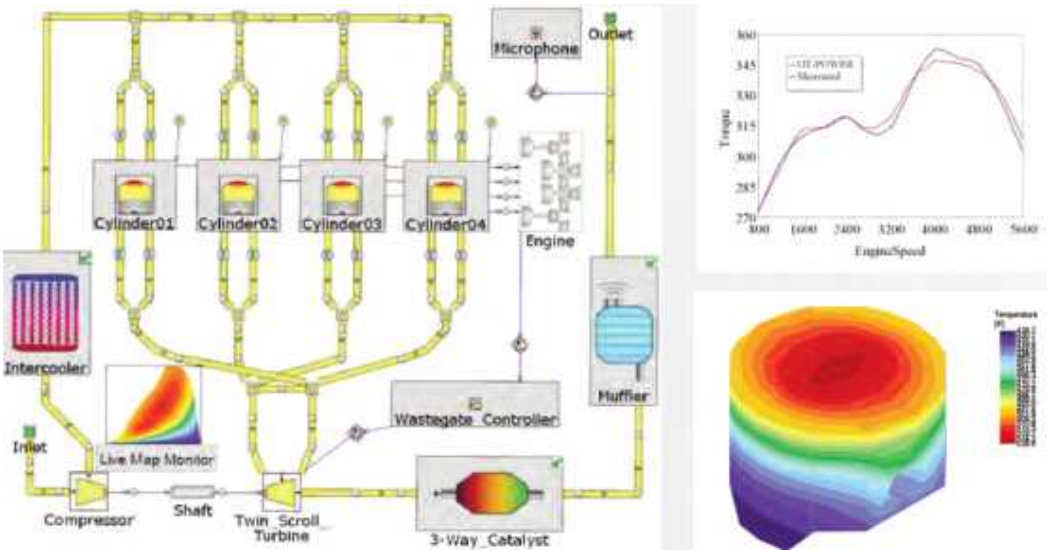


Figure 2.1 GT-Power software description[11]

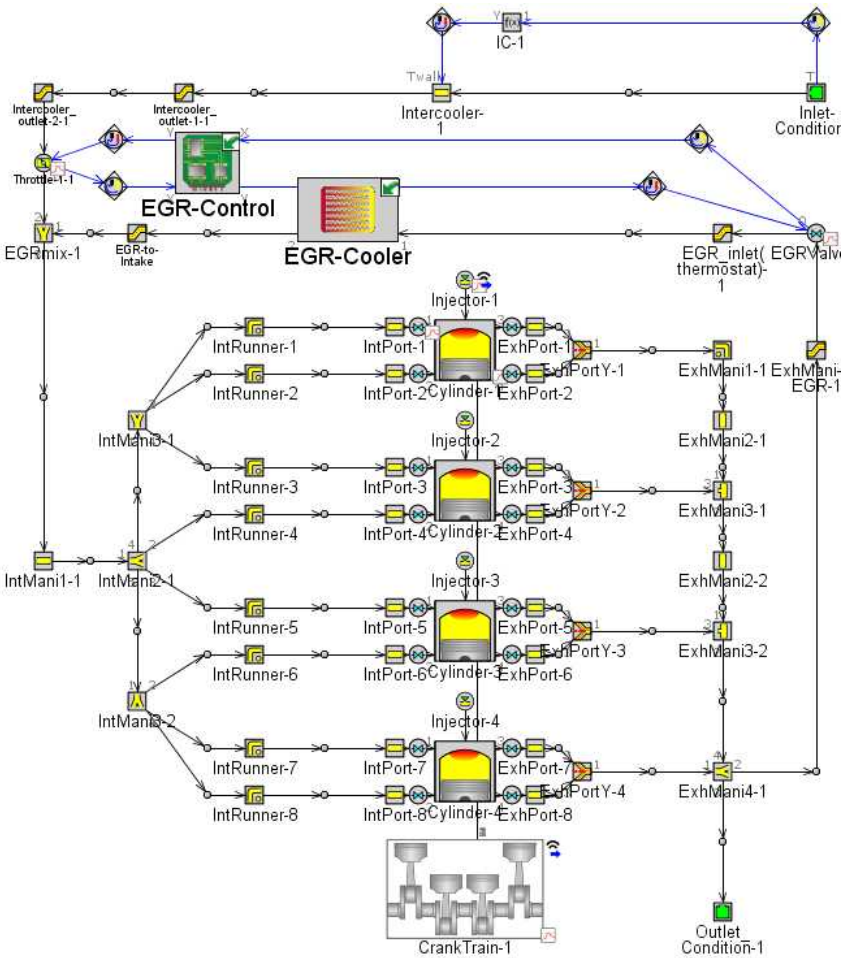


Figure 2.2 Engine model developed in GT-Power software

3. Model description

In this section, a few engine part models—EGR system, injectors, swirl control valve—are described which are critical to the NOx formation in the engine cylinders. Then, calibrating the engine combustion model by closed volume analysis based on the initial conditions for each operation points is explained.

3.1. EGR system

EGR system reduces NOx emissions from an engine by recirculating the exhaust gas and lowering the burned gas temperature. EGR cooler and EGR control system compose the EGR system implemented on our model. EGR cooler drops down the temperature of exhaust gas to be recirculated, and therefore cooler model is required to predict the cooler outlet temperature from the cooler inlet temperature. As shown in Figure 3.1(a), the temperature drop through the EGR cooler is linearly increasing as the temperature of exhaust gas gets higher. Since EGR cooler is water-cooled type with relatively high heat capacity, the coolant temperature is almost constant. Thus, the temperature difference between coolant and gas, and therefore heat transfer or temperature drop through cooler are greater as exhaust gas gets hotter. Based on the data shown in Figure 3.1(a), EGR cooler was implemented in the model with simple linear function of cooler

inlet temperature. Also, as described in Figure 3.1(b), this predictive model works well in transient conditions though the model was derived from data at steady-state conditions. In Figure 3.1(b), the red solid line is EGR cooler inlet temperature profile measured by auxiliary sensor, while blue solid line corresponds to the measured outlet temperature. Blue dotted line is cooler outlet temperature calculated from the linear function and the measure inlet temperature, and it shows good agreement with the measured data (blue solid line).

In Hyundai R engine, EGR fraction is controlled by lift-type valve that exists between the exhaust gas manifold and EGR cooler. Our model includes PI controller that can vary the EGR valve lift to change the flow rate of recirculated exhaust gas to follow the target EGR fraction, and therefore target EGR fraction map should be specified. EGR fractions at each operation point were calculated based on the measured temperatures. Since fresh air and recirculated exhaust gas are mixed at intake manifolds, energy equation below is valid. Description 'air', 'EGR', and 'mix' below each means intake air, recirculated exhaust gas, and mixed gas.

$$\dot{m}_{air} c_{air} T_{air} + \dot{m}_{EGR} c_{EGR} T_{EGR} = \dot{m}_{mix} c_{mix} T_{mix}$$

By assuming that specific heat capacity of gases are the same and using mass conservation, $\dot{m}_{air} + \dot{m}_{EGR} = \dot{m}_{mix}$,

$$EGR\ fraction = \frac{\dot{m}_{EGR}}{\dot{m}_{mix}} = \frac{T_{mix} - T_{air}}{T_{EGR} - T_{air}}$$

Hence, EGR fraction can be estimated by measured temperature data from sensors mounted at intercooler outlet, EGR cooler outlet, and intake manifold each. Figure 3.2 is the EGR fraction map generated from the equation above. X axis and Y axis of the graph each means engine RPM and BMEP, and tested engine operation points are marked with circles. X marks in the circles correspond to the operation points where EGR valve signal from OBD data is non-zero value which means EGR system operated at that point. It was also checked that T_{mix} is much higher than T_{air} when EGR valve signal is non-zero value. As we can see in the Figure 3.2, EGR system only operates at low RPM and low load conditions, and target EGR fraction increases as RPM and load decrease.

However, since temperature sensor at intake manifolds were located upstream, there is a possibility to measure the temperature of non-fully mixed gas. Thus, EGR fraction map was corrected by iterative approach. First, full engine model was implemented with the EGR fraction calculated from the energy equation above. Next, intake gas temperature at IVC timing was calculated as a model result, and EGR fraction can be recalculated considering pressure and volume of intake gas at IVC timing and mass of air in intake gas are known from MAF sensor. Then,

corrected EGR fraction map was used for full engine model again, and EGR fraction map was improved iteratively.

3.2. Injector

Fuel injection is directly related to the characteristics of combustion and engine performance, and therefore a few kinds of data which describes fuel injection should be specified in the model. First, the pressure of the fuel injected were known from the rail pressure measured as OBD data. As displayed on the Figure 3.3, rail pressure shows strong relation with the engine RPM and the mass of fuel injected.

Second, the timings of fuel injected were gauged at each engine operation point. While we tested the target vehicle with the combustion analyzer, the auxiliary sensor on the fuel injector measured the current signal. The number and the timing of each injection pulses in one cycles at each operation points were known by the injector current data as shown in the Figure 3.4. For every engine operation point, there were maximum two pilot injection pulses, one main injection pulse, and one post injection pulse. Pilot injection was conducted two times at low RPM regions, and one time at high RPM regions. Post injection was not conducted at low load regions.

Third, the mass of fuel injected at each operation points were calculated. The fuel injected per one cycle was measured as explained above, but the

mass of fuel injected during each pulse at each engine operation point should be identified since the combustion model we used was a pulse-based model. Therefore, we utilized a vehicle scanning device, named G-SCAN2 while running the target vehicle on dynamometer with combustion analyzer to get the ratio of fuel masses among pulses in one cycle. Based on the measured fuel mass ratio among injection pulses, the mass of fuel injected at each pulse was calculated and entered in the injector model.

3.3. Swirl control valve

In Hyundai R engine, swirl control valves are located between the intake manifolds and ports, and only one valve exists per two ports or one cylinder. Variable swirl actuator(VSA) controls the angle of the swirl control valve, and swirl motion of the intake flow is induced by the flow rate difference between the two ports as shown in Figure 3.5. Using the linear function between the SCV angle and the feedback signal output voltage from the valve, the angle of swirl control valve at each engine operation points was measured as shown in Figure 3.6(a). SCV does not operate in high RPM region (about over 2500rpm) and valve angle decreases at low RPM region as engine load increases, which means VSA induces more swirl motion in low load region.

Though it is known that swirl motion of intake gas enhances the fuel-air

mixing in the cylinder[12,13], the combustion model used in this study cannot consider the swirl motion in the engine cylinder since it is 0D based model. Rather manipulating the swirl ratio of intake gas directly in the combustion model, this effect was considered in the model through the 'entrainment rate multiplier', one of the combustion multipliers for calibrating the diesel combustion, which will be explained in the next section. The mass of air entrained into the fuel spray zone is calculated from the mass and momentum conservation of the spray, and the entrainment rate multiplier is used to modify the rate of the air entrainment into the spray area. As displayed on Figure 3.6(b), at low engine RPM region, the optimized entrainment rate multipliers are distributed at high values, while almost constant with lower values for the higher engine RPM region. Considering SCV operates aggressively at low engine RPM especially at low load, it can be understood that enhancement of fuel-air mixing by swirl motion induced by swirl control valve is reflected in the optimized value of the entrainment rate multiplier which calibrates the amount of air entrainment into the fuel spray zone in the combustion model.

3.4. Cylinder combustion model

A few kinds of diesel combustion model were provided in GT-Power

software, and we selected the model that predicts the combustion rate and associated emissions for direct-injection with multi-pulse injection events. This model includes the equations that describe the physical phenomenon related to the fuel injection, evaporation, mixing with surrounding air and combustion for each injection pulse. Therefore, the characteristics of fuel injection by pulses such as injected fuel mass and timing for each pulse should be defined, and these processes were already explained above.

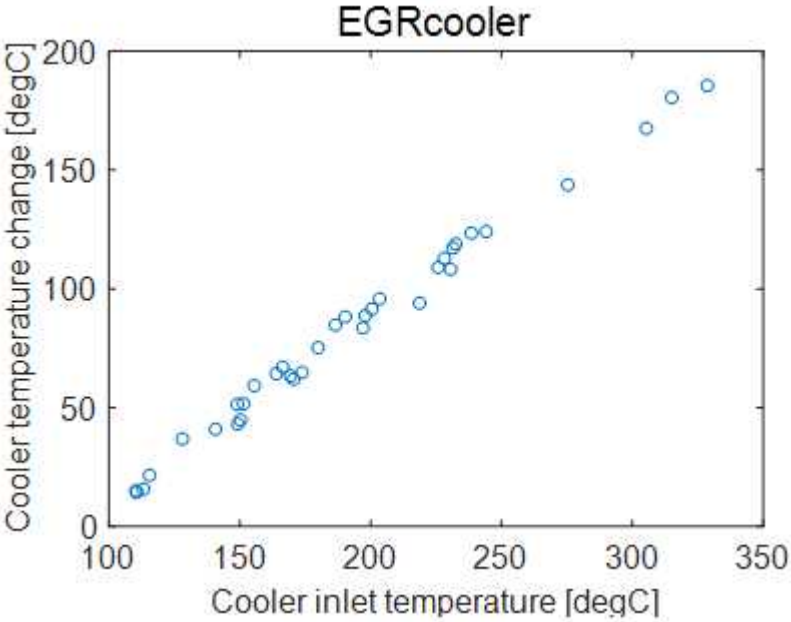
Also, four multipliers each related to the specific physical phenomenon during fuel injection and combustion were needed to be calibrated to precisely predict the diesel combustion. The first multiplier is 'entrainment rate multiplier' which is associated with air entrainment into the fuel spray. The second multiplier is 'ignition delay multiplier.' Ignition delay is calculated in the combustion model with an Arrhenius expression with temperature, and the multiplier is multiplied into this equation. The other two multipliers are 'premixed combustion rate multiplier', and 'diffusion combustion rate multiplier' which affect the rate of combustion. Both combustion rate is calculated as a function of gas mass, turbulent kinetic energy, and oxygen fraction, and each multiplier is producted with each equation to calibrate the rate of combustion.

Since thousands times of model running were required to tune these factors of the combustion model, the cylinder only model which is far smaller was segregated from the entire engine model for the optimization.

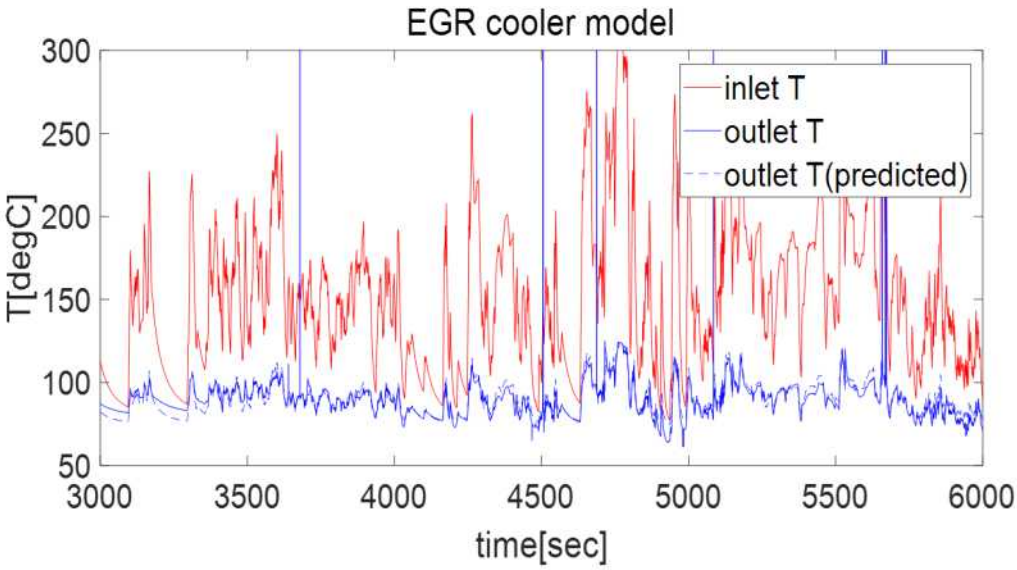
However, the intake and exhaust flow path are not included in this sub-model, thus this model only treats limited duration of an engine cycle, from IVC timing to EVO timing when the cylinder can be assumed as a closed volume. Closed volume analysis requires defining the initial state of the gas in cylinders at each engine operation point such as cylinder volume, temperature, volumetric efficiency, and the compositions of the in-cylinder gas at IVC timing. The volume of the cylinder at IVC timing was calculated by specification of the engine—displacement volume, compression ratio, bore, stroke, and connecting rod length. The pressure at IVC timing was included in the cylinder pressure profile data (Figure 3.7(a)) that will be used later for calibrating combustion model. The volumetric efficiency was calculated based on the mass flow rate of air measured by MAF sensor. The composition of the in-cylinder gas at IVC timing corresponds to the EGR fraction of intake gas, and target EGR fraction for each engine operation points was calculated in the 'EGR system' section.

After the initial states of the in-cylinder gas at each operation points were defined, the fuel burn rate was calculated from the measured cylinder pressure profile (Figure 3.79b)). This process is called 'reverse-run' that it is exactly the opposite procedure with general engine models do; they predicts the fuel burn rate based on their own combustion model, and then determine the pressure rise during

infinitesimal time and hence the pressure profile. Finally, optimization for four combustion multipliers is conducted to fit the burn rate with the reverse-run results. In other words, optimizer finds the best multiplier set that can match the burn rate profile which was predicted using the chosen multiplier set with the burn rate obtained from reverse-run for the all engine operation points. However, the entrainment rate multiplier exceptionally optimized separately for each operation points, since swirl motion induced by SCV should be reflected through this factor as explained above.



(a)



(b)

Figure 3.1 (a) Relation between EGR Cooler inlet and outlet temperature, (b) Prediction for EGR cooler outlet temperature

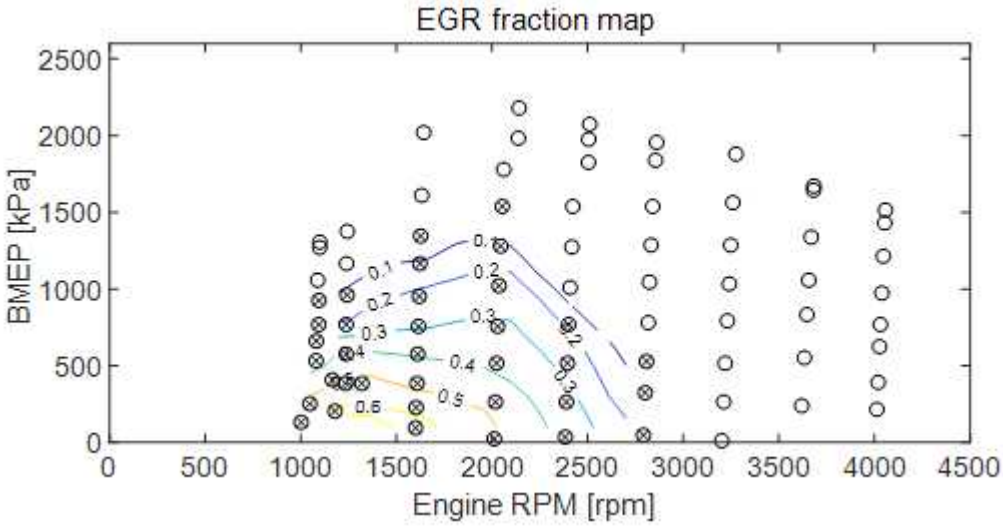


Figure 3.2 Estimated EGR fraction map

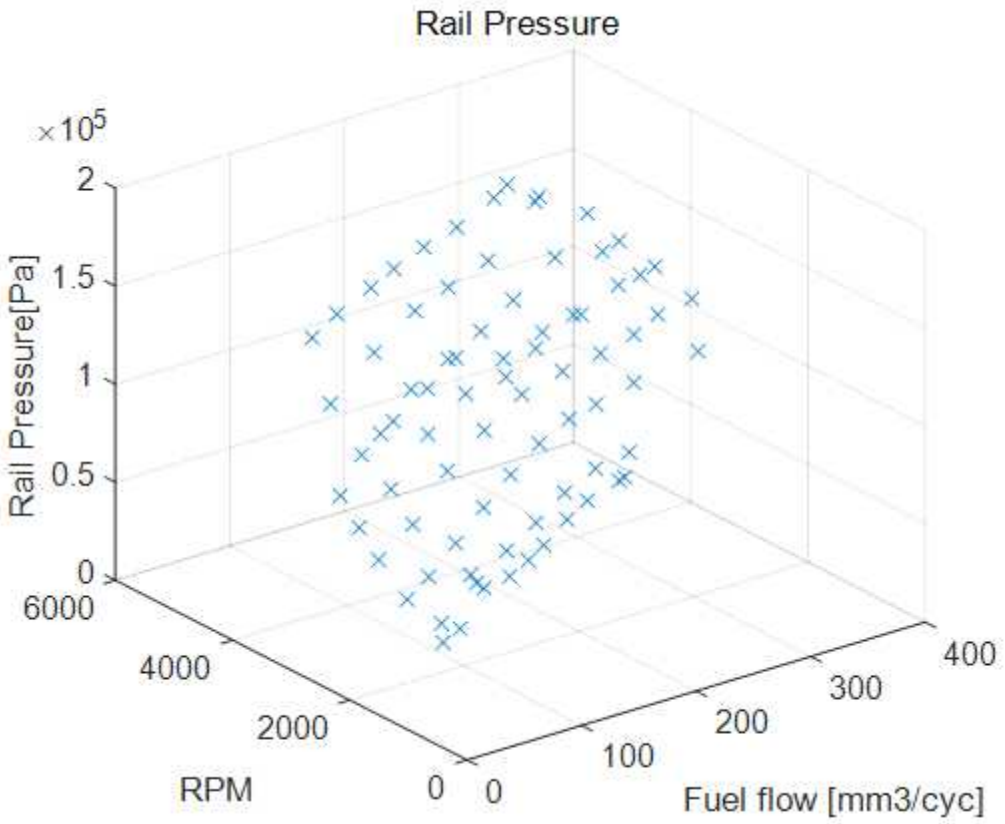


Figure 3.3 Rail pressure by engine RPM and the mass of fuel injected

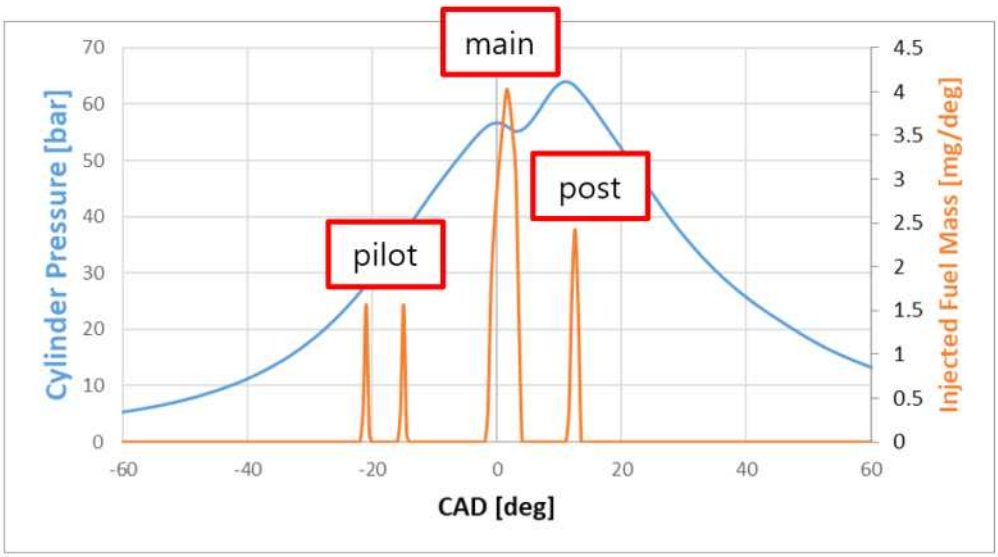


Figure 3.4 Cylinder pressure profile (blue)
& Injected fuel mass profile (orange)

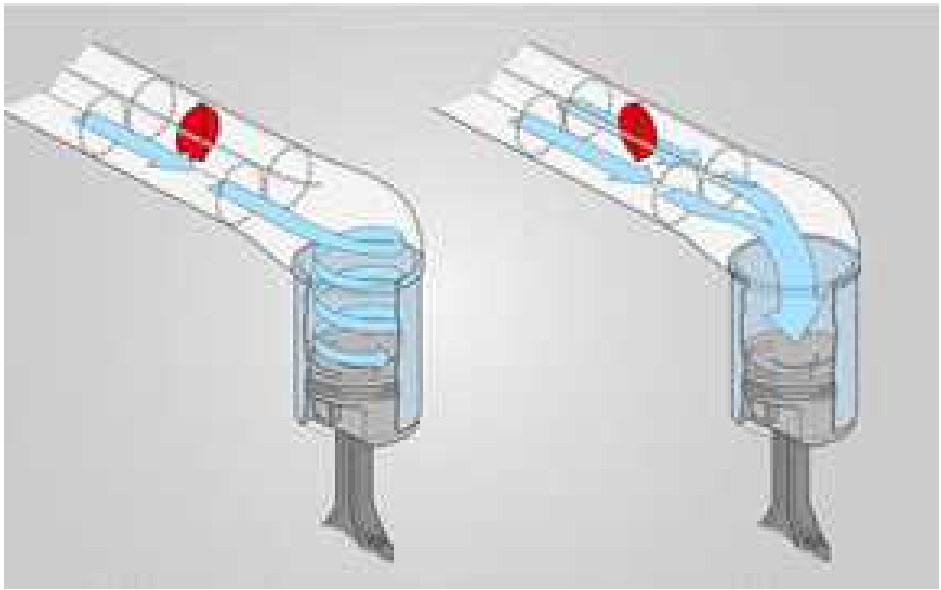
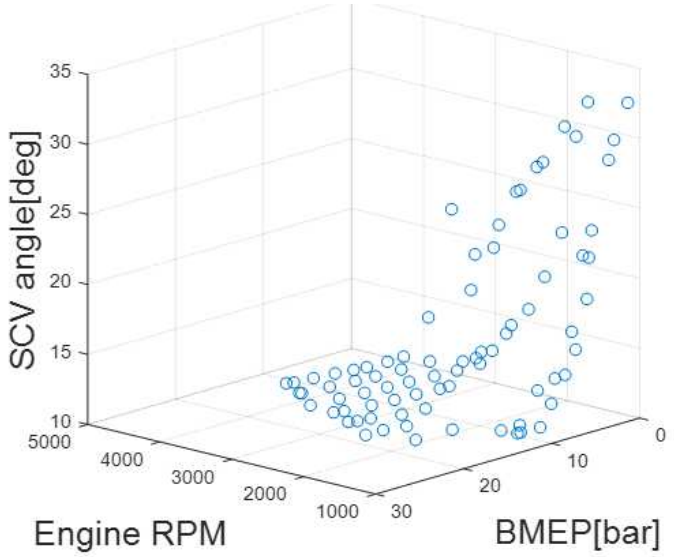
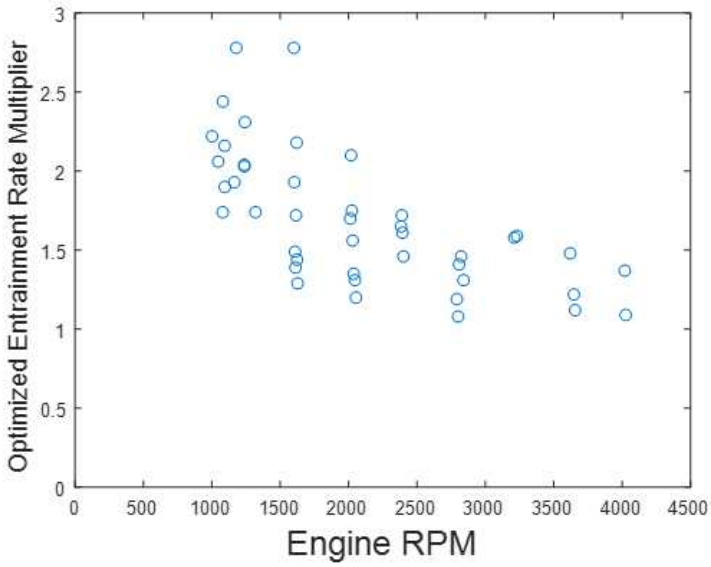


Figure 3.5 Swirl control valve(SCV) operation[14]

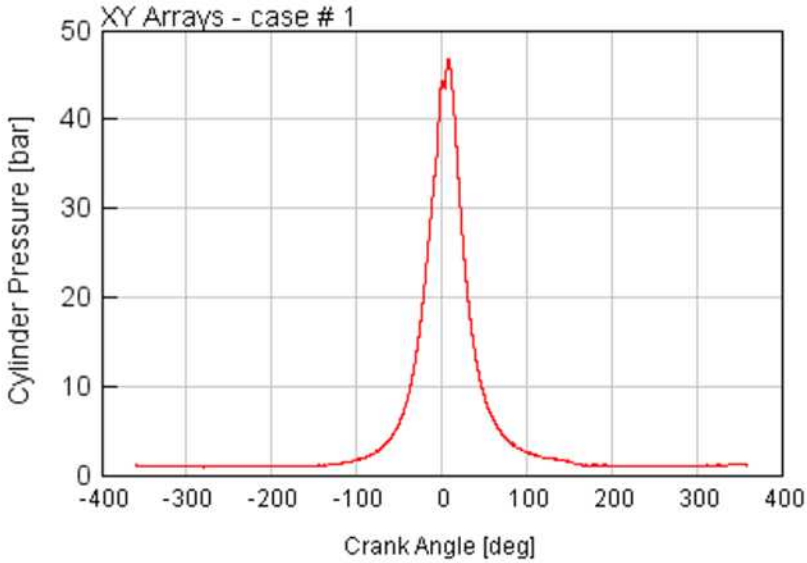


(a)

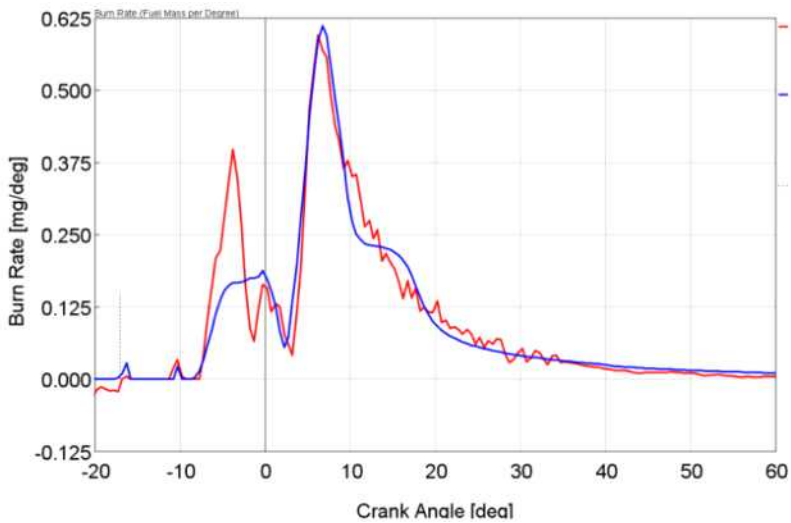


(b)

Figure 3.6 (a) Swirl control valve angle by engine RPM and BMEP, (b) Relation between optimized entrainment rate multiplier and engine RPM



(a)



(b)

Figure 3.7 (a) Measured cylinder pressure profile,
(b) Burn rate estimated based on pressure profile (red),
predicted by combustion model(blue)

4. Model results validation

Figure 4.1 shows the measured engine IMEP and the predicted engine IMEP by the model for each engine operation point. The predicted engine loads show good agreement with the measured ones, and it can be said that combustion phase in the engine cylinder was well predicted, since only the masses of fuel injected at each point were entered in the model, not controlling the fuel mass to target the engine load input.

The engine cylinder combustion model in GT-Power includes the NOx emissions model as sub-model. This model employs Extended Zeldovich mechanism which includes H, N, O, OH, N_2, O_2, NO as species and N_2 oxidation, N oxidation, OH reduction as reactions. Each reaction rate is affected by the temperature and the concentration of each species. The combustion model used was a two-zone model, and therefore temperature and the molecular concentration in the burned zone were utilized to calculate the NOx formation rate.

Figure 4.2 is the model result for NOx emissions from the cylinder model. Each plot on the graph corresponds to the engine operation points, and they are grouped with engine RPM—1000rpm, 1200rpm, 1600rpm, 2000rpm, 2400rpm, 2800rpm, and over 3000rpm. The bar graph with navy colour is the predicted NOx emissions from the cylinder model, and the graph with cyan colour on the background is the NOx emissions measured during the dynamometer test. The NOx emissions increase as the engine load

increases at every engine RPM region, and the predicted values also follow this trend quite well. Also, it can be seen that the NOx formation is directly related to the burned zone temperature from the model results Figure 4.3, which shows increasing burned zone temperature profile as engine load increases. Nonetheless, for some engine operation points, the cylinder model overestimate or underestimate the NOx emissions like high load region at 1200rpm and high load region at 1600rpm each. The main reason for the error with NOx emissions seems to be the uncertainty of the input data for injection. First, the calculated data for injected mass of the fuel at each pulse can hold some error. The number of significant figures of the mass ratio among pulses displayed on G-SCAN2 was only one, which induces inaccuracy for calculating the mass of fuel injected at main pulses. Second, since the injection timings were measured based on current signal from injectors, there can be an error for injection timings entered in the model due to the delay between the injection signal and real injection at injector. Third, the injection profile map for each operation points based on injected mass and engine speed was not specified in the model. The default database in the software for injection profiles was used, not defining the profile map for the injector mounted on the target vehicle. Since the NOx formation rate at burned zone is affected by burned zone temperature and the concentration of each species, the combustion related profiles by crank angle degree such as

combustion rate profile and oxygen species profile are critical to the prediction for the NOx emissions. Although the fuel burn rate was estimated from the cylinder pressure profile, it is hard to say that the combustion model can reproduce the combustion phase in the real engine if the injection parts are not correctly defined.

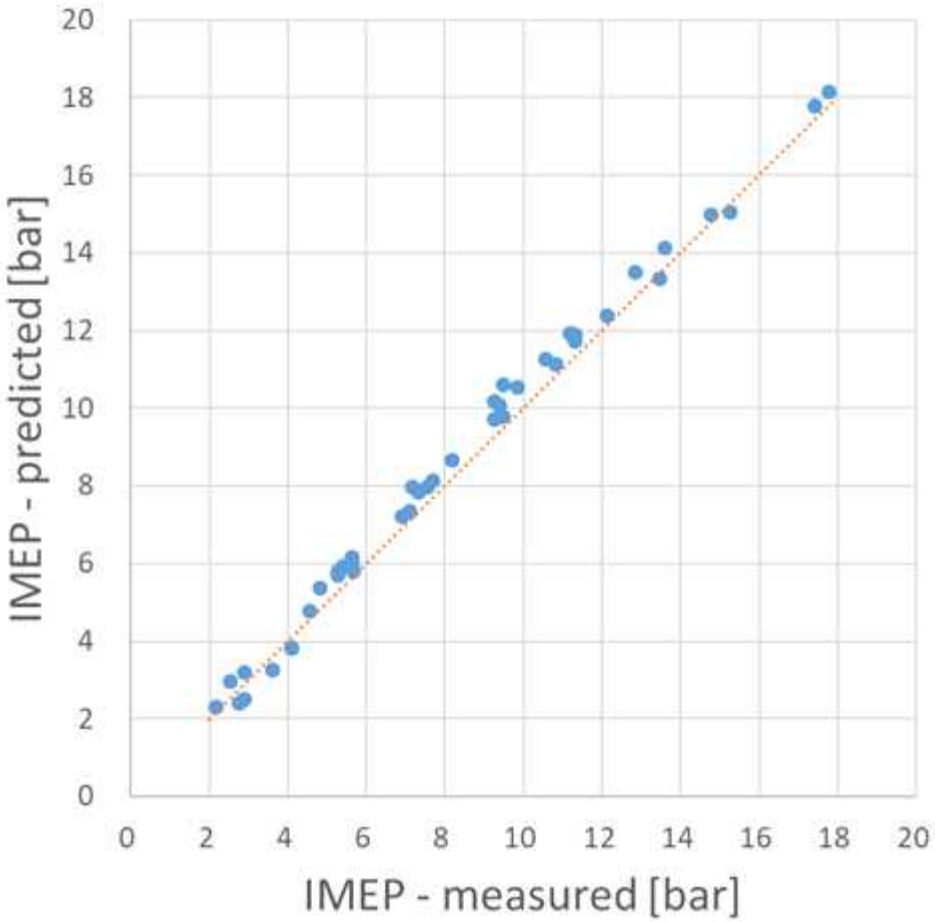


Figure 4.1 Model prediction for engine IMEP

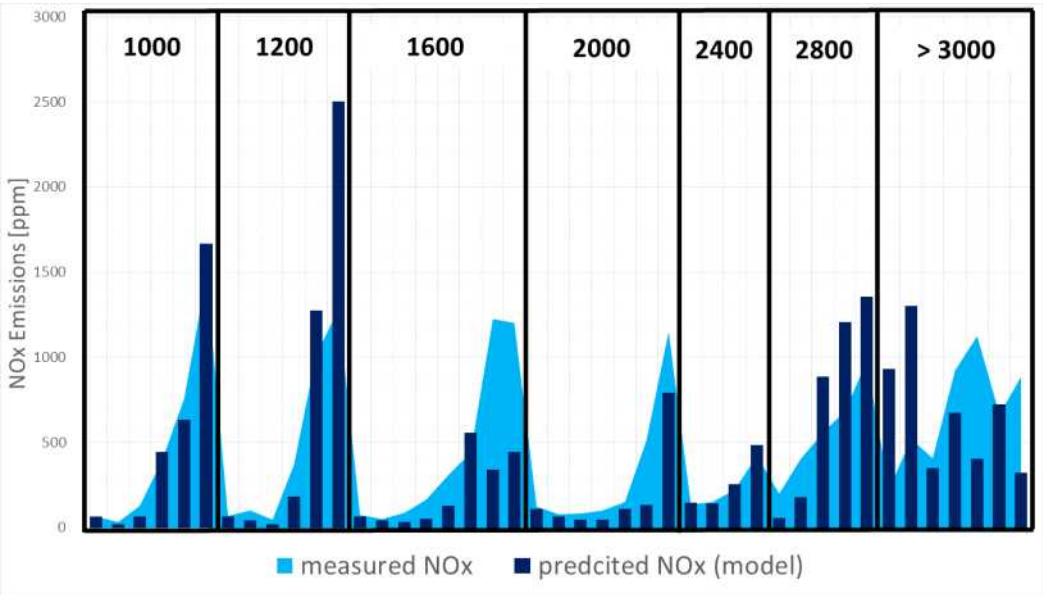


Figure 4.2 Model prediction for engine-out NOx emissions

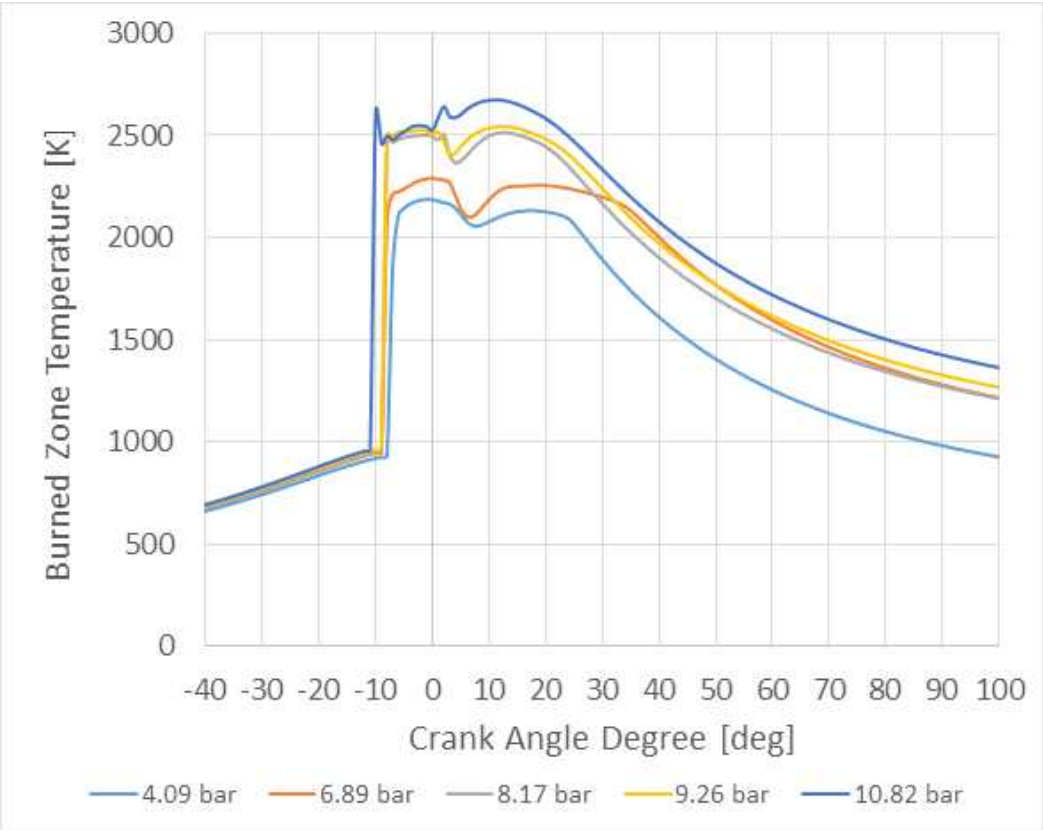


Figure 4.3 Burned zone temperature profiles by engine IMEP (1000rpm)

5. Conclusion

In this study, a physics-based diesel engine model was developed to predict the engine-out NOx emissions. The elemental data required to model the engine parts were measured while testing the target vehicle on the dynamometer. GT-Power software was utilized to model the engine, and a few engine parts which are critical to the NOx formation in the engine cylinders were implemented with physical model. First, EGR system which constitutes EGR cooler and EGR control system was modeled. EGR cooler was modeled simply with the relation between the temperature drop through the cooler and the cooler inlet temperature. The target EGR fraction at each engine operation point which EGR control systems require was calculated based on the temperature relation among intake air, exhaust gas, and the mixed gas. Also, additional iteration was conducted to improve the EGR fraction map using the equation of state in the engine cylinder at IVC timing. Second, the characteristics of the injector were specified. The mass of fuel injected at each injection pulse and the timing of fuel injection were measured and entered in the injector model. Third, the effect by swirl control valve located between the intake manifolds and ports was considered in the combustion model. Swirl control valves induce the swirl motion in engine cylinders especially in low speed and low load region. Swirl motion in the cylinder intensifies the fuel-air mixing and this enhancement was reflected in the entrainment

rate multiplier, which is one of the combustion multipliers. Finally, the combustion model in cylinder was developed. The initial states(IVC timing) of the cylinder at each operation point were defined. Then, fuel burn rate was estimated from the pressure profile data measured by combustion analyzer and the combustion multipliers were optimized, so that combustion model can reproduce the fuel burn rate in real engine cylinders. The engine output predicted by the engine model showed good agreement with the measured data for the all operation points covering wide range of engine RPM and BMEP. Also, the NOx emissions from the engine were well predicted except some points. It is expected that the additional improvement for the injector model can raise the accuracy of this predictive model.

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요약

NOx 배출량 예측을 위한 물리 기반의 EGR 시스템 포함 디젤 엔진 모델 개발

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자동차 배기가스가 대기 오염의 원인 중 하나로 꼽히면서 자동차 배출물에 대한 규제 기준이 더욱 강화되고 있다. 특히, NOx 배출량 규제치는 계속해서 낮아지고 있고, 배출량 인증 시험의 주행 환경은 더 넓은 엔진 운전 영역을 포함하도록 진화하고 있다. 거기다가, 최근에 Euro 6d에서는 실도로 배출량에 대한 규제까지 포함하려 하고, 이에 따라 자동차 NOx 배출량을 줄이기 위한 다양한 방안들이 제시되고 있다. 이러한 맥락에서 여러 엔진 운전 영역에서 NOx 생성량을 예측하고 그 경향성을 이해하려는 다양한 연구들이 진행되었다. 그러나 대부분의 연구들이 배출량 예측에는 성공했지만 엔진이 운전될 때의 물리적인 상황에 대한 이해가 부족하거나, 다양한 엔진 운전 영역을 연구하기에는 너무 많은 계산을 요구하는 등의 한계점이 있었다. 따라서 본 연구에서는 NOx 배출량을 예측하기 위한 EGR 시스템을 포함한 디젤 엔

진 모델을 개발하되, 0D/1D 기반 모델을 활용하여 엔진과 각 파트에서의 물리적인 상황을 파악할 수 있는 형태로 발전시켰다. 각 엔진 파트는 타겟 차량 주행 결과에서 얻은 데이터를 기반으로 구현되었다. 첫째, EGR 시스템은 EGR 쿨러와 EGR 제어 장치로 구성하였고, EGR 제어 장치는 측정 데이터를 기반으로 얻은 타겟 EGR율 맵을 기반으로 작동되도록 하였다. 둘째, 펄스별 분사량과 분사 타이밍 같은 인젝터 운행맵을 측정 데이터로부터 얻어서 모델에 반영하였다. 셋째, 스월 제어 밸브에 의한 연료-공기 혼합 증대 효과를 연소 모델에 반영하였다. 마지막으로, 연소 해석기에서 얻은 운전점별 실린더 압력 프로파일과 같은 데이터를 통해 엔진 실린더 내 연소 모델을 보정하였다. 엔진 IMEP과 NOx 생성량에 대한 모델 결과값은 전 운전 영역에 걸쳐서 측정치와 비슷한 수준의 값을 내었다.

주요어: 디젤 엔진, NOx(질소 산화물), 연소 모델, EGR 시스템

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