

Simulation and Experiment Investigation on Performance Development of Gasoline Engine

Xiaomao Zhang¹, Weimin Gao¹, Jun Xin², Ming Chen²

¹ Tongji University, School of Automotive Studies, Shanghai, China

² SAIC Motor Technical Center, Engine Department, Shanghai, China

zhangxiaomao@saicmotor.com

Abstract. In order to meet the requirements of high fuel combustion efficiency, balanced torque and power output, the performance development of a 1.5L small gasoline engine has been carried out. The technologies for high power and low fuel consumption include optimized combustion system and optimized gas exchange system. During the process of engine development, advanced engine simulation tools were applied to optimize design parameters in order to minimize the experimental iterations. The performance experiment of the engine has been verified. The test results show that the power is 80kW and the maximum torque is 136N.m. The brake specific fuel consumption (BSFC) at part load condition of 2000r/min & 2bar is 378g/kW/h. The engine performance achieves the development target.

Keywords: Gasoline engine; Performance development; CFD simulation; Gas exchange optimisation

1. Introduction

The role of the automotive industry on a country's economy is of great importance. As China economy continues to grow, the nation's auto industry is expanding at an unprecedented rate. 2010 annual auto sales in China exceeded 18 million units. As a result, more than a half of annually consumed crude oil is imported, and exhaust emissions from automobiles have been the main contributor to the deterioration of air quality in big cities. Furthermore, it is also well recognized worldwide that green-house gas (carbon dioxides) is causing the global warming [1, 2].

In order to address the environmental issues and the issue of energy independence, EU IV emission limit has been adopted in big cities such as Beijing and Shanghai. To encourage consumers to purchase vehicles with less fuel consumption, various tax incentives were introduced in the last two years. Phase III fuel consumption regulation was proposed and had been confirmed. The newly proposed fuel consumption targets call for approximately 20% fuel economy improvement for vehicles with manual transmissions comparing to Phase II targets.

In China, the market demand of around 1300kg weight small passenger car is very vast because of balanced Chinese requirement and cost. This passenger car matches 1.4L~1.6L port fuel injection (PFI) gasoline engine in general. The engine typically runs at part load below 2500rpm speed conditions in city area and runs at medium or high load from 2500r/min~3500r/min speed conditions in rural area and on the highway[3]. To save the energy and protect the environment, a small gasoline engine has been developed to meet the vehicle requirements. This paper describes the technologies adopted for a 1.5L natural aspirated gasoline engine performance development.

2. Engine Configuration Description

The development engine specification is shown in Table 1. The specific power target is 52kW/L and specific torque is 90N.m/L, which locates medium or high level in comparison with similar competitive

engines. As we known, fuel consumption at part load conditions is very important for New European Drive Cycle (NEDC) of the passenger vehicle. The BSFC target of typical condition of 2000r/min & 2bar is 380g/kW/h, which is competitive.

Table 1. Engine Specifications

Engine type	L4 Port Fuel Injection Engine, Naturally Aspirated
Bore [mm]	75
Stroke [mm]	84.8
Compression Ratio	10.5 : 1
Valve train	DOHC, Intake VCT
Max. Power /kW (@r/min)	78 (@6000)
Max. Torque /N.m (@r/min)	135 (@4500)
Intake cam profile	7.7mm×226deg
Exhaust cam profile	8.2mm×236deg

3. Combustion System Development

The geometry of the combustion system of the conception engine is shown in Figure 1.

The conventional pent roof combustion chamber is adopted. In order to improve the performance and fuel economy of the engine, the intake and exhaust ports are analyzed with CFD and flow test techniques.

3.1. Intake Port Design and CFD Analysis

It is very important that intake port forms reasonable tumble air to enter into the cylinder for gasoline engine, which influences succeeding combustion quality greatly [4, 5, 6]. According to the intake port test result, the tumble ratio of the conception intake port is 50% lower than the competitive engines. Hence, the conception port should be optimized. The optimization intake port is shown in Figure 2. The shape of intake port was modified largely based on the conception port. Additionally, the machine cutter line between the port core and valve seat was redesigned to improve flowing smoothly. Moreover, the optimization port core was considered shift tolerance to increase production performance robust.

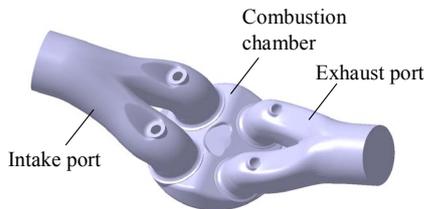


Figure1. Geometry Model of Conception Combustion System

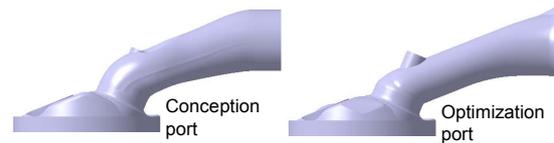


Figure2. Geometry of Conception and Optimization Intake Port

Steady state CFD simulation analysis of conception and optimization intake port is carried out. The CFD analysis is done under steady state condition with Star-CCM+ software. The simulation adopts steady state incompressible flow and k-epsilon turbulence model. The inlet is set total pressure (relative) 0 Pa and the outlet pressure is set -3430 Pa. The valve lift is set as 8 mm. The ambient pressure is 1.01bar and temperature is 300K. The flow coefficient is calculated based on the bore diameter. The tumble ratio is calculated at a half bore diameter location below the bottom surface of the cylinder head. The detailed steady state CFD assessment of intake port refers to document [7].

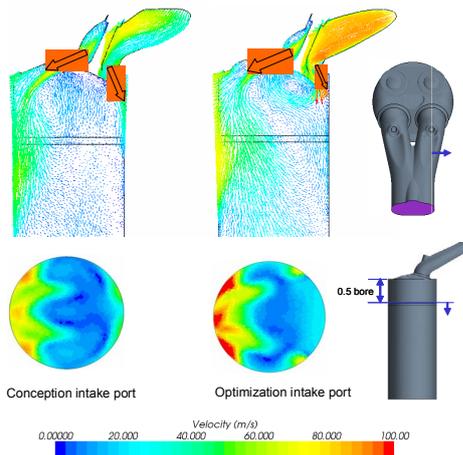


Figure 3. CFD Analysis of Conception and Optimization Intake Ports

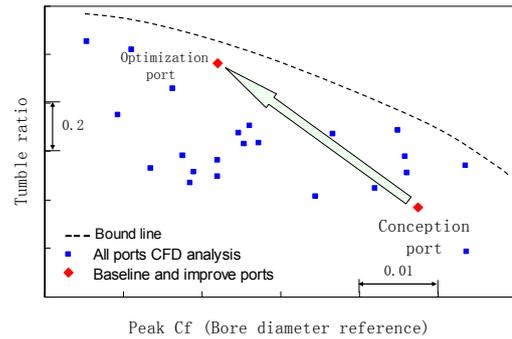


Figure 4. Comparison of CFD Analysis of Intake Ports

Figure 3 compares the velocity vectors of the conception and the optimization intake ports at 8mm. The intake air flow velocity through the upper side of the back valve is almost equal to the lower side of the back valve for the conception port, but for the optimization port, the air flow velocity through the upper side is much larger than the lower side. As a result, the tumble ratio of optimization port is higher than the conception port. Figure 4 shows the tumble ratio is relative to the flow coefficient, in which the conception port and the optimization port are compared with the benchmark ports. Although the flow coefficient of the optimization port decreases 14%, the tumble ratio increases 55% in comparison with the conception port. In general, there is a trade-off between the flow coefficient and the tumble ratio [8]. The optimization port increases tumble ratio obviously, which benefit flame propagation speed.

3.2. Exhaust Port Design and Analysis

Low flow resistance of exhaust port benefit exhaust gas scavenges from the cylinder which can decrease pump loss and fuel consumption.

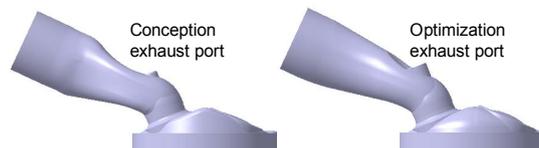


Figure 5. Geometry of Conception and Optimization Exhaust Port

The steady state flow test shows that the flow coefficient of the conception exhaust port locates medium level and the resistance can decrease through the optimization design. Figure 5 shows the geometry difference of conception and optimization exhaust ports. The shape of optimization exhaust port is smoother than the conception exhaust port. The port test results shown as Figure 6 indicate that the flow coefficient of the optimization port increase 10% or so in comparison with the conception port at typical exhaust valve lift that is equal to 0.35 minimum exhaust valve inner seat diameters. The improved effect is obvious.

3.3. In-cylinder Transient flow CFD Analysis

The in-cylinder transient flow CFD analysis of the intake and compression strokes for the conception and the optimization combustion system has been carried out. The optimization combustion system includes new intake port analyzed above. The inlet and exhaust boundary conditions adopt pressure and temperature, which are from one dimension gas exchange simulation results. The analysis condition is at full load at 2000r/min engine speed. The crank angle of the piston top dead center (TDC) of the compression stroke is 0 crank angles ($^{\circ}\text{CA}$).

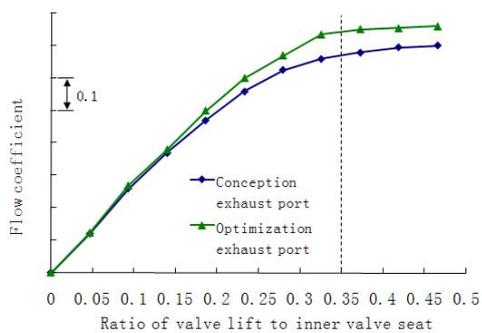


Figure 6. Comparison of Flow Test of Exhaust Ports

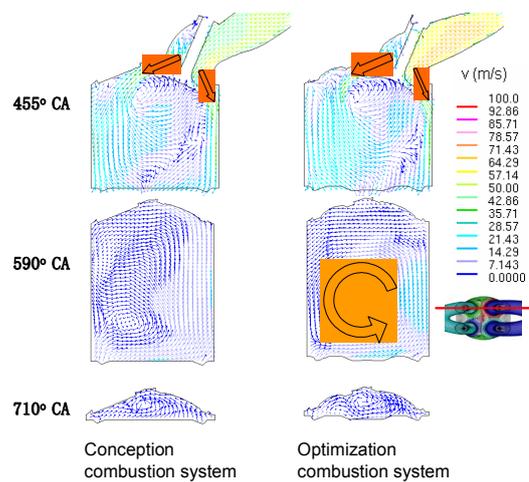


Figure 7. Comparison of Velocity Magnitude

Figure 7 shows the comparison of velocity magnitude between the conception and the optimization combustion systems at several typical crank angles. At the maximum lift of the intake valve (455 °CA), the intake air flow velocity through the upper side of the back valve is almost equal to the lower side of the back valve for the conception port, but for the optimization port, the upper air flow velocity is much larger than that of the lower side, which is similar to steady state CFD analysis results. Therefore the transient tumble ratio of the optimization port is higher than the conception port. When the intake valve closes at 590 °CA, the optimization combustion system generates more structured large-scale eddy than the conception combustion system, which keep this phenomenon near the piston top dead center of the compression stroke. Figure 8 is the tumble ratio comparison of two combustion system. The transient tumble ratio of the optimization combustion system is higher than the conception because of the structured large-scale eddy. The maximum tumble ratio of the conception combustion system is 0.9357 and the optimization combustion system is 1.8862 at the maximum intake valve lift. The optimization combustion system significantly increases the tumble ratio in cylinder, in comparison with the conception system.

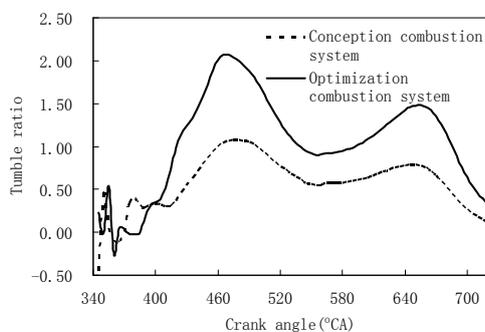


Figure 8. Comparison of Tumble Ratio

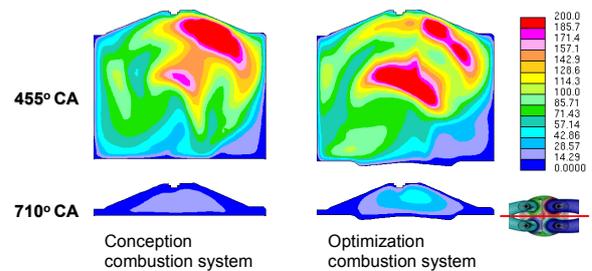


Figure 9. Comparison of TKE at 455 °CA and 710 °CA

Figure 9 shows that the turbulence kinetic energy (TKE) distribution of the optimization combustion system is higher and wider in cylinder than that of the conception combustion system because of the higher tumble ratio. The high TKE allows a faster propagation of flame [9]. The TKE of the optimization combustion system around spark plug is higher than the conception combustion system at ignition. As a whole, the optimization combustion system significantly improves the combustion efficiency.

4. Gas Exchange Optimization

Intake and exhaust cam profiles have significant impact on volumetric efficiency (VE) and in-cylinder flow motion, which affects combustion efficiency. Therefore, it is critical to optimize the cam profiles to achieve torque and power targets. One dimensional gas dynamic simulation software WAVE is used to simulate the gas exchange. Figure 10 is the diagram of engine's WAVE model. Figure 11 is the comparison of measured and calculated volumetric efficiency over the operating engine speed range. It appears that good

agreement between simulation and measurement is achieved. By the within 5% difference of volumetric efficiency between the simulation and the measurement is meant the calibration completion which is reasonable [10].

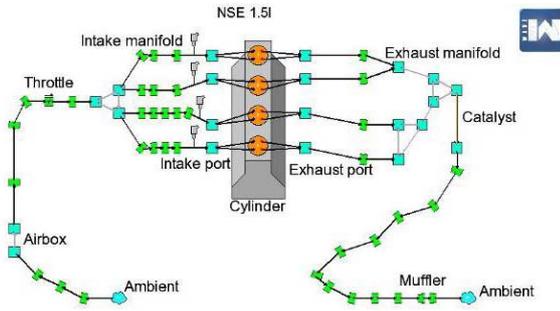


Figure 10. WAVE Model of Engine

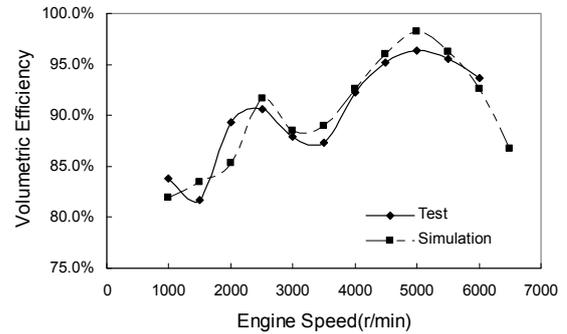


Figure 11. Calibration of WAVE model with test results

Table 2. Proposals of the intake cam profile

Proposals	Valve lift(mm)	Valve duration(°CA)
Option 1	6.5	208
Option 2	6.9	214
Option 3	7.3	220
Option 4	7.7	226
Option 5	8.0	230

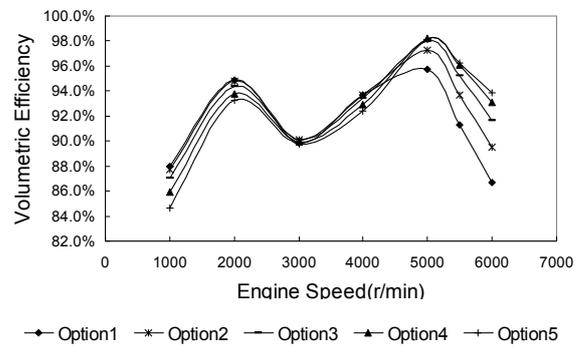


Figure 12. Comparison of VE for Different Intake Cam Profiles

The valve lift and duration, which influence greatly the engine volumetric efficiency has been analyzed. There are five options of intake cam profiles, as shown in Table 2. The results are shown in Figure 12. With the increasing of the valve lift, the volumetric efficiency at high engine speed increases. There is a trade-off between rated power and low end torque. For the optimization engine, the moderate option 4 is chosen for balance between end torque and rated power.

The intake manifold optimization mainly concentrates on two aspects: runner length and plenum volume. Four intake manifolds with different plenum volume are tested. Figure 13 shows the effect of the plenum volume on engine torque curve. At low engine speed, the engine torque increases as the intake manifold volume is reduced. The engine torque from 4000rpm to 5500rpm decreases as the plenum volume is reduced. Considering the low end torque and high speed power, the plenum volume of 0.67L is chosen for baseline and improved engines.

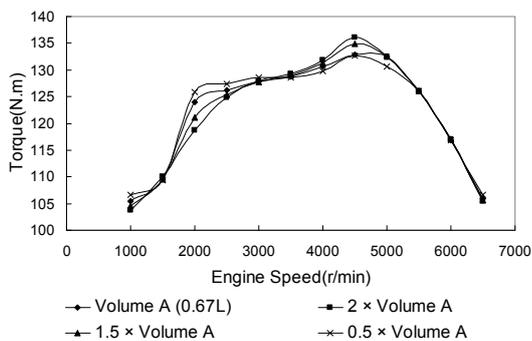


Figure 13. Effect of Runner Length of Intake Manifold on Engine Torque

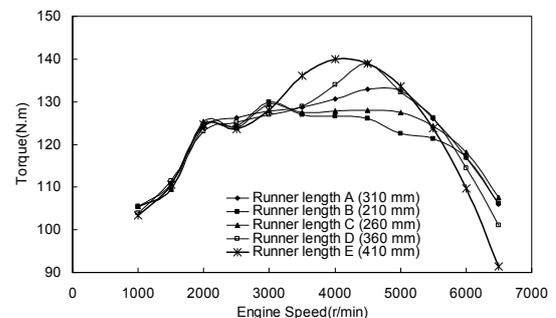


Figure 14. Effect of Runner Length of Intake Manifold on Engine Torque

The runner length of the intake manifold also influences the engine performance greatly. The volumetric efficiency curve of the engine moves to low speed (or high speed) if increase (or decrease) the intake manifold runner length because of resonance effects. In general, long runner length will benefit engine torque at low speed and short runner length will benefit high speed performance.

Figure 14 shows that below 3000rpm the engine torque changes little when changing the runner length. From 3000 rpm to 5500 rpm, the torque increases for longer runner length, but the torque decreases much at the speed over 5000rpm. Short runner length will only benefit engine power at high speed. The runner length 310mm gives a balanced performance at low and high speed. It gives a moderate performance at medium speed. After the optimization, the runner length of 310mm is adopted for the engine.

5. Engine Experiment Results

The engine performance tests of the engine have been verified. Figure 15 shows the experimental setup.

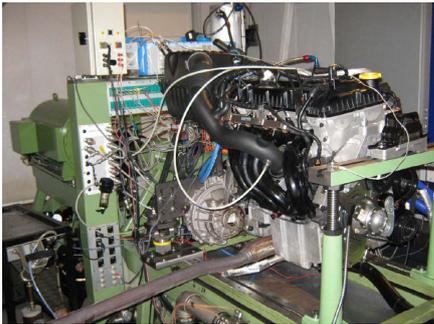


Figure 15. Engine Experimental Setup

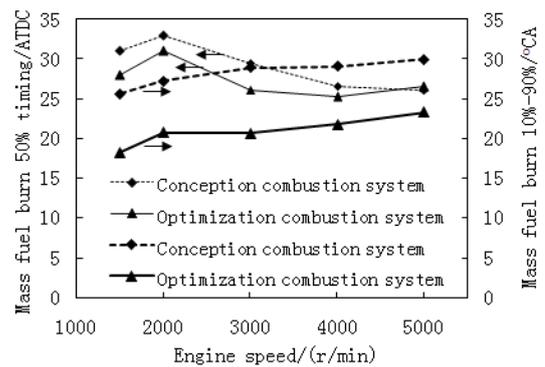


Figure 16. Comparison of fuel mass burn rate

During the test at full loads, ignition timing is adjusted to the maximum torque without knock occurrence, at the same time, controlling the same air-fuel ratio (AFR) and exhaust gas temperature below 850°C before the catalyst. Figure 16 shows that 50% burn fraction of fuel mass of the optimization combustion system advances about 2°CA, mass burn duration is 7°CA shorter than that of the conception combustion system at full load conditions. Figure 17 shows that the power is 80kW and the maximum torque is 136N.m. The fuel consumption (BSFC) at part load condition of 2000r/min & 2bar is 378g/kW/h, which achieves the performance development target.

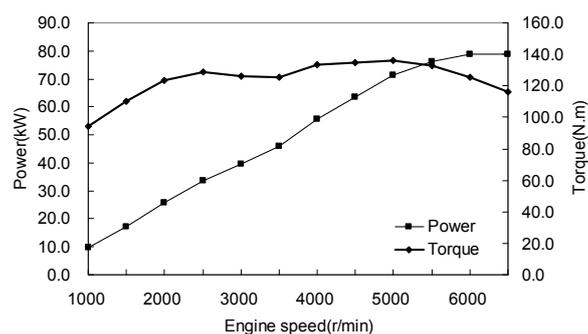


Figure 17. Power and Torque at Full Load

6. Conclusions

The performance development of a 1.5L gasoline engine was carried out. In order to improve the engine performance, the combustion system has been optimized and analyzed with CAD/CFD technologies. The intake manifold and cam profile have also been optimized with CAE technologies. The experiment shows that the power is 80kW and the maximum torque is 136N.m. The fuel consumption (BSFC) at part load condition of 2000r/min & 2bar is 378g/kW/h. The performance of the engine achieves the development target.

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