Acta Scientific Ophthalmology

Volume 2 Issue 1 February 2019

Review Article

Combustion System Development of GDI-T Engine for China Stage III Fuel Consumption Regulation

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Received: January 17, 2019; **Published:** February 04, 2019

Abstract

Chinese fuel consumption regulation will be more and more strict. China will implement Stage III fuel consumption regulation by the end of 2015. First Automobile Works (FAW) has developed a new GDI-T gasoline engine to meet this fuel consumption regulation for A class car. The rated power of this engine is 100 kW, and maximum torque of this engine is 220 Nm. The BSFC at 2000 r/min-2 bar BMEP reaches 373 g/kW.h. The acceleration time from 0 to 90% maximum load at 1500 r/min is less than 1.7 s. Matching with the curb weight 1265 kg of one A class car, maximum vehicle speed is 210.5 km/h and acceleration time of 0~100 km/h is 10 s. NEDC integrated fuel consumption is 5.85 L/100km, which is 15% lower than Chinese Stage III fuel consumption regulation meanwhile meeting the dynamic performance of the vehicle.

Keywords: Gasoline; GDI-T; Combustion System; Optical Engine Test; Pre-Ignition

Introduction

In China, with increasing of oil price and strict emissions regulations, high efficiency, low carbon, and low emissions have been necessary condition for developing a new gasoline product. It means that in-cylinder direct injection turbocharged gasoline engine with high rated power, low fuel consumption, low emission and fast response will be mainstream products in future.

In order to promote automobile factory developing energy-saving car, Chinese government has introduced corresponding fuel consumption regulations according to the actual situation of our country. Chinese government has published "Passenger car fuel consumption evaluation method and target" in 30th December, 2011. This regulation has come into effect in 1st January, 2012 in Beijing firstly. And the first year is introduction stage, then gradually strict every year, finally will be come into effect in the whole nation in 2015. Table 1. is the fuel consumption requirement of passenger car. Figure 1 is the milestone of fuel consumption target.

Vehicle kerb mass (CM) (kg)	Target value of fuel consumption (L/100 km)
CM≤750	5.2
750<CM≤865	5.5
865<CM≤980	5.8
980<CM≤1090	6.1
1090<CM≤1205	6.5
1205<CM≤1320	6.9
1320<CM≤1430	7.3
1430<CM≤1540	7.7
1540<CM≤1660	8.1
1660<CM≤1770	8.5
1770<CM≤1880	8.9
1880<CM≤2000	9.3
2000<CM≤2110	9.7
2110<CM≤2280	10.1
2280<CM≤2510	10.8
2510<CM	11.5

Table 1: Fuel consumption requirement of passenger car for China stage III.

Main parameters and design target of CA4GB GDI-T engine

The main parameters and performance targets are as below:

Table 2 is main parameters and performance targets. We can see that specific power of this engine is above average level and

Figure 1: Milestone of fuel consumption target.

Main Parameters and Performance Target	
Bore (mm)	76.5
Stroke (mm)	75.6
Displacement (L)	1.39
Cylinder Number	4
Firing Order	1-3-4-2
Compression Ratio	9.6
Max. Cylinder Pressure (MPa)	11
Fuel	RON 93
Power/Speed (kW/ r/min)	100/4500-5500
Max. Torque/Speed (N·m/ r/min)	220/1650-4350
Specific Power (kW/L)	72
Specific Torque (N·m /L)	158.2
Emission Level	EU V

Table 2: Main parameters and performance targets.

specific torque of this engine is at first class level from figure 2 and figure 3.

Figure 2: Benchmark of specific power.

Figure 3: Benchmark of specific torque.

Combustion system design

Figure 4 is the whole layout of CA4GB GDI-T combustion system. Fuel injector is side mounted.

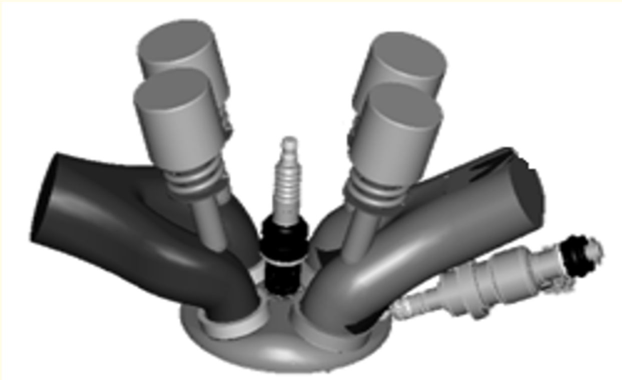

Figure 4: Layout of combustion system.

We use the high tumble intake port in this engine. The intake port flow characteristic is shown in figure 5. The average flow coefficient of intake port is 0.44 and the tumble ratio is 2.5. The flow and tumble of each cylinder is uniformity, the difference is within 1%.

The top surface shape of piston is shown in figure 6.

The maximum injection pressure is 15MPa. Hole patterns for 3 injectors are in table 3.

Table 4 shows main cam parameters. Maximum lift of intake valve is 9.5 mm and the duration of 1mm lift is 195°CA. Maxi-

Figure 5: Intake port flow characteristic.

Figure 6: Shape of piston chamber.

	Case 1	Case 2	Case 3
Spray Scheme			
Hole Number	6	7	6

Table 3: Hole pattern for 3 injectors.

mum lift of exhaust valve is 9.0 mm and the duration of 1mm lift is 181°CA. Cam profile of intake and exhaust valve are shown in figure 7.

	Lift/ mm	VC/1mm /°CA	VC/1mm /°CA	MOP/°CA
In Base Timing	9.5	382	577	478
Ex Base Timing	9.0	157	338	248
In Max Timing	9.5	332	527	428
Ex Max Timing	9.0	207	388	298

Table 4: Main cam parameters.

Figure 7: Cam profile of intake and exhaust valve.

Figure 8 is engine operation line in compressor map 1, named as case 1. Figure 9 is engine operation line in compressor map2, named as case 2. These two turbochargers both can meet engine performance requirement. Case 1 has more surge margin than case 2, but case 1 should be easy to over speed than case 2 in altitude environment. So we will make comparisons about two cases in performance test.

Figure 8: Operation line in turbocharger 1.

Figure 9: Operation line in turbocharger 2.

Optical engine test

In order to shorten the performance development period and choose the most proper injector, firstly we did optical single cylinder engine test.

Test was carried out at 3 engine operating conditions in optical engine including 1650 r/min-full load, 2000 r/min-2bar BMEP and 1200 r/min-1bar BMEP. We measured and recorded spray in cylinder, process of mixture formation and flame propagation and distribution after ignition on different injector scheme. We compared quality of mixture in cylinder for different timing, analyzed effect of flame kernel shape and soot, finally determined injector scheme and injection timing.

Figure 10 is the comparison of spray at 1650 r/min-full load with 3 injectors. We can see that there is wall hitting phenomenon with injector of case 2 at -225°CA ATDC. Wall hitting will lead to instantaneous dry friction between piston ring and cylinder wall, meanwhile increase oil dilution. There is also wall hitting phenomenon with injector of case 3 at -210°CA ATDC. There is wall hitting phenomenon with injector of case 2 or case 3 at -200°CA ATDC. There is no obvious wall hitting phenomenon with injector of case 1 in the whole process of spray.

Figure 10: Comparison of spray at 1650 r/min-full load.

Figure 11 is the comparison of deposition on piston surface with 3 injectors. We can see very clearly that there is obvious carbon deposition phenomenon on valve pit between intake and exhaust valve with injector of case 2 or case 3. It is means that fuel spreading is not good with injector of case 2 or case 3. Partial fuel stored in piston top, which lead to diffusion combustion in cylinder, then carbon deposit.

Figure 11: Comparison of deposition on piston surface.

Figure 12 is the comparison of spray at 2000 r/min-2bar BMEP with 3 injectors. We can see that there is no obvious wall hitting phenomenon with 3 injectors in the whole process of spray. 3 injectors all meet the requirements on this condition.

Figure 12: Comparison of spray at 2000 r/min-2bar BMEP.

Figure 13 and Figure 14 are the comparisons of spray at 1200 r/min-1bar BMEP with 3 injectors. We use split injection strategy on this condition. Injection timing of first injection is -250°CA ATDC and duration is 1.2ms. Injection timing of second injection is -110°CA ATDC and duration is 1.2ms. In Figure 13, we can see that there is wall hitting phenomenon with injector of case 3 at -238°CA ATDC. There is also wall hitting phenomenon with injector of case 2 at -234°CA ATDC. There is no wall hitting phenomenon with injector of case 1 in the whole first injection process of spray.

Figure 13: First injection comparison of spray at 1200 r/min-1bar BMEP.

Figure 14: Second injection comparison of spray at 1200 r/min-1bar.

From Figure 14, we can see that there is wall hitting phenomenon with injector of case 2 and injector of case 3 at -82°CA ATDC. There is no obvious wall hitting phenomenon with injector of case 1 in the whole second injection process of spray.

Finally, we choose injector of case 1 by optical engine test. Because there is no wall hitting and carbon deposit phenomenon on various conditions. So we use injector of case 1 in the following performance development test.

Performance development test

Turbocharger Test Results

Figure 15 is full load results of the two turbochargers. We can see that both turbochargers can achieve 220N·m maximum torque target value at 1500 r/min which is lower than target speed 1650 r/min. It can reduce lagging when turbocharged engine accelerate at low speed. Figure 16 is boost comparison of these two turbochargers. Torque of turbo case 2 from 1000 r/min to 2500 r/min is higher than turbo case 1, torque of turbo case 1 and case 2 from 2500 r/min to 5500 r/min are equivalent. But boost of turbo case 2 is lower than turbo case 1. Performance at low speed of turbo case 2 is better than turbo case 1.

Figure 15: Full load results of turbocharger.

Figure 17 is the comparison of fuel consumption at part load with two turbochargers. We choose four part load conditions as 2000 r/min-2bar BMEP, 2000 r/min-4bar BMEP, 3000 r/min-3bar BMEP, 4000 r/min-6bar BMEP. We can see that fuel consumption of turbo case 1 is better than turbo case 2.It will help to reduce vehicle fuel consumption.

Figure 16: **Boost comparison of turbocharger.**

Figure 17 : BSFC at part load with two turbochargers.

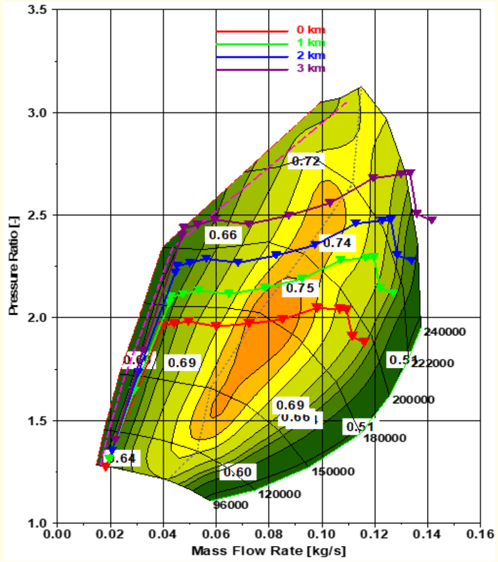

Figure 18 : Altitude capacity of turbo 1.

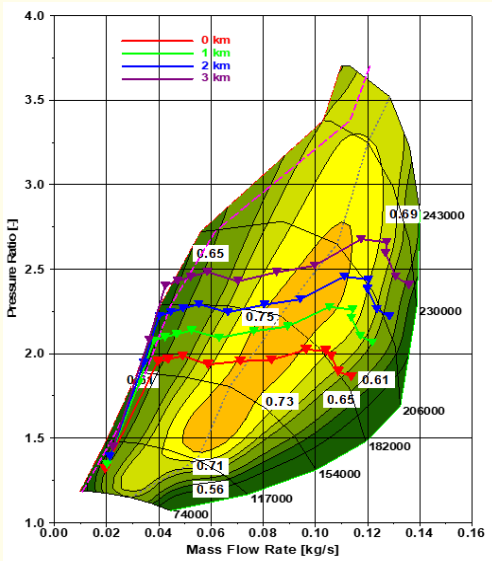

Figure 19 : Altitude capacity of turbo 2.

We predict operation line of altitude based on test results in the plain. Figure 18 and Figure 19 are altitude capacity of two turbochargers. The red line is at sea level, the green line is at altitude 1000m, the blue line is at altitude 2000m, the purple line is at altitude 3000m. We consider that there is 10% surge margin at sea level. Two turbochargers both meet this limit value. But turbo 2 is much easier to happening surge than turbo case 1. And turbo 1 is much easier to over speed than turbo 2 at altitude environment.

Turbocharger can not provide air and boost pressure of engine required in time when engine change the operating condition. There is lagging time for turbocharger. Acceleration capability of engine is worse as longer lagging time. Lagging phenomenon is more evident because of wider speed range and smaller inertia. Figure 20 and Figure 21 are transient response of two turbochargers. Engine speeds are at 1500 r/min and 2000 r/min. We change engine load from 10% maximum load to 90% maximum load, and then comparison response time of two turbochargers.

Table 5 is transient response time comparison results of two turbochargers. Response time at 1500 r/min of turbo case 1 is shorter than turbo case 2, but response time at 2000 r/min of turbo case 1 is longer than turbo case 2.

Figure 20 : Transient response of turbo 1.

Figure 21 : Transient response of turbo 2.

Speed /r/min	1500	2000
Torque /N·m	10%→90%	10%→90%
Turbo Case 1 /s	1.61	1.28
Turbo Case 2 /s	1.83	1.26

Table 5: Transient response time comparison results of two turbochargers.

Figure 22 is torque slop from knee to 90% maximum torque. We can see that this engine has very good transient response capacity.

Figure 22: Torque slope comparison of two turbochargers.

Overall consideration of full load performance, part load fuel consumption, altitude capacity and transient response capacity, we decide to choose turbo case 1.

Catalyst heating test performance

We choose 1200 r/min-1bar BMEP to represent catalyst heating behaviour. Lower specific HC emission and higher port specific heat flow is better for emission. Higher heat flow can heat catalytic converter faster, so that the catalyst can fast light off. We control 20°C coolant temperature and 20°C oil temperature. Figure 23 is results of catalyst heating. We can see that this engine has very low HC emission and very high heat capacity.

Figure 23: Catalyst heating results.

Oil dilution test results

Oil dilution not only impact life of lubricant oil but also thin oil film between liner and piston. Oil dilution is very important for direct injection engine. The oil dilution test including 5500 r/min-full load, 1500 r/min-full load, 2500 r/min-12bar BMEP on engine of cold condition and a cycle test of 6 hours duration. Figure 24 is the results of oil dilution test. We can see that the oil dilution ratio at every test condition is less than 6%.

Figure 24: Results of oil dilution test.

Final results of performance development

After completing whole performance test, maximum power of this engine achieve 100 kW, specific power achieve 72 kW/L. Maximum torque of this engine achieve 220 N·m from 1500 r/min to 4350 r/min, specific toque achieve 158.2 N·m/L. The result is shown in figure 25. BSFC at 2000 r/min-2 bar BMEP achieve 373 g/kW.h as figure 26.

Figure 25: Full load curve of this engine.

Figure 26: BSFC at 2000 r/min-2 bar BMEP (FEV Benchmark).

Irregular combustion test research

Per-ignition is an abnormal combustion phenomenon in cylinder before normal spark plug ignition. Super knock from pre-ignition is more destructive than conventional knock, and the mechanism is still not very clear. Pre-ignition easily appear at low engine speed and high load. In this research we use optical fibre testing system named VisioSet. This system can measure conventional knock, pre-ignition, post-ignition and flame propagation speed for direct injection engine. Fibre transducer is shown in figure 27. This is key components in the test system and is integrated in spark plug. There are 70 channels in fibre transducer including 4 layers. It is shown in figure 28. There are 28 channels in first layer and second layer. There are 7 channels in third layer and fourth layer. From first layer to third layer are mainly used for measuring knock and the fourth layer is mainly used for flame kernel formation and flame propagation direction after ignition.

Figure 27: Fibre transducer integrated spark plug.

Figure 28: Channel of transducer.

Lower diagram of Figure 29 is the test cycle for pre-ignition test. We can see that maximum cylinder pressure is no more than 10MPa in the first 5400s time. So we think that there is no pre-ignition phenomenon during this period. After operating condition is changed to 1500 r/min-full load, there are 3 times 5 cycles that maximum cylinder pressure exceed 10MPa. Generally, we think that it is acceptable for 1 time pre-ignition in 1 hour. So we think that there is pre-ignition phenomenon if we use single injection strategy.

Figure 29: Pre-ignition status with single injection.

Figure 30 is light intensity distribution in cylinder. We can see that there is uneven distribution of light intensity phenomenon in cylinder obviously. The light is stronger on pre-ignition position than others position. That shows that there is intense burning on pre-ignition position. We also see that the position of pre-ignition is not fixed, it means that pre-ignition is sporadic and uncertainty.

Figure 30: Light intensity distribution in cylinder.

Then we change the injection strategy into split injection. The results are shown in figure 31. We can see that maximum cylinder pressure is no more than 10MPa in the two 5400s durations. It means that there is no pre-ignition phenomenon in cylinder. Because when we use split injection strategy, distance of fuel penetration should be shorter. It can avoid fuel wall wetting on single injection condition. Meanwhile using split injection strategy can improve air movement in spark plug, promote the mixing of fuel and air, reduce the pre-ignition tendency. But because pre-ignition is sporadic and uncertainty, we can not control the pre-ignition by using split injection strategy completely. So split injection only can be a calibration method for restraining pre-ignition.

Simulation results in vehicle

Table 6 is simulation results of dynamic performance and fuel consumption in vehicle. The maximum vehicle speed is 210.5 km/h, acceleration time of 0~100 km/h is 10 s. NEDC integrated fuel con-

Figure 31: Pre-ignition status with split injection strategy.

sumption is 5.85 L/100km which is 15% lower than China`s Stage III fuel consumption regulation [1-4].

Item	Unit	Target Value	Simulation Results
Max. Vehicle Speed	km/h	208	210.5
Acceleration Time [0~100 km]	s	10.2	10
Fuel Consumption [NEDC Cycle]	L/100 km	6.9 (China`s Stage III)	5.85

Table 6: Simulation results in vehicle.

Conclusion

We have finished engine calibration work and have not finished the vehicle calibration. So the vehicle performance data is the simulation data.

Summary and Limitations

We have developed a new GDI-T gasoline engine by thermodynamic simulation, optical engine test and performance development test etc. Finally, NEDC integrated fuel consumption matched this GDI-T engine is 15% lower than China`s Stage III fuel consumption regulation meanwhile meeting the dynamic performance of the vehicle.

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