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Design and Selection of Piston for Engines with Compression Release Brake System

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Abstract

This paper introduces the process of piston design and selection during the development of compression-release brake on a heavy-duty engine. The design process of piston valve avoidance pit is introduced in detail; In order to reduce the negative impact caused by the increase of harmful volume due to the deepening of valve avoidance pit, keep the compression ratio unchanged and optimize the piston shape; Evaluate the five piston schemes optimized by simulation from the perspective of performance and emission, select the two pistons with the best performance, make samples and carry out subsequent test verification, and finally determine the piston with the best power, economy and emission by comparing the new piston scheme with the test results of the original engine piston. The product has been applied to the market, and the customer's brake performance of the engine Both power and economy are highly evaluated.

Keywords

Engine auxiliary brake; compression release brake; piston; valve avoiding pit.

1. Introduction

Engine assisted braking, especially compression-release braking, has become the standard configuration of heavy-duty engines due to its high braking power, low cost, small size, high reliability and high quality performance-price ratio.

The principle of compression-release braking is shown in Fig. 1. When the piston is close to the top dead center during the compression stroke, the exhaust valve opens to release the energy of the compressed gas stored in the engine cylinder into the exhaust system. Therefore, there is no energy to back pressure the piston [1] during the expansion stroke, and the cylinder is in a negative pressure state, which can absorb the kinetic energy of the whole vehicle, and finally achieve the purpose of braking the whole vehicle.

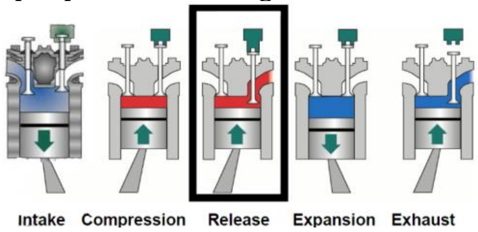

Fig. 1. Schematic diagram of compression-release braking

The main reason why the brake mode of compression-release braking can achieve higher braking power is that at the end of the compression stroke, the exhaust valve opens, and the valve lift at this time is large, which can often reach 3mm-4mm, which can release the compressed gas in the cylinder as much as possible. Because of the large lift of the brake

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exhaust valve and the opening of the accessory at the top dead center of compression, this brings the risk of the valve colliding with the piston. In order to avoid this risk and prevent the exhaust valve colliding with the piston, the position of the exhaust valve on the piston is generally corresponding to increase the valve avoidance pit. Although the compression-release braking can produce higher braking power, the valve avoidance pit is often harmful to the positive power performance of the engine. If the design is not good, it will often affect the performance and emissions of the engine under the normal ignition state.

This paper introduces the piston development process of an engine when improving its braking performance, which is roughly as follows:

1) During the development of compression-release braking, the lift of the brake exhaust valve at TDC increased by 3.02 mm, and the depth of the closed valve pit of the piston increased by 1.97 mm. The reduced volume of the closed valve pit is harmful to the performance and emissions of the engine in the ignition state. In order to minimize this hazard and optimize the combustion chamber, five new combustion chambers were developed;

2) By means of performance simulation, two piston samples with the best performance are selected from the aspects of performance and emission;

3) Calibrate the newly manufactured sample on the bench and compare it with the original machine, finally select the best piston and complete the selection.

2. Design of piston valve avoidance pit

2.1. Causes and influencing factors of valve escape pit

For four-stroke engines, in order to inhale as much fresh air as possible and exhaust more combustion gases when working under normal ignition conditions, the intake and exhaust valves will open early before the top dead center (TDC) and close late after TDC when designing the valve timing. In the TDC accessory, the clearance between the intake and exhaust valves and the piston has a minimum value, and the change process is shown in Fig. 2 [2].

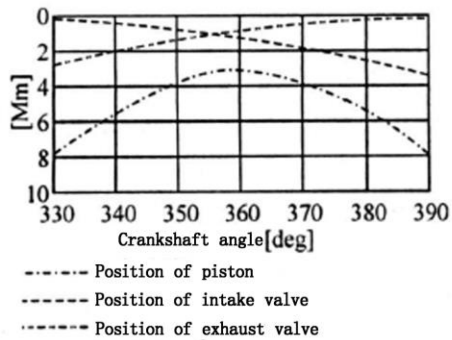

Fig. 2. Clearance between piston top surface and valve bottom surface

The factors that affect the design of the piston head anti-valve pit include: piston lift, intake and exhaust valve lift, valve timing, cylinder cushion compression thickness, the distance between the top surface of the piston and the top surface of the cylinder block at TDC, the distance from the lowest point of the valve to the bottom surface of the cylinder head, valve inclination, etc. [2].

2.2. Valve avoidance pit design of this engine

When upgrading the brake power of a heavy-duty engine product applied in the market, the compression-release brake device is used. The exhaust valve starts to open before TDC of the compression stroke, reaches the maximum value at TDC, and then closes slowly. Therefore, at

TDC, the distance between the brake exhaust valve and the piston is the minimum. According to the evaluation needs, the valve escape pit is set. The detailed process is as follows:

After the compression release brake device is added to the engine, in the braking state:

TDC: piston top surface is 0.2mm from the upper plane of the engine body (piston protrusion)

TDC: brake exhaust valve lift 3.02mm

Compression thickness of cylinder gasket: 1.2mm

Distance between exhaust valve bottom and cylinder head bottom: 0.05mm (valve sinking)

Minimum allowable theoretical clearance between piston and valve: 1.35mm

According to the calculation, the depth of valve escape pit is 3.02-(1.2-0.2-0.05)+ 1.35=3.42(mm). Fig. 3 shows the optimized depth of valve escape pit, and Fig. 4 shows the original depth of valve escape pit.

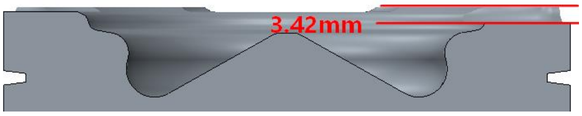

Fig. 3. Optimized depth of valve avoidance pit

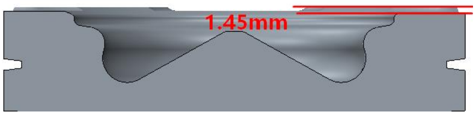

Fig. 4. Original depth of valve avoidance pit

2.3. Impact assessment of deepening of valve escape pit

The depth of the valve escape pit increased from 1.45mm before the engine upgrade to 3.42mm, resulting in an increase of 0.2ml in the combustion chamber volume, a decrease of 0.35 in the compression ratio, and a decrease from 17 in the original engine state to 16.65 [3,4,5].

The compression ratio is one of the important parameters to evaluate the engine performance. When the compression ratio decreases, it will cause the performance change of the engine in the positive power state. In order to reduce the harm of the harmful volume increase caused by the increase of the valve pit, the scheme of adjusting the compression ratio and optimizing the combustion chamber has been adopted. However, the change of the combustion chamber will still affect the positive power performance of the engine. Therefore, the dynamic performance of the optimized piston will be re-evaluated Economy and emission performance.

2.4. Piston combustion chamber design

Optimization scheme of piston: the valve pit is deepened to 3.42 mm, the combustion chamber is optimized, and the compression ratio is adjusted to 17.

A total of 5 combustion chamber schemes are designed, and the combustion chamber shape of each scheme is shown in Fig. 5.

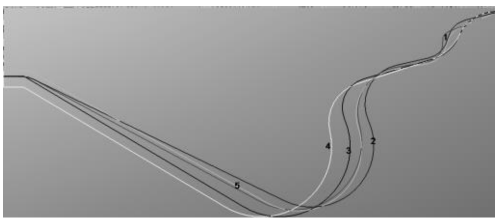

Fig. 5. Piston combustion chamber scheme

3. Performance simulation

From two aspects of performance and emission, compare the five newly designed schemes with the original machine, and recommend two schemes to make samples. In performance simulation, power, fuel consumption and emissions are mainly considered.

3.1. Power

Fig. 6 shows the power comparison between the five newly designed piston combustion chamber schemes, CASE1-CASE5, and the original engine scheme YJ. A total of 12 operating points are selected for power: low speed 1000r/min, high torque speed 1300 r/min and rated speed 2200 r/min, and the load rate is 25%, 50%, 75% and 100% of the speed and load rate combination. After weighting each working condition, the difference between CASE 1-CASE 5 and the original YJ is shown in Fig. 7. CASE 3 is the best, followed by CASE 4 and CASE 5, and CASE 1 and CASE 2 are the worst.

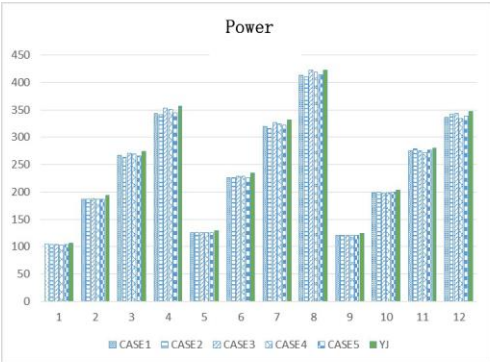

Fig. 6. Power of each combustion chamber scheme

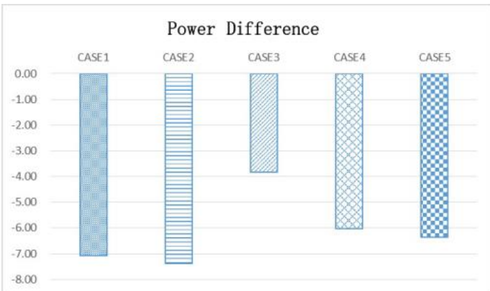

Fig. 7. Power difference

3.2. Fuel consumption

The operating point and power selected for the fuel consumption calculation are the same. The calculated fuel consumption is degraded to different degrees compared with the original engine. The difference of the degradation rate is shown in Fig. 8. After weighting each operating condition, the deterioration rate of each scheme and the difference of the original engine are obtained, as shown in Fig. 9. CASE 3 is the best, followed by CASE 4 and CASE 5, and CASE 1 and CASE 2 are the worst.

The comparison results of power and fuel consumption are consistent. CASE 3 is the best, and CASE 4 and CASE 5 are equivalent. Considering that the profile of CASE 5 is the closest to that of the piston combustion chamber of the original engine, CASE 3 and CASE 5 are selected for emission verification.

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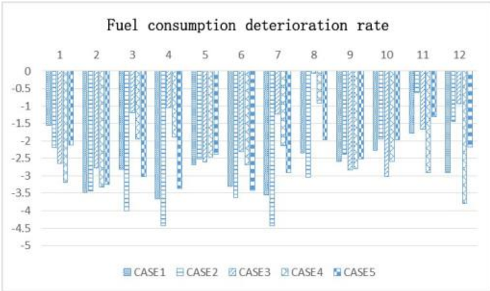

Fig. 8. Fuel consumption deterioration rate

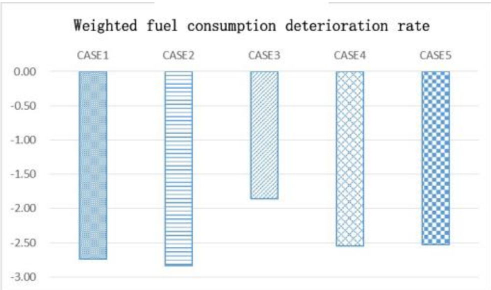

Fig. 9. Weighted fuel consumption deterioration rate

3.3. Emission analysis

Through simulation, only qualitative analysis can be carried out on emissions. The approximate evaluation results are shown in Fig. 10 and Fig. 11. The NO of CASE 3 scheme increases more and the SOOT decreases; In CASE 5 scheme, NOx decreases and Soot increases.

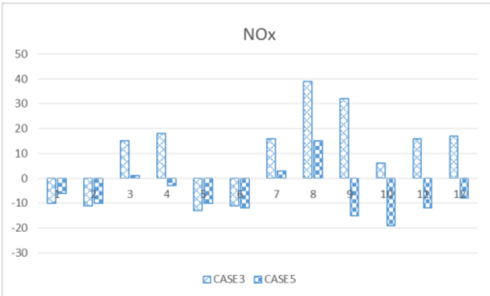

Fig. 10. NOx of CASE 3 and CASE 5

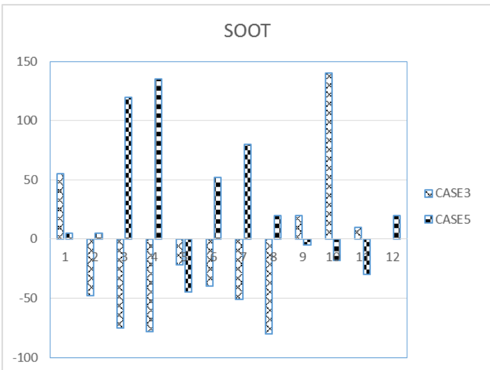

Fig. 11. Soot of CASE 3 and CASE 5

The conclusion of the performance simulation evaluation is that the power and fuel consumption of CASE 3 and CASE 5 are less degraded, and sample pieces can be made to further test and verify the performance and emissions. Fig. 12 shows the comparison of the combustion chamber shapes of CASE 3, CASE 5 and the original engine.

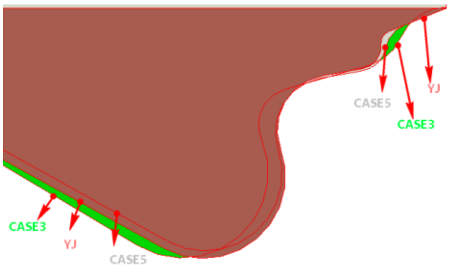

Fig. 12. Comparison of combustion chamber shapes of CASE 3, CASE 5 and the original engine

4. Performance test comparison

The performance tests were carried out on two samples of CASE 3 and CASE 5, and compared with the power, economy and emission performance of the original engine piston.

The test equipment is shown in TABLE Ⅰ:

TABLE I. TEST INSTRUMENTS AND METERS

Serial number	name	model	accuracy	range
one	Opacimeter	4390G004	±2%	(0-10) m-1
two	Fuel consumption meter	FC2212L	±0.20%	(0-120) KG/H
three	Engine measurement and control system	FC2000	0.5%	-
four	Filter paper smokemeter	415SG002	±0.1%	(0-10)FSN
five	Fuel temperature control	753CH	±1℃	(10-80)℃

4.1. ESC

The difference between the ESC emission results of CASE 3 and CASE 5 and the original machine is shown in TABLE Ⅱ. The BSFC of the two schemes are deteriorated, and CASE 5 is better than CASE 3; NOX is deteriorated, and CASE 5 is better; PM trends are different. CASE 5 is better than the original machine, and CASE 3 is degraded. From the perspective of ESC emissions, CASE 5 is preferred.

TABLE II. ESC EMISSION RESULTS

Difference	BSFC (g/kwh)	Nox (g/kwh)	PM (g/kwh)
CASE5-YJ	zero point two one	zero point zero eight	-0.0003
CASE3-YJ	zero point two five	zero point two eight	zero point zero zero four four

The comparison of ESC single-point fuel consumption is carried out. The results are shown in Fig. 13. The weighted fuel consumption of the three schemes is not different. Compared with the original machine, CASE 5 is slightly worse and CASE 3 is slightly better.

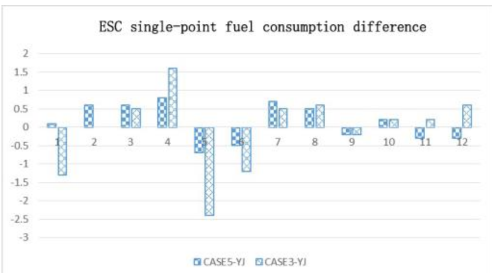

Fig. 13. ESC single-point fuel consumption difference

4.2. DOE test

In order to further compare the differences, DOE test was designed for comparison, keeping the rail pressure and advance angle unchanged, and comparing the fuel consumption of each scheme. As shown in Fig. 14, both schemes were degraded, with CASE 5 being (0-1) g/kwh worse than the original engine, CASE 3 being (1-3) g/kwh worse than the original engine, and CASE 5 being significantly better than CASE 3.

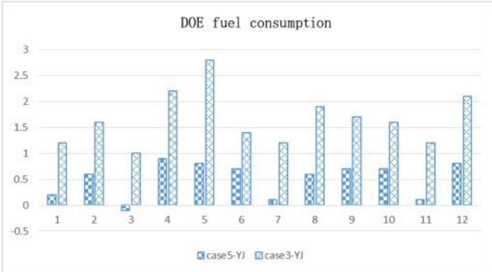

Fig. 14. DOE fuel consumption

4.3. External characteristic

Conduct the external characteristic test to test the torque and fuel consumption difference between each scheme and the original engine, as shown in Fig. 15 and Fig. 16. Compared with the original engine, the power and economy of CASE 3 and CASE 5 are deteriorated, and the deterioration of CASE 5 is small: in the external characteristic, the fuel consumption increases (0.5-1.5) g, and the external specific torque decreases within 10 Nm, which is acceptable.

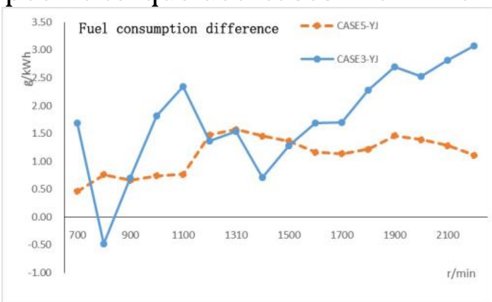

Fig. 15. Fuel consumption difference between CASE 3 and CASE 5 and the original engine in external characteristics

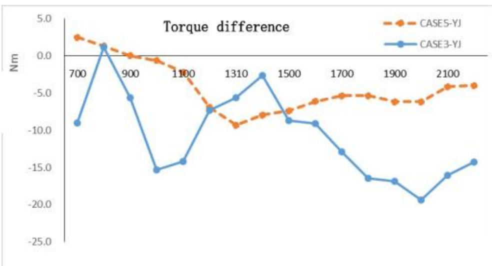

Fig. 16. Torque difference between CASE 3 and CASE 5 and the original engine in external characteristics

5. Conclusions

In order to meet the increasing demand of the market for brake power, the development of engine assisted braking, especially compression-release braking, is the standard equipment of heavy-duty trucks. In order to obtain higher brake power, the brake exhaust valve lift is larger, and the larger brake valve lift requires deeper valve avoidance pit. The increase of harmful volume of piston valve avoidance pit has brought about the deterioration of engine positive

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performance. In order to minimize this negative impact, Five new piston optimization schemes have been designed to keep the compression ratio unchanged and optimize the shape of the piston. Through simulation and test, the power, economy and emissions of the newly optimized piston have been comprehensively evaluated. The final selected piston can not only meet the requirements of high braking performance of the engine, but also has no deterioration in the performance of the positive power. The product has been applied to the market, and customers are very satisfied with the market performance of engine braking, power and economy.

Acknowledgments

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