Effects of Injection Rate Profile on Combustion Process and Emissions in a Diesel Engine

Fuqiang Bai[1], Zuowei Zhang[1], Yongchen Du[2], Fan Zhang[1], Zhijun Peng[3, 2*]

[1]Key State Lab of Engines, Tianjin University, China
[2]School of Engineering and Technology, University of Hertfordshire, UK
[3]Faulty of Creative Arts, Technologies and Science, University of Bedfordshire, UK

*Corresponding author: Faulty of Creative Arts, Technologies and Science, University of Bedfordshire, UK, jun.peng@beds.ac.uk

ABSTRACT

When multi-injection is implemented in diesel engine via high pressure common-rail injection system, changed interval between injection pulses can induce variation of injection rate profile for sequential injection pulse, though other control parameters are same. Variations of injection rate shape which influence the air-fuel mixing and combustion process will be important for designing injection strategy. In this research, CFD numerical simulations using KIVA-3V were conducted for examining the effects of injection rate shape on diesel combustion and emissions. After the model was validated by experimental results, five different shapes (including rectangle, slope, triangle, trapezoid and wedge) of injection rate profiles were investigated. Modelling results demonstrate that injection rate shape can have obvious influence on heat release process and heat release traces which cause different combustion process and emissions. It is observed that the baseline - rectangle (flat) shape of injection rate can have better balance between NOx and soot emissions than other investigated shapes. As wedge shape brings about the lowest NOx emissions due to retarded heat release, it produces highest soot emissions among five shapes. Trapezoid shape has the lowest soot emissions, while its NOx is not the highest one. The highest NOx emissions was produced by triangle shape due to higher peak injection rate.

INTRODUCTION

High Pressure Common Rail (HPCR) fuel injection system have provided significant benefit for optimizing air-fuel mixing and controlling ignition, combustion and emissions in diesel engines. With the aid of electronic controllers, HPCR system allows adjustment of injection pressure, fuel injection amount, injection timing and injection pulse number in each combustion cycle very flexibly [1-3]. Then optimised injection strategies can improve diesel engine combustion for low combustion noise, high combustion efficiency and low emissions [4, 5]. It can also provide post injection for help after-treatment system for regeneration. In recent years, as piezo injectors are widely used for HPCR injection systems [6, 7], injection timing and injection duration for each pulse can be controlled more precisely. Then multi-injection strategies are employed more and more for minimising emissions and fuel consumption.

When multi-injection strategies are employed, it has been realised that pressure waves that exist in HPCR system can result in significant effects on injection pressure and injection rate of sequent injection pulses [8, 9]. When an injection pulse is completed then the injector valve is closed, a pressure waves inside the fuel pipe does oscillate between the injector valve and the common rail. Even the pressure in the rail is rather steady during an injection, there is still pressure wave induced by a close of injector valve. Then the actual injection pressure and injection rate of sequent injection pulse will be changed by the pressure wave [10, 11]. When the start timing of a sequent injection pulse is just at the peak of pressure wave, its injection pressure at the beginning will be higher than rail pressure. Otherwise, the injection pressure will be lower than the rail pressure. Then intervals between two adjacent injection pulses play very important role for managing the injection pressure and injection rate of sequent injection pulse.

Currently, most injection strategies for diesel engines are designed to avoid different injection pressure between injection pulses, in order to have simple control to combustion processes. However, this kind of arrangements may not the best option for achieving optimal combustion and lowest emissions. Increased or reduced injection pressure and then varied injection rate even injection rate shape for a sequent injection pulse may result in required improvement to air-fuel mixing and combustion rate under specific engine operation condition. For this issue, more investigations are still needed for achieving a clear idea how varied pressure, varied injection rate and varied injection rate shape between injection pulses affect the combustion process and emissions.

Previous researches have suggested that effects of injection rates and their shape on air-fuel mixing and combustion processes are significantly important to organising diesel combustion [12, 13]. Suh [13] has conducted experiments on a high-speed direct-injection (HSDI) diesel engine for which compression ratio is 15.3:1 reduced from 17.8:1 by modification of combustion chamber shape to investigate the effects of the twin-pilot-injection strategies on combustion performance and exhaust emissions. The study shows lower NOx emissions (up to 45.7% were observed) whereas soot generation level almost unaltered in the multiple-injection case. Desantes et al. [14] investigated the effects of boot-type rate-shapes on engine performance and emissions. From their two-part study, they concluded that long boot length and low boot pressure decreased NOx emissions but increased brake specific fuel consumption (BSFC) and soot emissions. Moreover, they discovered that boot-type rate-shapes caused substantial change to the diffusion combustion regime as compared to the premixed combustion regime.

There have been a number of researches for examining injection rate of HPCR system, in particular for those single injection pulse. In multiple injection, it still lacks adequate understanding to the

injection rate with flexible injection interval, in particular with consideration of pressure wave's effects. The present research is applying CFD numerical simulation to examine how varied injection rate will affect combustion process and emissions in diesel engines. A full combustion model of diesel engine including sub-models of fuel injection, evaporation, air-fuel mixing, combustion and emissions was validated by required experimental results. Then combustion process and emissions were investigated by considering five injection rate shapes which are possibly produced by HPCR systems under influence of pressure wave. Simulation results provide an insight into effects of injection rate shapes on diesel combustion and emissions.

NUMERICAL MODEL

Model Description

Numerical simulations were conducted by using KIVA-3V code [15], which was improved by introducing several sub-models, as shown in Table 1. The sub-models introduced have been tested by previous researchers [16] and it has been suggested that these new sub-models are appropriate for diesel combustion. For resolving the turbulent flows in cylinder, the Renormalization Group (RNG) k—ε turbulence model [17] was used. The heat transfer from the wall was computed by the model developed by Han and Reitz [18] which counted the variations of both gas density and the turbulent Prandtl number in the boundary layer.

The spray process was modeled by a particle method, where the break-up processes of injected droplets were simulated by a Kelvin-Helmholtz Rayleigh-Taylor (KH-RT) model [19]. The collision model used here was one developed by Nordin [20] with improved grid independence. The interaction between spray and wall was represented by the model introduced by Han *et al.* [21], which considered the effects of gas density variation in simulating the size of secondary droplets in splashing.

The CHEMKIN [22] solver was coupled with KIVA-3V code to compute the chemical reaction. A reduced n-heptane reaction mechanism [23] was used to simulate diesel fuel chemistry, where the soot formation was solved by a phenomenological model, and NOx formation was represented with extended Zeldovich mechanism. From simulation results [24], it has shown fair agreements with experimental results could be achieved when the simulating method was used.

Table 1 Computational Submodels.

Turbulent Model	RNG k-ε Model [17]
Break Up Model	KH-RT Model [19]
Collision Model	Nordin Model [20]
Splash Model	Han *et al.* Model [21]
Heat Transfer (wall)	Han – Reitz Model [18]
Combustion	CHEMKIN [22]
Fuel Chemistry	Reduced N-Heptane Mechanism [23]
Soot Model	Phenomenological Model [24]

NOx Mechanism	Extended Zeldovich Mechanism 24]

Engine Specifications

The engine used in this numerical study was a HSDI (High Speed Direct Injection) diesel engine, same as used for experimental investigation by Herfatmanesh *et al* [9]. The engine has four valves and a displacement of 0.55 litre per cylinder. Most of the engine parameters were maintained the same as those used in original experiments for the model validation except for some modifications made on the injector for further combustion simulations. The specifications of the engine are listed in Table 2.

Table 2 Engine simulation specifications for CFD

Parameter	Unit	Detail
Engine type		2.2 L 4 cylinder turbocharged
Bore	mm	86
Stroke	mm	94.6
Valve number		4/cylinder
Compression ratio		18:1
Turbocharger		VGT
Fuel injection system		Common rail
Injector hole diameter	mm	0.12
Injector hole number		6/injector
Peak torque	Nm	360 @1500 rpm
Peak power	kW	155 @3800 rpm

Computational Grid

Since the diesel injector used in the research has 6 equally distributed orifices, the combustion chamber was represented by a 60º sector grid with periodic boundary conditions, as shown in Figure 1. The grid for the cylinder volume at TDC (Top Dead Centre) has approximately 25,231 hexahedral cells. The typical cell size is smaller than 3 mm in three dimensions, which is at the similar level used by Kim et al. [25]. Kim et al. also employed KIVA-3V for their simulation with grids (2.2x2.2x3.0 mm^3) and their results have demonstrated adequate confidence of precision for combustion

simulation. Therefore the grid used here is considered to yield reasonable accuracy for those predictions.
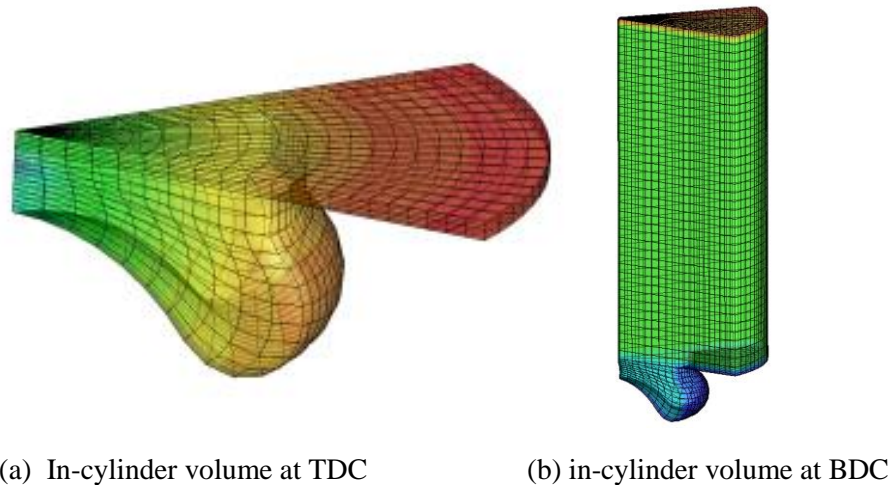

(a) In-cylinder volume at TDC   (b) in-cylinder volume at BDC

Figure 1 Computational mesh ((a) and (b) are not in same scale)

Model Validation

The validation presented here was mainly combined with engine experimental results presented by previous researchers [28]. The fuel mass per cycle for the validation was 1 mg for pilot injection and 21 mg for main injection. The engine speed is 1800 rpm and the injection pressure is 180 MPa. Other operating conditions were maintained same as described in [28]. Figure 2 shows that the comparison of the in-cylinder pressure between the simulation and experimental results. From the Figure, it can be observed that a good agreement has been achieved between experimental and simulation results. For emissions which are shown in Figure 3, both NOx and soot emissions have been compared between experimental results and simulation by varying the main injection timing. Those results show that the model can predict emissions with necessary accuracy.
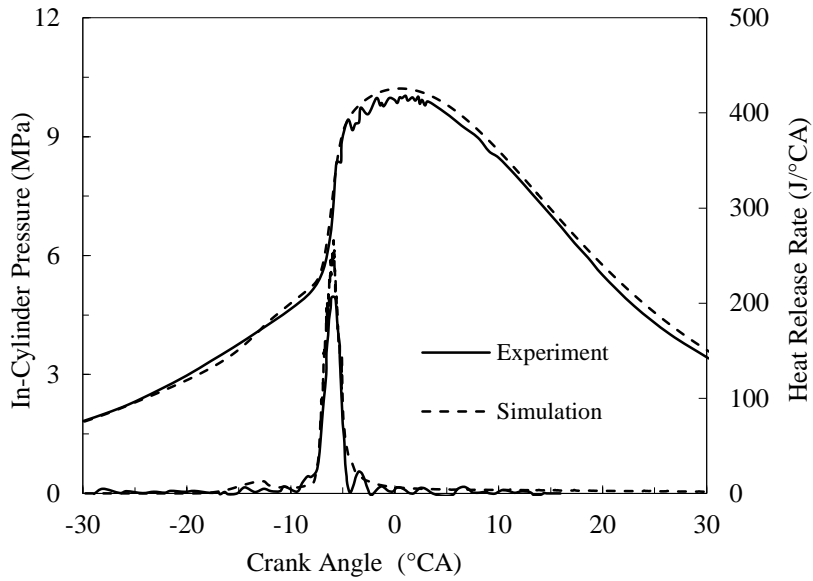
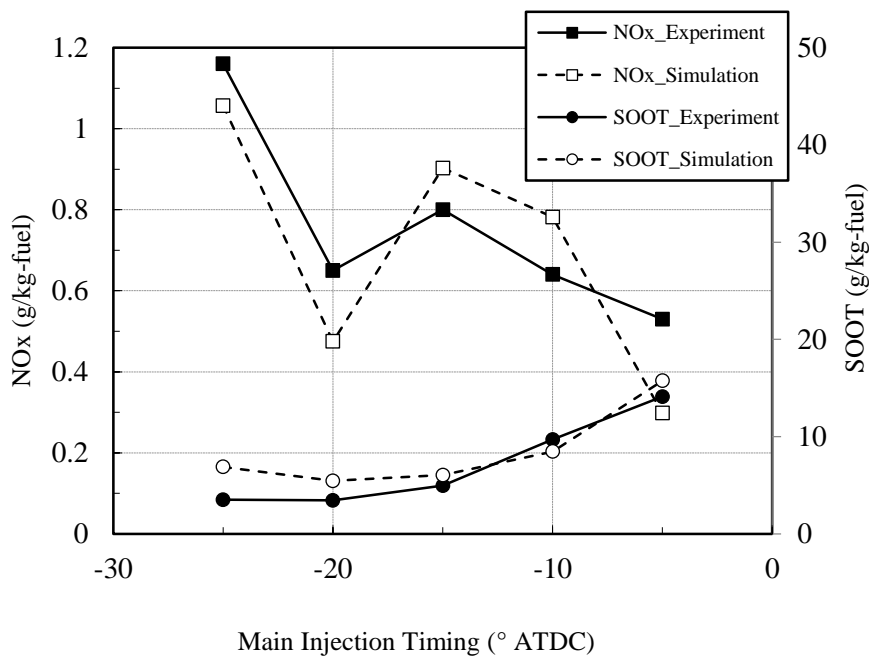Figure 2 Comparison of in-cylinder pressure traces between experimental results and simulation

Figure 3 Comparison of NOx and Soot emissions between experimental results and simulation

SIMULATION CONDITIONS

From previous experimental investigation, it has demonstrated that the sequential injection pulse can have very different injection rate profile. While other parameters are constant, increasing interval between injection pulses can result in the injection rate shape of sequential pulse gradually becoming

more flat. The injection rate shape of sequent pulse for interval increased from 750 μs to 2500 μs are very different, as shown in Figure 4,.

It can be seen in Figure 4 that average injection rates for different injection intervals are very different. With the increase of interval, the average injection rate keeps decreasing, in particular from 1250 to 1500 and then to 1750 μs. Those injection rates of intervals less than 1250 μs have similar trend and shape. They have a high peak at the start point, then second low peak after some reduction. The final stage has a smooth decreasing slope. The injection rate of 1500 μs has some delay at the beginning, then displays similar trend as those of less than 1250 μs. After the interval increases over 1750 μs, all traces of injection rates have very similar profile – after a longer delay than that of 1500 μs, then having very slow increase and long and flat persistence, and finally decreasing quickly.
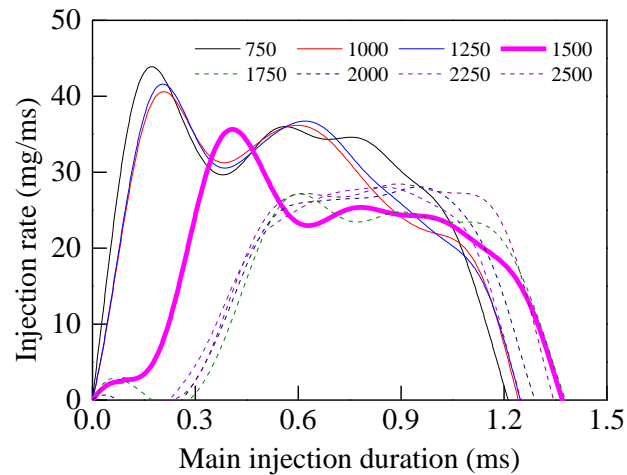

Figure 4 Experimental results of injection rates of second injection pulse (pulse width of 600 μs – 600 μs, interval shown in the figure, 80 MPa injection pressure)

Considering practical injection rate shape's variation, in this research, five different injection rate shapes were selected to investigate the effects. As shown in Figure 5, among the five different injection rate shapes, the total fuel amount, pulse width and average injection rate are maintained same. While the rectangle shape has totally flat injection rate, the slope one has smooth increase across all pulse, and the wedge one has a little more steep increase at the beginning then keeps flat at late stage. The triangle shape consists of only an increase stage and then a decrease stage with highest peak rate at the middle point. The trapezoid has the faster increase at beginning and fast decrease at finish stage but have some flat stage in between.

Although those shapes are considered here more for theoretical analysis, resemblant profiles may be produced under certain injection conditions, in particular the trapezoid which is more close to most

practical injection rate profiles. In order to study the effect of injection rate shapes, it was assumed that all the fuel injected before TDC with the injection timing at 8.6 ºCA BTDC. The fuel amount is 49.2 mg.
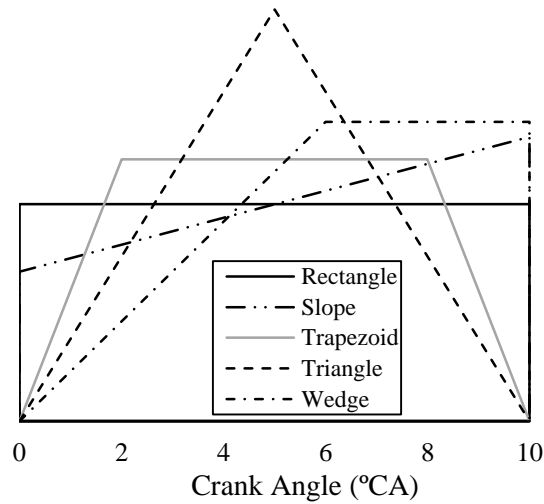

Figure 5 Different injection rate shape used for CFD simulation

RESULT AND DISCUSSIONS

Effects on Combustion Process and Heat Release Rate

The in-cylinder pressure traces and heat release rate traces for five cases are presented in Figure 6. As the rectangle one has the highest injection rate at the start point, its ignition timing is the earliest one and at the beginning its heat release rate has a faster increase than other shapes. As the triangle one has the highest injection rate close to TDC, it produces a highest heat release rate near TDC.
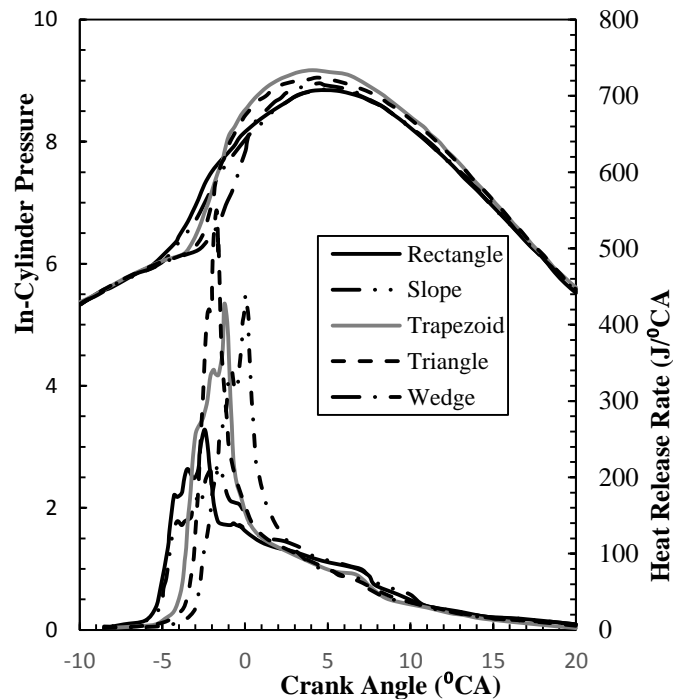
Figure 6 Simulated in-cylinder pressure traces and heat release rate traces for five different injection rate shapes

During late stage of combustion, as the rectangle one and slope one do not get further increase too much of injection rate, increases in heat release rates are not so obvious compared to other three shapes. The wedge one has a later peak at late stage because of its highest injection rate for a long time at the late injection stage.

Basic process can be analysed by inspecting the in-cylinder temperature distribution. The results at four different crank angles are shown in Figure 7. At the beginning and late stage of fuel injection, it shows that in-cylinder temperature distributions are very similar for all the five shapes. But by the mid-stage combustion (about 5 ºCA ATDC), triangle and trapezoid have obviously bigger areas of high temperature than other three shapes. This may be due to the higher injection rate during middle injection stage for those two shapes.
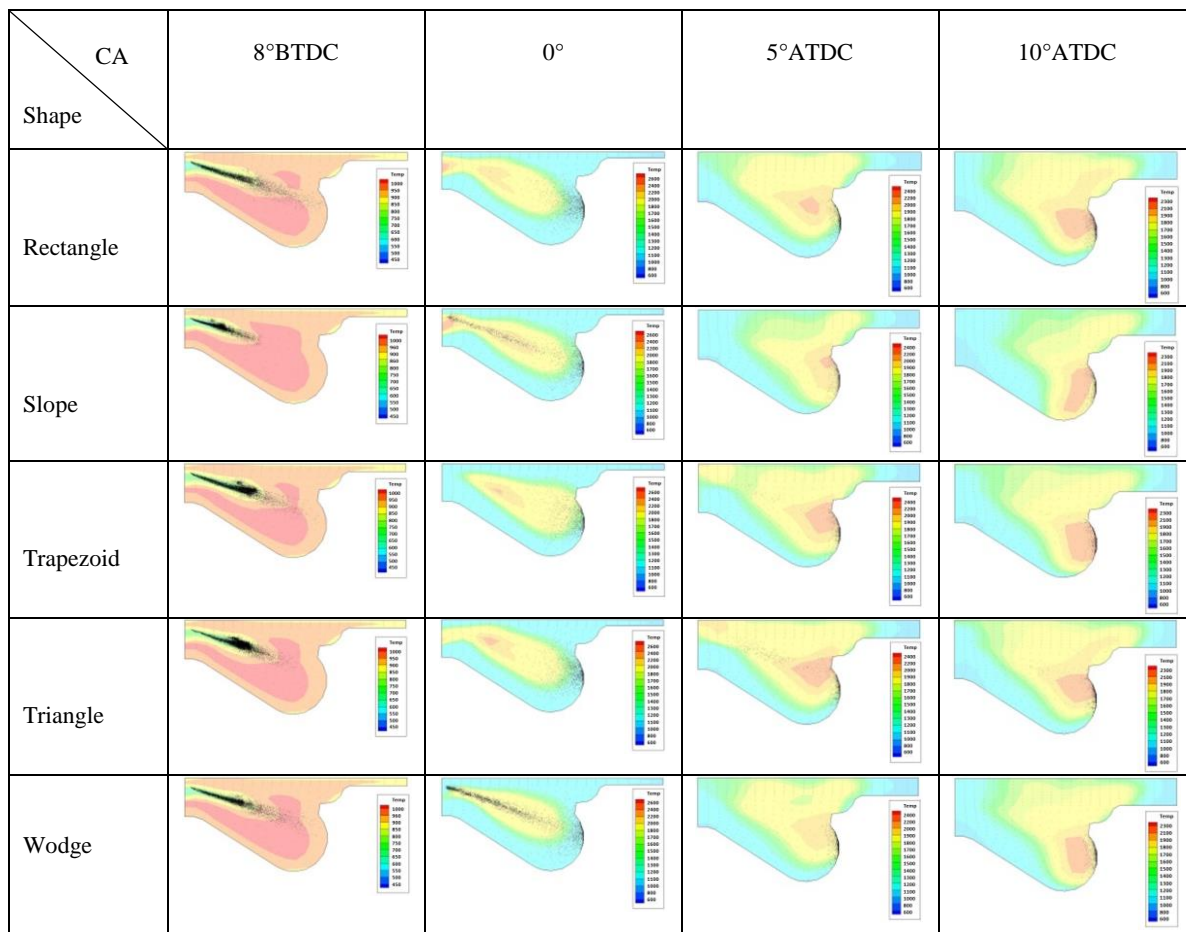
CA \ Shape	8°BTDC	0°	5°ATDC	10°ATDC
Rectangle				
Slope				
Trapezoid				
Triangle				
Wodge				

Figure 7 In-cylinder temperature distributions at 8 ºCA BTDC, TDC, 5 ºCA ATDC, 10 ºCA ATDC for five different injection rate shapes

At the late stage of combustion (such as 10 ℃A ATDC as shown in Figure 7), all five have similar temperature distribution, although wedge has the smallest area of high temperature (red colour) and slope has the smallest area of mid-high temperature (yellow colour). In Figure 7, it can be observed that slope and wedge shapes have still obvious fuel spray at TDC, while other shapes' injection have almost totally stopped. This may be due to higher injection rate at late injection stage for slope and wedge shape.

By analysing the in-cylinder temperature traces from in-cylinder pressure traces for five shapes, from results as shown in Figure 8, it can be observed that they are consistent with information shown in Figure 7. Although the initial temperature increase did not take place too early for trapezoid and triangle shapes, their temperatures increase very rapidly and maintain high values than other three shapes. For wedge shape, its temperature is not so high although its peak value of heat release is about twice higher than that of rectangle shape. The main reason may be that its most heat release takes place at a later stage then other shapes.
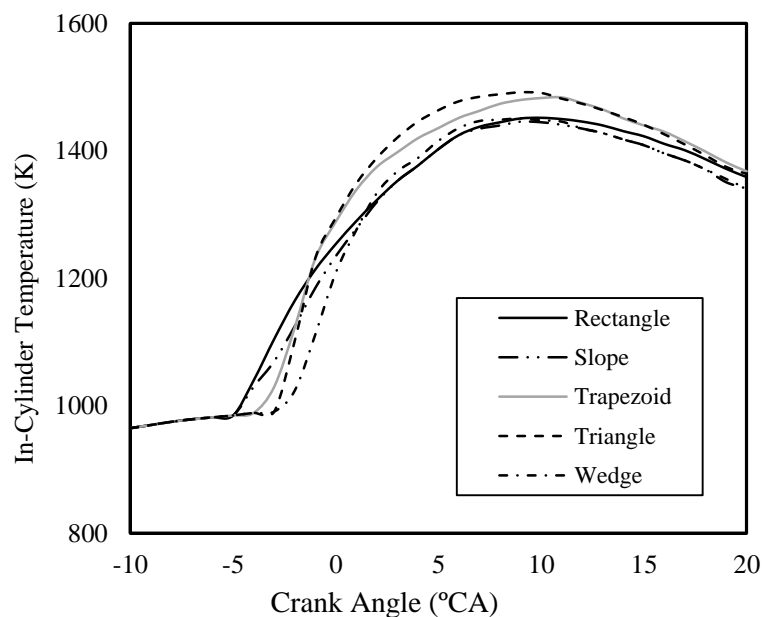

Figure 8 Simulated in-cylinder pressure traces and heat release rate traces for five different injection rate shapes

Effects on Emissions

Based on the difference of in-cylinder peak temperature (as shown in Figure 8), the triangle shape shows the highest NOx emissions and the wedge shows the lowest NOx emissions, as shown in

Figure 9. A special point is that the rectangle shape has lower peak temperature than the slope one, but its NOx emissions are higher. By checking the in-cylinder temperature distributions of two shapes (as shown in Figure 7), it can be observed that found the high temperature area of the rectangle shape is bigger than the slope one at the crank angle where the combustion will completes.
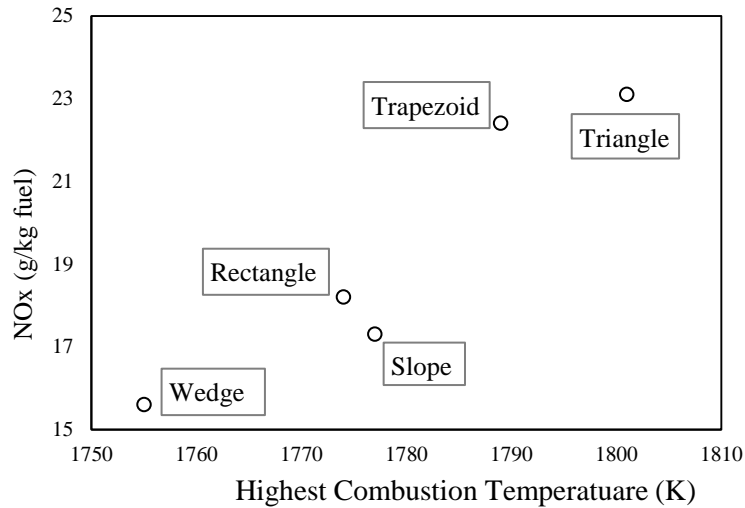

Figure 9 Simulation results of NOx emissions for five different injection rate shapes

Soot and NOx emissions as shown in Figure 10 basically show their relation as trade-off. But triangle one which has the highest NOx emissions does not follow the trend to have lowest soot emissions. By reviewing the heat release rates, it can be seen that its peak heat release rate too early than other. The reason for the more soot emissions are that a lot of fuel was injected due to the highest injection rate. Hence, a liquid film may be formed inside the piston bowl as shown in figure 7. If both NOx and soot emissions are considered as important factors, the rectangle shape can have better compromise between them, while the triangle shape is the worst case. For practical applications, this means that high peak in injection rate shape should be avoided.
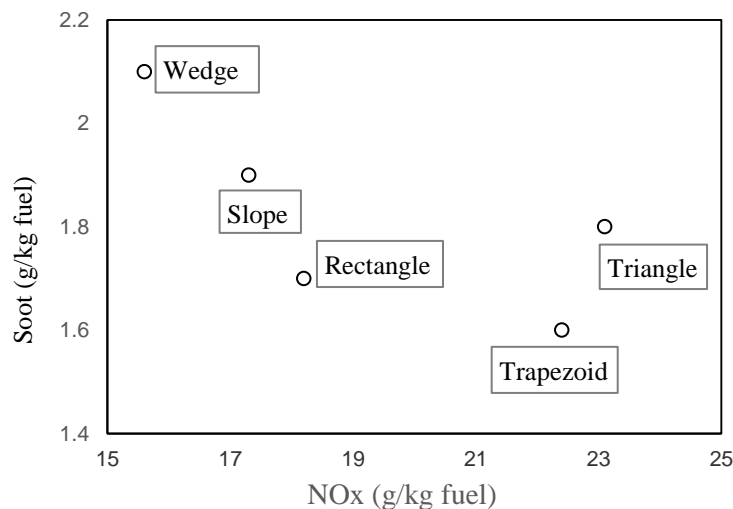
Figure 10 Simulation results for soot and NOx emissions of five different injection rate shapes

Figure 11 shows the results of CO and HC emissions. For triangle shape, because it has the highest in-cylinder temperature, it HC emissions is the lowest one. But its CO emissions are the highest one. This may be due to the poor air-fuel mixing while most fuel is injected during middle stage with very high injection rate. The wedge shape has also very high CO emissions since it has high injection rate at the late injection stage. It's HC emissions are the highest one. This is obviously due to its low combustion temperature. Slope and rectangle have low CO emissions but high HC emissions. This may be because their injection rates are flat than other shapes. In the one hand, the flat injection rate may result in better mixing, then produce lower CO emission. On the other hand, their low combustion temperatures contribute to high HC emissions. It can be noticed that slope has both lower CO and HC emissions than rectangle shape. This suggests slow increase of injection rate can benefit both HC and CO emissions, compared to totally flat injection rate.

By having this set of simulations, it suggests that different injection rate shape for individual injeciton pulse can also have influence on combustion process and emissions. The rrectangle shape can have better balance between NOx and soot emissions, because other shapes with higher injection rate at some point (temporal) will result in higher NOx or soot emissions.
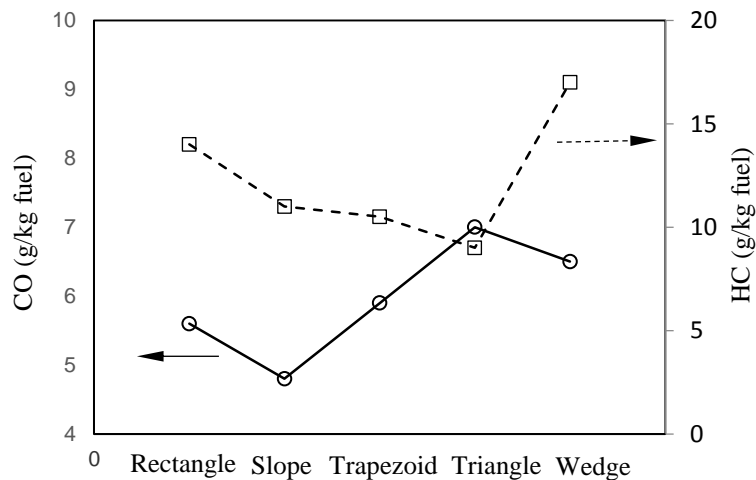
Figure 11 Simulation results for CO and HC emissions of five different injection rate shapes

CONCLUSIONS

In the research work presented, influences of variations of injection rate shape caused by different injection intervals were examined by numerical simulation based on KIVA-3V CFD code. From the results, the following conclusions have been derived.

- Injection rate shape has an influence on diesel combustion process, in-cylinder mixing, heat release rate and in-cylinder temperature distribution. The triangle shape has the highest peak combustion temperature due to too high injection rate at some injection point. The Wedge shape's combustion temperature is the lowest across most combustion stage due to its late ignition timing.

- Rectangle shape can have better balance between NOx and soot emissions, while other shapes with higher injection rate at some point (temporal) will result in higher NOx or soot emissions.

- The triangle shape produces the highest NOx emissions due the highest combustion temperature. The trapezoid shape shows the lowest soot emissions due to both better mixing and high combustion temperature.

- The slope shape has the lowest CO emissions and not very high HC emissions. Just for CO and HC emissions, it is better than rectangle shape.

CONFLICT OF INTEREST

The author(s) declare(s) that there is no conflict of interest regarding the publication of this paper.

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ACRONYMS

ATDC	after top dead centre
BTDC	before top dead centre
CA	crank angle
CFD	computational fluid dynamics
CO	carbon monoxide
EOI	end of injection
HC	hydrocarbon
HPCR	high pressure common rail
HRR	heat release rate
HSDI	high speed direct injection
NOx	nitrogen oxides
PM	particulate matters
TDC	top dead centre