Applied Mathematics and Nonlinear Sciences

<https://www.sciendo.com>

Effects of internal/external EGR and combustion phase on gasoline compression ignition at low-load condition

Jingyu Gong¹, Binbin Yang^{1†}, Leilei Liu¹, Yue Liang¹, Zhifa Zhang¹, Fan Zhang²¹School of Transportation and Vehicle Engineering, Shandong University of Technology, Zibo 255000, China²State Key Laboratory of Engines, Tianjin University, 135 Yaguan Rd, Tianjin 300350, China

Submission Info

Communicated by Juan Luis García Guirao

Received December 20th 2020

Accepted January 31st 2021

Available online April 8th 2021

Abstract

A single-cylinder test engine model was built by GT-Power software, and the effects of internal exhaust gas recirculation (i-EGR), external EGR (e-EGR), i-EGR/e-EGR coupling and the crank angle degree at which 50% of total heat loss has taken place (CA50) on combustion and emission characteristics of gasoline compression ignition at low-load condition were analysed. The results show that the ignition delay period with e-EGR was extended slightly with the increased EGR ratio, while that with i-EGR strategy first shortened and then extended, and that the optimised indicated thermal efficiency could be achieved using a small amount of i-EGR. With the same EGR ratio, nitrogen oxide (NO_x) emission is more likely to be suppressed by i-EGR, while soot emission was more deteriorated, and the superior trade-off relationship between carbon monoxide (CO)/hydrocarbon (HC) emissions and NO_x emission was attained by the combination of lower i-EGR ratios and CA50 closed to top dead centre. When using i-EGR/e-EGR coupling with total EGR ratio being fixed, the indicated thermal efficiency was decreased by increasing i-EGR ratio, while the lower NO_x, CO and HC emissions could be realised, but only with the consequence that soot emission increased.

Keywords: gasoline compression ignition, internal EGR, external EGR, combustion phasing

1 Introduction

With great concern about the enormous demand for energy, several industry and policy initiatives (mainly in Europe and the USA) aim at shifting the powertrain from internal combustion engine (ICE) to gasoline/plug-in hybrid or fuel cell [1]. However, despite the growing number of new power sources, it is expected that by 2040, oil-based conventional energy conversion devices will provide about 90% of energy for transportation [2]. Thus, ICE will still be the primary power plant and widely used from small-scale application to a high-power

[†]Corresponding author.Email address: yangbinbin@sdut.edu.cn

generator (e.g. power plants) for a long time [3]. But the increasingly severe international energy situation and high-standard emission regulations have put forward new requirements for the ICE industry.

In the attempt for high efficient and clean combustion, the researchers have demonstrated that higher thermal efficiency can be obtained by compression ignition (CI) engines due to the high compression ratio and a lack of pumping loss as compared with spark-ignition engines [4,5], but the spray diffusion combustion always results in higher nitrogen oxide (NO_x) and soot emissions [6,7]. Low-temperature combustion (LTC) combustion realised by injection close to TDC, higher EGR ratio and injection pressure was first proposed by Akihama et al., and the combustion temperature and local equivalence ratio are controlled to a sufficiently low level for avoiding the NO_x and soot formation regions on the Φ -T map [8]. Then, Kalghatgi et al. [9,10] proposed to realise partially premixed combustion by the direct injection of gasoline fuel using a high-pressure common rail system, referred to gasoline compression ignition (GCI). Compared with diesel fuel, applying gasoline fuel with high octane number (long ignition delay period) and volatility to CI engines is more likely to obtain well-premixed or properly stratified mixture without very high injection pressure and EGR ratio [11–13], while the combustion is transited from fuel–air mixing control of the conventional diesel combustion to the cooperative control of fuel–air mixing and chemical kinetics. Numerous relevant studies [14–16] have shown that, GCI combustion can expand the high-load limit effectively with the improved thermal efficiency and has the potential to reduce NO_x and soot emissions significantly under medium to high load conditions. However, considering the poor ignition quality of gasoline, it is difficult for the auto-ignition function to work, especially under low-load conditions, owing to the low in-cylinder temperature and pressure, which lead to the increase of cycle-to-cycle variation and even misfires. Also, it is not suitable for the use of after-treatment technology because of the low exhaust temperature under such operation conditions [17].

To solve the above problems, the researchers have studied the technologies for improving the combustion stability of GCI combustion under low-load conditions such as wide-distillation fuels [18], intake preheating [19], intake boost [20], injection strategy [21] and internal exhaust gas recirculation (i-EGR) [22,23]. Both intake preheating and intake boost can improve the combustion process directly but quite laborious in application due to the low exhaust energy and temperature [19,20]. As for injection strategies, a single injection is widely used in low-load GCI combustion to avoid excessive fuel mixing [21]. i-EGR can be realised by the variable valve actuation (VVA) system, which makes full use of the hot residual gas to improve the initial thermodynamic state of the in-cylinder charge. Meanwhile, the fuel can also be reformed to a large number of such smaller molecules as acetylene with high reactivity, which can promote the auto-ignition [22–24]. Borgqvist et al. [22], have explored the effect of i-EGR realised by both negative valve overlap (NVO) and exhaust valve rebreathing (2-EVO) strategies on GCI combustion. The results have shown that i-EGR realised by NVO can effectively improve the combustion stability of GCI combustion under low-load conditions, but the decreased gas-exchange efficiency leads to the deterioration of indicated thermal efficiency [25]. By contrast, 2-EVO strategy could achieve better fuel economy along with the improved exhaust temperature. However, i-EGR will accelerate the combustion reaction rate, resulting in higher NO_x emission. In case external EGR (e-EGR) could be coupled, the in-cylinder temperature and residual gas ratio can be cooperatively controlled to optimise the low-load GCI combustion process [26]. Therefore, it is essential to further study i-EGR/e-EGR coupling strategy under specific operating conditions.

Nomenclature			
ICE:	Internal combustion engine	HC	Hydrocarbon
2-EVO:	Exhaust valve rebreathing strategy	GCI:	Gasoline compression ignition
ATDC:	After top dead centre	i-EGR:	Internal exhaust gas recirculation
CA:	Crank angle	IVC:	Intake valve close
CA10:	The crank angle degree at which 10% of total heat loss has taken place	IVO:	Intake valve open
CA50:	The crank angle degree at which 50% of total heat loss has taken place	LTC:	Low-temperature combustion
CA90:	The crank angle degree at which 90% of total heat loss has taken place	NO _x :	Nitrogen oxide
CI:	Compression ignition	NVO:	Negative valve overlap strategy
CO	Carbon monoxide	rpm:	Revolutions per minute
e-EGR:	External exhaust gas recirculation	SI:	Spark ignition
EGR:	Exhaust gas recirculation	TDC:	Top dead centre
EVC:	Exhaust valve close	VVA:	Variable valve actuation
EVO:	Exhaust valve open	IMEP	Indicated mean effective pressure

As an important parameter for describing the combustion process, combustion phasing (the crank angle degree at which 50% of total heat loss has taken place [CA50]) is widely used in the research of advanced combustion technologies [27–30]. Tan et al. [27] have made a detailed analysis of the factors affecting the process of LTC, which shows CA50 is an important factor for the thermal efficiency and closely related to the emissions. From the experimental research with wide-distillation fuel conducted by Du et al. [29], under the load of indicated mean effective pressure (IMEP) at 0.51 MPa, the indicated thermal efficiency can reach the peak value when the CA50 is at 5°CA after top dead centre (ATDC), and the lowest fuel consumption rate can be achieved by blending 40% gasoline. Based on the analysis of the emission formation mechanism, Ickes [30] has concluded that NO_x emission mainly depends on CA50 and EGR ratio under LTC mode.

The behaviour of GCI combustion is perceptibly different from that of the traditional diesel engines mainly because it is affected not only by the heating and dilution effects but also by the specific heat capacity and the active products [31]. To achieve high-efficient and stable GCI combustion, EGR ratio and CA50 have to be optimised. Therefore, one-dimensional numerical simulation has been carried out in this article to further investigate the effect of CA50 on combustion and emission characteristics of GCI combustion under low-load condition, with i-EGR, e-EGR and i-EGR/e-EGR coupling strategies, respectively.

2 Materials and methods

In the present study, a single-cylinder, water-cooled and four-stroke engine is used. The engine is equipped with a typical high-pressure common rail system and a VVA system, and the geometric compression ratio is 16:1. Detailed engine and injector specifications are listed in Table 1. The high-pressure common rail system

Table 1 Engine and injector specifications

Parameter	Value
Bore (mm)	105
Stroke (mm)	125
Displacement (L)	1.08
Geometric compression ratio	16:1
Connecting rod length (mm)	210
Squish height (mm)	0.85
Valves/cylinder	4
Swirl ratio	1.5
Number of holes	8
Hole diameter (mm)	0.15
Injection pressure (MPa)	50
Cone angle (°)	150

Table 2 Engine operation parameters

Parameter	Value
Speed (rpm)	1500
IVO (°CA ATDC)	-377
IVC (°CA ATDC)	-133
EVO (°CA ATDC)	125
EVC (°CA ATDC)	Variable
P intake (MPa)	0.15
T intake (K)	323
Intake O ₂ concentration	21%
Coolant temperature (°C)	85

enables the flexible settings of injection timing, common rail pressure and injection mass. The VVA system is used to control both the timing and lift of intake and exhaust valves.

The engine speed was kept at 1500 rpm, and the load of nearly all test points was about 0.5 MPa IMEP. The intake pressure and temperature were maintained at 0.15 MPa and 323 K after intake boosting. The coolant temperature was maintained at 85°C. More details about engine operation condition are listed in Table 2. The test fuel in this study is gasoline, with 92 research octane number. The physical and chemical properties of the test fuel are listed in Table 3.

Table 3 Physical and chemical properties of the test fuel

Property	Value
Fuel mass (mg/cycle)	28
Molecular formula	C ₈ H ₁₈
Density (kg/m ³ @ 20°C)	740
Lower heating value (MJ/kg)	44.3
Research octane number	92
Self-ignition temperature (°C)	420
Heat of vaporisation (kJ/kg @ 20°C)	308
Critical temperature (K)	500
Absolute entropy (J/kgK @ 20°C)	3704

Table 4 Verification of emission results

Property	Experiment results	Simulation results
NO _x	13.76	11.12
Soot	0.005	0.055
CO	21.47	20.85
HC	7.87	3.96

A single-cylinder diesel engine model was established by GT-Power software based on the engine specifications, as shown in Figure 1. Quasi-dimensional multi-zone combustion model ‘EngCylCombDIJet’, Woschni heat transfer model and flow model ‘EngCylFlow’ are used in numerical simulations. The energy conservation equation of the working fluid in the combustion process is shown in formula (1).

$$\frac{d(m_b e_b)}{dt} = -p \frac{dV_b}{dt} - Q_b \left(\frac{dm_f}{dt} h_f + \frac{dm_a}{dt} h_a \right) \quad (1)$$

Where, m_b , m_f and m_a are burned zone mass, fuel mass and air mass, respectively; e_b is burned zone energy; p is cylinder pressure; V_b is volume; Q_b is heat transfer; h_f is the enthalpy of fuel.

The heat transfer Q of each wall of the combustion chamber can be calculated by heat transfer formula (2):

$$\frac{dQ_\omega}{d\phi} = \sum_1^3 \frac{dQ_w}{d\phi} = \frac{1}{\omega} \sum_1^3 \alpha_g \times A_i (T - T_w) \quad (2)$$

Where, ω is angular velocity, α_g is instantaneous average heat transfer coefficient, A is heat transfer area, T and T_w are temperatures of in-cylinder charge and cylinder wall.

Figure 2 and Table 4, respectively, show the comparisons of cylinder pressure and emissions between numerical simulation and engine test. Except for soot emission, the results have a high degree of agreement, which indicates the model can accurately reflect the actual engine operation.

Fig. 1 Engine simulation model.

i-EGR was realised through 2-EVO strategy, and the lift curves of intake and exhaust valves are shown in Figure 3. The i-EGR ratio was adjusted by changing the opening degree of the back pressure valve (back valve in Figure 1). As the opening degree of back pressure valve increasing, more residual gas would be trapped in the cylinder, which result in the increase of i-EGR ratio. The calculation formula of i-EGR ratio is as follows [32]:

$$R_{EGR} = \frac{m_{e-EGR} + m_{i-EGR}}{m_{fresh} + m_{e-EGR} + m_{i-EGR} + m_{residual}} \quad (3)$$

Fig. 2 Model verification.

Fig. 3 Valve lift profiles.

Where, m_{e-EGR} is the exhaust mass (kg) passing through the external intercooler; m_{i-EGR} is the exhaust mass entering the cylinder during the 2-EVO; m_{fresh} is the fresh intake mass (kg); $m_{residual}$ is the mass (kg) of residual gas in the cylinder at IVC. When using the 2-EVO strategy, the external intercooler valve (EGR valve in Figure 1) is closed, and m_{e-EGR} remains at 0. The ratios of i-EGR and e-EGR are supposed to be equal by means of the same in-cylinder oxygen concentration.

3 Result and discussion

In this article, the results are divided into three sub-sections, and the test points of the three groups are listed in Table 5. Injection timing was varied to control CA50 in terms of EGR strategies and ratios. Group 1 and Group 2 were set up to investigate the effects of i-EGR/e-EGR and CA50 on GCI combustion and emission characteristics under low-load condition, respectively. With injection timing fixed at -18°CA ATDC , Group 3 was set up to investigate the effects of i-EGR and e-EGR ratios on GCI combustion behaviour under the i-EGR/e-EGR coupling strategy, while the total EGR ratio was kept at 50%, and i-EGR/e-EGR ratio of each case was set to 5%/35%, 10%/30%, 15%/25%, 20%/20%, 25%/15%, 30%/10% and 35%/5%.

Table 5 Test points of three groups

Control parameter	Group 1	Group 2	Group 3
Injection timing ($^{\circ}\text{CA}$ ATDC)	-38~ -12	-38~ -12	-18
i-EGR ratio	0~ 50%	0%	5%, 10%, 15%, 20%, 25%, 30%, 35%
e-EGR ratio	0%	0~ 50%	35%, 30%, 25%, 20%, 15%, 10%, 5%

3.1 Effects of EGR and CA50 on GCI combustion characteristics

Figure 4 shows the variations of the ignition delay period as a function of i-EGR ratio and e-EGR ratio under different injection timings. The ignition delay period is defined as the crank angles between injection timing and the crank angle degree at which 10% of total heat loss has taken place (CA10). It is evident that the ignition delay period is increased gradually by advancing injection timing, so the mixture formation process and fuel atomisation quality can be improved, thanks to allowing more time for fuel–air mixing.

It should also be noted that the trends of variation in ignition delay period as a function of i-EGR ratio and e-EGR ratio are quite different. The ignition delay period is slightly extended with the increase of e-EGR ratio and remains almost unchanged when the injection timing is within the range of $-18 \sim -14^{\circ}\text{CA}$ ATDC. On the other hand, with i-EGR ratio increases, the ignition delay period first prolongs and then shortens. For example, with injection timing fixed at -14°CA ATDC, the ignition delay period is reduced approximately from 15.9 to 14°CA as i-EGR ratio is increased approximately from 0 to 35%. This is mainly because the low-temperature reaction is accelerated on account of the heating effect of the hot residual gas. With i-EGR ratio continues to be increased to 50%, the residual gas contains a large number of combustion products with high specific heat capacity, such as CO_2 and H_2O , which lead to the increase of specific heat capacity of the mixture. At the same time, EGR also dilutes the in-cylinder oxygen concentration. Therefore, there is always a competitive relationship between heating effect and dilution and heat capacity effects, but in the case of i-EGR ratio being higher than 35%, the effect of heating is weaker than those of dilution and heat capacity, which can explain the prolonged ignition delay period.

Figure 5 shows the variations of the equivalence ratio as a function of i-EGR ratio and e-EGR ratio under different injection timings. With the increase in EGR ratio, the equivalence ratio increases gradually as well. When EGR ratio is lower than 15%, the equivalence ratios with both i-EGR and e-EGR are almost the same. As EGR ratio is increased approximately from 15% to 50%, the difference of equivalence ratio between i-EGR and e-EGR is significantly enlarged, and the equivalence ratio with i-EGR is gradually higher than that of e-EGR. This is mainly because of the fact that, as compared with the e-EGR case, when the in-cylinder oxygen concentration is the same based on Eq. 1, the heating effect of i-EGR leads to more expansion in charge volume, which prevents the intake airflow and causes a greater reduction in volumetric efficiency.

Figure 6 shows the variation of combustion duration as a function of i-EGR ratio and e-EGR ratio under different injection timings. The combustion duration is defined as the crank angles between CA10 and CA90 (the crank angle degree at which 90% of total heat release has taken place). For each of i-EGR and e-EGR cases, there occurs a minor difference of combustion duration with variable injection timings. As EGR ratio increases, the extension of combustion duration with i-EGR is more obvious than that with e-EGR, and the gap between i-EGR and e-EGR cases is enlarged as EGR ratio exceeds 25%. This can be attributed that applying e-EGR strategy allows more air into the cylinder with the same EGR ratio, so the dilution effect is lower than that of i-EGR, which also leads to an increase in oxygen concentration and the consequent burning rate.

Figure 7 shows the variations of CA50 as a function of i-EGR ratio and e-EGR ratio under different injection timings. Combined with the above results, CA50 shows a similar trend with the ignition delay period as a function of EGR ratios, mainly due to the almost unchanged combustion duration.

Figure 8 shows the variation of indicated thermal efficiency as a function of CA50 under different i-EGR and e-EGR ratios. The delayed CA50 leads to a reduction in the degree of constant volume, and the indicated ther-

Fig. 4 Effects of i-EGR and e-EGR ratios on ignition delay period.

Fig. 5 Effects of i-EGR and e-EGR ratios on equivalence ratio.

mal efficiency decreases as a consequence. As e-EGR ratio raised approximately from 0% to 45%, the indicated thermal efficiency is reduced slightly by about 2%. With the increase of i-EGR ratio, the indicated thermal efficiency first increases and then decreases. In the case of i-EGR ratio at 15%, where a slight increase of indicated thermal efficiency is observed as compared to the no EGR case; but there is an obvious decrease in the indicated thermal efficiency as i-EGR ratio raised approximately from 30% to 45%. The difference between i-EGR and e-EGR strategies on combustion process mainly depends on the heating effect and equivalence ratio. Compared with the e-EGR cases, the combustion behaviour with i-EGR has higher heat transfer loss and equivalence ratio, where the indicated thermal efficiencies are more deteriorated. From the above, the optimised indicated thermal efficiency could be available by combining a small amount of i-EGR under low-load GCI combustion.

From the above phenomenon, it can be concluded that there is indeed a competitive relationship between heating effect and dilution and heat capacity effects when i-EGR strategy is used. When the i-EGR ratio remains at a low level, the combustion process and burning rate are mainly affected by heating effect. Once the i-EGR ratio is increased to a certain value, the dilution and heat capacity effects begin to play a dominant role in affecting the combustion process.

Fig. 6 Effects of i-EGR and e-EGR ratios on combustion duration.

Fig. 7 Effects of i-EGR and e-EGR ratios on CA50.

3.2 Effects of CA50 and EGR on GCI emission characteristics

Figure 9 shows the variation of NO_x emission as a function of CA50 under different i-EGR/e-EGR ratios. With CA50 advanced, the combustion reaction rate and the peak combustion temperature increase, more NO_x is generated as a result. It can also be figured out that i-EGR has more influence on NO_x emission as compared with e-EGR. This could partially be attributed to that, with the same CA50, the shorter ignition delay period with i-EGR is against the fuel–air mixing and brings in a reduction in combustion rate, both of which are favourable for the suppression of NO_x emission. Furthermore, the previously mentioned increased equivalence ratio of i-EGR case, also causes the oxygen-enriched condition, is destroyed. Therefore, as compared with e-EGR case, NO_x emission with i-EGR is lower due to the above two factors. Even though the hot residual gas leads to an increase in the cylinder temperature, the dilution effect of exhaust gas still plays a dominant role in the combustion process under the current operating conditions.

Figure 10 shows the variation of soot emission as a function of CA50 under different i-EGR/e-EGR ratios. From the diagram, an increase in soot emission is observed by the more advanced CA50. This is mainly because

Fig. 8 Effects of CA50 and EGR ratio on indicated thermal efficiency.

the advanced CA50, or rather the early injection leads to an excessive fuel–air mixing, so that the regions of in-cylinder lean mixture are expanded, which benefits the soot polymerisation and condensation processes. The soot emission of i-EGR case is always slightly higher than that of e-EGR case, because the favourable conditions for fuel cracking and dehydration processes in the late combustion phase can be easily attained due to the heating effect of i-EGR. As i-EGR ratio increased from 30% to 45%, the soot emission shows a more obvious increase especially with CA50 closed to TDC. Thus, when using high i-EGR ratio, the retarded CA50 can effectively control the soot emission.

Fig. 9 Effects of CA50 and EGR ratio on NO_x emission.

Figures 11 and 12 show the variations of carbon monoxide (CO) and hydrocarbon (HC) emissions as a function of NO_x emission under different CA50, respectively. It can be seen that both CO and HC emissions have an obvious trade-off relationship with NO_x emission, and lower CO and HC emissions can be achieved by i-EGR with the same ratio as e-EGR. This is mainly because of the fact that the conditions which are associated with the formation of NO_x are in direct contradiction to those corresponding to the formation of CO and HC in the combustion with EGR involved, but the higher average combustion temperature caused by i-EGR can

Fig. 10 Effects of CA50 and EGR ratio on soot emission.

improve the oxidation of CO and HC in the late combustion process, and the advanced CA50 always result in the higher combustion temperature and the more complete combustion as well. Besides, the thinner quenching layer and the shorter ignition delay period shortened resulted by i-EGR are also beneficial to the control HC emission. Therefore, the optimised trade-off relationship between CO/HC and NO_x can be attained by the combination of i-EGR and more advanced CA50.

Fig. 11 Effects of CA50 and EGR ratio on CO emission.

3.3 Effect of coupling of i-EGR and e-EGR on GCI combustion and emission characteristics

Figure 13 shows the variation of equivalence ratio as a function of i-EGR/e-EGR ratio. With the total EGR ratio kept at 40% and the injection timing fixed at -18°CA ATDC, for example, when e-EGR ratio is increased from 5% to 35%, the lower equivalence ratio is achieved by the increase of volumetric efficiency based on the previous analysis, which also results in a decreasing in the flow intensity [32].

Figure 14 shows the variations of the cylinder pressure and heat release rate as a function of i-EGR/e-EGR ratio. As i-EGR ratio increases, the cylinder pressure before ignition decreases gradually due to the lower vol-

Fig. 12 Effects of CA50 and EGR ratio on the HC emission.

umetric efficiency as mentioned above. In addition, the combustion process is advanced owing to the improved reactivity of fuel–air mixture by the heating effect of i-EGR. However, the slower burning rate caused by the decreased oxygen concentration results in a reduction in the peaks of cylinder pressure and heat release rate.

Fig. 13 Effect of i-EGR ratio and e-EGR ratio on equivalence ratio.

Figure 15 shows the variations of exhaust temperature and indicated thermal efficiency as a function of i-EGR/e-EGR ratio. It is found that the indicated thermal efficiency is deteriorated by the increase of i-EGR ratio, but which is beneficial for the improvement of exhaust temperature and consequently the application of the after-treatment system. With i-EGR ratio increases, the amount of hot residual gas from the previous cycle increases; besides, the heat capacity of in-cylinder charge is decreased by the reduction in volumetric efficiency, and the average combustion temperature increases as a result. Both of the two factors play a significant role in the increase of exhaust temperature. The extension of combustion duration also leads to an increase in exhaust temperature; however, when i-EGR ratio increases, the heat transfer loss is increased by the higher average combustion temperature, and the longer combustion duration also results in a decrease in the degree of constant volume. Therefore, higher i-EGR ratios have a negative effect on the indicated thermal efficiency.

Figure 16 shows the variations of NO_x and soot emissions as a function of i-EGR/e-EGR ratio. NO_x emission remains at a low level resulted in the total EGR ratio of 40%. With the increase of i-EGR ratio,

Fig. 14 Effect of i-EGR ratio and e-EGR ratio on cylinder pressure and heat release rate.

Fig. 15 Effect of i-EGR and e-EGR ratios on exhaust temperature and indicated thermal efficiency.

NO_x emission shows an overall decreasing trend but increases slightly when i-EGR ratio is increased from 20% to 35%. Based on the previous analysis, the increased average combustion temperature caused by i-EGR is unfavourable for NO_x emission control. However, the lower oxygen concentration leads to a reduction in burning rate owing to the increased equivalence ratio, which is beneficial for the reduction of NO_x emission. Therefore, NO_x emission first decreases and then increases for these two factors. It is well known that, as the incomplete combustion product, soot is formed during the fuel cracking and dehydrogenation process under high temperature and hypoxia conditions. The suppression of soot emissions is mainly negatively affected by the heating effect and the increasing of local equivalent ratio caused by higher i-EGR. For this reason, soot emission increases continuously with high i-EGR ratio as shown in Figure 16.

Figure 17 shows the variations of CO and HC emissions as a function of i-EGR/e-EGR ratio. As i-EGR ratio raised from 5% to 35%, CO and HC emissions are reduced by 53.49% and 64.50%, respectively. This could be attributed to the diminished local low-temperature regions and the accelerated combustion rate as a result of the increase of average combustion temperature, which leads to a reduction in CO production due to the more complete combustion. Besides, the reactivity of fuel–air mixture is improved by the heating effect, the HC emission is, therefore, reduced due to the more accessible auto-ignition of gasoline and the diminished flame quenching regions. Furthermore, the oxidation processes of both CO and HC are promoted by the increase of residual gas fraction. Hence, applying higher i-EGR ratio is quite useful in the further control of CO and HC emissions.

Fig. 16 Effect of i-EGR and e-EGR ratios on NO_x and soot emission.

Fig. 17 Effect of i-EGR ratio on CO and HC emissions.

4 Conclusion

In the current research, the effects of i-EGR, e-EGR, i-EGR/e-EGR coupling and CA50 on GCI combustion and emission characteristics at the low-load condition are investigated. The main conclusions are listed as follows:

- With the increase of EGR ratio, the ignition delay period with e-EGR is extended slightly while that of i-EGR strategy first shortened and then extended. The higher equivalence ratio and longer combustion duration could be achieved by i-EGR strategy, and the optimised indicated thermal efficiency can be achieved using 15% i-EGR ratio.
- Compared with e-EGR case, NO_x emission is more likely to be suppressed by i-EGR with the same EGR ratio, while soot emission was more deteriorated. The superior trade-off relationship between CO/HC emissions and NO_x emission was achieved using i-EGR with lower ratios, and CO and HC emissions could be further reduced by CA50 closed to TDC.
- When using i-EGR/e-EGR coupling with total EGR ratio kept at 40%, the higher equivalence ratio, the lower indicated thermal efficiency and the higher exhaust temperature were attained by an increase in the i-EGR ratio; NO_x, CO and HC emissions are reduced by 62.35%, 53.49% and 64.50%, but soot emission increased.

Acknowledgement: This study is supported by the projects of the National Natural Science Foundation of China (Grant Number 51806127 and 52075307) and the Natural Science Foundation of Shandong Province (Grant Number ZR2016EEQ09).

References

- [1] Kalghatgi G, Levinsky H, Colket M (2018), Future transportation fuels, *Progress in Energy and Combustion Science*, 69, pp.103-105.
- [2] Araghi Y, Maarten K (2017), Identifying reasons for historic car ownership and use and policy implications: an explorative latent class analysis, *Transport Policy*, 56, pp.12-18.
- [3] Kalghatgi G (2018), Is it really the end of internal combustion engines and petroleum in Transport?, *Applied Energy*, 225, pp.965-974.
- [4] Johnson TV(2009), Diesel emission control in review, SAE Technical Paper 2009-01-0121.
- [5] Heywood J B(1988), *Internal Combustion Engine Fundamentals*, McGraw-Hill Inc., New York, USA.
- [6] Yao M, and Liu H (2012), Review and prospect of the combustion technology of homogeneous charge compression ignition and low temperature combustion, *Chinese Journal of Automotive Engineering*, pp. 79-90.
- [7] Jiang C, Li Z, Liu G, Qian Y (2019), Achieving high efficient gasoline compression ignition (GCI) combustion through the cooperative-control of fuel octane number and air intake conditions, *Fuel*, 242, pp.23-34.
- [8] Akihama K, Takatori Y, Inagaki K, Sasaki S, Dean AM (2010), Mechanism of the Smokeless Rich Diesel Combustion by Reducing Temperature, SAE Technical Papers, 20012001-01-0655.
- [9] Kalghatgi G (2005), Auto-ignition quality of practical fuels and implications for fuel requirements of future SI and HCCI engines, SAE Technical Paper 2005-01-0239.
- [10] Kalghatgi G, Risberg P, Ångström H-E (2006), Advantages of fuels with high resistance to auto-ignition in late-injection, low-temperature, compression ignition combustion, *ResearchGate*.
- [11] Cracknell R F, Ariztegui J, Dubois T (2014), Modelling a gasoline compression ignition (GCI) engine concept, SAE Technical Paper 2014-01-1305.
- [12] Solaka H, Aronsson U, Tuner M and Bengt J (2012), Investigation of partially premixed combustion characteristics in low load range with regards to fuel octane number in a light-duty diesel engine, SAE Technical Paper 2012-01-0684.
- [13] Borgqvist P, Tunestal P and Johansson B (2012), Gasoline partially premixed combustion in a light duty engine at low load and idle operating conditions, SAE Technical Paper 2012-01-0687.
- [14] Manente V, Zander C, Johansson B and Tunestal P (2010), An advanced internal combustion engine concept for low emissions and high efficiency from idle to max load using, SAE Technical Paper 2010-01-2198.
- [15] Dempsey AB, Curran SJ, and Wagner RM (2016), A perspective on the range of gasoline compression ignition combustion strategies for high engine efficiency and low NO_x and soot emissions: Effects of in-cylinder fuel stratification, *International Journal of Engine Research*, 17, pp.1-21.
- [16] Liu X, Tong L, and Wang H (2017), Experimental and modelling investigations of the gasoline compression ignition combustion in diesel engine, SAE Technical Paper 2017-01-0741.
- [17] E Parks James, Prikhodko V, Storey J M.E, Teresa L (2010), Emissions from premixed charge compression ignition (PCCI) combustion and affect on emission control devices, *Catalysis Today*, 151(3/4) pp.278-284.
- [18] Tesfa B, Mishra B, Zhang C, Gu F, and Ball AD (2013), Combustion and performance characteristics of CI (compression ignition) engine running with biodiesel, *Energy* 51, pp.101-115.
- [19] An Y, Raman V, and Tang Q (2019), Combustion stability study of partially premixed combustion with low-octane fuel at low engine load conditions, *Applied Energy*, 235, pp.56-67.
- [20] Zhang X, Zheng Z. and Huang S (2018), Effects of Intake Pressure on Gasoline Low Temperature Compression Ignition, *Chinese Internal Combustion Engine Engineering*, 39, pp.1-7.
- [21] Zhu Y, Wang H, Zhang X, Tong L, and Yao M (2016), Gasoline Compression Ignition Low-Load Combustion Optimization Based on Double Exhaust Valve Opening and Injection Strategy, *Transactions of CSICE*, 34, pp.415-422.
- [22] Borgqvist P, Tunestal P, and Johansson B (2013), Comparison of negative valve overlap (NVO) and rebreathing valve strategies on a gasoline PPC engine at low load and idle operating conditions, SAE Technical Paper 2013-01-0902.
- [23] Zhou L, Hua J, Wei H, and Han Y (2019), An experimental investigation on low load combustion stability and cold-firing capacity of a gasoline compression ignition engine, *Engineering*, 5, pp.558-567.
- [24] Weall A and Nick C, Gasoline Fuelled Partially Premixed Compression Ignition in a Light Duty Multi Cylinder Engine: A Study of Low Load and Low Speed Operation, SAE Technical Paper 2009-01-1791.
- [25] Wolters P, Salber W, Geiger J, and Duesmann M (2003), Controlled Auto Ignition Combustion Process with an Electromechanical Valve Train, SAE Technical Paper 2003-01-0032.
- [26] Zhang X, Wan H, and Zheng Z (2016), Effects of late intake valve closing (LIVC) and rebreathing valve strategies on diesel engine performance and emissions at low loads, *Applied Thermal Engineering*, 98, pp.310-319.

- [27] Tan Y, Zheng M, and Reader GT(2009), An Enabling Study of Diesel Low Temperature Combustion via Adaptive Control, SAE Technical Paper 2009-01-0730.
- [28] Jing, E(2012). Analysis of Characteristic Parameters of Combustion Process on Diesel Engine, MA. thesis, Jilin University, Changchun, China.
- [29] Du J, Sun W, and Xiao S (2005), Influence of Combustion Phase and EGR on Combustion and Emissions of Premixed Compression Ignition with Wide Distillation Range Fuels, Transactions of CSICE, 33, pp.110-116.
- [30] Ickes, A (2009). Fuel Property Impact on a Premixed Diesel Combustion Mode, Ph.D. thesis, Michigan: University of Michigan, Michigan.
- [31] Zhou L, Hua J, and Liu F (2018), Effect of internal exhaust gas recirculation on the combustion characteristics of gasoline compression ignition engine under low to idle conditions, Energy, 164, pp.306-315.
- [32] Zhang X, Zheng Z, Li Y, and Zhu Y (2017), Effects of Internal and External EGR and Injection Pressure on Low Temperature Combustion of Diesel Engine, Vehicle engine, 5, pp.1-5,10.