A Study on the Combustion System of a Spark Ignition Natural Gas Engine

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ABSTRACT

This paper describes the development of a natural gas engine with a prechamber. The comparison between a conventional gas engine and the engine with the prechamber is given. The testing results show that the new combustion system is reliable and practical. The performance of the engine, compared to the conventional gas engine, is greatly improved due to stable ignition and quick flame propagation in the prechamber and a high burning rate and short burning period in main chamber. The analyses of the combustion processes and lean combustion potential in the engine are further discussed.

INTRODUCTION

The importance of energy conservation and environmental protection has been generally realized in recent years due to the decrease of oil resources and serious pollution of the atmosphere in the world. The engine running on natural gas, at high thermal efficiency and low exhaust emissions, will be one of the engines that are most competitive by the end of this century. Research on natural gas engines has been increasing since the 1980's, especially on combustion, and a lot of significant results have been given [1,2,3].

The main constituent of natural gas is methane which has some oxidation characteristics that are different from other hydrocarbons. Because of high ignition energy, narrow combustible limits and a low burning speed of methane in air, the combustion period in spark ignited natural gas engines might become too long, resulting in high exhaust temperature and low thermal efficiency.

This paper describes a new prechamber combustion system in a conventional spark ignited natural gas engine to increase burning speed and improve combustion processes. With the condition that the original engine cylinder heads were not changed the authors designed a number of key parts, such as the prechamber assembly, intake check valve, etc. The research results show that these parts are practicable, reliable, and easily developed . As the mixture in the prechamber is ignited stably and the flame propagates quickly, the combustion duration in main chamber is considerably shortened. As a result, the performance of the engine with the prechamber is clearly improved.

DESIGN FOR PRECHAMBER COMBUSTION SYSTEM

OVERALL ARRANGEMENT

The prototype refers to a conventional natural gas engine 2190T produced by Shengli Power Machinery Plant. The specifications of the test gas engine have been shown in Table 1. The overall arrangement of the prechamber combustion system has been showed in Figure 1. The induction system of the main chamber was not been changed, but the main mixer was adjusted while testing so as to provide a lean mixture for the main chamber during the intake stroke. The partial pure gas from the front of the main mixer enters into the submixer and forms a rich mixture with the air introduced in the submixer. The rich mixture enters into the prechamber through the check valve and then forms a stoichiometric mixture with a fraction of the lean mixture forced from the main chamber during the compression stroke. Then the mixture in the prechamber is ignited via a conventional spark plug and ignition system.

Table 1 Engine specifications

Engine	2190T
No. Cylinders	2
Bore	190 mm
Stroke	220 mm
Induction	Naturally Aspirated
Compression Ratio	8.6:1
No. Valves	4
Piston Type	Bathtub

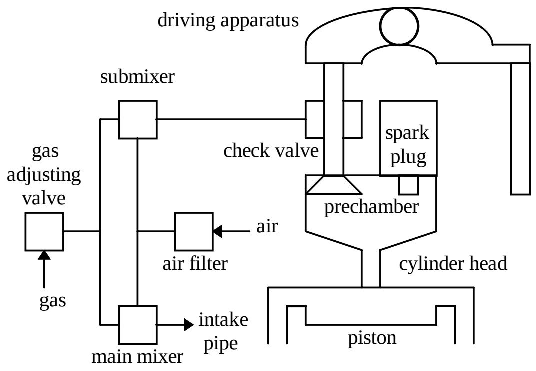

Figure 1. Overall arrangement of prechamber combustion system

The driving apparatus of check valve is mechanical, in which a special press block directly drives the check valve while the press block moves up and down simultaneously with the intake valves. A spark plug is located in the top of the prechamber where the mixture is ignited. After the mixture in the prechamber burns rapidly, a high temperature and high pressure flame quickly propagates through the prechamber and initiates combustion of the lean mixture in the main chamber. This process has the ability to operate under overall leaner conditions than open chamber configurations due to the higher ignition energy jetted from the prechamber when compared to conventional spark ignition [3].

PRECHAMBER

It is important to determine the volume of the prechamber. Too small of a prechamber volume results in too low ignition energy which is not strong enough to ignite the lean mixture in the main chamber. But too large volume of prechamber might make the prechamber assembly hard to install in the cylinder head. The ratio of the volume of prechamber to the total clearance volume is approximately (2~5)% according to the results of W. E. Snyder [1] and T. Nakazono [2] . The ratio was designed to 4% in this paper so that the cylinder heads of the prototype were not changed.

The prechamber assembly is installed in the place of the cylinder head instead of the spark plug coolant jacket of the prototype. The cross section of the prechamber assembly is shown schematically in Figure 2. The prechamber assembly consists of the upper part and the lower part. The spark plug, check valve, and pressure transducer are located in the top of the upper part. The prechamber orifice is fixed in the lower part, which is seated on a modified portion of the cylinder head. The inner wall of the lower part is designed to an approximately spherical surface to facilitate formation of a vortex in the prechamber.

The two parts are securely connected together by the threads. The assembly is compressed down in an annular groove of the cylinder head from the top surface of the upper part by six bolts which secure the assembly and allow sufficient clamp load. During testing, various forms of the lower part may be used to study the effect of the number, the diameter and the direction of orifices on combustion processes.

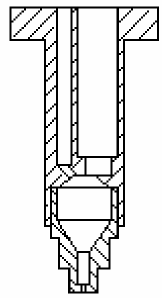

Figure 2. Cross section of the prechamber assembly

MAIN CHAMBER

The configuration of the main chamber in the piston crown was designed to a deep bathtub type instead of a shallow one like the original piston. There are two obvious merits in this design . One of them is to keep the production of the piston crown from changing too much. The other one is to increase the squish areas on the surface of the piston crown, which is conducive to forming the turbulence of the mixture in the main prechamber.

To prevent detonation, it is important to determine a reasonable compression ratio according to the chemical composition of natural gas provided by the local gas utility. Low compression ratio makes the thermal efficiency of engine decrease, while high compression ratio results easily in detonation. The compression ratio used in this study was 10:1. In this case the ratio of the squish area of the piston crown to the area of the piston cross section was 64%, compared to 46% for the original piston.

PARTS

The check valve group was composed of the check valve, the valve guide with the valve seat, the valve spring, the spring retainer, and the valve cotter as shown in Figure 3. The reliability of check valve is especially important for normal combustion in the prechamber because misfires can be caused if this valve sticks for any reason. The check valve used in the study has operated reliably for 100 hours without any sign that the check valve was sticking, by means of the reasonable design and the high process precision.

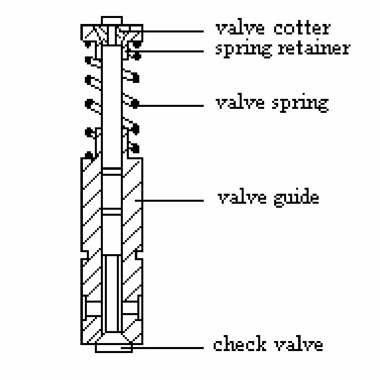

Figure 3. Check valve group

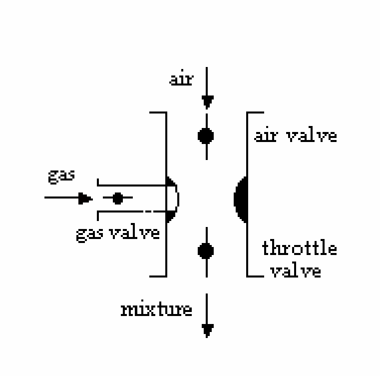

Figure 4. Scheme of the submixer

The submixer operates in such a way like a venturi carburetor. The scheme of the submixer is shown in Figure 4. The rich mixture with suitable flow can be provided for the prechamber by adjusting the air valve, the gas valve, and the throttle valve.

The press block used to drive the check valve was fixed in a bridge driven by the inlet rocker arm. Therefore the block moves up and down simultaneously with the inlet valves. The gap between the block and the subvalve may be adjusted by twisting the regulating screw in the block.

The rich mixture is introduced into the prechamber against the electrodes of the spark plug. By means of the arrangement the exhaust gas in the gaps between the electrodes can been cleaned out in time. At the same time, the central electrode of the spark plug is cooled to some extent. As a result, the performance of ignition and the reliability of operation of the spark plug are improved.

EXPERIMENTAL RESULTS

The following are the comparisons between the engine with the prechamber combustion system (PCC) and the original engine (2190T) at same rated condition.

IGNITION ADVANCE

Figure 5 shows the ignition advance curves of the both engines for power and exhaust temperature at the rated condition. It can be seen that the optimal ignition advance of the PCC engine is 19°C crank angle lower than that of the 2190T engine, while the exhaust temperatures decrease approximately 100°C for every cylinders. This means that the combustion duration of the PCC engine is shorter than that of the 2190T engine owing to the higher velocity of flame propagation in the combustion chambers. There have been some problems for the original engines in use as a result of high exhaust temperature. Accordingly, the use of PCC can improve evidently the combustion processes in 2190T engines.

(a) power

(b) exhaust temperature

Figure 5. Performance of ignition advance

PERFORMANCE

The performance results of both engines tested are shown in Figure 6 and Figure 7. Figure 6 shows the load curves for thermal consumption ratio and exhaust temperatures. Although the methane numbers (MN) of natural gases used in the two engines are basically same, the thermal consumption ratio and the exhaust temperatures of the PCC engine decrease 6.5% and 70°C or more, respectively, on the whole than those of the 2190T engine. At the rated condition the thermal consumption ratio of the PCC engine decreases 10%, and the mean exhaust temperature is 527°C, decreasing 85°C, compared with the 2190T engine.

(a) thermal consumption ratio

(b) exhaust temperature

Figure 6. Load curve (1000 rpm)

Figure 7 is a comparison of excess air ratio vs. load for the cylinders of the two engines. It has been found that the mixtures of the PCC engine were leaner than those of the 2190T engine on the whole. This can easily explain for the PCC engine why the thermal consumption ratio and exhaust temperatures were both decreased.

Figure 7. Excess air ratio vs. load

EMISSION

Figures 8~10 show the emissions results taken for the two engines tested at 1000 rpm under the same loads as shown in Figure 7. Examination of Figure 8 reveals that the CO production from the cylinder 1 of the PCC engine is reduced to almost one half than that of the 2190T engine, which used the richer mixtures below 0.90 excess air ratio. The CO production from the cylinder 2 of the PCC engine is as low as that of the 2190T engine, which used the mixtures above 0.90 excess air ratio.

Examination of HC emissions as shown in Figure 9 yields almost inverse results whether from the cylinder 1 or the cylinder 2. It stands to reason that the PCC engine produces more HC emissions as a result of its lower exhaust temperatures than the 2190T engine.

As for NOx emissions, we see from Figure 10 that the PCC engine exhibits lower NOx. Again we see that the trends of NOx as a function of load are quite smooth for the PCC engine. Generally speaking, NOx production for an engine tested depends on the peak cylinder temperature, the excess air ratio and the combustion duration. Hence, it may be confirmed that the lower NOx emission is a result of the lower peak cylinder temperature in the PCC engine.

CYLINDER PRESSURE

Figures 11~12 show the cylinder pressure and the pressure rise in the cylinder 1 of the PCC engine. Though the cylinder pressures and the pressure rises in the main chamber and the prechamber decrease as load lessens, the peak pressure rise in the prechamber is high even at light load. At 14% load (10kW), for instance, the peak pressure rise is up to 0.52 (MPa/°CA) as shown in Figure 12(b). This means that the ignition and the flame propagation in the prechamber are normal and reliable, which play important roles of initiating combustion of the lean mixture in the main chamber.

(a) Cylinder 1

(b) Cylinder 2

Figure 8. CO emissions

Figure 9 (a). Cylinder 1 HC emissions

Figure 9 (b). Cylinder 2 HC emissions

(a) Cylinder 1

(b) Cylinder 2

Figure 10. NO_X emissions

(a) cylinder pressure

(b) pressure rise

Figure 11. Cylinder pressure and pressure rise (1000 rpm, 30kW)

Figure 12(a) Cylinder pressure (1000 rpm, 10kW)

Figure 13(b). Pressure diagram (1000 rpm, 40 kW)

Figure 12(b). Cylinder pressure rise (1000 rpm, 10kW)

Figure 13(a). Pressure diagram (1000 rpm, 70 kW)

Figure 13(c). Pressure diagram (1000 rpm, 10 kW)

Figure 13 is a comparison of the cylinder pressure traces in cylinder 1 for the two engines at 1000 rpm and under the different loads. It follows from this that the beginning of combustion in the main chamber almost are same, approximately 3° BTDC, for the two engines under the different loads. Yet the ignition advance of the PCC engine has been lessened 19° crank angle. That is to say, the duration, from the ignition of the spark plug to the beginning of combustion in the main chamber, of the PCC engine is much shorter than that of the 2190T engine. This is due to the good characteristics of the prechamber on ignition, flame propagation and jet penetration.

At any load the peak cylinder pressures in the main chamber for the PCC engine nearly occur at same point, while those for the 2190T engine decrease 6° CA as load lessens from 70 kW to 10 kW. Therefore, we have the reason to believe that the burning duration in the main chamber for the PCC engine also is shorter than that for the 2190T engine at any condition.

Figure 14. Exhaust temperature

EXCESS AIR RATIO

Excess air ratio is an important parameter to affect performances of a S.I. gas engine. The following is a brief examination regarding the effect of excess air ratio on the performance of the PCC engine at the rated condition. Figure 14 shows the exhaust temperature measured as a function of the excess air ratio for the two cylinders. The exhaust temperature reaches the top around 1.0 excess air ratio, while whether rich or lean mixture may make the exhaust temperature drop. This is very useful while regulating the cylinder-cylinder mixtures for a multicylinder gas engine.

Figure 15. Power for the given level position

Figure 16. Fuel consumption for the given level position

Figure 15~16 show the effect of mean excess air ratio on power and fuel consumption when the governor level was fixed in the position at which the engine may operate at the rated condition. As the mean excess air ratio is decreased or increased from 1.0, the power of the engine becomes low. Yet the fuel consumption always decreases as the mean excess air ratio is increased whether for rich or lean mixture. This result reveals that lean combustion may make both maximum power and fuel consumption decrease for a certain type engine. Therefore, the appropriate degree of lean combustion must be determined in terms of the uses of a natural gas engine.

Figure 17 illustrates the fuel consumption curve vs. the excess air ratio in cylinder 2 at the condition that the rated power was not changed. This graph represents the potential of lean combustion for the PCC engine. It has been proved by testing that if the rated power is allowed to lessen, the use of leaner mixture, above 1.45 excess air ratio in cylinder 2, still obtained the normal combustion without the appearance of misfire.

Figure 17. Fuel consumption

CONCLUSION

1. The authors have developed successfully the prechamber combustion system in a type 2190T natural gas engine. The operation of 100 hours showed that the PCC engine was reliable and practical.

2. The PCC system tested had excellent characteristics, that is, stable ignition and quick flame propagation in prechamber and high burning rate and short burning duration in main chamber.

3. At the rated condition the thermal consumption ratio of the PCC engine decreased 10%, and the mean exhaust temperature was 527°C, decreasing 85°C, compared with the 2190T engine. NOx and CO emissions were decreased, while HC emissions and peak pressure rise were increased for the PCC engine.

4. At the condition of the rated power unchanged, lean combustion made fuel consumption, exhaust temperature and NOx decrease, but maximum power was decreased for a certain type engine. Future work could examine improvement in lean operating engines.

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